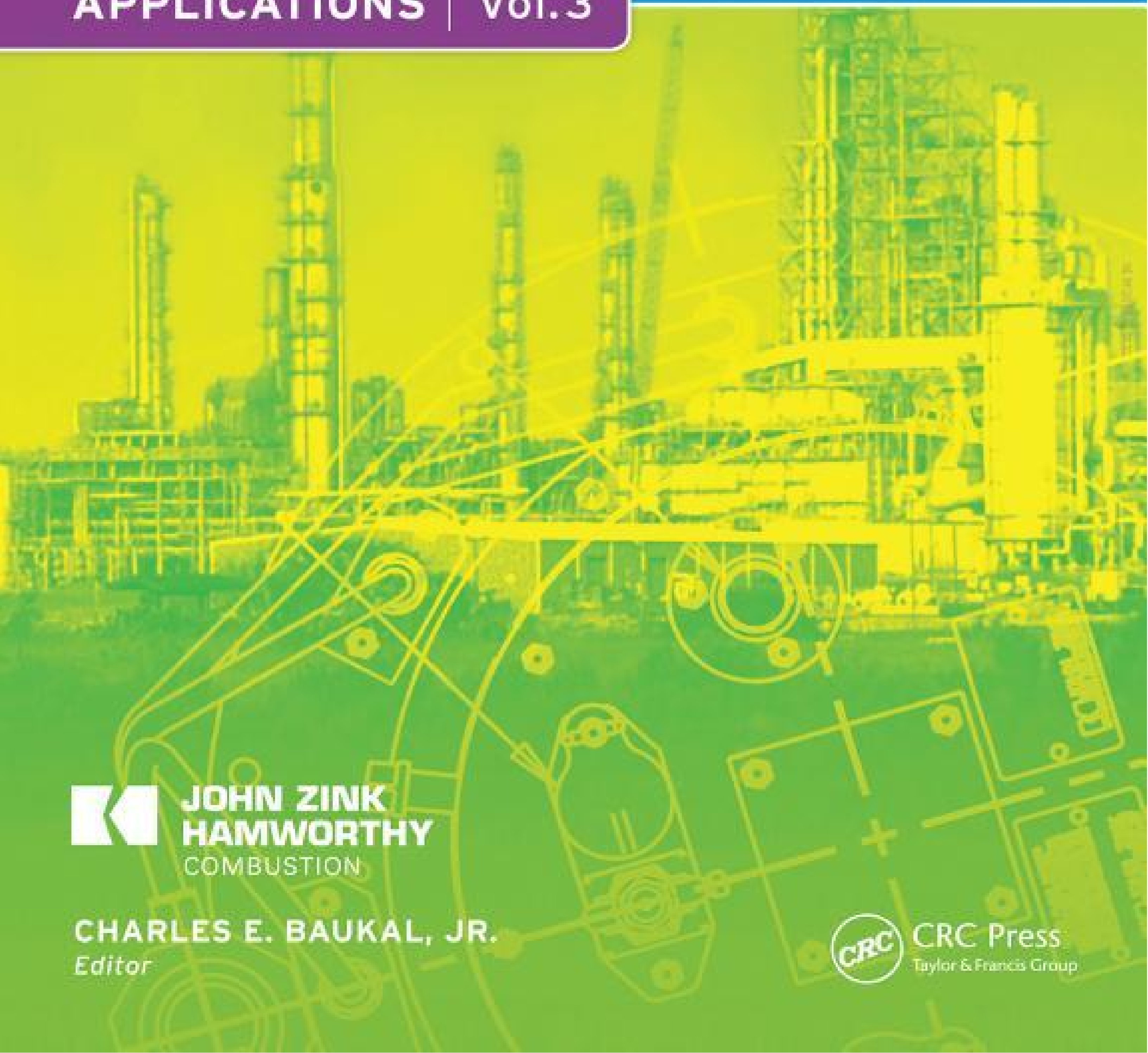
2nd Edition

THE JOHN ZINK HAMWORTHY COMBUSTION HANDBOOK

APPLICATIONS | Vol.3



JOHN ZINK
HAMWORTHY
COMBUSTION

CHARLES E. BAUKAL, JR.
Editor

CRC Press
Taylor & Francis Group

THE JOHN ZINK HAMWORTHY

COMBUSTION HANDBOOK

SECOND EDITION

Volume 3

APPLICATIONS

INDUSTRIAL COMBUSTION SERIES

Series Editors:
Charles E. Baukal, Jr.

The Coen & Hamworthy Combustion Handbook:
Fundamentals for Power, Marine & Industrial Applications
Stephen Londerville and Charles E. Baukal, Jr.

The John Zink Hamworthy Combustion Handbook, Second Edition
Volume 1 — Fundamentals
Volume II — Design and Operations
Volume II1 — Applications
Charles E. Baukal, Jr.

Industrial Burners Handbook
Charles E. Baukal, Jr.

The John Zink Combustion Handbook
Charles E. Baukal, Jr.

Computational Fluid Dynamics in Industrial Combustion
Charles E. Baukal, Jr., Vladimir Gershtein, and Xianming Jimmy Li

Heat Transfer in Industrial Combustion
Charles E. Baukal, Jr.

Oxygen-Enhanced Combustion
Charles E. Baukal, Jr.

THE JOHN ZINK HAMWORTHY

COMBUSTION HANDBOOK

SECOND EDITION

Volume 3

APPLICATIONS

Edited by

Charles E. Baukal, Jr.

CRC Press
Taylor & Francis Group
Boca Raton London New York

CRC Press is an imprint of the
Taylor & Francis Group, an **informa** business

CRC Press
Taylor & Francis Group
6000 Broken Sound Parkway NW, Suite 300
Boca Raton, FL 33487-2742

© 2014 by Taylor & Francis Group, LLC
CRC Press is an imprint of Taylor & Francis Group, an Informa business

No claim to original U.S. Government works
Version Date: 20130524

International Standard Book Number-13: 978-1-4398-4075-7 (eBook - PDF)

This book contains information obtained from authentic and highly regarded sources. Reasonable efforts have been made to publish reliable data and information, but the author and publisher cannot assume responsibility for the validity of all materials or the consequences of their use. The authors and publishers have attempted to trace the copyright holders of all material reproduced in this publication and apologize to copyright holders if permission to publish in this form has not been obtained. If any copyright material has not been acknowledged please write and let us know so we may rectify in any future reprint.

Except as permitted under U.S. Copyright Law, no part of this book may be reprinted, reproduced, transmitted, or utilized in any form by any electronic, mechanical, or other means, now known or hereafter invented, including photocopying, microfilming, and recording, or in any information storage or retrieval system, without written permission from the publishers.

For permission to photocopy or use material electronically from this work, please access www.copyright.com (http://www.copyright.com/) or contact the Copyright Clearance Center, Inc. (CCC), 222 Rosewood Drive, Danvers, MA 01923, 978-750-8400. CCC is a not-for-profit organization that provides licenses and registration for a variety of users. For organizations that have been granted a photocopy license by the CCC, a separate system of payment has been arranged.

Trademark Notice: Product or corporate names may be trademarks or registered trademarks, and are used only for identification and explanation without intent to infringe.

Visit the Taylor & Francis Web site at
http://www.taylorandfrancis.com

and the CRC Press Web site at
http://www.crcpress.com

Dedication

The staff at John Zink Company dedicates this book to Robert E. Schwartz. Bob, as he is known by his friends, worked for John Zink from 1967 till his retirement in 2011. During his tenure at John Zink, Bob served in many key roles, including vice president of the company. His expertise was wide ranging and included significant technological achievement in the areas of process burners, flares, and vapor recovery systems. Bob authored numerous publications and is named as an inventor on 52 U.S. patents. As a result of his proficiency in these areas, Bob is widely considered an expert in the industry. Active in the American Petroleum Institute, Bob made significant contributions to several standards, and in 2011 he received a lifetime achievement award for his many years of valuable service. Bob was a major contributor to the design, construction, and operation of the extensive combustion testing facility at John Zink Company headquarters located in Tulsa, Oklahoma. As a colleague, Bob always made himself available to anyone seeking his advice and took particular care with inexperienced engineers to provide them not only the advice they sought but also to teach them critical thinking and problem-solving skills. Bob is known and respected by combustion experts the world over. He left an indelible mark on the company he called home for over 44 years, and his legacy will be long lasting. Thank you, Bob, for your dedicated service. Your knowledge, wisdom, and friendship will be treasured for years to come.

Contents

List of Figures

List of Tables

Foreword to the First Edition

As we enter the twenty-first century, the importance of energy for industry, transportation, and electricity generation in our daily lives is profound. Combustion of fossil fuels is by far the predominant source of energy today and will likely remain that way for many years to come.

Combustion has played major roles in human civilization, including both practical and mystical ones. Since man discovered how to create fire, we have relied on combustion to perform a variety of tasks. Fire was first used for heating and cooking, and later to manufacture tools and weapons. For all practical purposes, it was not until the onset of the Industrial Revolution in the nineteenth century that man started to harness power from combustion. We have made rapid progress in the application of combustion systems since then, and many industries have come into existence as a direct result of this achievement.

Demands placed on combustion systems change continuously with time and are becoming more stringent. The safety of combustion systems has always been essential, but emphasis on effective heat transfer, temperature uniformity, equipment scale-up, efficiency, controls, and—more recently—environmental emissions and combustion-generated noise has evolved over time. Such demands create tremendous challenges for combustion engineers. These challenges have been successfully met in most applications by combining experience and sound engineering practices with creative and innovative problem-solving.

Understanding combustion requires knowledge of the fundamentals: turbulent mixing, heat transfer, and chemical kinetics. The complex nature of practical combustion systems, combined with the lack of reliable analytical models in the past, encouraged researchers to rely heavily on empirical methods to predict performance and to develop new products. Fortunately, the combustion field has gained considerable scientific knowledge in the last few decades, and such knowledge is now utilized in industry by engineers to evaluate and design combustion systems in a more rigorous manner. This progress is the result of efforts in academia, government laboratories, private labs, and companies like John Zink.

The advent of ever-faster and more powerful computers has had a profound impact on the manner in which engineers model combustion systems. Computational fluid dynamics (CFD) was born from these developments. Combined with validation by experimental techniques, CFD is an essential tool in combustion research, development, analysis, and equipment design.

Today's diagnostic tools and instrumentation—with capabilities unimaginable just a few years ago—allow engineers and scientists to gather detailed information in hostile combustion environments at both microscopic and macroscopic levels. Lasers, spectroscopy, advanced infrared, and ultraviolet camera systems are used to nonintrusively gather quantitative and qualitative information, including combustion temperature, velocity, species concentration, flow visualization, particle size, and loading. Advanced diagnostic systems and instrumentation are being transferred beyond the laboratory to implementation in practical field applications. The information obtained with these systems has considerably advanced our knowledge of combustion equipment and has been an indispensable source of CFD model validation.

Oil refining, chemical processes, and power generation are energy-intensive industries with combustion applications in burners, process heaters, boilers, and cogeneration systems, as well as flares and thermal oxidizers. Combustion for these industries presents unique challenges related to the variety of fuel compositions encountered. Combustion equipment must be flexible to be able to operate in a safe, reliable, efficient, and environmentally responsible manner under a wide array of fuel compositions and conditions.

Combustion is an exciting and intellectually challenging field containing plenty of opportunities to enhance fundamental and practical knowledge that will ultimately lead to the development of new products with improved performance.

This book represents the tireless efforts of many John Zink engineers willing to share their unique knowledge and experience with other combustion engineers, researchers, operators of combustion equipment, and college students. We have tried to include insightful and helpful information on combustion fundamentals, combustion noise, CFD design, experimental techniques, equipment, controls, maintenance, and troubleshooting. We hope our readers will agree that we have done so.

David H. Koch
Executive Vice President
Koch Industries

Preface to the First Edition

Combustion is described as "the rapid oxidation of a fuel resulting in the release of usable heat and production of a visible flame."[1] Combustion is used to generate 90% of the world's power.[2] Regarding the science of combustion, Liñán and Williams wrote the following:

> Although combustion has a long history and great economic and technical importance, its scientific investigation is of relatively recent origin. Combustion science can be defined as the science of exothermic chemical reactions in flows with heat and mass transfer. As such, it involves thermodynamics, chemical kinetics, fluid mechanics, and transport processes. Since the foundations of the second and last of these subjects were not laid until the middle of the nineteenth century, combustion did not emerge as a science until the beginning of the twentieth century.[3]

Chomiak wrote the following: "In spite of their fundamental importance and practical applications, combustion processes are far from being fully understood."[4] In Strahle's opinion, "combustion is a difficult subject, being truly interdisciplinary and requiring the merging of knowledge in several fields."[5] It involves the study of chemistry, kinetics, thermodynamics, electromagnetic radiation, aerodynamics, and fluid mechanics, including multiphase flow and turbulence, heat and mass transfer, and quantum mechanics to name a few. Regarding combustion research,

> The pioneering experiments in combustion research, some 600,000 years ago, were concerned with flame propagation rather than ignition. The initial ignition source was provided by Mother Nature in the form of the electrical discharge plasma of a thunderstorm or as volcanic lava, depending on location. … Thus, in the beginning, Nature provided an arc-augmented diffusion flame and the first of man's combustion experiments established that the heat of combustion was very much greater than the activation energy—i.e., that quite a small flame on a stick would spontaneously propagate itself into a very large fire, given a sufficient supply of fuel.[6]

In one of the classic books on combustion, Lewis and von Elbe wrote the following:

> Substantial progress has been made in establishing a common understanding of combustion phenomena. However, this process of consolidation of the scientific approach to the subject is not yet complete. Much remains to be done to advance the phenomenological understanding of flame processes so that theoretical correlations and predictions can be made on the basis of secure and realistic models.[7]

Despite the length of time it has been around, despite its importance to man, and despite vast amounts of research, combustion is still far from being completely understood. One of the purposes of this book is to improve that understanding, particularly in industrial combustion applications in the process and power generation industries.

This book is generally organized in two parts. Part I deals with the basic theory of some of the disciplines (combustion, heat transfer, fluid flow, etc.) important for the understanding of any combustion process and consists of Chapters 1 through 13. While these topics have been satisfactorily covered in many combustion textbooks, this book treats them from the context of the process and power generation industries. Part II deals with specific equipment design issues and applications in the process and power generation industries.

References

1. Industrial Heating Equipment Association, *Combustion Technology Manual*, 5th edn. Combustion Division of the Industrial Heating Equipment Association, Arlington, VA, 1994, p. 1.
2. N. Chigier, *Energy, Combustion, and Environment*. McGraw-Hill, New York, 1981, p. ix.
3. A. Liñán and F.A. Williams, *Fundamental Aspects of Combustion*. Oxford University Press, Oxford, U.K., 1993, p. 3.
4. Chomiak, *Combustion: A Study in Theory, Fact and Application*, p. 1.
5. W.C. Strahle, *An Introduction to Combustion*. Gordon & Breach, Langhorne, PA, 1993, p. ix.
6. F.J. Weinberg, The first half-million years of combustion research and today's burning problems, in *Fifteenth Symposium (International) on Combustion*, The Combustion Institute, Pittsburgh, PA, 1974, p. 1.
7. B. Lewis and G. von Elbe, *Combustion, Flames and Explosions of Gases*, 3rd edn. Academic Press, New York, 1987, p. xv.

Preface to the Second Edition

The first edition of the *John Zink Combustion Handbook* was published in 2001. It replaced the previous industry standard book (*Furnace Operations*, 3rd edition, Gulf Publishing, Houston, 1981) written by Dr. Robert Reed, who was the former technical director of the John Zink Company. The first edition of the *Zink Handbook* consisted of 800 oversized pages, was in full color, and was written by 30 authors as compared to *Furnace Operations*, which consisted of 230 pages, was in black and white, and was written by a single author. The first edition of the *Zink Handbook* was a major expansion compared to *Furnace Operations*. The second edition of the *Zink Handbook* is another major expansion compared to the first edition.

The second edition consists of three volumes, collectively over twice as large as the single-volume first edition. Volume I concerns the fundamentals of industrial combustion such as chemistry, fluid flow, and heat transfer. While the basic theory is presented for each topic, the unique treatment compared to standard textbooks is how these topics apply to industrial combustion. Volume II concerns design and operations and includes topics related to equipment used in industrial combustion such as installation, maintenance, and troubleshooting. It also includes an extensive appendix with data relevant to industrial combustion equipment and processes. Volume III concerns applications and covers topics such as process burners, boiler burners, process flares, thermal oxidizers, and vapor control. It shows how the information in volumes I and II is used to design and operate equipment in particular industry applications.

There were several reasons for writing a second edition. The first is the natural improvement in technology with time. For example, NOx emissions from process burners are lower than ever and continue to decrease with advancements in technology. A second reason for the new edition is to make improvements to the first edition as recommended by readers. One example is to have more property data useful for the design and operation of combustion equipment. Another reason for the new edition is to expand the coverage to include technologies not covered in the first edition such as metallurgy, refractories, blowers, and vapor control equipment.

While these three volumes represent a significant expansion of the first edition, some topics could have been covered in greater detail and some topics have received little if any attention. There is still much to learn on the subject of industrial combustion, which is far more complicated than the average person would ever imagine. This is what makes it such an exciting and dynamic area of technology that has a significant impact on society because it affects nearly every aspect of our lives.

Acknowledgments

The authors would like to collectively thank the John Zink and Coen companies for their help and support during the preparation of these materials. Many colleagues helped with ideas, content, and the preparation of figures and tables. The authors would also like to thank Rick Ketchum, Andrew Walter, Vincent Wong, and Jeffrey Ma for their help in preparing the materials for this book.

Chuck Baukal thanks his wife, Beth, and his daughters, Christine, Caitlyn, and Courtney, for their continued support. He also thanks the good Lord above, without whom this would not have been possible. Wes Bussman thanks his family, Brenda, Sean, and Zach, for their support. He also thanks all of his colleagues at the John Zink Company LLC, Tulsa, Oklahoma, for their encouragement and for the knowledge they have shared with him throughout his career. Jason McAdams thanks his wife, Heather, and his sons, Ian and Nathan, for their support and for their understanding of the extra time and effort that being in the combustion business sometimes takes. He also thanks all of those who have given their time to help him learn throughout his career. Bill Johnson thanks his father and mother, Harold and Mary Johnson, for giving him a solid foundation and great work ethics. His wife, Judy, deserves a special and loving thank you for allowing him to pursue a career in combustion, which has involved traveling worldwide and a lot of time away from home over the years. He also wants to thank his great mentors at John Zink, including Hershel Goodnight, Don Iverson, Charles Summers, Ronnie Williams, Jake Campbell, and Sam Napier. It has always amazed him how a bunch of "cowboys" from Oklahoma built the largest combustion company in the world. A special thanks to Dr. Richard Waibel and Mike Claxton for being his "day-to-day" mentors for many years. Last but not least, thanks to Earl Schnell and Charlene Isley for being supervisors who allowed him to grow and expand his knowledge and get this knowledge in the best place to benefit the company. Vladimir Lifshits acknowledges Coen Company, Inc. and his long-term boss Steve Londerville for a stimulating work environment and opportunity to expand combustion technology during his 23-year career with Coen and for the invitation to write chapters of this book on burners and controls. Steve Londerville acknowledges Chet Binasik for hiring him at Coen in 1977 and providing a long fulfilling career in R&D; Temple Voorhees for his never ending guidance in the drive for new products with a passion that was unparalleled in this industry; Sherman Eaton for mentoring him for many years and paving the way for his career and accomplishments; Jim Church for driving into his mind that engineers need to understand financials and their many discussions; his wife Pure who never complained about, "Sorry I have work this weekend"; and finally John Baxter, his lifelong friend since childhood, who always said, "Just get it done" and then we can have some fun. Roger Blanton gratefully thanks and acknowledges his loving wife Brenda for faithfully supporting him during many weeks while away from home on business travel; Harley Cooper for professional support and encouragement; the John Zink Company for many years of challenging opportunities; and his Lord and Savior Jesus Christ who makes all things possible.

Editor

Charles E. Baukal, Jr., is the director of the John Zink Institute for the John Zink Co. LLC, Tulsa, Oklahoma, where he has been since 1998. He has also been the director of R&D and the director of the R&D Test Center at Zink. He previously worked for 13 years at Air Products and Chemicals, Inc, Allentown, Pennsylvania in the areas of oxygen-enhanced combustion and rapid gas quenching in the ferrous and nonferrous metals, minerals, and waste incineration industries. He has also worked for Marsden, Inc. (a burner supplier in Pennsauken, New Jersey) for five years in the paper, printing, and textile industries and for Selas Corp. (a burner supplier in Dresher, Pennsylvania) in the metals industry, both of whom make industrial combustion equipment. He has over 30 years of experience in the fields of industrial combustion, pollution control, and heat transfer and has authored more than 100 publications in those areas. Dr. Baukal is an adjunct instructor for Oral Roberts University, the University of Oklahoma, the University of Tulsa, and the University of Utah. He is the author/editor of thirteen books in the field of industrial combustion, including *Oxygen-Enhanced Combustion* (1998), *Heat Transfer in Industrial Combustion* (2000), *Computational Fluid Dynamics in Industrial Combustion* (2001), *The John Zink Combustion Handbook* (2001), *Industrial Combustion Pollution and Control* (2004), *Handbook of Industrial Burners* (2004), *Heat Transfer from Flame Impingement Normal to a Plane Surface* (2009), *Industrial Combustion Testing* (2011), *Oxygen-Enhanced Combustion*, 2nd edition (2013), *Coen & Hamworthy Combusiton Handbook* (2013) and *The John Zink Hamworthy Combustion Handbook*, 3 Volumes (2013).

He has a PhD in mechanical engineering from the University of Pennsylvania, Philadelphia, Pennsylvania, an MBA from the University of Tulsa, and is a licensed professional engineer in the state of Pennsylvania, a board-certified environmental engineer (BCEE), and a qualified environmental professional (QEP). He has also served as an expert witness in the field of combustion, is an inventor on 11 U.S. patents, and is a member of numerous honorary societies and *Who's Who* compilations. Dr. Baukal is a member of the American Society of Mechanical Engineers, the American Society for Engineering Education (ASEE), and the Combustion Institute. He serves on several advisory boards, holds offices in the Combustion Institute and ASEE, and is a reviewer for combustion, heat transfer, environmental, and energy journals.

Contributors

Adam Bader is the manager of thermal oxidized aftermarket sales and service for John Zink Company LLC, Tulsa, Oklahoma. He has worked at John Zink since 2005 and has over eight years of experience in flare and thermal oxidizer systems. He has given numerous training courses and authored a publication on the subject of proper flare operation, design, and selection. He holds a BS in mechanical engineering technology from Oklahoma State University.

Peter F. Barry is the former director of Duct Burners for the John Zink Company LLC, Tulsa, Oklahoma. He has a BS in mechanical engineering from Lehigh University.

John Bellovich is the manager of the Combustion Rental Group at the John Zink Company LLC, Tulsa, Oklahoma. He received his BSME from the University of Tulsa and has more than over 20 years of experience in the industrial combustion industry. He has written or cowritten three published articles.

Roger E. Blanton, PE, is the business development manager in the Vapor Control Group for John Zink Company LLC, based in Tulsa, Oklahoma. In addition, Roger serves as the John Zink Vapor Control School Coordinator. He is a registered professional engineer working for over 30 years with rotating equipment and petrochemical processes. Roger holds a BS in mechanical engineering from the University of Tulsa and an MBA with marketing and management emphasis from Oklahoma State University. He has specialized in technical product design, application, and sales; new product development and market introduction; sales and distribution channel management; and international business development. Roger is a member of the American Society of Mechanical Engineers (ASME), member of Tau Beta Pi Engineering Honor Society Oklahoma Beta Chapter, ordained deacon of the First Baptist Church of Tulsa, past member of the University of Tulsa Mechanical Engineering Advisory Board, and past president of the Greater Ozarks International Trade Association. Roger has authored many published technical papers and articles.

David Brown is a cracking furnace expert working for Total Research and Technology Feluy (TRTF) in Belgium. Prior to joining TPRF in 2009, David worked for Shaw Energy & Chemicals (formerly Stone & Webster) for 28 years in a leading role in the development, sales, design, commissioning, and operation of cracking furnaces. David received his MA and MEng from the University of Cambridge and his DPhil from the University of Oxford. He is a chartered engineer and a fellow of the (British) Institution of Chemical Engineers. He served as session chair for Fundamentals & Technology (Ethylene Producers' Conference). He is also the joint holder of five patents and has presented papers on combustion in cracking furnaces on numerous occasions.

Wes Bussman, PhD, is a senior research and development engineer for John Zink Company LLC (Tulsa, Oklahoma). He received his PhD in mechanical engineering from the University of Tulsa. He has over 20 years of experience in basic scientific research work, industrial technology research and development, and combustion design engineering. He holds ten patents and has authored several published articles and conference papers. He has also been a contributing author to several combustion-related books. He has taught engineering courses at several universities and is a member of Kappa Mu Epsilon Mathematical Society and Sigma Xi Research Society.

I.-Ping Chung, PhD, is a senior development engineer in the Technology and Commercial Development Group at the John Zink Company LLC (Tulsa, Oklahoma). She has worked in the field of industrial combustion and equipment, fluid dynamics, atomization and sprays, spray combustion, and laser diagnosis in combustion and has a PhD in mechanical and aerospace engineering. She has 24 publications and holds 9 patents. She is a registered professional engineer of mechanical engineering in the states of California and Iowa.

Michael G. Claxton is a senior principal engineer in the Burner Process Group of the John Zink Company LLC, Tulsa, Oklahoma. He received his BS in mechanical engineering from the University of Tulsa and has worked for the John Zink Company LLC, Tulsa, Oklahoma, in the field of industrial burners and combustion equipment since 1974. He has coauthored a number of papers and presentations covering combustion, combustion equipment, and combustion-generated emissions and is coholder of several combustion-related patents.

Carl A. Connally, formerly senior design engineer for thermal oxidation systems, joined the John Zink Company LLC in 1977. He is a graduate of Texas Tech

University with a bachelor's degree in chemical engineering and an MBA from the University of Tulsa. Carl joined John Zink as an applications engineer, designing acid gas combustion equipment, low BTU gas burners, and pressurized air heaters for the Burner Division. In the last 20 years, he has been involved in the design of thermal oxidizers and acid gas equipment.

Harley Cooper is the director of the Flare Gas Recovery Group at John Zink Company LLC, Tulsa, Oklahoma. He has more than 20 years of vapor control and recovery experience in the petrochemical industry. Cooper holds a BS in mechanical engineering and an MBA, both from the University of Tulsa. He is a registered professional engineer in the state of Oklahoma.

Tami Fischer is a senior designer at John Zink Company. She has a BS from the University of Wisconsin–Madison and has worked at Zink for 25 years.

Juergen Foelting is the manager of the Controls and Automation Group in John Zink KEU GmbH, Krefel, Germany. He has a degree in automation engineering from the University of Applied Sciences Krefeld. He has been in that role for more than 10 years now and has over 16 years of experience with the development of combustion systems. In that time, he was significantly involved in the development of the "multipurpose philosophy" of the company and gained experience with the design of combustion plants as well as commissioning work.

Brandy S. Johnson, PE holds a bachelor of science degree in mechanical engineering from the University of Kansas. She has been with John Zink since 1998 and currently holds the position of Director, Biogas Systems. Brandy's areas of focus have included solid waste and wastewater treatment facilities in both private and public sectors worldwide. She is a registered professional engineer in the state of Oklahoma. Brandy is a previous author and presenter at the SWANA Landfill Gas Symposium in 2000 with "A Closer Look at Condensate Injection into Landfill Gas Flares" and in 2005 entitled "Specifying a Cost Effective Landfill Flare System." She is a coinstructor for the Combustion course at the University of Tulsa and the University of Utah.

William Johnson started in the industry in 1969 with Stanley Consultants in Muscatine, Iowa, where he helped design coal-fired steam-generating units up to 650 MW. He moved to Oklahoma in 1976 and joined Williams Brothers Engineering. At Williams Brothers, he was involved in oil and gas transmission and offshore unloading facilities. He then joined John Zink in November 1977 and worked as a project manager in the Burner Group.

Jay Karan (Jaiwant Jayakaran) is director of thermal oxidizer systems at the John Zink Company LLC, Tulsa, Oklahoma. He has over 25 years of experience in the fields of combustion, petrochemicals, and power, with responsibilities in R&D, plant operations, and engineering. Jay has an MS in mechanical engineering. He has authored several technical articles and papers over the years and has several patents.

Wolfgang Klaus is a senior process engineer at John Zink KEU GmbH, Krefeld, Germany. He has worked in the field of burner design and process technology for over 25 years. Of these, he spent 15 years in the research department and test center of KEU, focusing on new burner development, improvement and characterization of existing designs, and field testing of new developments, especially KEU DeNOx Technology. He has coauthored numerous technical articles and papers.

Vladimir Lifshits is a principal development engineer of the Design Engineering Group of Coen Company, Inc. He joined the company in 1989. For eight years up to 2008, he was a director of Coen Research and Development Department. Lifshits pioneered the development of low NOx burners with lean premix combustion. His many developments include the most advanced combustion products and systems for boilers, steam generators for enhanced oil recovery, and air heaters. He is the author of many technical papers and patents. He has a combined MS in mechanical engineering and physics from St. Petersburg Polytechnic Institute of Russia.

Tim W. Locke is the director of engineering and project management of the Vapor Control Group at John Zink Company LLC, Tulsa, Oklahoma. He has been with John Zink since 1989 and has fulfilled various roles for process burners, flare systems, and thermal oxidizer systems and most recently as product director for biogas systems. He has a BS in petroleum engineering from Oklahoma State University, has coauthored three patents for ultralow emission biogas flaring, has authored one article for *MSW Management*, and has authored and presented three technical papers for the Solid Waste Association of North America.

Steve Londerville is currently chief engineer at Coen Company, San Mateo City, California. Previous positions since 1978 at Coen were chief technical officer, vice president R&D, director R&D, and chief engineer. During the past 35 years, he has been involved with all aspects of product development at Coen Co. He received his mechanical engineering degree from San Jose State University in 1977. He holds 7 patents and has authored 16 publications. He is a member of ASME, AIChE, the Combustion Institute, Tau Beta Pi, and the Institute for

Liquid Atomization and Spray Systems (ILASS), including past officer in the board of directors of ILASS. He was recognized as the ASME 1990 Distinguished Mechanical Engineer of the year, Santa Clara Valley Section.

Rasik Patel is the vice president of process technology with OnQuest Inc., San Dimas, California. He has 36 years of experience in designing fired process heaters, coker heaters, steam methane reformers, and pyrolysis furnaces for the refining and petrochemical industries. He is an active member in the heat transfer subgroup of the API task force, working on fired heater standards. He received his MS in chemical engineering from the Imperial College of Science & Technology, University of London, England.

Jeff Peterson is a process engineering advisor for Flint Hills Resources at its Corpus Christi refinery at Texas. Peterson provides process engineering support for overpressure protection and flare systems as well as other projects. He holds a BS in chemical engineering from Michigan Technological University.

Erwin Platvoet is director of process burner engineering at John Zink Company LLC, Tulsa, Oklahoma. He has over 20 years of experience in CFD and heat transfer engineering for various companies in Europe and the United States. He received his MS in chemical engineering from Twente University of Technology in the Netherlands and currently holds six patents.

Richard Price is the sales director of John Zink Hamworthy Combustion for the marine and offshore industry with global responsibility and has over 18 years experience in marine-related combustion applications. He received his master's degree in mechanical engineering from Southampton University in the United Kingdom. He is a chartered engineer and member of the Institute of Marine Engineering, Science & Technology.

Bernd Reese is leader of process technology at John Zink KEU GmbH, Krefeld, Germany. He has worked in the field of burner design and process technology for over 25 years. Of these, he spent 15 years in the research department and test center of KEU, focusing on new burner development and the improvement of existing designs. He and his team improved the KEU DeNOx Technology. He has coauthored numerous technical articles and papers.

Jason P. Rolf, PE, is the manager of project engineering for the Biogas Division at John Zink Company LLC, Tulsa, Oklahoma, with over 15 years of experience. He earned a bachelor of science degree in mechanical engineering from Iowa State University and a master of business administration degree from Oklahoma State University. Jason is also licensed as a professional engineer in the State of Oklahoma.

Lothar Schulz is a senior sales and application engineer in the air heater and heat recovery business unit of the John Zink KEU GmbH, Krefeld, Germany. He has a degree in process engineering from the University of Applied Sciences (UAS), Krefeld, and has worked for John Zink KEU GmbH in the field of process heating, heat recovery, and thermal oxidizers for food applications since 1989.

Robert E. Schwartz (retired) PE, was a senior technical specialist at John Zink Company LLC, Tulsa, Oklahoma. He worked in the fields of combustion, flares, pressure relieving systems, fluid flow, and heat transfer for nearly 50 years, which include 42 years with John Zink, where he provided technical and business leadership in all product areas and has extensive international experience. He has 51 U.S. patents for inventions of apparatus and methods that are in use throughout John Zink. He is also the associate editor of *The John Zink Hamworthy Combustion Handbook*. His areas of technical expertise encompass development, design, fabrication, and operation of combustion equipment, including flares, incinerators, process burners, boiler burners, and vapor control; reduction of NOx and other emissions from combustion processes; fluid flow and heat transfer in process and combustion equipment; noise elimination and control; vapor emissions control using recovery processes; hazardous waste site remediation; and permitting and operation of hazardous waste storage and disposal sites. His professional organizations and awards include membership in the American Society of Mechanical Engineers, the American Institute of Chemical Engineers and Sigma Xi, the Scientific Research Society, Registered Professional Engineer in the state of Oklahoma, recipient of the University of Missouri Honor Award for Distinguished Service in Engineering, and election to The University of Tulsa Engineering Hall of Fame. He received his BS and MS in mechanical engineering from the University of Missouri.

Stephen L. Somers was a senior process engineer at the John Zink Company LLC, Tulsa, Oklahoma. He has over 30 years of experience in combustion and process design, with 15 of those years in the application and design of duct burners for heat recovery steam generator (HRSG) supplementary firing. He has an MS in chemical engineering from the University of Oklahoma and a BS in chemical engineering from the University of Tulsa.

Nick Tuttle was a consultant in the Flare Gas Recovery Group at John Zink Company LLC, Tulsa, Oklahoma. He has more than 40 years of industry experience and is a registered professional engineer. Tuttle holds a BS in chemical engineering from New Mexico State University and has completed additional work toward an industrial management degree from the University of Houston. He has authored numerous papers and is an inventor on four U.S. patents.

Timothy Webster is the president of Coen Company, Inc, Tulsa, Oklahoma. He has worked in the field of industrial combustion for 18 years and received his MS in engineering from the University of Wisconsin. He has over 40 publications and is a licensed professional mechanical engineer in California.

Jeff White is a senior flare design engineer at the John Zink Company LLC, Tulsa, Oklahoma. He has worked in the field of flare system design at John Zink Company for 30 years. He received his MS in mechanical engineering from The University of Texas at Austin. He has published two articles, one on flare radiation methods and the other on flow measurement by ASME nozzles.

Klaus-Dieter Zschorsch is a senior process and application engineer in the Catalytic and Thermal Oxidizer Engineering Group of the John Zink KEU GmbH, Krefeld, Hamburg, Germany. He has a degree in process engineering from the University of Applied Sciences (UAS), Hamburg, and has worked for John Zink KEU GmbH in the field of catalytic oxidizers, condensation units, and selective catalytic reduction since 1990 and in the field of thermal oxidizers since 2010. He has been appointed as a guideline committee member of Verein Deutscher Ingenieure e.V. (VDI). As a committee member, he serves in an honorary capacity for the VDI Guideline “Method of Catalytic Waste Gas Cleaning.” He supports the committee by formulating technical rules and coauthoring papers and presentations covering fundamentals of catalytic oxidizers, oxidative processes, and selective catalytic reduction.

Prologue

Fred Koch and John Zink: Pioneers in the Petroleum Industry

The early decades of the twentieth century saw the birth and growth of the petroleum industry in Oklahoma. Drilling derricks sprouted like wildflowers throughout the state, making it among the top oil producers in the nation and Tulsa the "Oil Capital of the World" by the 1920s.

Refining operations accompanied oil production. Many of the early refineries were so small that today they would be called pilot plants. They were often merely topping processes, skimming off natural gasoline and other light fuel products and sending the remainder to larger refineries with more complex processing facilities.

Along with oil, enough natural gas was found to make its gathering and sale a viable business as well. Refineries frequently purchased this natural gas to fuel their boilers and process heaters. At the same time, these refineries vented propane, butane, and other light gaseous hydrocarbons into the atmosphere because their burners could not burn them safely and efficiently. Early burner designs made even natural gas difficult to burn as traditional practice and safety concerns led to the use of large amounts of excess air and flames that nearly filled the fire box. Such poor burning qualities hurt plant profitability.

Among firms engaged in natural gas gathering and sales in the northeastern part of the state was Oklahoma Natural Gas Company (ONG). It was there that John Steele Zink (Figure P.1b), after completing his studies at the University of Oklahoma in 1917, went to work as a chemist. Zink's chemistry and engineering education enabled him to advance to the position of manager of industrial sales. But while the wasteful use of natural gas due to inefficient burners increased those sales, it troubled Zink and awakened his talents first as an innovator and inventor and then as an entrepreneur.

Seeing the problems with existing burners, Zink responded by creating one that needed less excess air and produced a compact, well-defined flame shape. A superior burner for that era, it was technically a premix burner with partial primary air and partial draft-induced secondary air. The use of two airflows led to its trade name, BI-MIX®. The BI-MIX® burner is shown in a drawing from one of Zink's earliest patents (Figure P.2).

ONG showed no interest in selling its improved burners to its customers, so in 1929 Zink resigned and founded Mid-Continent Gas Appliance Co., which he later renamed the John Zink Company.

(a)

(b)

FIGURE P.1
(a) Fred Koch and (b) John Zink.

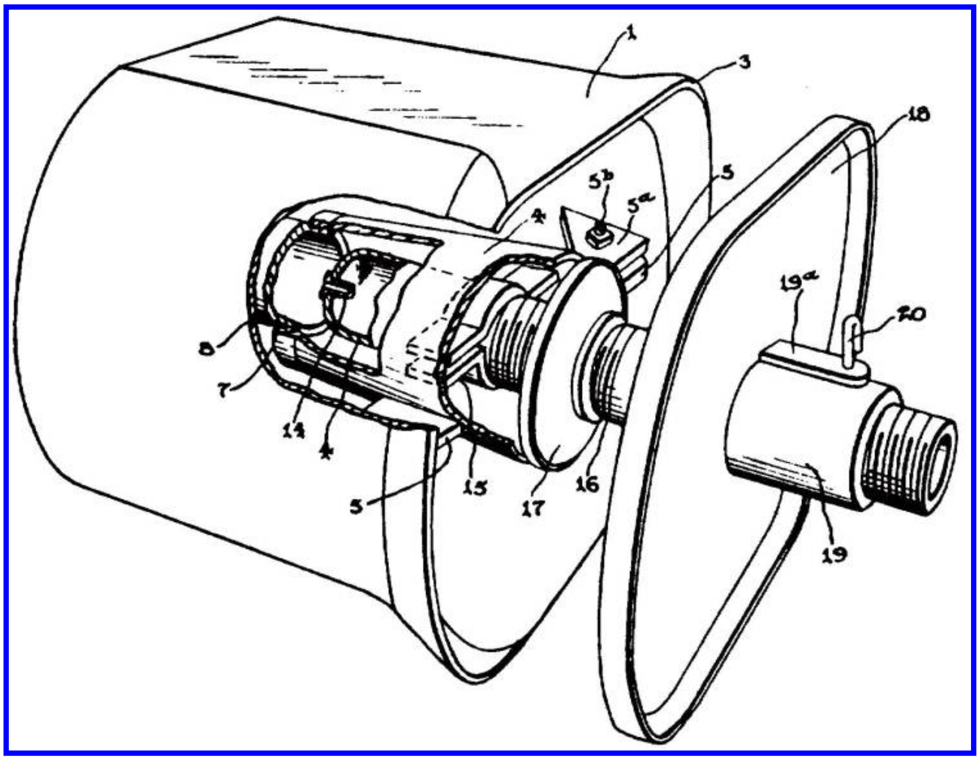


FIGURE P.2
Drawing of BI-MIX® From Zink's patent.

Zink's BI-MIX® burner was the first of many advances in technology made by his company, which to date has seen over 250 U.S. patents awarded to nearly 80 of its employees. He carried out early manufacturing of the burner in the garage of his Tulsa-area home and sold it from the back of his automobile as he traveled the Oklahoma oil fields, generating the money he needed to buy the components required to fabricate the new burners.

The novel burners attracted customers by reducing their fuel costs, producing a more compact flame for more efficient heater operation, burning a wide range of gases, and generally being safer to use. Word of mouth among operators helped spread their use throughout not only Oklahoma but, by the late 1930s, to foreign refineries as well.

Growth of the company required Zink to relocate his family and business to larger facilities on the outskirts of Tulsa. In 1935, he moved into a set of farm buildings on Peoria Avenue, a few miles to the south of the city downtown, a location Zink thought would allow for plenty of future expansion.

As time passed, Zink's company became engaged in making numerous other products, sparked by its founder's beliefs in customer service and solving customer problems. After World War II, Zink was the largest sole proprietorship west of the Mississippi River. Zink's reputation for innovation attracted customers who wanted new burners and, eventually, whole new families of products. For example, customers began asking for reliable pilots and pilot igniters when atmospheric venting of waste gases and emergency discharges was replaced by combustion in flares in the late 1940s. This in turn was followed by requests for flare burners and finally complete flare systems, marking the start of the flare equipment industry. Similar customer requests for help in dealing with gas and liquid waste streams and hydrocarbon vapor led the Zink Company to become a leading supplier of gas and liquid waste incinerators and also of hydrocarbon vapor recovery and other vapor control products.

Zink's great interest in product development and innovation led to the construction of the company's first furnace for testing burners. This furnace was specially designed to simulate the heat absorption that takes place in a process heater. Zink had the furnace built in the middle of the employee parking lot, a seemingly odd placement. He had good reason for this because he wanted his engineers to pass the test furnace every day as they came and went from work as a reminder of the importance of product development to the company's success.

Zink went beyond encouraging innovation and motivating his own employees. During the late 1940s, Zink and his technical team leader, Robert Reed (who together with Zink developed the first smokeless flare), sensed a need for an industry-wide meeting to discuss technologies and experiences associated with process heating. In 1950, they hosted the first of four annual process heating seminars in Tulsa. Interest in the seminars was high, with the attendance level reaching 300. Attendees of the first process heating seminar asked Zink and Reed to conduct training sessions for their operators and engineers. These training sessions, which combined lectures and practical hands-on burner operation in Zink's small research and development center, were the start of the John Zink Burner School®. The year 2010 marked the 60th anniversary of the original seminar and the 50th year in which the Burner School has been offered. Over the years, other schools were added to provide customer training in the technology and operation of hydrocarbon vapor recovery systems, vapor combustors, and flares.

Included among the 150 industry leaders attending the first seminar was Harry Litwin, former president and part owner of Koch Engineering Co., now part of Koch Industries of Wichita, Kansas. Litwin was a panelist at the closing session. Koch Engineering was established in 1943 to provide engineering services to the oil refining industry. In the early 1950s, it developed an improved design for distillation trays, and because of their commercial success the company chose to exit the engineering business. Litwin left Koch at that time and set up his own firm, the Litwin Engineering Co., which grew into a sizeable business.

During the same period that John Zink founded his business, another talented young engineer and industry innovator, Fred C. Koch, was establishing his reputation as an expert in oil processing. The predecessor to Koch Engineering Co. was the Winkler–Koch Engineering Co., jointly owned by Fred Koch with Lewis Winkler, which designed processing units for oil refineries. Fred Koch had developed a unique and very successful thermal cracking process that was sold to many independent refineries throughout the United States, Europe, and the former Soviet Union. One of the first of these processing units was installed in a refinery in Duncan, Oklahoma, in 1928, one year before Zink started his own company.

While the two men were not personally acquainted, Koch's and Zink's companies knew each other well in those early years. Winkler–Koch Engineering was an early customer for Zink burners. The burners were also used in the Wood River refinery in Hartford, Illinois. Winkler–Koch constructed this refinery in 1940 with Fred Koch as a significant part owner and the head of refining operations. Winkler–Koch Engineering, and later Koch Engineering, continued to buy Zink burners for many years.

Fred Koch and two of his sons, Charles and David, were even more successful in growing their family business than were Zink and his family. When the Zink family sold the John Zink Company to Sunbeam Corporation* in 1972, the company's annual revenues were $15 million. By that time, Koch Industries, Inc., the parent of Koch Engineering, had revenues of almost $1 billion. Since then, Koch has continued to grow; its revenues in the year 2011 were over $100 billion.

When the John Zink Company was offered for sale in 1989, its long association with Koch made Koch Industries a very interested bidder. Acting through its Chemical Technology Group, Koch Industries quickly formed an acquisition team, headed by David Koch, which succeeded in purchasing the John Zink Company.

Koch's management philosophy and focus on innovation and customer service sparked a new era of revitalization and expansion for the John Zink Company. Koch recognized that the Peoria Avenue research, manufacturing, and office facilities were outdated. The growth of Tulsa after World War II had made Zink's facilities an industrial island in the middle of a residential area. The seven test furnaces on Peoria Avenue at the time of the acquisition, in particular, were cramped, with such inadequate infrastructure and obsolete instrumentation they could not handle the sophisticated research and development required for modern burners.

A fast-track design and construction effort by Koch resulted in a new office and manufacturing complex in the northeastern sector of Tulsa that was completed at the end of 1991. In addition, a spacious R&D facility adjacent to the new office and manufacturing building replaced the Peoria test facility.

The initial multimillion dollar investment in R&D facilities included an office building housing the R&D staff and support personnel, a burner prototype fabrication shop, and an indoor laboratory building. Additional features included steam boilers, fuel storage and handling, data gathering centers, and measurement instrumentation and data logging for performance parameters from fuel flow to flue gas analysis.

Koch has repeatedly expanded the R&D facility. When the new facility began testing activities in 1992, nine furnaces and a multipurpose flare testing area were in service. Today, there are 14 outdoor test furnaces and 2 indoor research furnaces. Control systems are frequently updated to keep them state of the art.

Zink is now able to monitor burner tests from an elevated customer center that has a broad view of the entire test facility. The customer center includes complete automation of burner testing with live data on control panels and flame shape viewing on color video monitors.

A new flare testing facility (Figure P.3) was constructed in the early 2000s to dramatically expand and improve Zink's capabilities. This project represents the company's largest single R&D investment since the original construction of the R&D facility in 1991. The new facilities accommodate the firing of a wide variety of fuel blends (propane, propylene, butane, ethylene, natural gas, hydrogen, and diluents such as

---

* Sunbeam Corporation was primarily known as an appliance maker. Less well known was Sunbeam's group of industrial specialty companies such as John Zink Company.

FIGURE P.3
Flare testing facility.

nitrogen and carbon dioxide) to reproduce or closely simulate a customer's fuel composition. Multiple cameras provide video images along with the electronic monitoring and recording of a wide range of flare test data, including noise emissions. The facility can test all varieties of flare systems with very large sustained gas flow rates at or near those levels that customers will encounter in the field. Indeed, flow capacity matches or exceeds the smokeless rate of gas flow for virtually all customers' industrial plants, giving the new flare facility a capability unmatched in the world.

These world-class test facilities are staffed with engineers and technicians who combine theoretical training with practical experience. They use the latest design and analytical tools, such as computational fluid dynamics, physical modeling, and a phase Doppler particle analyzer. The team can act quickly to deliver innovative products that work successfully, based on designs that can be exactly verified before the equipment is installed in the field.

Koch's investment in facilities and highly trained technical staff carries on the tradition John Zink began more than 80 years ago: providing our customers today, as he did in his time, with solutions to their combustion needs through better products, applications, information, and service.

Robert E. Schwartz
Tulsa, Oklahoma

1

Process Burners

Erwin Platvoet, I.-Ping Chung, Michael G. Claxton, and Tami Fischer

CONTENTS

1.1 Introduction

The term "process burners" is used for those burners that operate in heaters and furnaces in the refining, petrochemical, and chemical industries. Each of these industries has specific requirements, as outlined in Volume 1, Chapter 2. However, they all have the following typical characteristics in common:

Heat release	1–15 MM Btu/h (0.29–4.4 MW)
Firebox pressure at the burner	−0.25 to −0.75 in H_2O (−0.62 to 1.87 mbar)
Burner pressure loss	0.25–8 in H_2O (0.62–19.9 mbar)
Excess air ratio	10%–25%

Process burners are all built in a similar manner (see Figure 1.1). In a natural-draft burner, the air enters through a noise suppression section whose primary function is to dampen the noise from the burner. The muffler is connected to a section that houses the air control, which serves to adjust the amount of air through the burner. This air control is typically a set of louvers or blades that can be rotated to partially or completely close the entrance into the plenum. The plenum, or wind box, is the section immediately before the burner tile. It serves to distribute air to the burner throat and dampen the noise from the firebox. The burner tile is a refractory piece that shapes and stabilizes the flame. One or more burner tips are used to inject the fuel into the air stream. They are connected to the fuel risers, which, in turn, are connected to the fuel manifold. A small pilot burner is often used to provide an ignition source for the main burner.

Although most process burners have quite common operating characteristics, they can be classified in a number of different ways (see Table 1.1). In the following sections, examples from each category are highlighted and their process applications are shown.

1.2 Classification Based on Emissions

An important way to distinguish burners from each other is by their emissions of nitrogen oxides (NOx) (see Volume 1, Chapter 15). This is likely the most important distinguishing characteristic, because the manner in which burner designs reduce emissions has a great impact on their flame shape and behavior. The different categories that currently exist based on NOx are

- Conventional
- Low NOx
- Ultralow NOx
- Next-generation ultralow NOx

Unfortunately, there are no industry standards for the definition of each of these, so they are generally

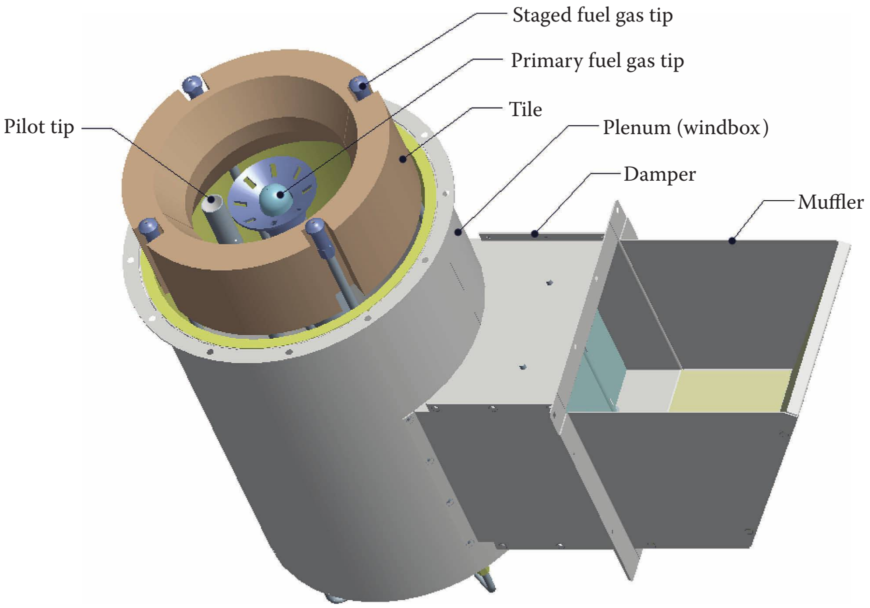


FIGURE 1.1
General burner construction.

TABLE 1.1
Burner Classifications

By motive force	Forced draft	Natural draft	Self-inspirated	
By NOx emissions	Conventional	Low NOx	Ultralow NOx	
By flame shape	Round	Flat		
By placement in the firebox	Floor, free standing	Floor, against wall	In the side wall	In the roof
By fuel type	Gas	Oil	Combination	
By NOx reduction type	Fuel staging	Air staging		
By fuel/air mixing	Diffusion type	Premix type		

determined by the burner manufacturer. Examples from each category are highlighted in the next sections.

1.2.1 Conventional Burners

Conventional burners have been designed with no other criteria than flame shape and stability in mind. Until the mid-1980s when emissions became very important, this was the dominant style of burner. Emissions of NOx, carbon monoxide (CO), unburned hydrocarbons, and particulate matter did not play a role in this style of burner. This typically means that fuel and air are mixed as rapidly as possible to create a compact flame. This is also why they are called conventional burners. Process heater burners designated raw gas burners, partial premix burners, premix burners, and combination gas/oil burners predating 1970 are almost always conventional burners. Using these burners has allowed furnace designers to design very compact fireboxes with high heat densities.

1.2.1.1 Natural-Draft and Forced-Draft Burners

An example of a conventional natural-draft gas-only burner is the John Zink PVYD burner (see Figure 1.2). This burner is shown configured with an integral burner plenum, inlet air control damper, and inlet noise suppression muffler.

A flat-flame conventional burner is shown in Figure 1.3. This is the John Zink forced-draft PFFG burner. Again, this burner is shown using the integral plenum design. This burner is shown without an inlet air muffler, allowing the burner to be connected to forced-draft air supply ducting.

A round flame conventional gas and/or liquid burner is shown in Figure 1.4. This is the John Zink PMA burner. This burner is the conventional “dual block” design, utilizing both the common main burner tile and a primary tile used for stabilization of the liquid firing. It is shown using the integral plenum design. This burner is shown without an inlet air muffler, allowing the burner to be connected to forced-draft air supply.

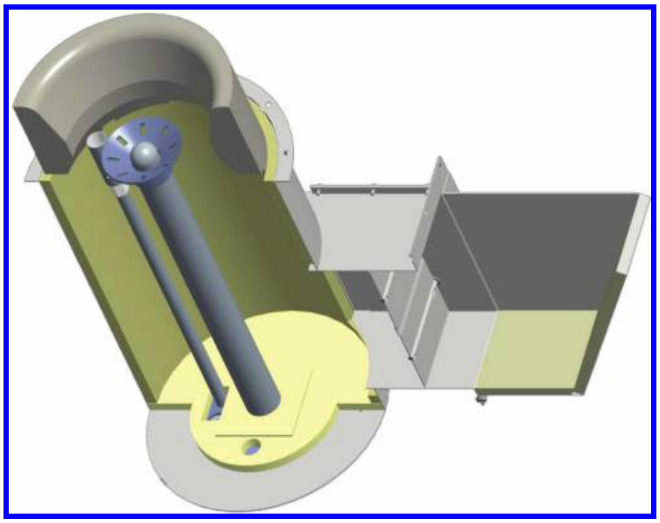


FIGURE 1.2
PVYD-M natural-draft, gas-only burner (with air inlet noise suppression).

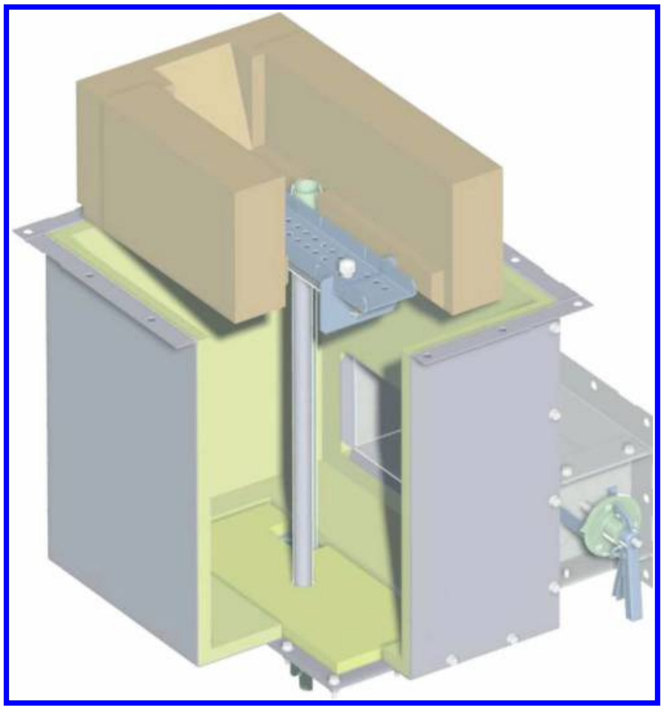


FIGURE 1.3
PFFG gas-only flat-flame burner.

1.2.1.2 *Premix Gas Burners*

An example of a conventional premix floor burner is the John Zink HEVD burner (see Figure 1.5). The fuel is used to entrain the air into a venturi. The mixture exits through a multitude of fuel ports on a spider-type gas head. The same principle can be used in a radiant wall burner (see Section 1.4).

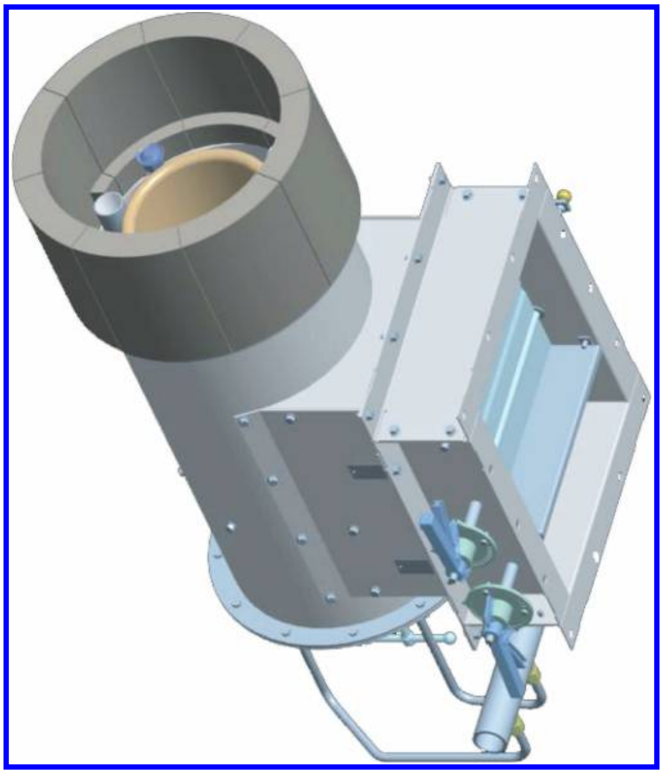


FIGURE 1.4
PMA round flame combination gas and liquid burner.

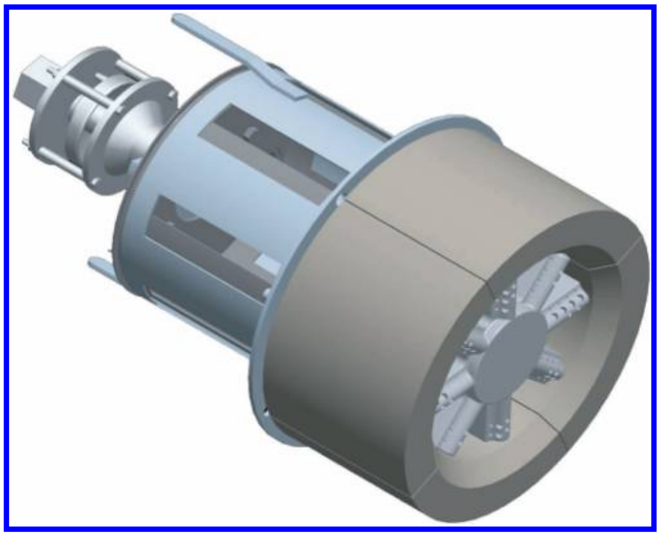


FIGURE 1.5
HEVD premix burner assembly.

1.2.2 Low-NOx Burners

Low NOx emissions are obtained by delaying the combustion. The fuel air mixture is deliberately made too rich or too lean for quick combustion. This delay lowers the peak flame temperatures and reduces the thermal NOx production. In low-NOx burners, the delay is achieved by staging the air or staging the fuel in two zones.

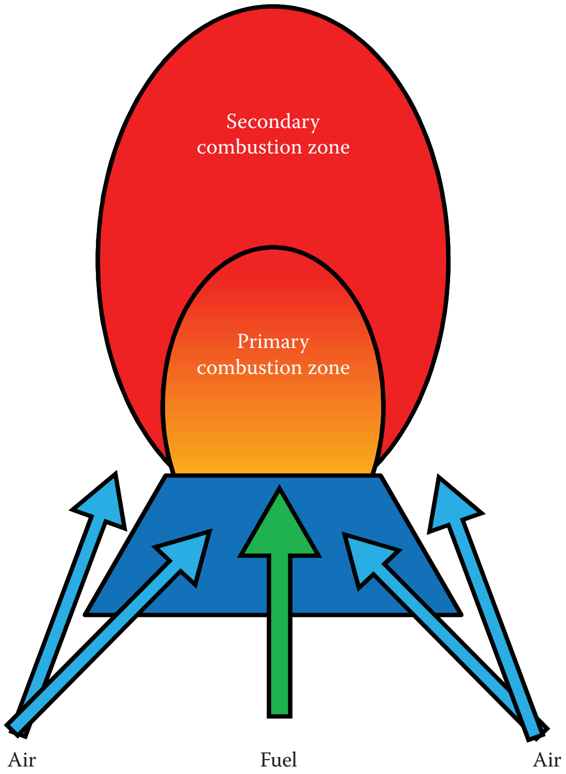


FIGURE 1.6
Staged-air combustion.

1.2.2.1 Staged-Air Burners

In staged-air burners, the primary air (approximately 40%–45% of the total air) is mixed with the total quantity of fuel producing a fuel-rich flame ("primary zone") that is both relatively cool and deficient in oxygen—conditions that inhibit NOx formation. It is the primary air that is used to control flame stability and NOx reduction.

Secondary air is introduced downstream of the primary flame zone at which point combustion is completed in an environment that is sufficiently cool to limit thermal NOx production. It is the secondary air that is used to complete combustion and provide the excess air. A schematic is shown in Figure 1.6.

1.2.2.2 HAWAstar Burner

An example of a staged-air, gas-fired burner is the John Zink HAWAstar burner (Figure 1.7). The ratio of primary to secondary air in a HAWAstar burner is fixed by the burner design. The air control handle can only be used to control the total amount of air to the burner.

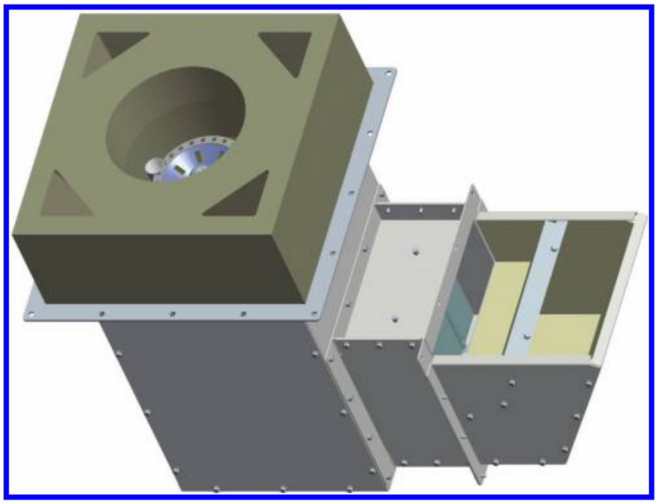


FIGURE 1.7
HAWAstar staged-air, gas-fired burner assembly.

1.2.2.3 PLNC Combination Gas and Liquid Staged-Air Burners

The John Zink series LNC burner (Figure 1.8) is a staged-air, combination gas and/or liquid burner. The burner is specifically designed for firing either refinery fuel gas, or heavy (high viscosity) residual oils, or a combination of the two simultaneously. The burner is designed as a "dual block" burner. This means that there are two burner tiles used to stabilize the primary combustion

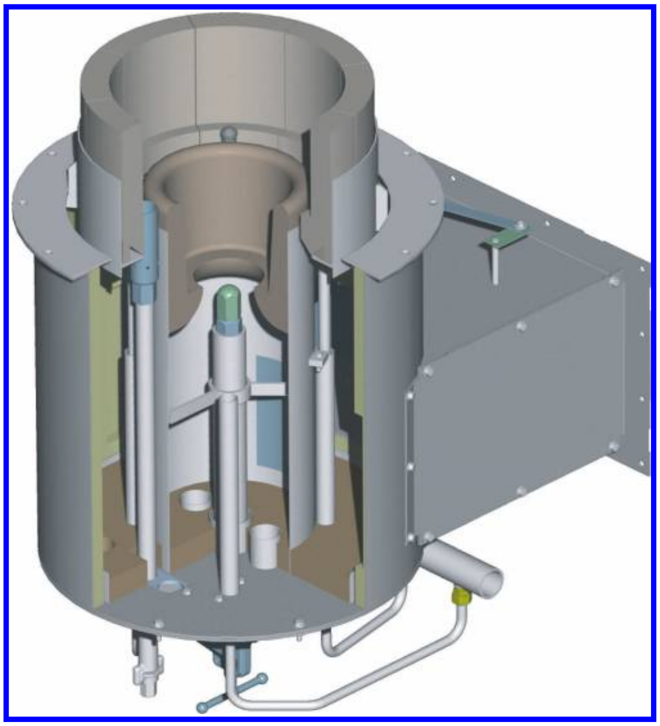


FIGURE 1.8
LNC staged air combination gas/liquid burner assembly.

of the fuels, one each for the oil (primary oil tile) and the gas (secondary tile). This design allows for independent stabilization zones and enhanced stabilization for heavy, high-viscosity fuel oils. The oil liquid-firing capability is handled by a centrally located atomizing gun, located in a refractory-lined primary oil tile. This oil tile provides a small amount of ignition air and recirculation of the initial liquid flame for stabilization. The gas-firing capability is handled through a set of raw gas tips located around the perimeter of the secondary tile throat. The secondary throat is formed by the outside of the primary oil tile and the inside of the secondary tile.

Central to the design of the LNC burner is the separation of the combustion into two zones. The first combustion zone includes all of the fuel and the air supplied through the primary oil tile and the secondary tile throat. This air supply is designed to be substoichiometric for the fuel supplied. In the case of the LNC, which is designed to handle high-carbon-content residual oils, this first combustion zone is designed to be 70% stoichiometric. The second combustion zone is developed downstream from the first by adding air. This air is provided by an annular space around the outside of the secondary tile. This secondary combustion zone air is designed to provide 45%–60% of stoichiometric air. The result is a burner that is "sized" for 15%–30% excess air while maintaining staged combustion.

There are two alternate designs for air control:

1. The first design provides separate air control for the primary oil tile. The secondary tile throat and staged-air annulus air control are in common. This design allows trimming of the primary oil air to aid in stability, especially at low oil-firing rates. In this design, the ratio of the first combustion zone to the second combustion zone is effectively fixed by the burner design.
2. The second design provides adjustable air capabilities to all three air throats. The primary oil throat will have an adjustable door. The secondary tile throat will have an adjustable air door. The staged-air annulus will have an adjustable air door. In this design, it is possible to "move" combustion air from any of the air throats to any other throat. This provides another level of control but also adds complexity. Moving air from the staged-air annulus to the primary oil and/or secondary throat increases the stoichiometry to the first combustion zone. This improves flame quality, but has a negative impact on the NOx emissions (increased emissions). Moving air from the primary oil and/or the secondary tile throat to the staged-air annulus decreases the stoichiometry of the first combustion zone. This improves NOx emissions (decreased emissions), but has a negative impact on the flame quality and, if taken too far, could induce instability.

1.2.2.4 Hamworthy Enviromix 2000 Burner

The Hamworthy Enviromix 2000 burner is an example of a staged-air burner that has separate dampers for primary and secondary air control (see Figure 1.9). It has been designed for the process and petrochemical industries where staged air and low emissions are required. The burner can fire a wide range of fuel gases (hydrogen rich to butane) and fuel oils such as naphtha, diesel, or heavier-type fuels, singularly or in combination.

The air staging is controlled within the purposely designed wind box and muffler block assemblies, the wind box being fully adjustable to allow for any alterations required due to future changeable site conditions. The wind box design was established utilizing flow modeling in order to achieve an even air distribution throughout the burner ensuring optimum performance combined with minimal NOx formation.

The optimum burner setup is therefore to adjust the primary air to "make the flame unstable" and then

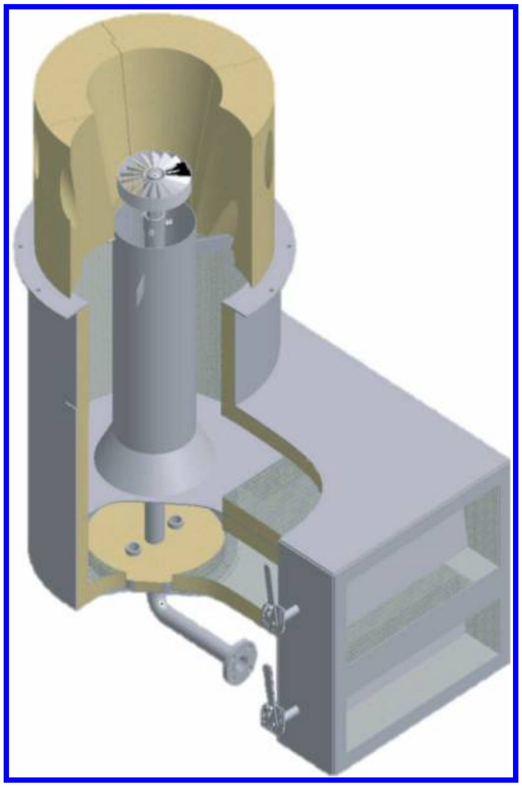


FIGURE 1.9
Hamworthy Enviromix 2000 staged-air burner assembly.

open slightly to make the flame stable. At this setting, the optimum trade between acceptable flame stability and lowest NOx performance is determined. The primary damper will need adjusting differently for differing fuels: hydrogen is a stable gas; therefore less air is required, and natural gas is more of an unstable gas; therefore more air is required. The secondary air is then adjusted to obtain the desired excess air level.

The flexibility of the Enviromix burner design allows the burner to be suited to many different applications including flat-flame burners designed to fire upward against the furnace wall or in between tube rows.

Another feature of the Enviromix is the low noise emissions that are generally less than 82–85 dB(A) SPL at 1 m (3 ft) from a single burner, dependent on operating parameters and can be further reduced with additional silencing measures.

1.2.2.5 *Hamworthy EEP Burner*

The staged-air burner concept can be extended to different tile shapes in order to meet flat-flame requirements. The Hamworthy EEP burner (Figure 1.10) is a wall-fired flat-flame version of the Enviromix 2000, developed specifically for coker heaters and steam-cracking furnaces.

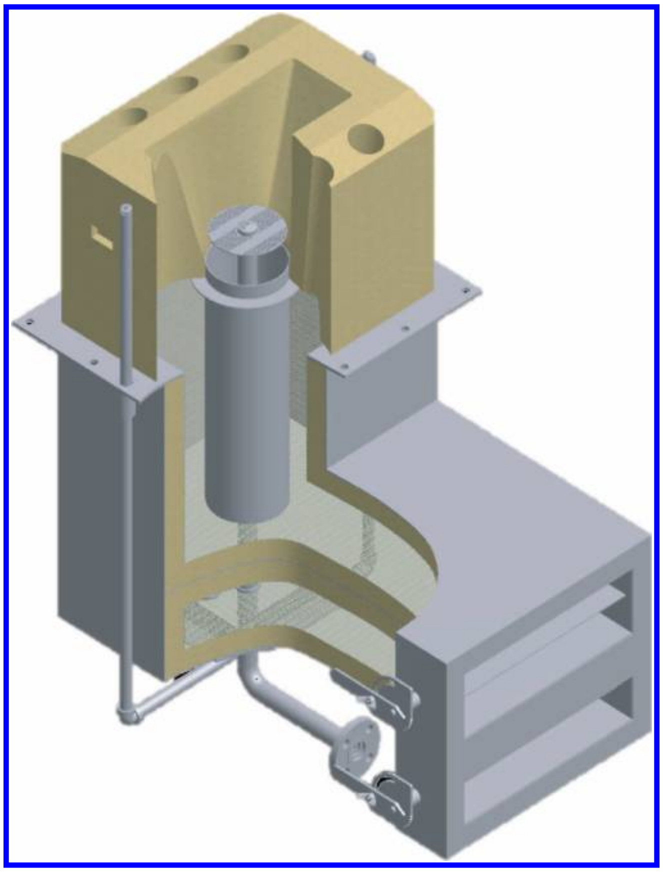


FIGURE 1.10
Hamworthy EEP flat-flame staged-air burner assembly.

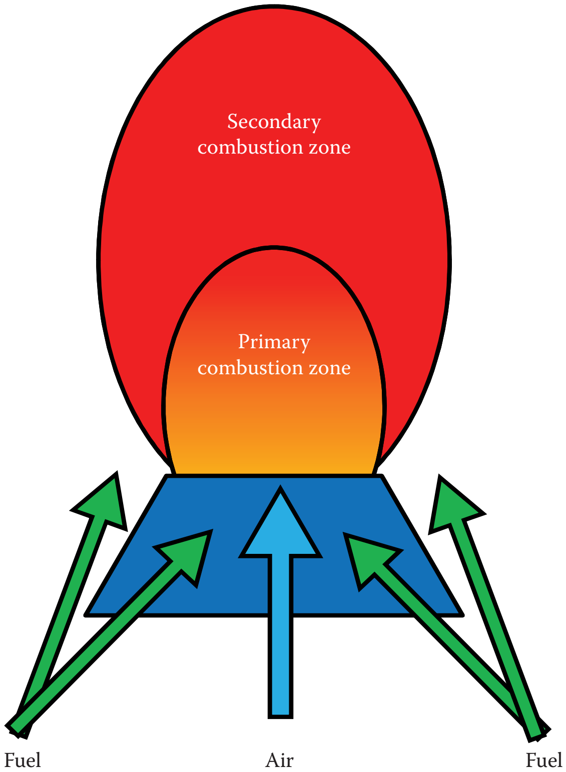


FIGURE 1.11
Staged-fuel combustion.

1.2.2.6 *Staged-Fuel Burner*

In staged-fuel burners, the fuel is separated into two zones, the primary fuel and the secondary fuel, while the air is introduced into the flame all at once, as shown in Figure 1.11.

1.2.2.7 *PSFG Burner*

In the John Zink PSFG burner (see Figure 1.12), the primary fuel is injected through a primary fuel gas tip, and staged or secondary fuel is injected by the four staged-fuel gas tips that are located around the burner tile perimeter.

1.2.2.8 *PSFFG Burner*

The flat-flame version of the PSFG is called the PSFFG (Figure 1.13). It has been used in coking, ethylene, reforming, and steam superheater applications.

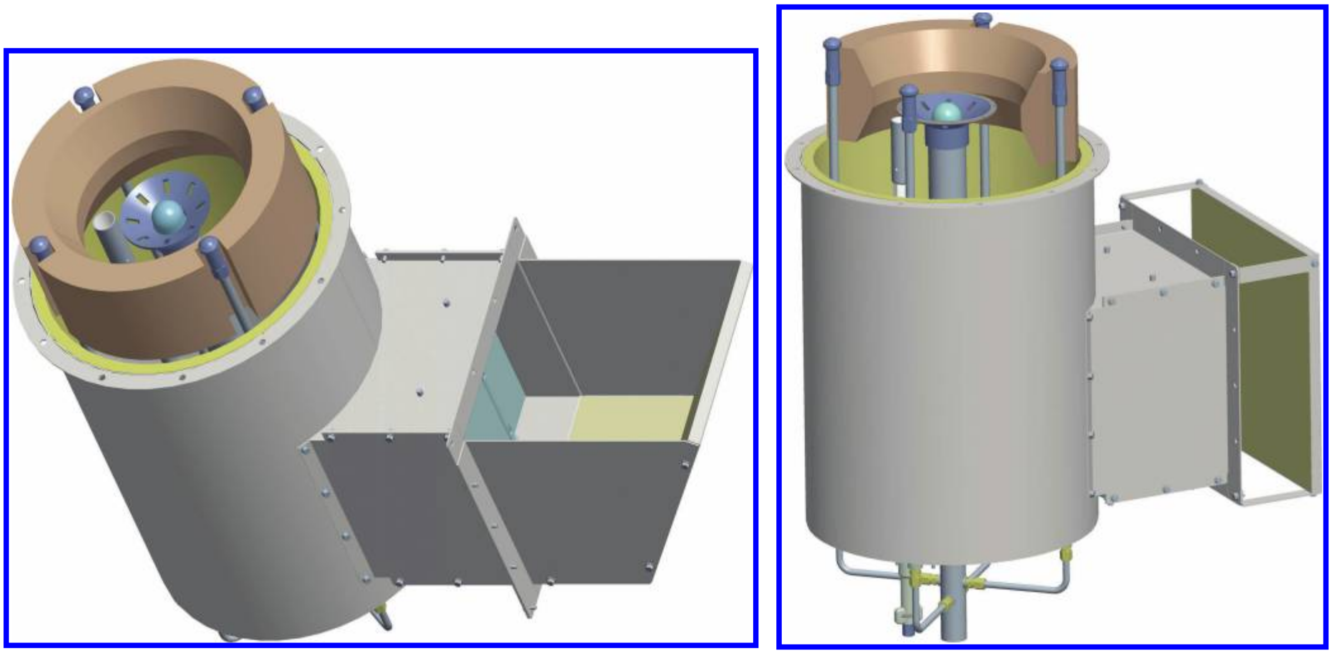


FIGURE 1.12
PSFG burner assembly.

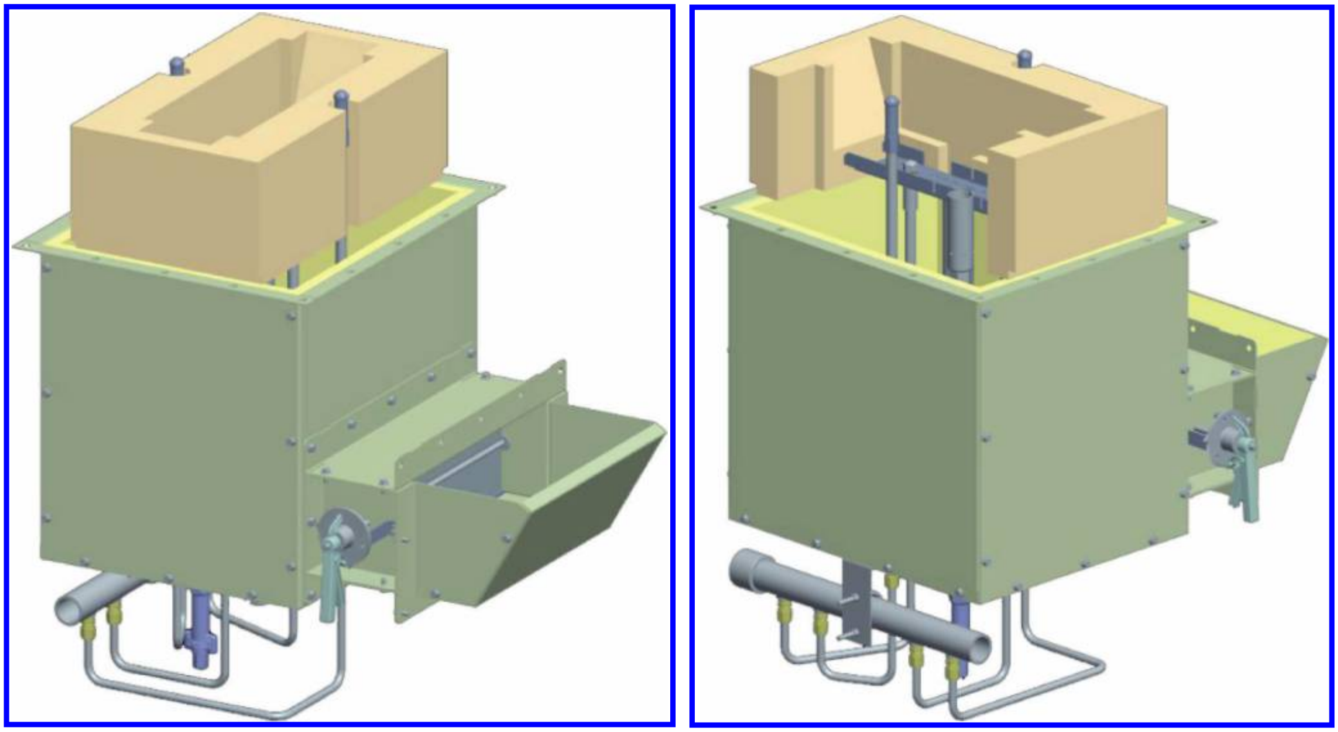


FIGURE 1.13
PSFFG flat-flame staged-fuel burner assembly.

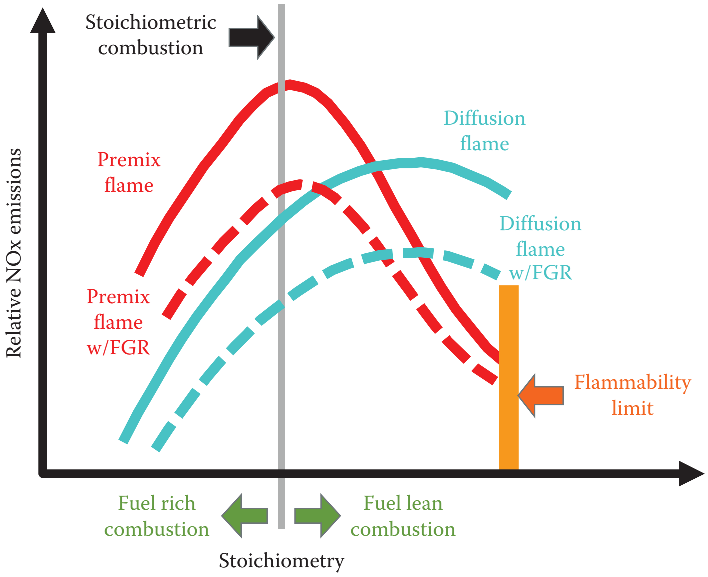


FIGURE 1.14
Ultralow-NOx stoichiometry.

1.2.3 Ultralow-NOx Burners

1.2.3.1 Internal Flue Gas Recirculation: INFURNOx™ Technology

The bold lines in Figure 1.14 show how thermal NOx emissions vary with stoichiometry for a diffusion flame compared with a premix flame. Under sub-stoichiometric or fuel-rich conditions, both types of flames show a rapid drop in NOx as the flames burn under richer and richer conditions. On the other hand, the premix flame shows a peak in NOx emissions very near to stoichiometric conditions and shows a reduction in NOx as the flame is operated under increasingly lean conditions. For a diffusion flame, the peak NOx level is greater than that for the premix flame, and the peak occurs well into the lean operating regime.

Introducing flue gas recirculation (FGR) reduces NOx emissions for both types of flames. This is shown by the dashed lines in Figure 1.14. The FGR dilutes local oxygen concentrations, and more importantly, it reduces flame temperatures. These two factors combine to reduce NOx emissions. On this basis, if one were only interested in limiting NOx, it would be advisable to operate either very lean with a premix flame or very rich with either type of flame and to include FGR.

Unfortunately, operating very lean or very rich would be inefficient and would lead to increases in other pollutants. However, if one burns part of the fuel in a lean premixed flame and the remainder in a rich flame, then combine the products of combustion from these two flames in a final burnout zone, one can achieve very low emissions and an acceptable overall excess air level.

If the lean and rich combustion zones are organized properly, the burner can employ an additional NOx control technique that is often referred to as reburning (see Figure 1.15). It is possible to arrange for the products of combustion from the lean primary flame to pass through the rich secondary flame prior to the final burnout zone. This "lean–rich–lean" combustion scheme allows for the NOx in the primary zone combustion products to either react with the N atoms in the rich secondary flame to form N_2 or to react with the hydrocarbon radicals and thus to recycle back to HCN as shown in the prompt/fuel NOx mechanism (see Volume 1, Chapter 15). This permits the N molecules that were once in the form of NO to have another chance to end up as N_2 rather than remain as NO. At the same time, they are providing a source of

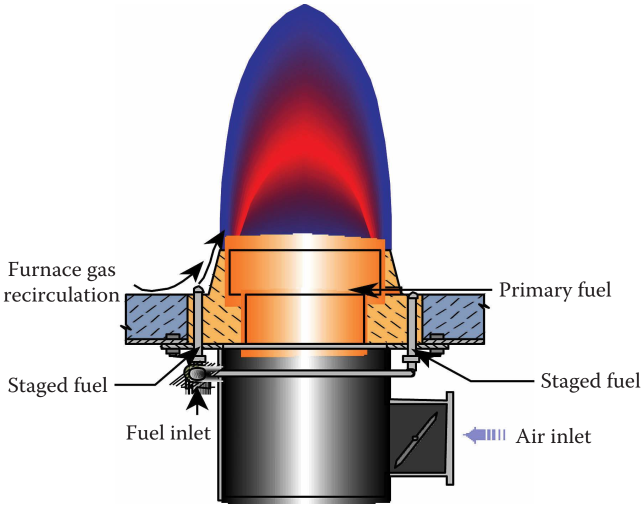


FIGURE 1.15
Ultralow-NOx flame development.

NO molecules to help the N molecules in the fuel-rich region to form N_2. John Zink Company has termed this unique combination of NOx control techniques "INFURNOx™" technology.

1.2.3.2 PSMR Burner

Figure 1.16 shows a John Zink PSMR ultralow-NOx burner that employs INFURNOx technology. The primary fuel is injected from the periphery of the burner to the interior of the burner and entrains furnace gases prior to burning in a lean primary flame zone internal to the burner tile. The secondary fuel, also fired from the periphery of the burner, entrains furnace gases as it passes up the external tile prior to burning at the outlet of the primary flame zone.

1.2.3.3 Lean Premix Burner

Figure 1.17 depicts the variation in NOx with flame stoichiometry for two different types of flames. This figure characterizes the NOx emissions expected

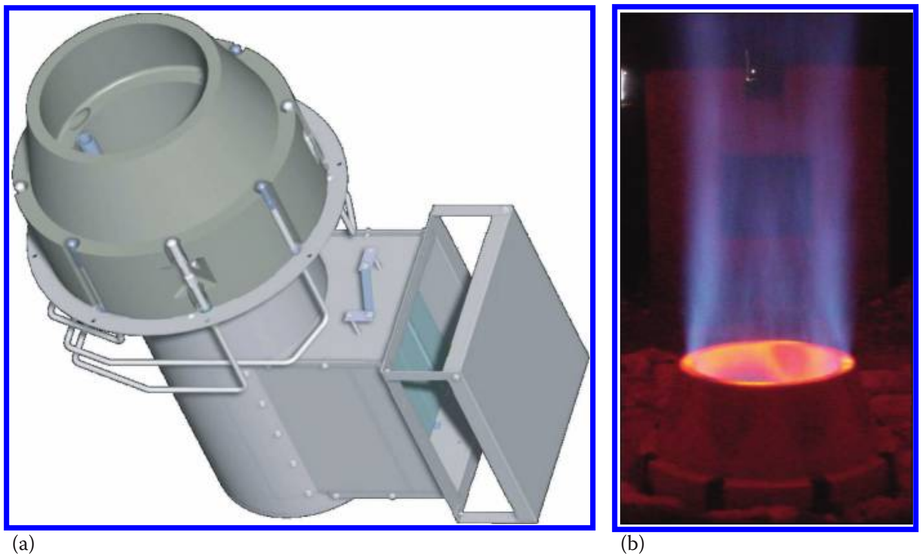

(a) (b)

FIGURE 1.16
(a) PSMR ultralow-NOx burner assembly and (b) flame photo.

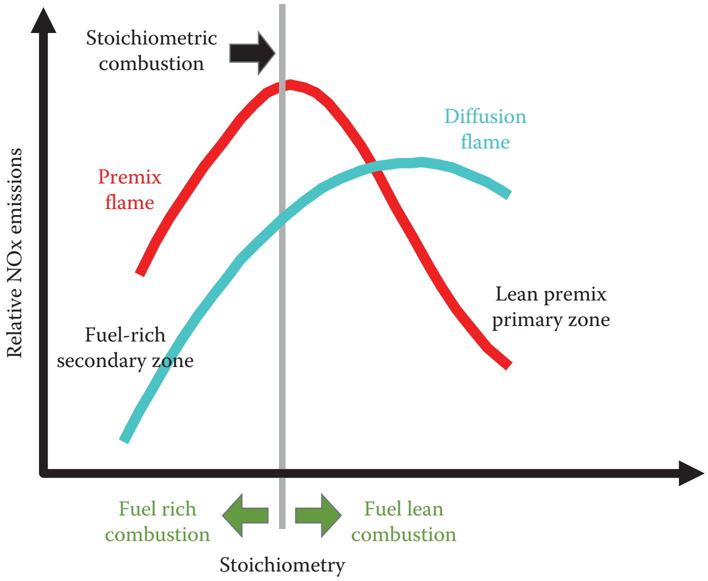


FIGURE 1.17
Relative NOx emissions versus flame stoichiometry.

with a non-premixed diffusion flame and a premixed flame. The curves show that NOx emissions are depressed for either premix or diffusion flames under fuel-rich or sub-stoichiometric conditions and also for very lean premixed flames. Operating a burner excessively rich or lean results in excessive energy losses. In the case of fuel-rich combustion, CO and unburned hydrocarbon emissions are the results due to incomplete combustion. Operating a burner excessively lean can result in CO and unburned hydrocarbon due to quenching of the flame prior to completion of combustion. Figure 1.17 shows that operating with more than one combustion zone can lead to very low NOx emissions with reasonable overall excess air in the final combustion products. It shows that firing a part of the fuel as a lean premixed flame will produce low NOx in that combustion zone. Firing the remaining fuel under fuel-rich conditions with FGR will produce low NOx emissions in another combustion zone. Then properly combining the lean and rich combustion products to form a final burnout zone ultimately allows operation with low excess air and very low NOx emissions.[1]

1.2.3.4 LPMF Burner

The LPMF burner (Figure 1.18) is a gas-fired burner that utilizes lean premix, internal FGR, and fuel staging to achieve reduced NOx emissions. Typical applications have been natural draft, but in theory the burner can be applied as a forced-draft version with or without air preheat as well.

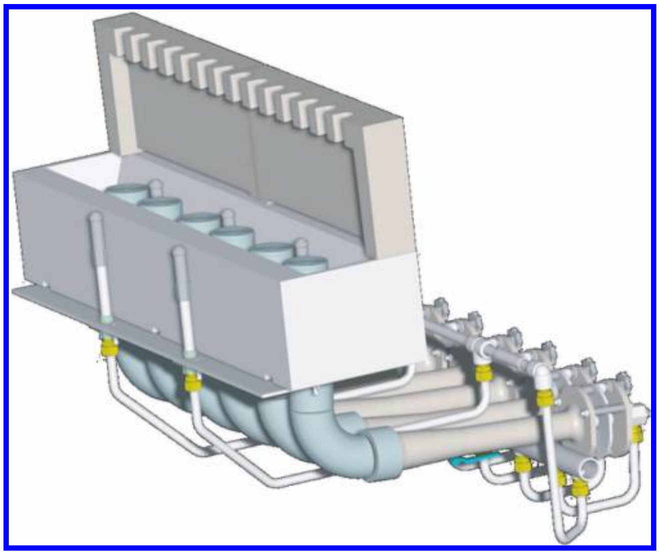


FIGURE 1.18
LPMF lean premix staged-fuel burner assembly.

This type of burner has been successfully designed and applied in furnace configurations requiring burners firing vertically, freestanding or against a wall, and horizontally, free field or along a floor. Additional design and testing was performed with some freestanding models; however, all field applications to date are configured with the burner mounted and firing adjacent to a refractory floor or wall. For the vertically fired models, both vertically mounted and horizontally mounted configurations exist.

The burner design was initially developed and later refined to achieve the lowest possible NOx emissions in an effort to be competitive with post-combustion treatment alternatives to meet the most stringent NOx emissions requirements. This technology is inherently designed to be optimized for NOx emissions performance with relatively high furnace operating temperatures (i.e., above 1400°F [760°C]). In order to achieve good operability and stability in cold operating conditions (e.g., during startup), the burner is equipped with one or more startup tips that are in operation when the firebox temperature is below 1300°F–1400°F (700°C–760°C).

1.2.3.5 Enhanced Internal Flue Gas Recycle Burners

Engineers have continued refining ultralow-NOx burner technology. New developments have focused on increasing the amount of flue gas that is entrained by the fuel into the flame as well as improving the mixing of air, flue gas, and fuel.

1.2.3.6 COOLstar® Burner

The COOL*star* burner is a patented[2] ultralow-NOx burner with compact dimension, high turndown ratio, and reasonably short flame length. The uniqueness of this burner is the tile design. The COOL*star* tile can generate a folded flame pattern like a flower shape if looking down from the top of the tile as illustrated in Figure 1.19.

The COOL*star* tile is illustrated in Figure 1.20. The tile circumference is divided into four or more sections depending on the burner size. The larger the burner size, the more the sections.

Each section is confined by two radially protruding baffles. The baffles have three functions: (1) to stop fuel traveling beyond the tile periphery in order to limit the flame width, (2) to create folded flame pattern, and (3) to stabilize the flame. The computational fluid dynamics (CFD) results for the COOL*star* burner also confirm a folded flame pattern as shown in Figures 1.21 and 1.22. Figure 1.21 shows the CFD

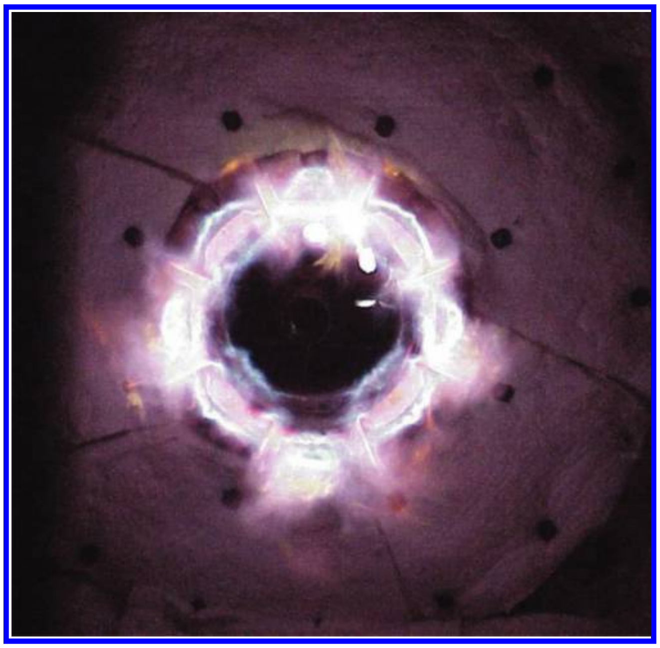


FIGURE 1.19
The COOL*star* folded flame (flower shape when looking down from the top of the tile).

simulation results for the CO concentration contours of a COOL*star* burner. CO, indicated by the color scale on the left, is a generally accepted measure for a flame envelope. Figure 1.22 shows the gas temperature contours of the COOL*star* burner. The warmer color, to the red end of the scale, represents the higher gas temperature. They are all shaped like a flower or a folded flame pattern.

The folded flame creates more area for the combustion zone (or flame sheet) to contact with the surrounding flue gases. When flame mixes with flue gases, the combustion flame temperature is reduced, resulting in low-NOx generation. Hot flue gas also helps stabilize the flame. This is the reason that the COOL*star* burner produces ultralow NOx while keeping a high turndown ratio. A high turndown ratio makes the burner easy to operate. The flue gases are forced to circulate back by several high-pressure fuel jets, which are installed around the tile as illustrated in Figure 1.20.

Due to its design and absence of any physical blockage in the center of the burner, the COOL*star* burner occupies a very small footprint, which means it requires a relatively small heater cutout diameter. The small footprint along with the good air/fuel mixing results in compact flame dimensions as illustrated in Figure 1.23. Since the COOL*star* burner tile is constructed as one solid piece, it can be used for up-firing, horizontal-firing, or down-firing applications.

1.2.3.7 HALO® Burner

A novel new way to enhance the mixing of the flue gas with the fuel and air is by using a Coanda surface. The principle of a Coanda surface is explained in Figures 1.24 and 1.25.

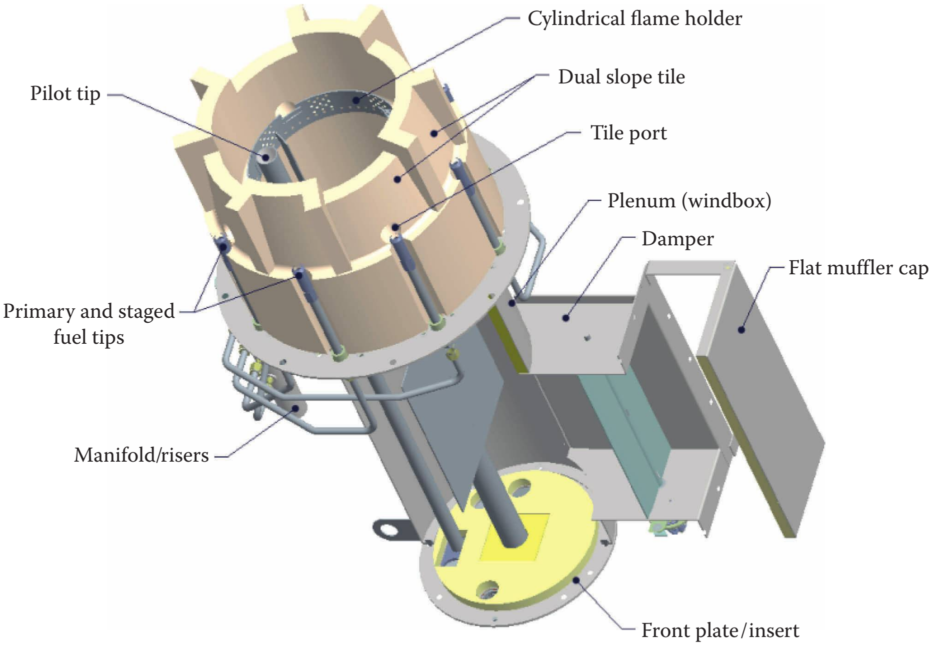


FIGURE 1.20
COOL*star*® burner cutaway.

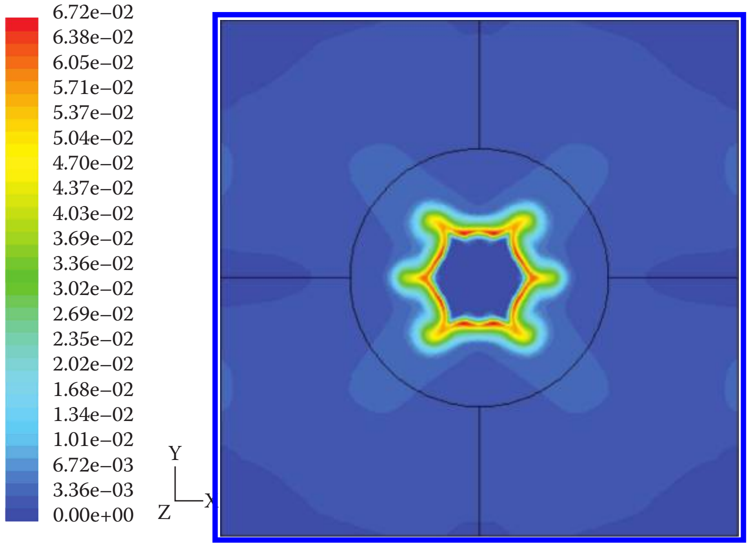


FIGURE 1.21
CFD simulation results for CO concentration contours on COOL*star* burner. The color scales on the left represent the CO mole fractions in the combustion zone.

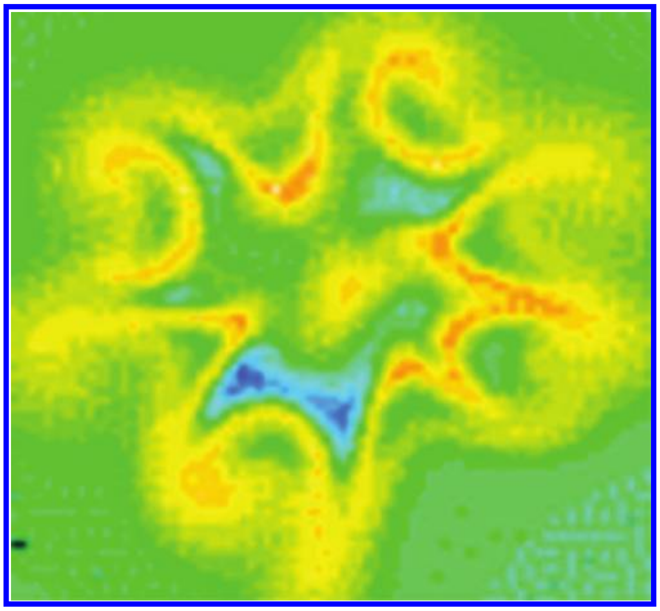


FIGURE 1.22
CFD simulation results for temperature contours of COOL*star* burner.

A fluid that is flowing over a curved surface will preferentially follow the curvature of that surface. The thin film of fluid that flows over the curved surface entrains the surrounding fluid and mixes with it in a very efficient manner. This phenomenon is utilized in the John Zink HALO burner (see Figure 1.26). The gas fuel is injected near a curved burner tile in such a manner that the fuel, air, and flue gas are mixed very efficiently. This allows for compact flames while achieving very low NOx and CO emissions.[3,4]

FIGURE 1.23
COOL*star* flame photo.

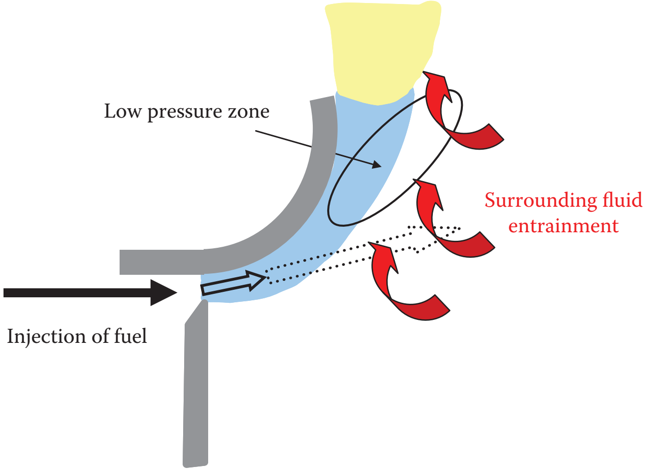


FIGURE 1.24
Entrainment around a Coanda surface.

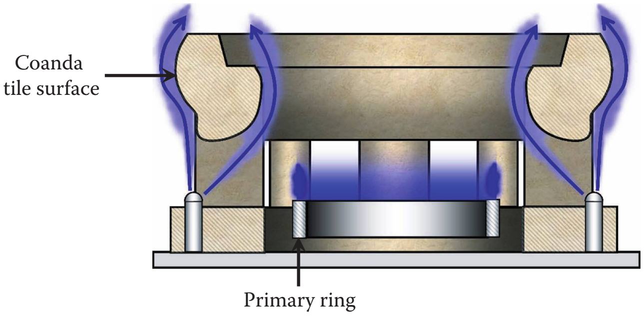


FIGURE 1.25
Coanda tile surface.

FIGURE 1.26
HALO flame photo.

1.3 Flame Shape: Round versus Flat and Freestanding versus Wall-Fired

The main two flame shapes are round and flat. A round flame is typically associated with freestanding burners, that is, there are individual unsupported flames in the middle of the firebox with radiant tubes mounted against the firebox walls (see Figure 1.27). This is a cost-effective way to build a firebox since the amount of tube surface per unit firebox volume can be quite high, but typically results in the tubes being heated only from one side. Firing tubes from one side only is therefore restricted to applications where the tube circumferential heat flux distribution is not critical.

A wall-fired burner, on the other hand, typically produces a flat flame. The idea behind it is to heat the refractory wall and use it as a radiating plane toward the tubes that are located in the center of the firebox. Flue gas just by itself is typically a poor radiative emitter as it emits and radiates only in certain wavebands (see Volume 1, Chapter 7). A solid wall does not have this restriction and is therefore capable of a more efficient transfer of radiant heat. This type of firebox arrangement with burners against the walls and tubes in the center is more costly to build since the amount of tube surface per unit

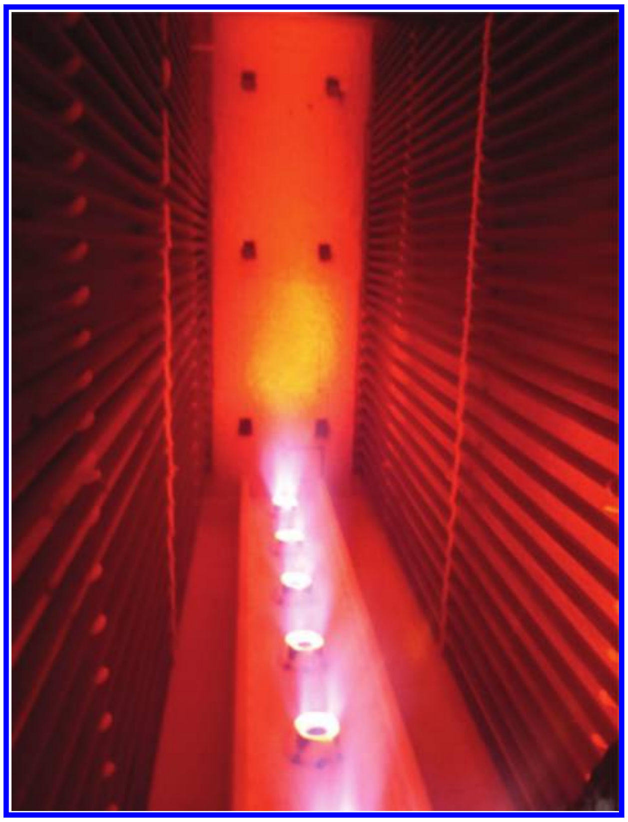


FIGURE 1.27
Freestanding burners.

firebox volume is low. It does allow much better control of heat flux to the tubes and is therefore the preferred solution in those applications where the flux and temperature profile, both longitudinal and circumferential, are critical. These are used in, for example, coker heaters and steam-cracking furnaces.

In some firebox designs, a mix of round and flat flames is used. In such cases, the flat flames fire against the wall, and the round flame burners are arranged in rows between rows of tubes.

1.3.1 Round Flame Burners

A number of round flame burners have already been presented in the preceding sections. In addition to the low-NOx upward-firing burners, round flames are typically used in applications where horizontal flames are required or where burners are firing in the downward direction. A typical downward-firing application is in steam methane reformers, where a natural gas feed is passed over a catalyst with steam in order to produce "syngas," a gas made up primarily of hydrogen and CO. For more information on this process, refer Volume 1, Chapter 2.

1.3.1.1 MDBP Burner

A typical burner for down-firing reformer service is the John Zink MDBP, shown in Figure 1.28. Waste gas from the pressure swing adsorption (PSA) unit, containing large amounts of CO_2, is mixed with makeup fuel gas and injected into the flame through a center gas gun. The design of the burner tile and internals is aimed at creating a compact flame while yielding low NOx emissions due to the presence of the inert components in the fuel gas.

FIGURE 1.28
MDBP burner firing PSA off gas (view looking up at burner).

1.3.1.2 PDSMR Mk-II Burner

Another way to inject the PSA gas into the flame is by individual fuel tips for the makeup and PSA gas. This allows for staging of both these gases and yields additional reductions in NOx emissions. One such burner to employ this concept is the John Zink PDSMR Mk-II, shown in Figures 1.29 and 1.30.

1.3.2 Flat-Flame Wall-Fired Burners

1.3.2.1 PXMR Burner

Flat-flame wall-fired burners, as mentioned in Section 1.1, use the firebox wall to create a certain heating pattern that is compatible with the process. A traditional flat-flame, wall-fired burner for John Zink has been the PXMR burner, shown in Figure 1.31. This relatively simple concept produces good flame patterns and low NOx emissions by injecting primary fuel from the sides and staged fuel from the front of the burner. This burner is most suited for applications with low heat release per burner, such as delayed coker heaters.

1.3.2.2 PSFFR Burner

Applications that require higher heat release per burner, such as steam cracking, are better served with the John Zink PSFFR burner, shown in Figure 1.32.

The PSFFR is used for heat releases up to 10 MM Btu/h (2.9 MW) and produces typical NOx emissions of 45 ppm (corr. to 3 vol% O_2, dry) in ethylene applications. By injecting primary fuel into two internal venturis, the burner entrains flue gas into a primary combustion zone. The resulting flame produces a heat flux profile that is compatible with the most stringent process requirements.

1.3.2.3 LPMF Burner

The LPMF burner (Figure 1.33) is often used in ethylene applications that require even more stringent NOx emissions. This burner uses very lean premixed air/fuel technology to create a quasi-flameless combustion zone. The resulting NOx levels are typically in the range of 25–30 ppm in cracking furnaces with arch temperatures of ~2200°F (1200°C). Section 1.2.3.4 offers more detailed information on the LPMF burner design.

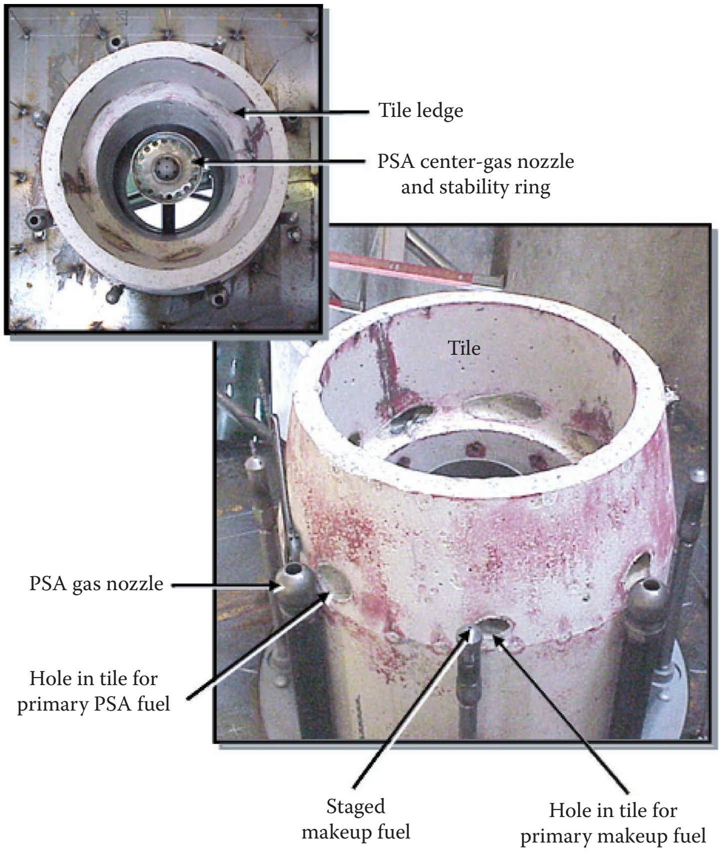


FIGURE 1.29
PDSMR Mk-II tile and fuel tips.

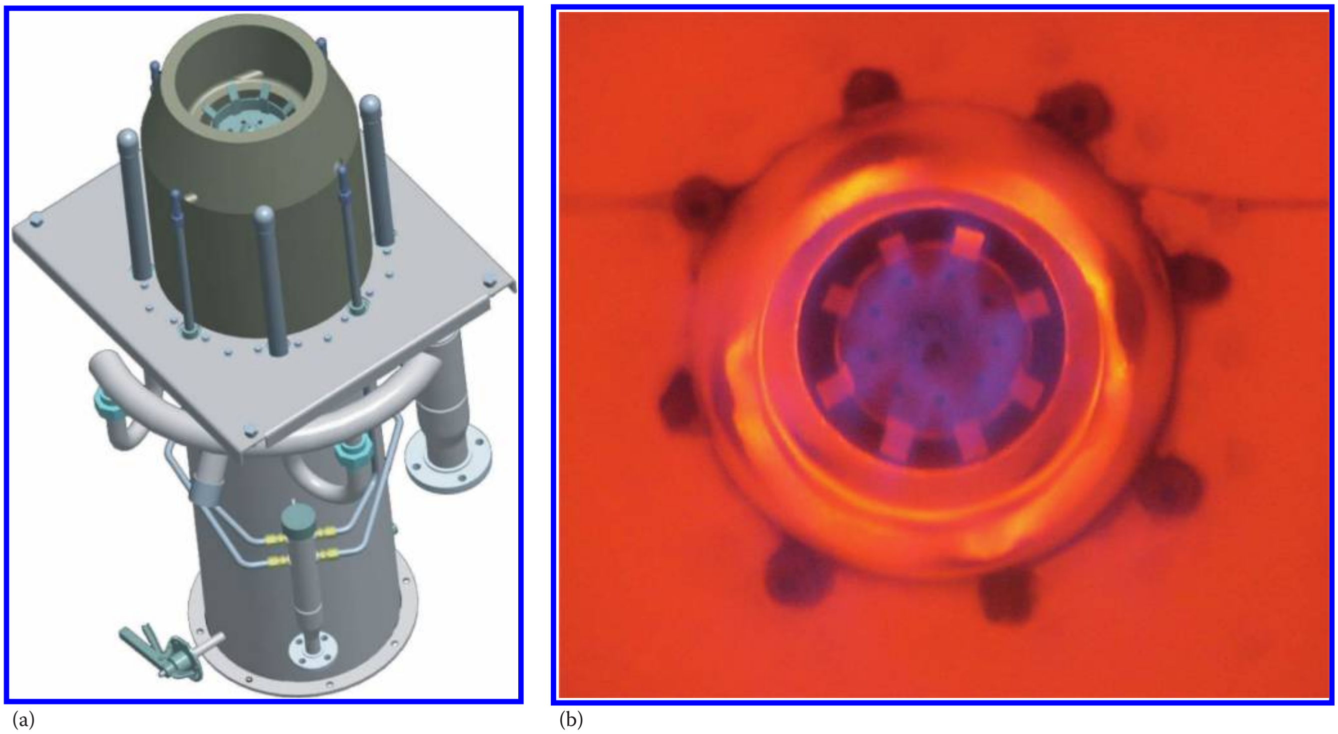

(a) (b)

FIGURE 1.30
(a) PDSMR Mk-II burner assembly and (b) flame photo.

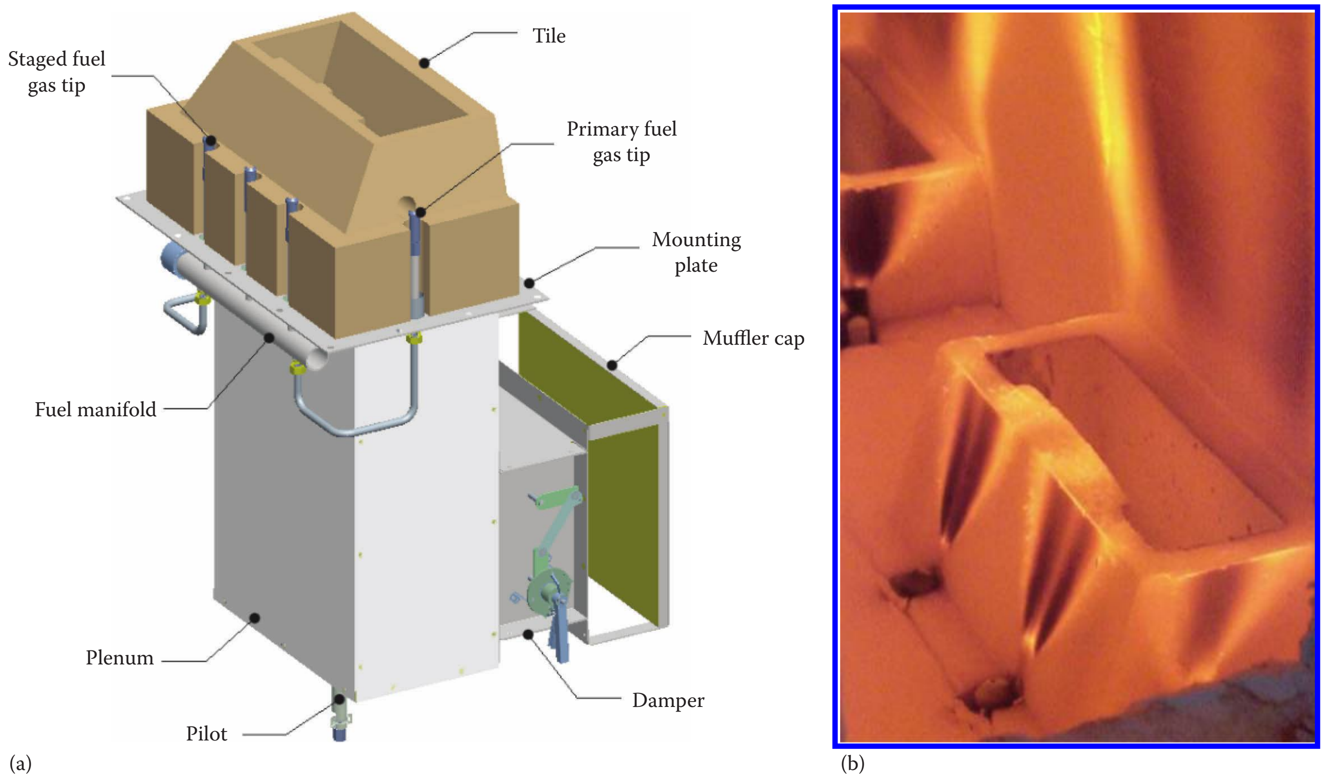


FIGURE 1.31
(a) PXMR burner assembly and (b) flame photo.

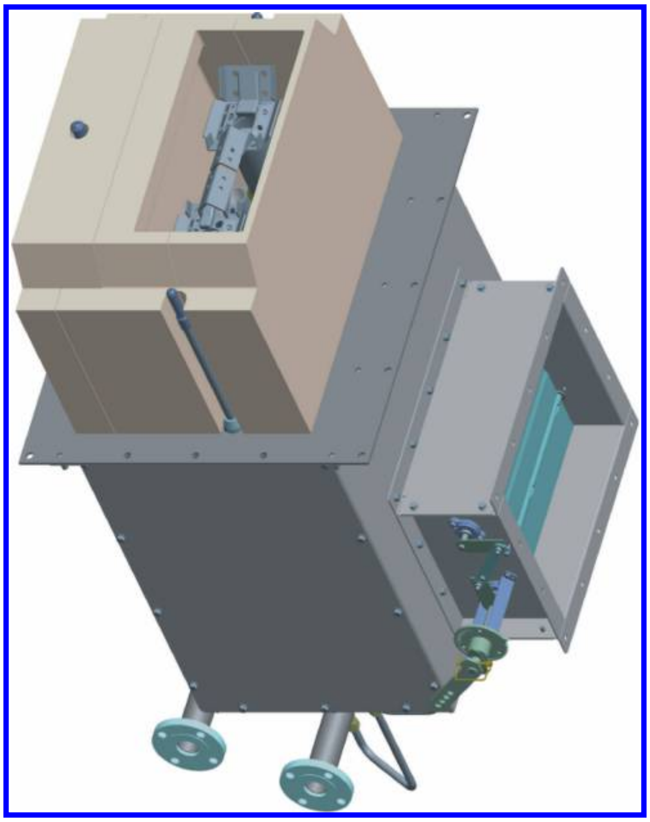


FIGURE 1.32
PSFFR burner assembly.

1.3.2.4 RTW Burner

The RTW burner has been designed specifically for wall-fired coker applications. Built for low burner pressure drop (less than 0.3 in H_2O [0.75 mbar]) and heat releases ranging from 1.5 to 3 MM Btu/h (0.44 to 0.88 MW), it produces very short and wide flames. During the burner design process, CFD was used to ensure optimal air and fuel distribution. With only four gas tips, the burner produces a very uniform heating of the refractory wall, resulting in optimal flux control. The NOx emission level ranges from 20 to 35 ppm (corrected to 3% O_2) depending on air preheat temperature. Figures 1.34 and 1.35 compare photos of the test burner and CFD-calculated result.

1.3.3 Flat-Flame Freestanding Burners

1.3.3.1 PXMR-DS Burner

Applications that require freestanding flat flames, such as steam superheaters in styrene monomer plants, are served well with the John Zink PXMR-DS (Figure 1.36). This burner is essentially a rectangular version of the John Zink COOL*star* burner (Section 1.2.3.6). NOx emissions from this burner are comparable to the COOL*star* burner.

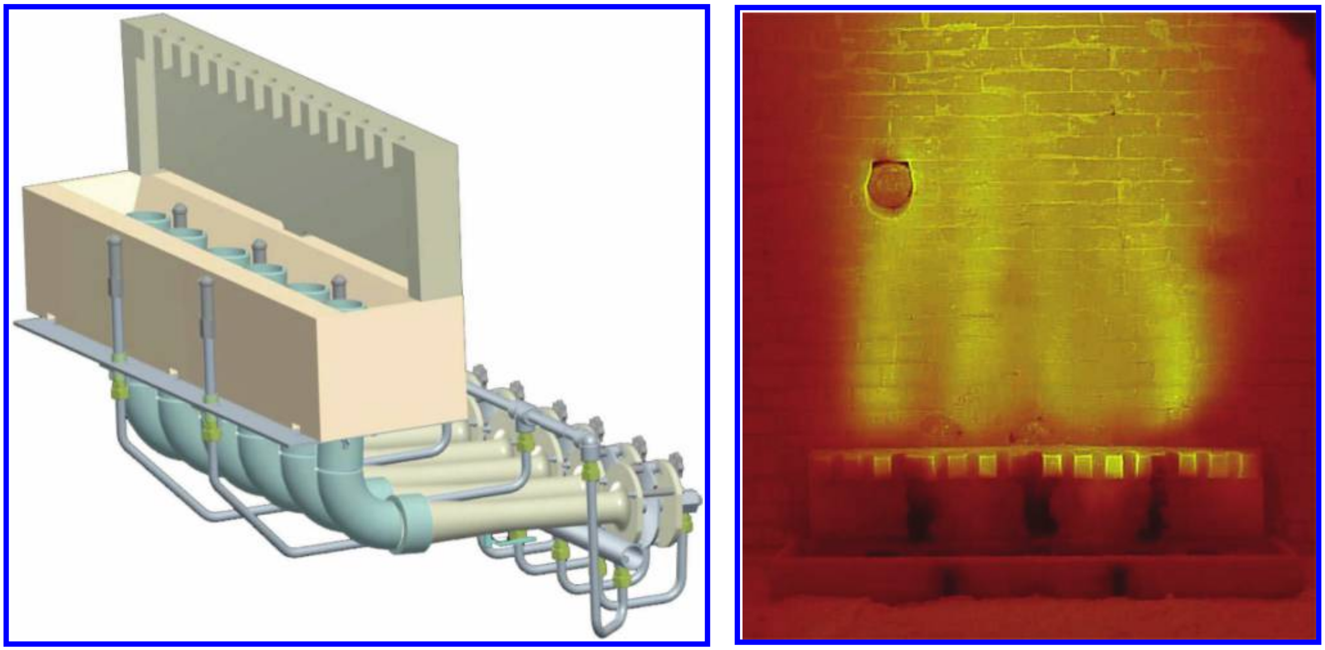
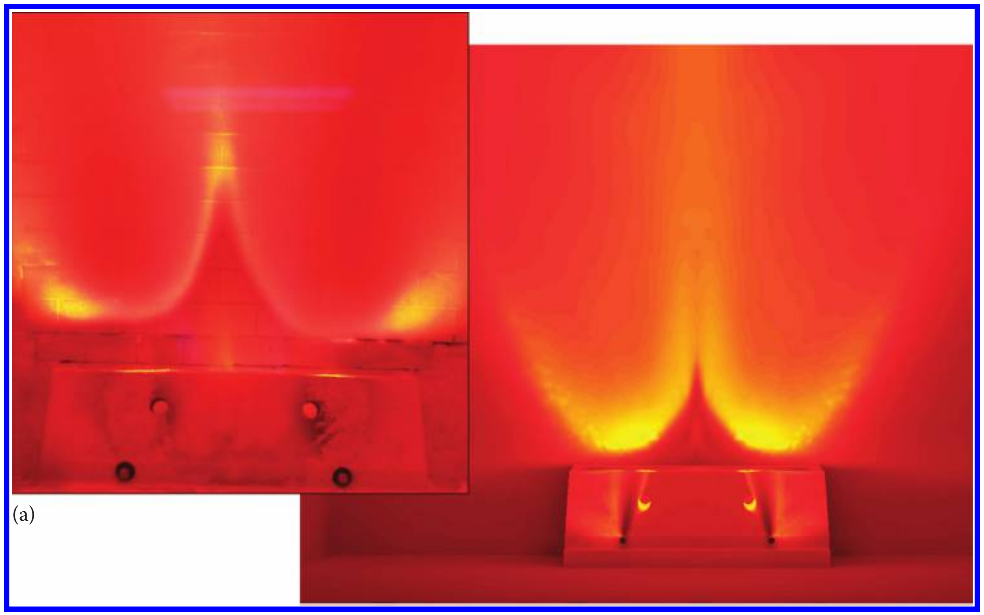

(a) (b)

FIGURE 1.33
(a) LPMF burner assembly and (b) flame photo.



(b)

FIGURE 1.34
(a) RTW test burner and (b) CFD-calculated result.

1.4 Radiant Wall Burners

1.4.1 Premix Radiant Wall Burners

Applications that require very even heat flux to process coils, such as hydrogen reforming and hydrocarbon cracking to ethylene, often use double-fired process coils. The radiant wall burner is a common method used to ensure uniform heat distribution. Premix radiant wall burners are the most common (see Figure 1.37).

1.4.1.1 PMS Burner

The John Zink series PMS radiant wall, premix gas burner has been utilized in hydrogen reformers and

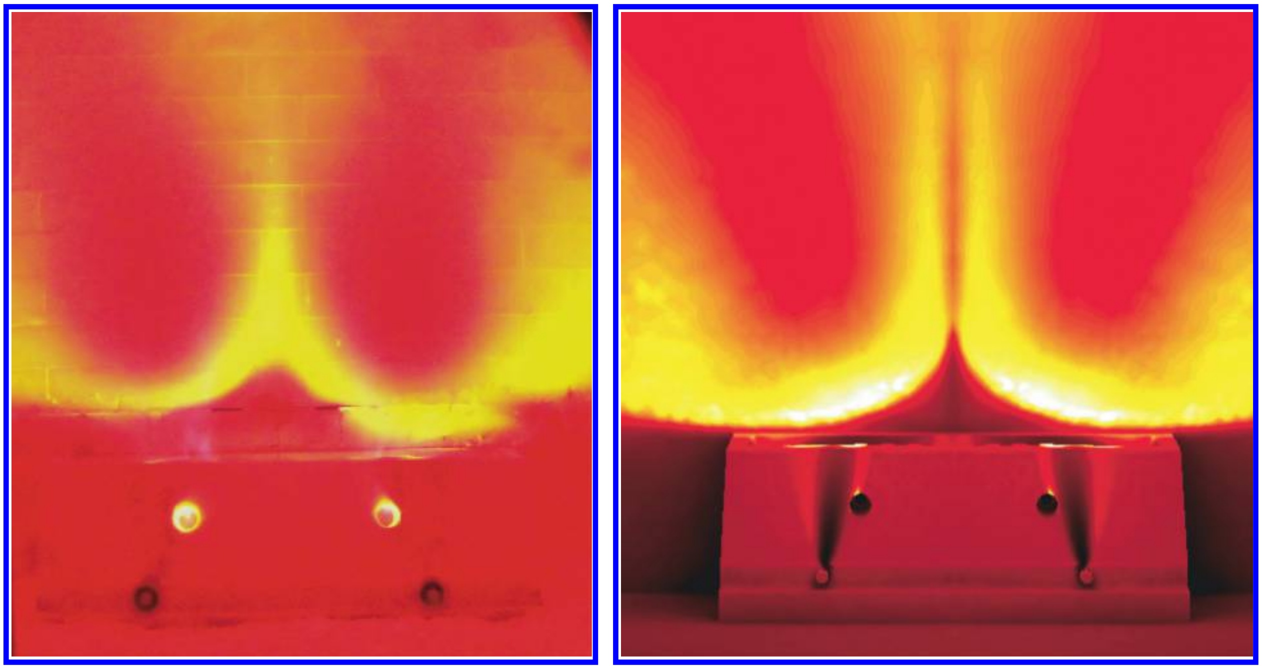

(a) (b)

FIGURE 1.35
(a) Modified RTW burner and (b) CFD-calculated result.

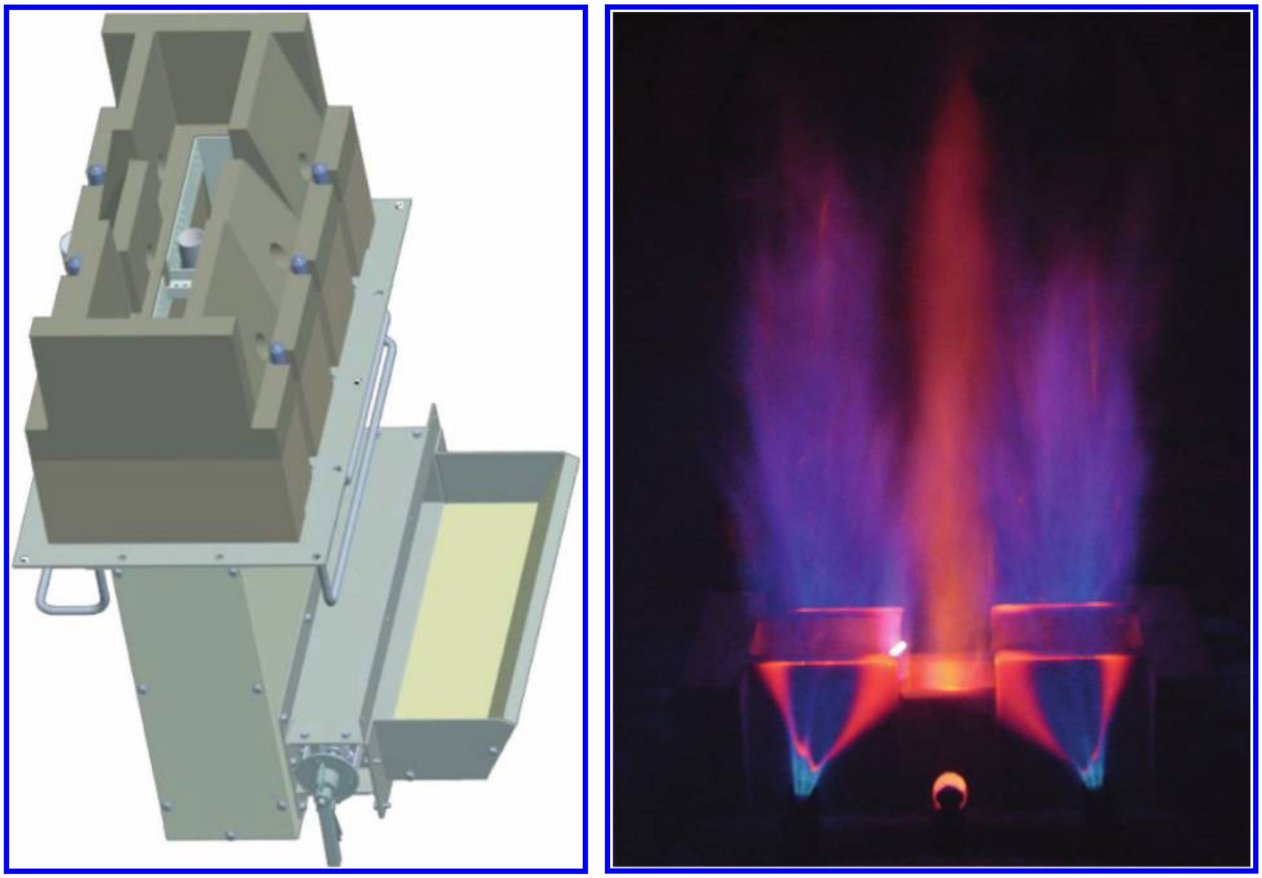

(a) (b)

FIGURE 1.36
(a) PXMR-DS burner and (b) flame photo.

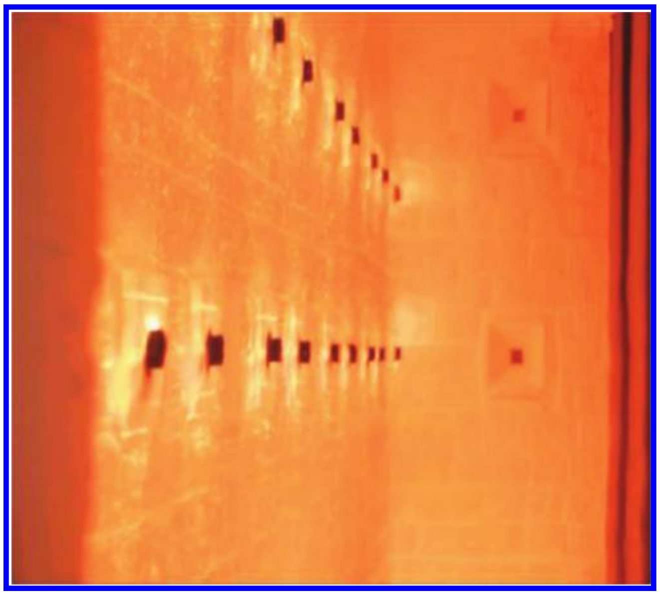


FIGURE 1.37
Premix radiant wall burner array.

ethylene-cracking applications since the late 1950s. These burners have evolved over time from their original low-capacity, low-efficiency premix assemblies. The older designs used sand castings for the venturi and tips. These castings, with their rough internal surfaces, the venturi inlet, and tip outlet configurations were not optimum. The present-day design of the PMS burner can reach higher capacities with higher premix stoichiometries. This increase in capacity and premix efficiency is due to the new spin-cast or extruded venturis and higher-quality, higher-efficiency outlet premix gas tips. The new venturis have smoother interior surfaces, reducing frictional losses, and better inlet configurations, reducing inlet losses. The new tips are higher-quality castings, are smoother, and have rounded surfaces in all flow passages for reduction of losses. Computer modeling or CFD (see Volume 1, Chapter 13) and cold flow testing (see Volume 1, Chapter 11) have been used extensively to ensure minimal losses and maximum conversion of the fuel gas motive energy into mass of air entrained.

The basic PMS burner, shown in Figure 1.38, is comprised of five parts: (1) the venturi, (2) the primary premix tip, (3) the fuel orifice, (4) the primary air door, and (5) the secondary lighting and sighting port(s). Burner tile assemblies including tile mounting plates, high-temperature hot face, and an insulating backup block are also available. The primary insert starts with a venturi, or fuel/air mixer, and the premix distribution tip. The fuel gas is metered and injected into the venturi through the fuel orifice that is positioned at the entry to the venturi. The PMS burner fuel metering is located externally to the heat-affected furnace zone. The fuel is used to provide the motive energy to draw ambient air into the venturi for premixing with the fuel prior to the ignition zone. Primary air can be manually adjusted by opening or closing the primary air door assembly, also located at the entry to the venturi. If required, secondary air can be introduced to the burner, outside the premix system, through the lighting and sighting port(s) located in the annular space around the venturi and tip assembly.

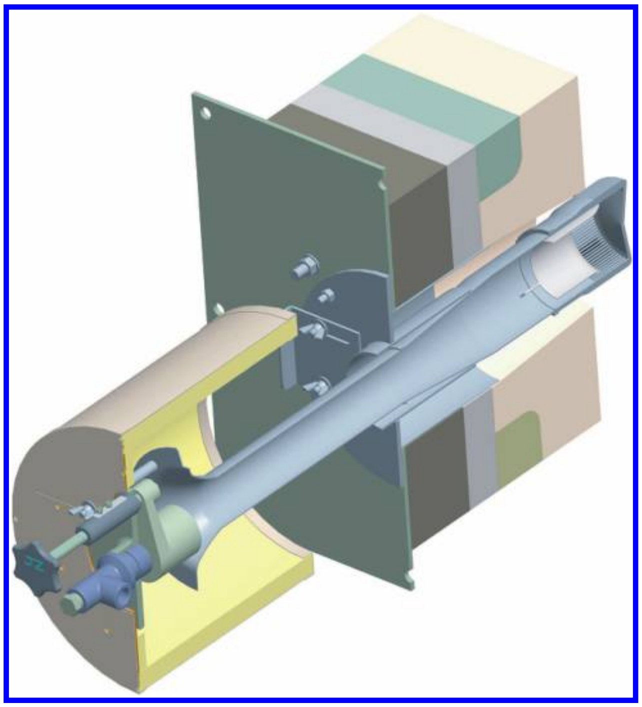


FIGURE 1.38
PMS premix gas burner assembly.

The PMS burner is a high-efficiency premix burner that provides a flat, radially projected flame pattern (see Figures 1.39 and 1.40). This burner is typically mounted horizontally but can be mounted in any direction. The burner is mounted flush through a furnace refractory-lined casing, with its tip projecting a short distance into the radiant chamber. The radial flame then heats the surrounding refractory, which then radiates to the process. Due to the high efficiency of its premixing of combustion air with the fuel, PMS burners are relatively immune to furnace draft variations. These burners can also operate through a range of firing rates with only small changes in stoichiometry.

The PMS burner can be sized for 100% premix capacity including excess air (all air through the venturi and tip). Capacities can range from 0.07 MW (240,000 Btu/h) up to 0.88 MW (3.0 × 10^6 Btu/h). As a premix burner, it is limited in fuel composition, range of fuel calorific value, and range of fuel molecular weight (see Volume 2, Chapter 6). Typically, a fuel with 80 mol% or greater hydrogen content will severely limit the range of operation due to flashback. Additionally,

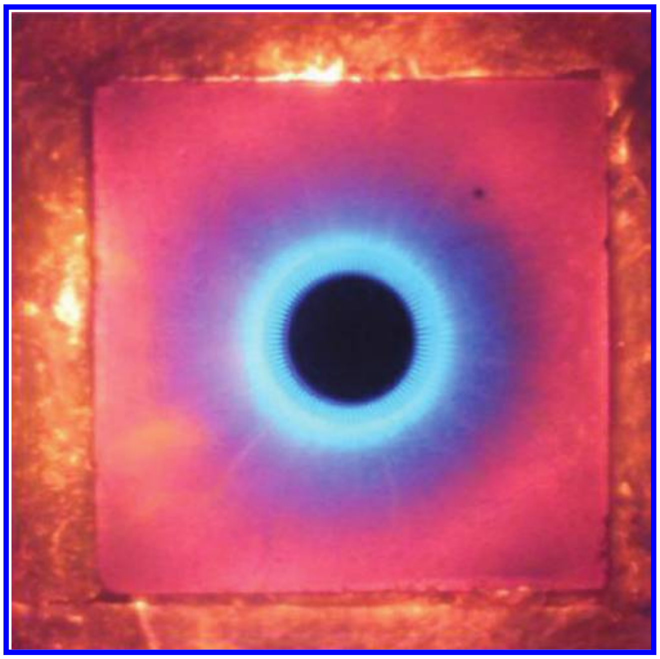


FIGURE 1.39
PMS flame (front view).

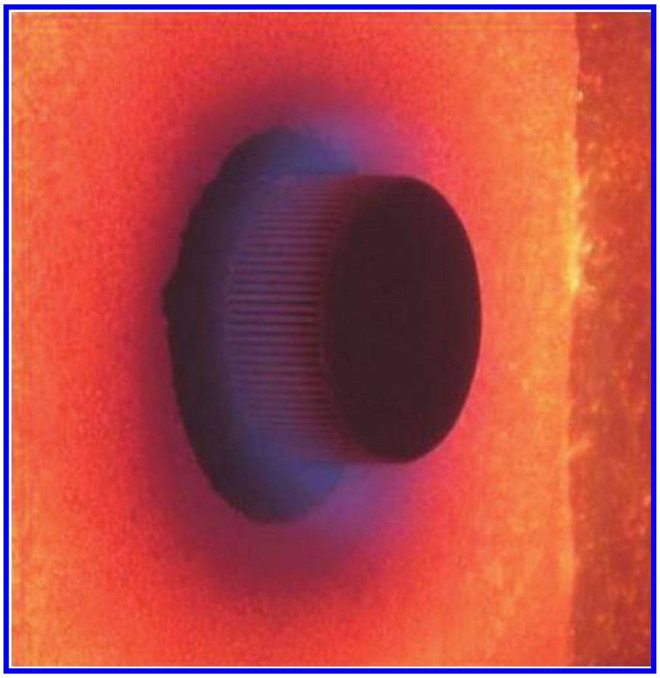


FIGURE 1.40
PMS flame (side view).

if a burner is designed for a typical midrange calorific value fuel of 23.6–47.3 MJ/N-m^3 (600–1200 Btu/scf), attempting to operate on higher calorific gases (such as propane or butane) will result in a limit to the maximum capacity. This limited capacity is due to the loss in air entrainment due to the lowered pressure required for fuel supply. A typical turndown capacity for these burners is 25% of burner maximum design capacity. This is often increased (higher percent of design rate/less turndown), dependent on the maximum design rate, the fuel composition, and the design fuel pressure.

1.4.1.2 Hamworthy Walrad Burner

Hamworthy's Walrad series burner (Figure 1.41) has been designed for the process industry where an aspirating radiant wall burner is required. The burner fuel pressure is used for aspiration and mixing of the combustion air with the fuel. Due to this, these burners are generally accepted as limited in gas composition range to less than 60 mol% hydrogen. Beyond this point, the possibility of flashback (preignition within the burner insert) is vastly increased. The momentum of the fuel gas is used to entrain atmospheric air using a jet and venturi. The standard Walrad burner utilizes the fuel to entrain 100% of the total combustion air; therefore, the gas injector is designed for the highest pressure available to maximize the momentum of the fuel gas. Secondary air ports are typically closed or slightly open to allow a cooling air path and have been included in the base design to allow small changes in the burner design operation or for operational flexibility. The low-NOx version uses the staged-fuel principle of NOx reduction taking approximately 20% of the total gas through a single auxiliary fuel injector.

Over the years, the burner insert design has been improved to allow a greater flexibility with regard to performance. During the development of the new design (in particular the main gas nozzle), full use was

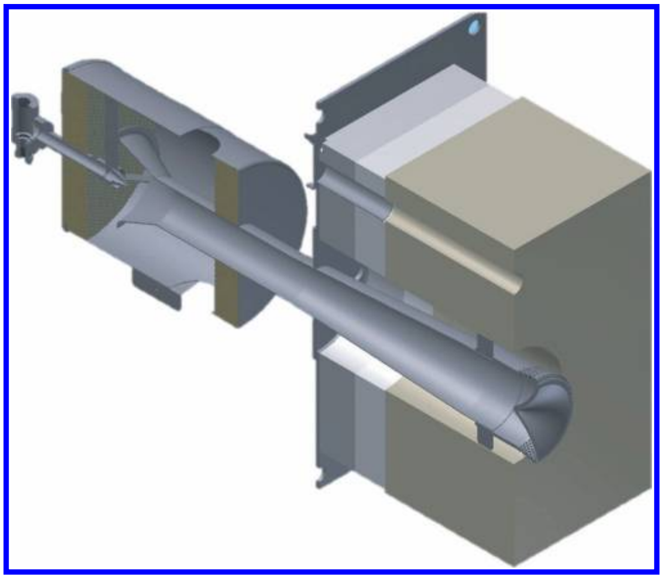


FIGURE 1.41
Hamworthy Walrad burner assembly.

made of Hamworthy's flow modeling and test facilities. Improvements were made to the internal gas/air flow patterns, eliminating the air pockets and dead zones that were present due to internal recirculation.

In addition, during the "hot" firing tests, it was seen that the nozzle produced skin temperatures in excess of 870°C (1600°F) in what was considered critical areas. The outlet nozzle was therefore redesigned to reduce its temperature. This redesign included the following:

1. Reduction in total mass
2. Redesign of the nozzle drilling pattern—creating a flame front further from the nozzle, reducing direct heating
3. Reduction of the setting dimension (protrusion into the furnace)
4. Improved flow characteristics through internal profile change

The old, heavy cast components have been replaced with a fabricated stainless steel construction. This results in a significant weight reduction and increased corrosion resistance. This construction offers the following benefits to the end user:

- Total weight of the burner insert of 15 kg (33 lb)
- No necessity for complex paint systems and continued repainting
- Reduction in burner standoff from the furnace

The standard Walrad burner is comprised of five parts: (1) the burner body (venturi), (2) the main gas nozzle (primary premix tip), (3) the fuel gas jet, (4) silencer and damper, and (5) the secondary air port(s). The primary insert starts with a venturi, or fuel/air mixer, and the premix distribution tip. Burner tile assemblies including tile mounting plates, high-temperature hot face, and insulating backup block are also available.

The burner can operate with a wide range of fuel gases from 60 mol% hydrogen to a highly inert purge gas through propane. It should be noted that high gas pressures are required at the extreme cases. Lower fuel gas pressures will limit the burner's ability to fire higher-hydrogen fuels. The Walrad burner can be designed for duties ranging from 0.1 to 0.7 MW (341,000 Btu/h to 2.39×10^6 Btu/h). A typical turndown would be 33% of design (3:1) and higher fuel gas pressures can result in greater turndown.

1.4.1.3 LPMW Burner

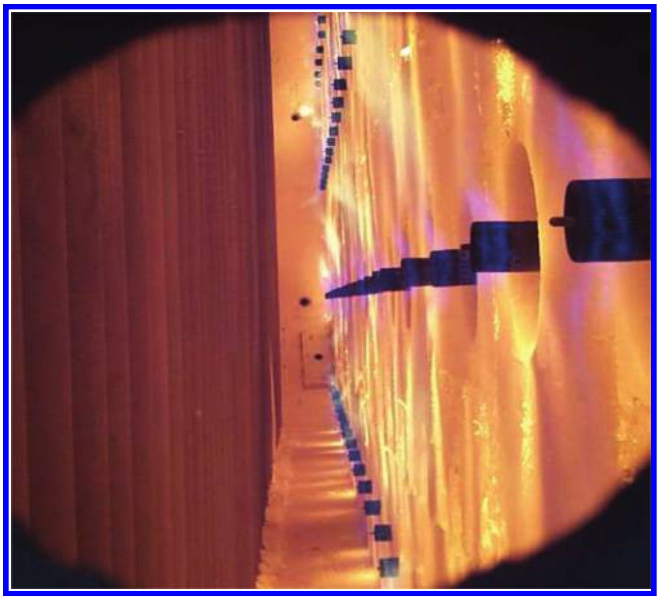


FIGURE 1.42
LPMW radiant wall burner array.

The John Zink series lean premix wall (LPMW) burner shown in Figure 1.42 is a gas-fired, radiant wall, ultralow-NOx burner. As with other radiant wall burners, it is mounted through the refractory-lined casing of a radiant chamber and provides a minimal projection radial flame to heat the surrounding refractory. It uses lean premix primary fuel technology and high-temperature diffusion flames for stability. It uses a centrally located, radially fired staged-fuel injector in conjunction with the lean premix primary to achieve reduced NOx emissions.

The LPMW fuel metering orifices are located external to the furnace, placing the smallest orifices outside of the heat-affected zone. The venturi and primary premix distribution tip are "sized" dependent on the capacity required, the fuel composition, and the fuel pressure available. The primary fuel orifice is typically sized for 50%–65% of the burner's design capacity. This reduced quantity of the total fuel must entrain all of the combustion air required for both the primary and staged fuel, including any design excess air. The staged insert provides the remaining fuel required to meet the design capacity. The staged fuel is introduced into the combustion zone in such a way that it mixes with the low-oxygen furnace gases prior to being incorporated into the flame.

There are three basic designs of LPMW burner that can be provided[5–8]:

1. The first design (Figure 1.43) provides the venturi offset at an angle from the primary premix and staged tips. In this design, the staged-fuel insert is mounted through an elbow. This configuration provides access to the staged insert,

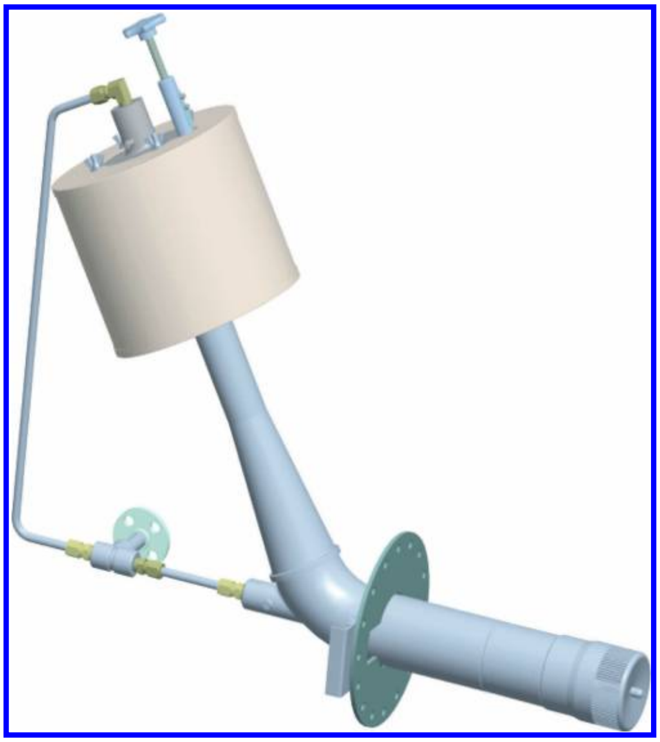


FIGURE 1.43
LPMW with elbowed venturi.

enabling maintenance of the staged tip without dismounting the main premix burner from the tile or furnace. Full shutdown of the burner is required, but the burner does not have to be dismantled, and the main fuel piping to the burner does not have to be disconnected.

2. The second design (Figure 1.44) provides a more compact in-line venturi and tip configuration. In this design, John Zink has developed an in-line adapter that allows the venturi to remain in direct line of flow with the primary tip, increasing the premix efficiency. This in-line adaptor incorporates flow vanes and provides the flow channel for supply of fuel to the center-mounted staged-fuel tip. Maintenance of the staged tip now requires the disconnection of the fuel supply and the dismounting of the burner insert from the tile or furnace.

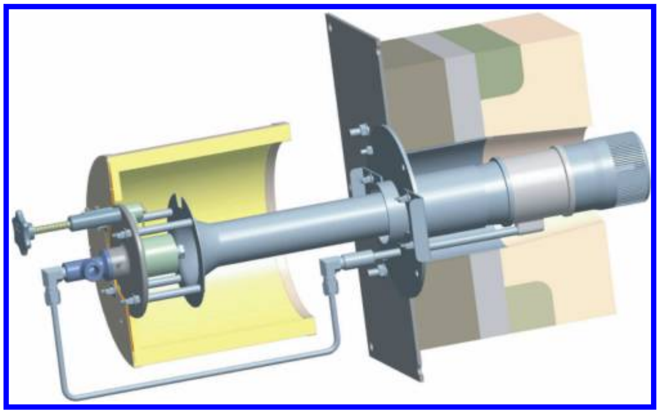


FIGURE 1.44
LPMW with in-line venturi and staged-fuel adaptor/tip.

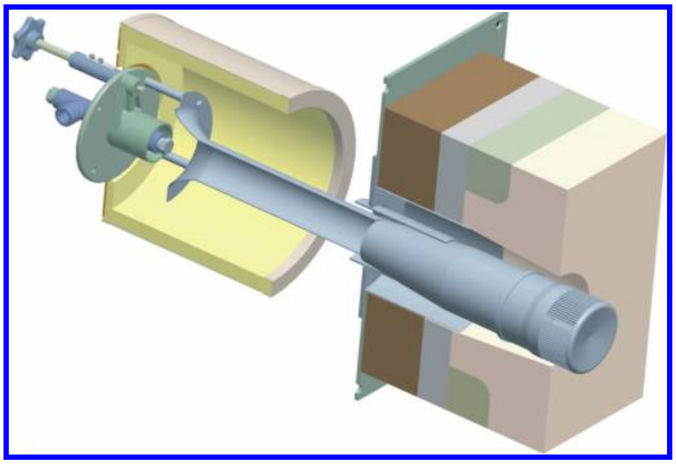


FIGURE 1.45
LPMW with in-line venturi for RFS integration.

3. The third design (Figure 1.45) is similar to the second. In this design the venturi remains in-line with the primary premix tip. However, the adaptor and the staged-fuel tip are omitted. This design is specifically for the purposes of integrating John Zink's patented RFS (remote fuel staging) insert. The primary fuel remains the same sub-100% of the design capacity. The premix stoichiometry remains the same as if the burner had staged fuel, in the range of 170%–220%. The staged fuel required to complete the burners' design capacity will be injected into the combustion zone from a totally separate injection point not contained within this burner.

1.4.2 Raw Gas Radiant Wall Burners

1.4.2.1 FPMR Burner

The FPMR burner, seen in Figures 1.46 and 1.47, is designed to operate with cold or preheated combustion air under forced-draft conditions. The FPMR burner is designed in such way that fuel and air are introduced into the furnace at separate locations. The fuel is injected into the furnace through a central gas tip. The combustion air is provided in an annular region around this tip and is injected in a radial pattern into the furnace, parallel to the furnace wall. Prior to the initiation of the combustion reaction, both the fuel and the air mix with high-temperature, inert furnace gases.

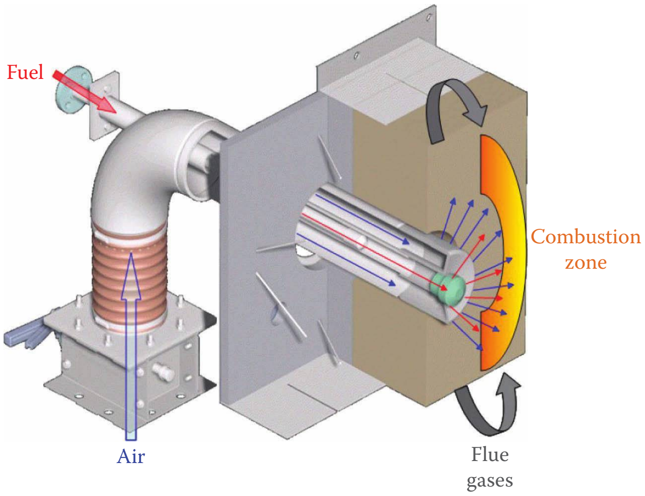


FIGURE 1.46
FPMR radiant wall burner cutaway.

The fuel is finally oxidized in a quasi-flameless combustion environment.

In order to ensure that the distribution of the air is as even and as parallel as possible, the exit velocity of the air tip slots is kept higher than the velocity of the air in the annulus.

The furnace gases are entrained into the combustion zone by the high exit velocity of the air leaving the air tip and by the fuel gas. By entraining inert gases into the combustion zone, the burner reduces NOx emissions by reducing the actual flame temperature that, in turn, reduces the amount of thermal NOx formation. Additionally, by keeping the air and fuel streams separate for as long as possible, the combustion surface is increased, and the heat per unit volume produced is decreased, further reducing the flame temperature. The burner can be designed to use high-pressure gas (typical 2 barg or 29 psig), low-pressure gas (typical, PSA at 0.2 barg or 2.9 psig), and vaporized heavy fuels such as propane, butane, pentane, and naphtha. FPMR burners have been supplied for use in steam/methane reforming furnaces and could also be applied in ethylene dichloride crackers (EDC).

1.5 Combination Gas and Oil Firing

Because of availability, ease of transport, and flexibility in storage, many refining and petrochemical plants use liquids as fuel, especially outside the United States. Design of a liquid-fired burner is significantly different from that of any gas-fired burner. In addition, the operation of a liquid-fired burner presents many more problems than those of a gas-fired burner. For those reasons, a majority of burners designed for liquid firing are not liquid fired only but are also equipped with gas-firing capability. Therefore, oil burners for process heaters are most commonly combination gas/oil burners.

Firing a liquid fuel requires atomization of the fuel. In order to efficiently combust, the liquid fuel needs to be broken up into small droplets. Smaller droplets allow fast surface vaporization, providing the required gas phase for mixing with air. To break the fuel into small droplets, an atomization system (oil gun) is required. The process of oil atomization is discussed in detail in Volume 1, Chapter 10. Here, the most commonly used oil guns are presented.

With the requirement of liquid and gas dual-fuel firing capability, the configuration of combination burners

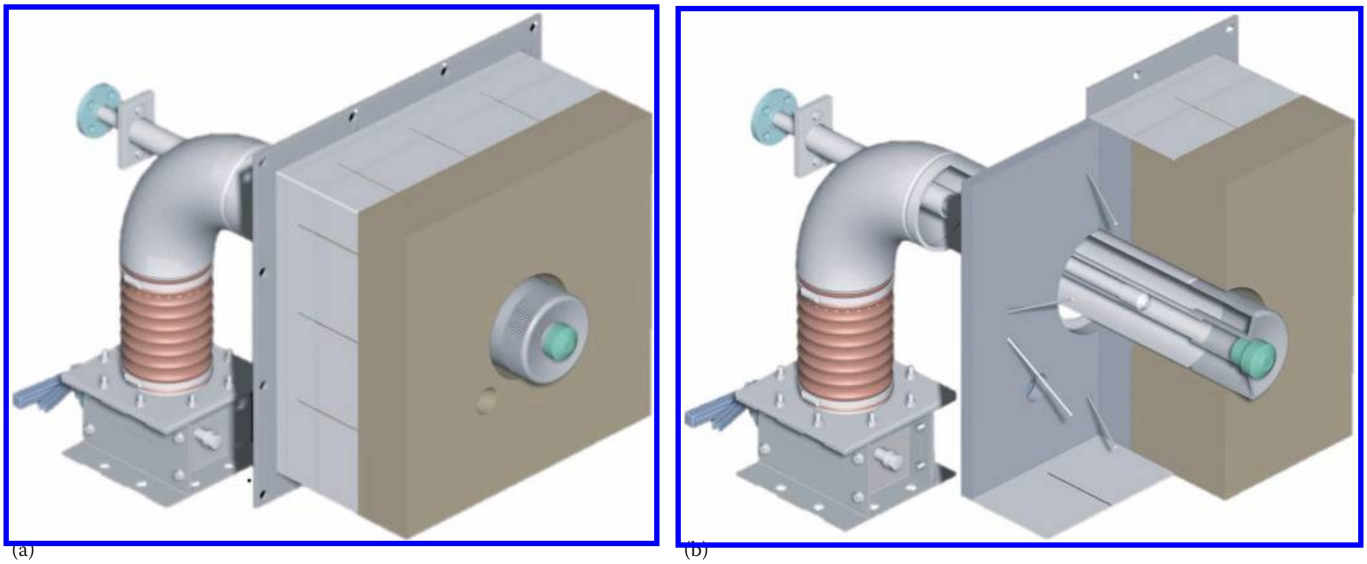

(a) (b)

FIGURE 1.47
(a) FPMR burner assembly and (b) cutaway.

becomes complicated. Gas/oil combination burners are discussed in Volume 2, Chapter 6, and are discussed in more detail in Chapter 2 of this book. As follows, different designs of atomization systems and combination burners are introduced.

1.5.1 Atomization Systems (Oil Guns)

Most oil guns in industry are of a concentric tube design. In most oil guns designed with concentric tubes, the oil flows within the inner tube, while the atomizing medium, most commonly steam, flows through the annular area formed by the inner and outer tubes (see Figure 1.48).

The concentric tube design is well suited for the high-viscosity residual oil commonly called No. 6 oil, or other high-viscosity liquid fuels. Viscosity is a function of temperature. Higher temperature results in lower liquid hydrocarbon oil viscosity. Low-viscosity liquid is much easier to atomize. A concentric tube arrangement atomized with steam has the advantage of heating the oil and keeping the fuel at low viscosity.

Concentric tube oil gun designs can be used for light oil, or low-viscosity distillate oils. The fuel pressure versus fuel flow characteristics will be different, as discussed in detail in Chapter 2. However, if the light oil contains high vapor pressure (volatile or low boiling point) components, concentric tube oil guns utilizing steam for atomization are not recommended. The heat imparted by high-temperature steam can vaporize volatile components and cause vapor lock resulting in pulsation or instability of the oil flame. In this case, a dual-tube oil gun design as shown in Figure 1.49 is recommended.

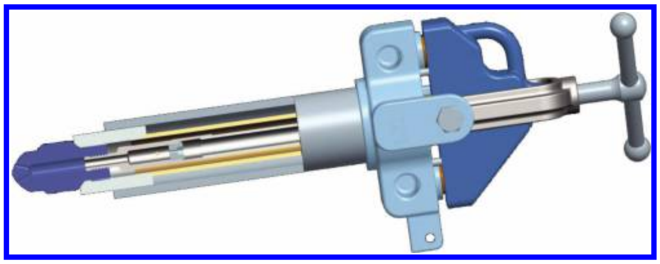


FIGURE 1.48
Oil gun with concentric tube design.

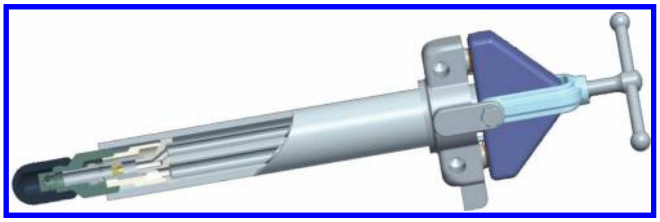


FIGURE 1.49
MEA oil gun with dual (parallel)-tube design.

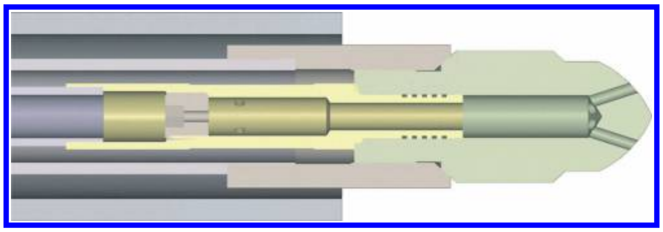


FIGURE 1.50
John Zink EA oil atomizer and tip.

1.5.1.1 John Zink EA Oil Gun (Internal Mixing Chamber)

The JZ EA oil gun with the concentric tube design is shown in Figure 1.50. In this process oil gun, the atomizing medium pressure is controlled at a positive differential above the oil pressure. In an internal mixing chamber type of oil gun, the high-pressure steam used as the atomizing medium is injected through several atomization ports into the mixing chamber where the metered oil is flowing. For heavy oils requiring heating to achieve the correct viscosity, this differential is 20–30 psig (2.4–3.1 barg) higher than the oil pressure. The steam-oil mixture is then introduced into the furnace through the tip. In the EA series oil atomizer, the exit from the atomizer mixing chamber is axial. This design is especially good for residual oils with suspended particulates. It is also beneficial in locations where the oil viscosity is difficult to maintain due to inconsistent fuel supply or lack of oil temperature control. The exit ports on the tip can be arranged at various spray angles and different spray shapes, that is, hollow cone shape or fan shape, depending on the application. A detailed description can be found in Volume 1, Chapter 10 on oil atomization and in Chapter 2 of this book.

1.5.1.2 John Zink MEA Oil Gun (Internal Mixing Chamber)

The MEA oil gun design is similar to the EA gun with the exception of a modified mixing chamber outlet and tip design as indicated in Figure 1.51. Before reaching the tip, the steam-oil mixture is split into multiple

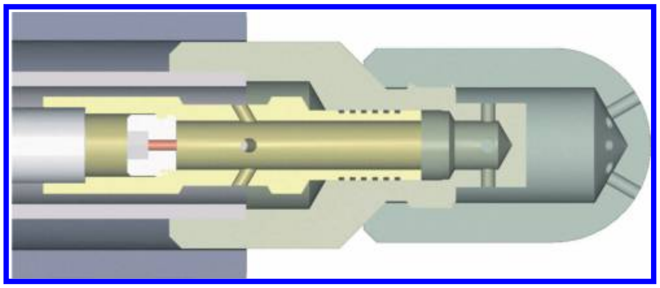


FIGURE 1.51
John Zink MEA oil atomizer and tip.

streams and turned 90°. The exiting atomized/mixed oil-steam stream impacts the walls of the tip exit chamber prior to exiting. These additional changes in flow direction and separation into multiple streams prevent large droplets from reaching the exit ports. This design is intended to reduce particulate emissions generated by low atomization quality.

1.5.1.3 Hamworthy SAR (Steam Atomized Residual) Oil Gun

The Hamworthy SAR oil atomization system is an internal mix atomization system used where sufficient atomizing pressure is available (see Figure 1.52). The gun is designed as a concentric tube arrangement. The fuel travels through the inner pipe to the core (mixing chamber) where it flows through the oil orifice into the outer tip chamber. The atomizing medium, most commonly steam, passes along the outer pipe to enter the core. Within the core and the outer tip, a thorough emulsification of the oil with the atomizing medium occurs. This mixed stream passes through at a high velocity to the cone tip, where it is ejected in a highly atomized state. The flame pattern generated is controlled by the exit port drilling pattern of the cone tip.

1.5.1.4 Hamworthy DS (Dual Stage) Oil Gun

The Hamworthy DS oil atomization system utilizes a fixed atomizing pressure; hence, it is an ideal choice when the atomizing medium pressure is limited (see Figure 1.53). The first stage of atomization is by fuel pressure. The oil entering the dual stage gun is forced through a mechanical atomizing assembly consisting of a fuel body, disk, and tip. The second stage of atomization utilizes the momentum of the atomizing medium, usually steam, but can be compressed air. The mechanically atomized oil is sprayed into the outer tip chamber where the atomizing medium exits, churning the oil into a complete emulsion and initiating vaporization. Finally, expansion of the two phase flow of atomized oil, oil vapor and atomizing medium through the cone tip completes the spray. The flame pattern generated is controlled by the exit port drilling pattern of the cone tip.

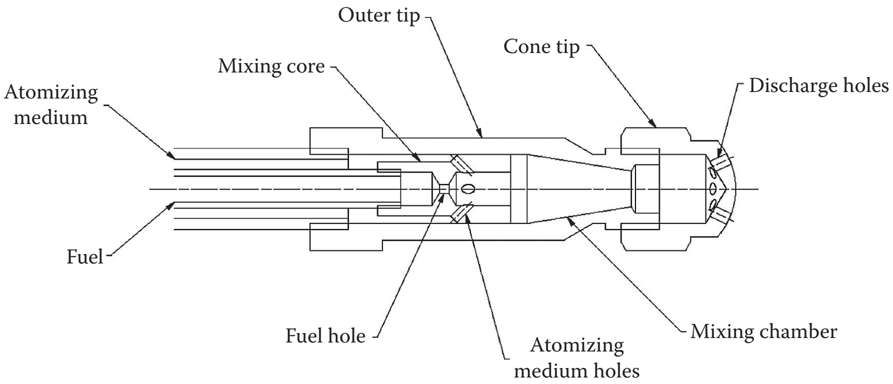


FIGURE 1.52
Hamworthy SAR oil atomizer and tip.

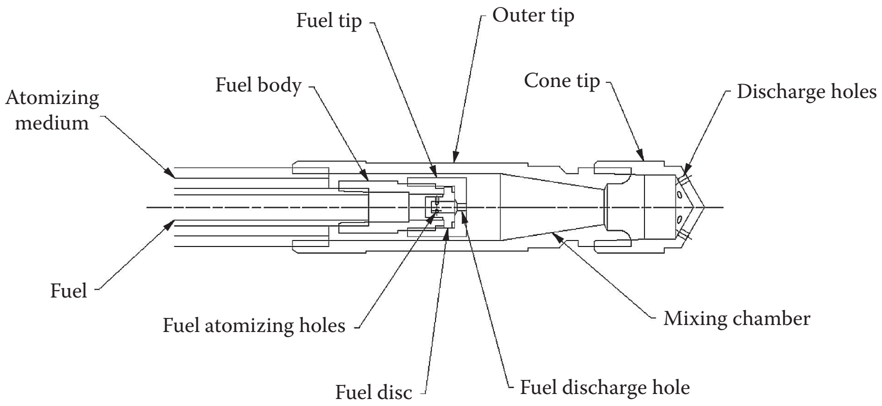


FIGURE 1.53
Hamworthy DS oil atomizer and tip.

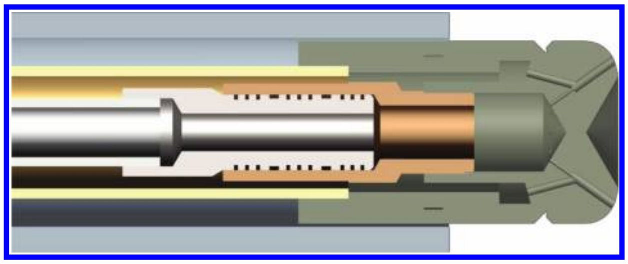


FIGURE 1.54
John Zink PM atomization system (port mix).

1.5.1.5 John Zink PM Oil Gun (Port Mix Atomization)

The John Zink PM oil atomization system is one form of the port mix, or Y-jet, design common throughout industry. This type of oil gun utilizes multiple "sets" of converging ports to break the oil down into small droplets (see Figure 1.54). Other similar atomizing system oil guns are supplied by COEN, TODD, and Hamworthy.

This design utilizes intersecting oil metering and atomizing medium (typically steam) ports. The angle of intersection is designed to develop shear forces on the oil. The point of intersection occurs at the entrance to an expanded area exit port. This impinging flow of oil and atomizing medium at an abrupt expansion results in a shearing of the viscous liquid, breaking it up into very fine droplets for dispersion. This atomization occurs almost immediately prior to exiting the tip. The exit port is limited in length to prevent re-combining of the droplets. The length of the exit port is only as long as required to provide the desired direction of the jet.

Since the oil metering is accomplished through multiple ports, this type of oil gun is typically used only for larger heat releases. John Zink will typically limit its use to heat releases greater than 5.9 MW (20 MM Btu/h).

The PMA, a dual-tube design of the PM oil gun, is also available.

1.5.1.6 HERO® Oil Gun

The HERO gun is an acronym of High Efficiency Residual Oil gun. It is a patented atomization device[9,10] with advantages of low steam consumption, good atomization, short flame length, low pollutant emissions including NOx and particulates. The HERO gun design is shown in Figure 1.55. It is a combination of a Y-jet[11] and the EA oil gun. The detailed description of HERO gun design and performance can be found in Volume 1, Chapter 10 on oil atomization and elsewhere.[12–14]

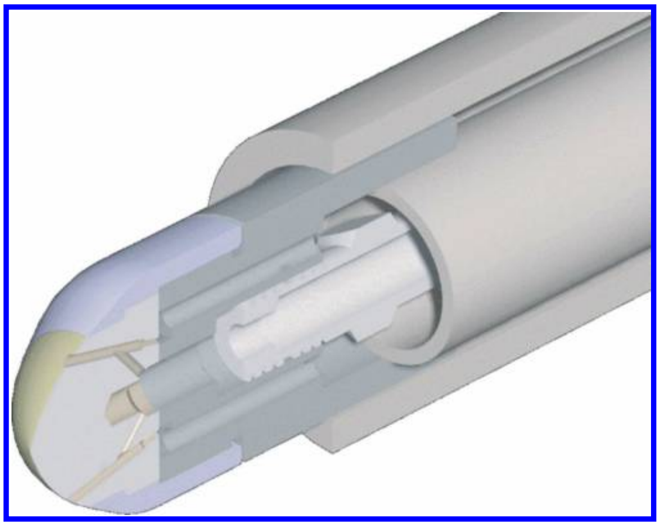


FIGURE 1.55
The HERO gun.

1.5.2 Combination Burner

A combination burner can be viewed as really two burners in one, or maybe even three in one. This is because, first, the combination burner must perform using oil exclusively as the fuel. Second, it must perform using gas exclusively as the fuel, and third using both gas and oil simultaneously as fuels. Good performance on one fuel does not imply any trend toward good performance or even adequate functioning on the other fuel or the combination. The design strategies that work well for either gas-only or oil-only burners actually end up conflicting in a combination burner.

1.5.2.1 Conventional Combination Burner

A typical combination burner contains primary oil tile (also called the regen tile) and secondary tile as indicated in Figure 1.56. The oil gun is located at the center of the regen tile and the gas tips are located between the regen tile and the secondary tile.

1.5.2.2 PLNC Burner

The PLNC is a first generation low-NOx combination burner (see Figure 1.57). Air or steam atomization of liquid fuels is difficult to stage into primary and secondary fuel zones. Therefore, to reduce NOx emissions, it is more practical to stage the combustion into multiple zones by using staged-air technology. The gas firing is the same as a conventional combination dual block burner. The oil gun is centrally located inside a primary tile, also called "regen tile." The gas firing is provided through multiple tips located around the oil tile, in the secondary tile zone. Staged air is provided through an

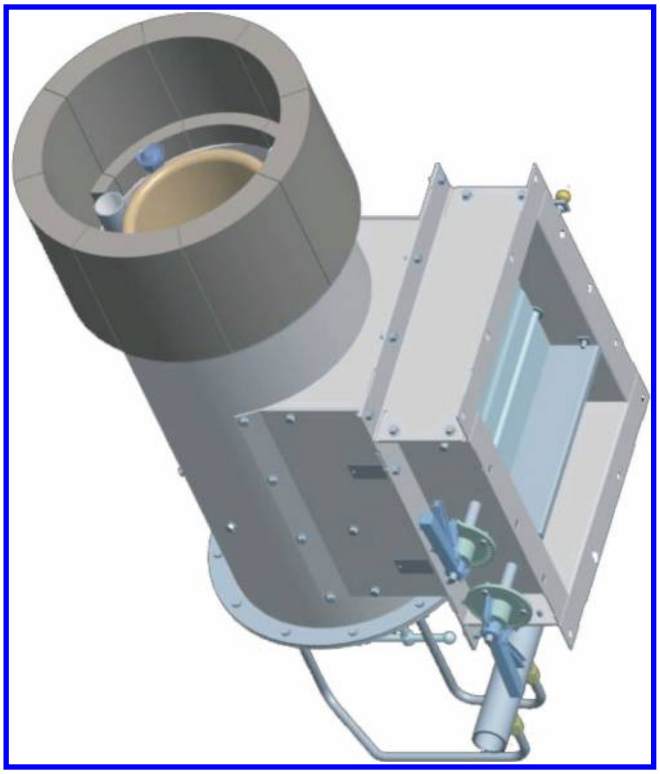


FIGURE 1.56
Standard combination burner.

open area around the burner tile. Different air dampers are used to control the various levels of air through each channel (see Section 1.2.2.1).

1.5.2.3 DEEPstar® Burner

The DEEPstar burner is a patented combination burner[15] (Figure 1.58) designed to achieve low NOx emissions for gas firing, oil firing, and combined gas/oil firing. The design uses staged fuel and furnace gas entrainment to reduce NOx emissions when firing gas.

Fuel gas firing is accomplished using two primary gas tips located adjacent to the ledge in the primary tile throat and four staged gas tips located in the refractory-shielded secondary tile sections positioned in the secondary air zone around the outer circumference of the primary tile. This refractory shielding of the staged tips creates the staged-fuel gas combustion. In addition, the staggered secondary air and shielded fuel gas injection allows a greater secondary air and furnace gas interface for mixing prior to the secondary combustion zone.

Oil firing is designed as staged air with furnace gas entrainment in the second combustion zone for reduction of NOx emissions.[16] Air for the first combustion zone is supplied through the primary tile throat. The oil gun is located at the center of the primary tile within the first air zone, the primary tile throat. Air for the second combustion zone is via the four slots formed by the secondary (staged) gas tip refractory shield tiles. The furnace gas entrainment occurs due to this staggered arrangement of the air slots and the gas tip shield blocks. This staggered secondary air arrangement allows the furnace gas to be entrained by the primary oil flame with minimal interference from the secondary air. It also allows greater air and furnace gas interface for mixing prior to entering into the secondary combustion zone.

Oil and gas flames from a DEEPstar burner are shown in Figure 1.59. The bright yellow color from the oil flame is due to soot particles and small droplets in the oil flame. These factors of liquid hydrocarbon combustion radiate in the visible light spectrum. The more transparent flame from the gas flame is due to a lower occurrence of solid particles raising the predominant radiation above the visible spectrum. It is often quite difficult to see a well-mixed gas flame in an operating furnace environment.

1.6 Pilot Burners

In the simplest terms, pilot burners are small, independently controlled burners used to ignite the larger burners that are used to provide heat to the process. These pilot burners are predominantly premixed, although there are raw gas and liquid hydrocarbon-fueled pilots. The scope of this section is limited to gas-fired premix pilots.

One of the earliest attempts at specifying pilots was presented by the National Fire Protection Association (NFPA). The NFPA 85c code specified pilots for use in power generation boilers. These pilots were initially centered on use in coal or heavy oil–fired large-capacity boiler burners. NFPA 85c was withdrawn in 1991. In 1995, the NFPA, in conjunction with the American National Standards Institute (ANSI), reissued a standard that again addressed pilots, ANSI-NFPA 8502. This standard was withdrawn in 1999. The most recently issued NFPA standards are NFPA 85 issued in 2004, latest edition 2011, and NFPA 87 latest edition 2011.

NFPA 85[7] is a standardized code covering boiler and combustion systems hazards. This code designates equipment used to ignite main burners as igniters. Specifically, this code breaks down boiler igniters into four classifications: (1) *class 1*, (2) *class 2*, (3) *class 3*, and (4) *class 3 special*. NFPA's definitions of these igniter classifications can be found in Section 3.3.85, "Igniter" of the 2011 Edition.

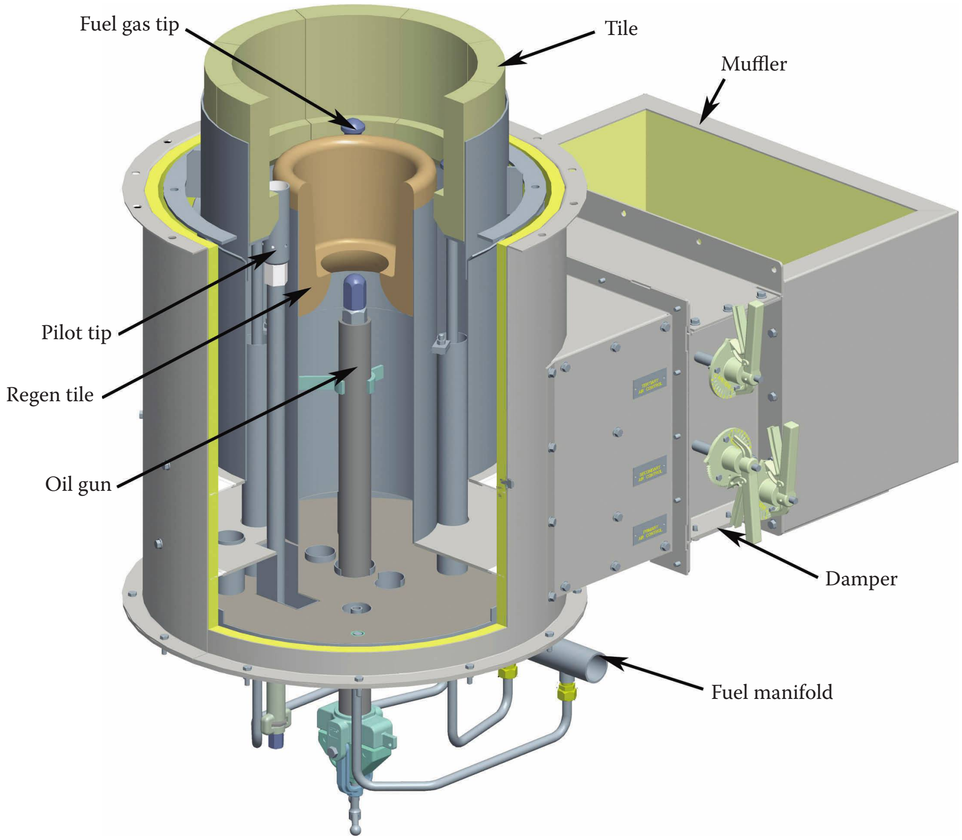


FIGURE 1.57
PLNC staged-air combination burner.

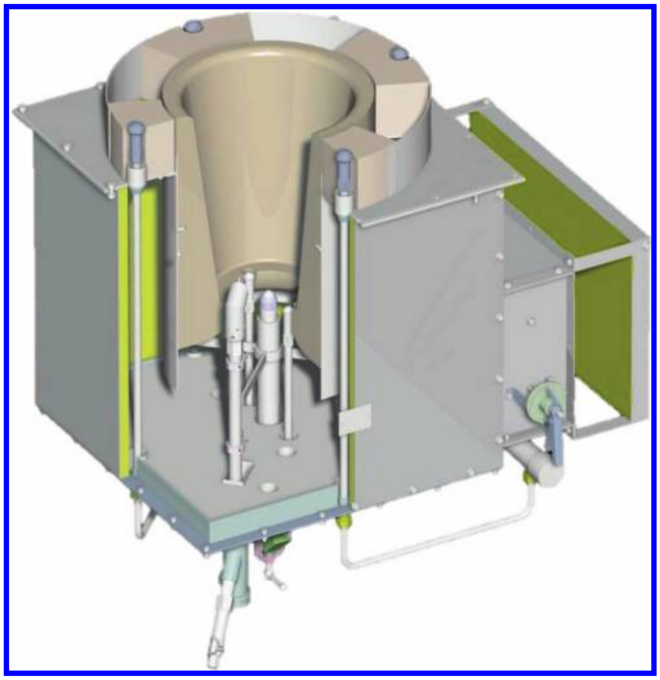


FIGURE 1.58
DEEPstar low-NOx gas/oil combination burner.

NFPA 87[8] is a standardized code for fluid heaters. In the administration chapter of this code, Section 1.1.3 states that this code does not apply to fired heaters in petroleum refineries and petrochemical facilities or reformer and cracking furnaces in the petrochemical and chemical industries.

The American Petroleum Institute (API) has two publications that address pilots. API Standard 560/ISO 13705 Fired Heaters for General Refinery Service[9] (August 2007) covers pilots in Section 14 Burners and Auxiliary Equipment, Sections 14.1.9 and 14.1.10. Here, the API simply references pilots as preferable to no pilot, unless otherwise specified by the owner/operator. Pilots for burners ranging in capacity up to 4.4 MW (15 × 10^6 Btu/h) are specified as having a minimum energy output of 22 kW (75,000 Btu/h), and larger burners output is to be specified by the owner/operator. Only continuous operation pilots are addressed in this standard. The standard requires the pilots to be able to ignite and re-ignite an individual main burner operating over the full range of its fuels. API 560 also allows that ignition or re-ignition may require regulation of the combustion air flow to achieve burner ignition.

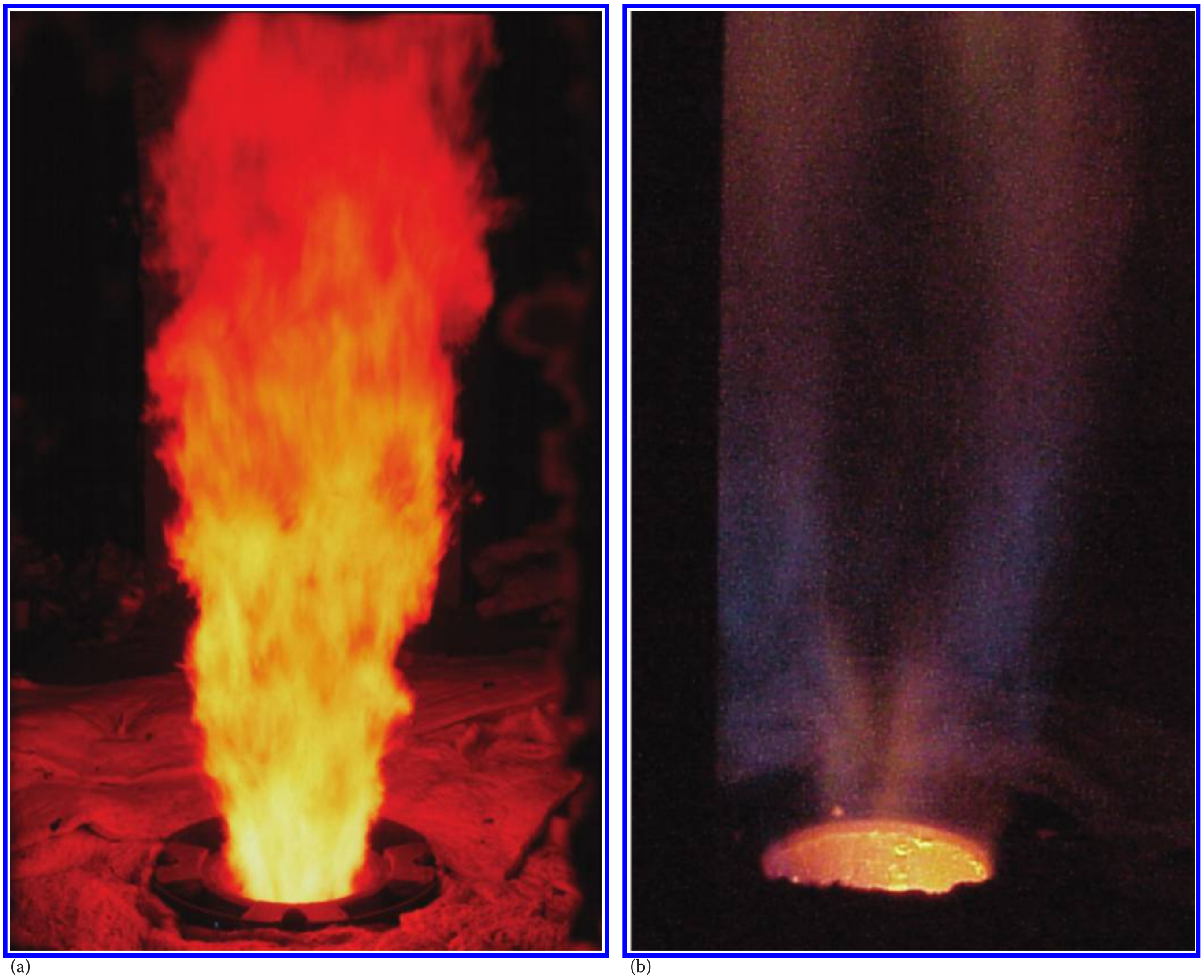

(a) (b)

FIGURE 1.59
(a) DEEPstar oil flame and (b) gas flame.

API Recommended Practice 535, Burners for Fired Heaters in General Refinery Services,[20] covers pilots in Section 7, Pilots and Ignitors. Here, pilots are defined as gas-fired burners used to ignite and re-ignite the main burner over its full operating range. This recommended practice specifies the minimum heat release of continuous operating pilots to be 19 kW (65,000 Btu/h) for burners up to 4.4 MW (15 × 10^6 Btu/h) capacity. It states that the pilots must be provided with a continuous supply of combustion air, whether the main burner is in service or not. By design, pilots should remain stable over the full range of the main burner's design operation and should not be affected by the loss of the main burner flame. API 535 also allows that pilot stability may be affected by out of design high draft or high furnace pressure.

Engineers concerning themselves with pilots and pilot operation and control should familiarize themselves with the full content of these referenced publications. These publications have the status of being industry-accepted norms.

1.6.1 Premix Pilots

1.6.1.1 John Zink ST-1-S Premix Gas Pilot

The most common pilot supplied by John Zink is presently the ST-1-S, which is shown in Figure 1.60. This pilot is a manually ignited, premix gas pilot, designed for high

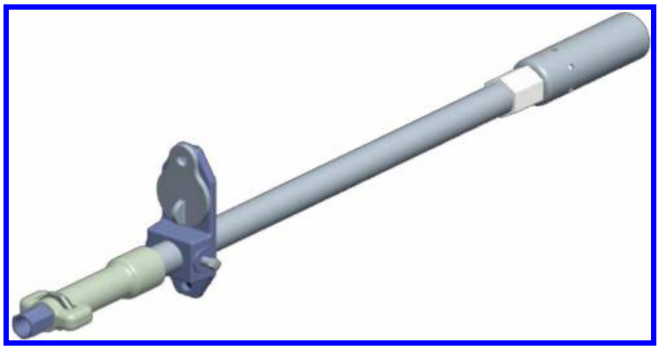


FIGURE 1.60
ST-1-S manual pilot.

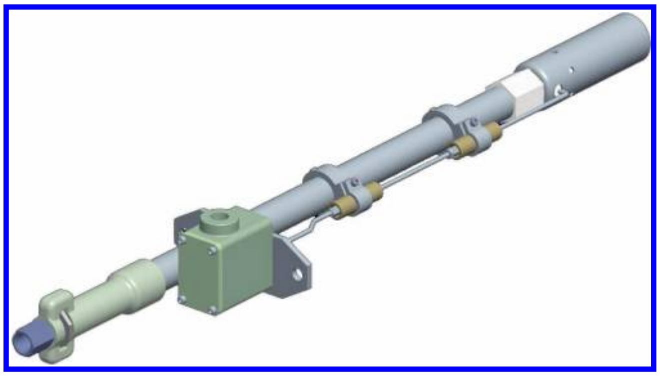


FIGURE 1.61
ST-1-SE electric ignition pilot.

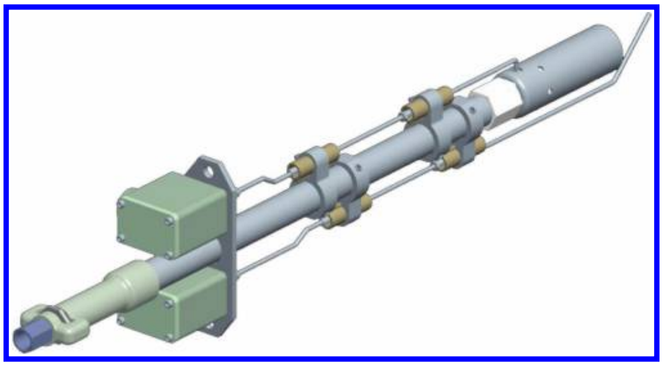


FIGURE 1.62
ST-1-SE-FR electric ignition pilot with an integral flame rod.

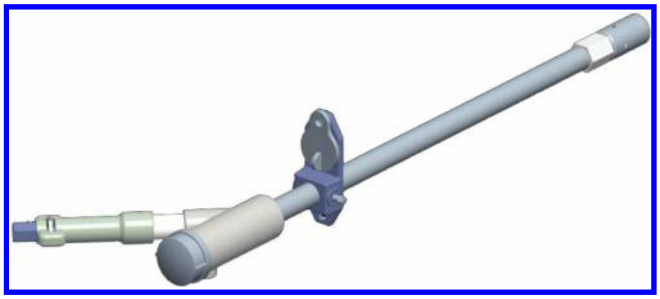


FIGURE 1.63
KE-1-ST electric ignition pilot.

efficiency on natural gas fuels. The pilot mixer is generally located external to the burner housing, making the combustion air for the pilot ambient air. The gas/air mixture tube is 1 in. (25 mm) sch. 40 pipe allowing a high percentage of stoichiometric air. The tip is a multiport premix tip utilizing impinging ports for initial flame stabilization. The tip also has a high-alloy cast shield providing additional stabilization and protection from variations in directional external flows (wind or draft effects).

The standard ST-1-S pilot utilizes a 1.59 mm (0.0625 in.) diameter fuel orifice. It has a normal operating pressure range of 0.5–1.1 barg (7–16 psig). On most methane-rich natural gas fuels, its capacity would be 21.4–31.6 kW (73,000–108,000 Btu/h). The ST-1-S pilot has a proven operational range from 0.2 to 1.75 barg (3 to 25 psig). This pilot's standard configuration will handle fuels ranging from LPG (liquefied petroleum gas, which usually consists of vaporized butane and propane) to 30% hydrogen with a single-tip design. There are two (2) additional reduced area tips, designed for progressively higher hydrogen content of up to 75 mol%. As each of the smaller area tips, suitable for higher hydrogen, is used, the pilot becomes less suitable for heavier pilot fuels.

The ST-1-S pilot is available with high-tension (transformer powered) electric ignition, the ST-1-SE (see Figure 1.61). It is available with frame rectification rods for flame proof, the ST-1-S-FR. It is available configured for both electric ignition and flame proof via rectification, ST-1-SE-FR (see Figure 1.62).

1.6.1.2 John Zink KE-1-ST Premix Gas Pilot

The KE-1-ST pilot, shown in Figure 1.63, is a variation on the ST-1-S pilot with an internal electric ignition rod supplied standard. This pilot is a high-tension (transformer powered) electrically ignited, premix gas pilot, designed for high efficiency on natural gas fuels. The pilot mixer is generally located external to the burner housing, making the combustion air for the pilot ambient. The electric ignition rod is housed within the gas/air mixture tube, giving a self-contained package. The gas/air mixture tube is 1 in. (25 mm) sch. 40 pipe allowing a high percentage of stoichiometric air. However, owing to the obstruction created by the electrode and insulators, the mixer efficiency is less than that of the ST-1-S. The tip is a multiport premix tip utilizing impinging ports for initial flame stabilization. The tip also has a high-alloy cast shield providing additional stabilization and protection from variations in directional external flows (wind or draft effects).

The standard KE-1-ST pilot utilizes a 1.59 mm (0.0625 in.) diameter fuel orifice. It has a normal operating pressure range of 0.5–1.1 barg (7–16 psig). On most methane-rich natural gas fuels, its capacity would be 21.4–31.6 kW (73,000–108,000 Btu/h). This pilot is capable of handling fuels ranging from LPG to 30% hydrogen with a single-tip design. Higher levels of hydrogen content can be accommodated, but the range of operating pressure and capacity are restricted. Owing to the required configuration for internal electric ignition sparking, there are no variations in tip areas available for higher-hydrogen-content fuels.

1.6.1.3 John Zink ST-2 Premix Gas Pilot

The John Zink ST-2 pilot is a manually ignited, premix gas pilot, designed for high efficiency on a broad range of refinery or chemical plant fuels (see Figure 1.64). The pilot mixer has been redesigned following ASME design criteria. The venturi is generally located external to the burner housing, making the combustion air for the pilot ambient. The gas/air mixture tube is 1½ in. (38 mm) sch. 40 pipe (2 in. or 51 mm nominal outside diameter) allowing even higher percentages of stoichiometric air. The tip is a multiport premix tip utilizing impinging ports for initial flame stabilization. The tip also has a high-alloy cast shield providing additional stabilization and protection from variations in directional external flows (wind or draft effects).

The standard ST-2 pilot can be designed to utilize a 1.59 mm (0.0625 in.) diameter or a 1.99 mm (0.0785 in./#47 MTD) diameter fuel orifice. It has a normal operating pressure range of 0.35–1.38 barg (5–20 psig). On most methane-rich natural gas fuels, the pilot's capacity using the smaller standard orifice would be 18.25–35.75 kW (62,300–122,000 Btu/h). The pilot's capacity using the larger standard orifice would be 28.8–56 kW (98,250–191,000 Btu/h). Test stand operation of the ST-2 pilot has shown an operational range from 0.2 to 1.75 barg (3 to 35 psig). This pilot has shown the ability to handle fuels ranging from LPG (vaporized butane and propane) to 85% hydrogen with a single-tip design. The higher levels of hydrogen content are accommodated by operating at greater than 0.7 barg (10 psig).

The ST-2 pilot is available with high-tension (transformer powered) electric ignition or high-energy (capacitance discharge) electric ignition, the ST-2-E or ST-2-EI. It is available with frame rectification rods for flame proof, the ST-2-FR. It is also available supplied with a modified flame shield for flame proof via optical or digital scanner. It is available configured for both electric ignition and flame proof via rectification, ST-2-E-FR/ST-2-EI-FR.

1.6.1.4 John Zink KE-2-ST Electric Ignition Premix Gas Pilot

The standard KE-2-ST pilot (Figure 1.65) utilizes a 1.59 mm (0.0625 in.) diameter fuel orifice. It has a normal operating pressure range of 0.5–1.1 barg (7–16 psig). On most methane-rich natural gas fuels, its capacity is 21.4–31.6 kW (73,000–108,000 Btu/h). The KE-2-ST pilot has a proven operational range from 0.2 to 1.38 barg (3 to 20 psig). This pilot is capable of handling fuels ranging from LPG to 30% hydrogen with a single-tip design. Higher levels of hydrogen content can be accommodated, but the range of operating pressure and capacity are restricted. Owing to the required configuration for internal electric ignition sparking, there are no variations in tip areas available for higher hydrogen content at normal operating pressure ranges.

The ignition system utilizes a fixed spark gap igniter. Those units supplied with flame rods utilize a fixed surface area ionization rod. Both elements, the igniter and flame rod, are encapsulated inside a stainless steel tube to help eliminate problems with moisture and dirt, which can result in shorting out the electronics.

The KE-2-ST pilot is available with high-tension (transformer powered) electric ignition KE-2-HT (high tension) or high-energy (capacitance discharge) electronic ignition KE-2-HE (high energy). It is available with flame rectification rods for flame proof, KE-2-HT-FR and KE-2-HE-FR. This pilot can also be supplied with a modified flame shield for flame proof via optical or digital scanner. This modified flame shield is angled, restricting the

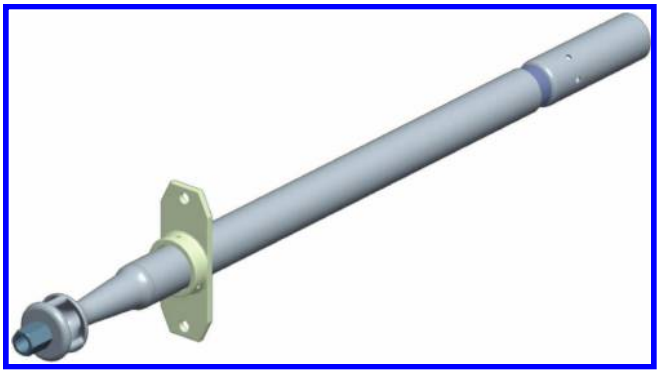


FIGURE 1.64
ST-2 manual pilot.

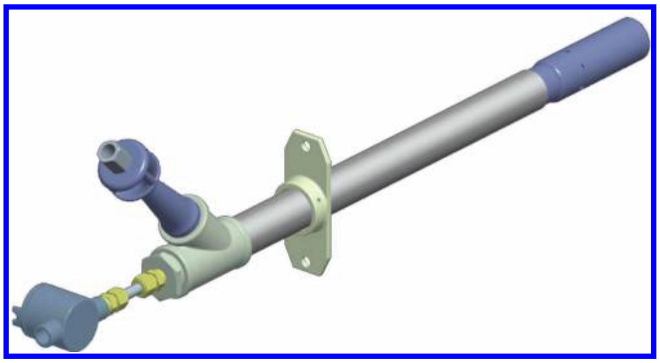


FIGURE 1.65
KE-2-ST electric ignition pilot.

scanner's view to the pilot flame only. Scanners are also available, KE-2-HT-SC and KE-2-HE-SC. The KE-2-ST series pilot can be engineered to fit most existing pilot mounting systems.

1.6.2 Flame Detection

Current technology for pilot flame detection includes flame rods and scanners. Within recent years, the most common method of pilot flame proof is through the use of a flame ionization rod and flame rectification circuits. In simple terms, placing a conducting element (the flame rod) within a flame zone, the ionized particles within the flame provide a conducting pathway between the isolated and insulated ground flame rod assembly and the grounded pilot and main burner assemblies. This design concept insists on three points: the flame rod must be immersed in the pilot flame, the flame rod assembly must be isolated and insulated from any grounding or contact with the pilot and main burner assemblies, and the pilot body must be adequately grounded.

The flame rectification circuit provides a low voltage source to the flame rod. With the presence of flame, the ionized zone within the flame allows current to flow. Sensing this current flow is proof of flame. That is, if the flame rod is within the flame zone and the flame is in contact with a sufficient portion of the grounded pilot tip, detection of the current flow becomes proof of flame. Therefore, when there is no flame, there is no ionization zone. If there is no ionization zone, then there is no current flow. If there is no current flow, then there is no flame. With a constant source of voltage supplied to the flame rod, the establishing of flame will allow the rectification circuit to register current flow almost instantly.

A less intricate, and less sure, method of flame rod use is to directly measure the potential (voltage) generated through the ionized zone and the two dissimilar materials at different temperatures, the flame rod and the pilot tip. This method requires no outside source of voltage but is more dependent on fuel composition, stoichiometry, flame positioning, and flame intensity for a signal to prove flame presence.

Optical scanning technology can include UV (ultraviolet) and/or IR (infrared) frequency ranges. IR detection is highly efficient for radiant frequencies typical of solid bodies, carbon dioxide, and water vapor. While a majority of non-diatomic combustion products are carbon dioxide and water vapor, the signal received by an IR detector can be of other hot radiating surfaces, such as surrounding refractory or metal objects. These other hot surfaces do not necessarily indicate flame in the location being monitored. Certain IR scanning devices compensate for this possible "false" signal by looking for the frequency shift, or "flicker," inherent in combustion. Hot, radiating solid bodies do not flicker. They are more constant in radiating frequency owing to their constant composition and relatively constant temperature.

UV detection is based on the radiation frequency achieved by the high temperatures produced within combustion. Detection of an UV signal assures the presence of combustion. The shortcoming of this detection method is that many products of combustion are UV absorptive or opaque. High levels of carbon dioxide, carbon monoxide, and water vapor between the emitting flame zone and the optic sensor will mask or shield the strength of the signal. This can result in detection levels lower than desired or required.

There are scanners available that monitor both UV and IR and also monitor the "flicker" function.

Most scanning flame detectors have a self-checking ability to prove that the sensor has not failed in the proved condition. This self-checking ability is not required, but, many safety protocols require either self-checking or multiple (redundant) scanners for flame proof assurance. Older, analog signal scanners used a mechanical shutter for implementation of the self-checking function. These mechanical shutters were subject to sticking or failure if the scanners were not mounted in the correct orientation or if the moving parts became dirty. The newer digital scanners achieve the same self-checking effect through the use of programmed signal interruption.

1.6.3 Ignition

Ignition of pilot burners and process main burners can be accomplished either locally or remotely. If the ignition is accomplished locally, either a torch (flame ignition) or electronic methods can be used. If ignition is accomplished from remote signal, electronic methods are used. For the purposes of this section, electronic methods are addressed, either locally activated or remotely activated.

Manual electric ignition devices presently available are a battery-powered high-energy (capacitor discharge) unit, a battery-powered motor-charged high-energy unit, and an external power supply high-tension (transformer discharge) unit.

Pilot-mounted ignition systems are generally provided with their primary energy source externally. This source can come from normal AC power sources or from control system DC supply. Both high-energy and high-tension systems are available.

High-energy ignition systems are characterized by pulsed sparks of very high current. A capacitor is charged and abruptly discharged at periodic intervals. The igniter is typically a coaxial construction; some constructions use semiconductor surfaces that can spark when completely submerged in water. The voltage of the energy storage capacitor is usually less than 3000 V; the spark voltage of a normal igniter tip is often less than 100 V. When the spark occurs at the igniter tip, it creates a short circuit across the charged capacitor, which generates high currents. High-energy spark currents can exceed 2500 A peak. Power levels during the spark can exceed 100 kW. This high-intensity spark will ignite natural gas and light and heavy oils. The spark is "self-cleaning" because the spark clears carbon buildup or particulate material located in the discharge path.[21]

High-tension ignition systems are characterized by continuous arcing of very high voltage with low amperage. A transformer is supplied with the primary voltage from an external source and that "stepped up" to the final voltage. The primary voltage is typically 110–240 VAC with the secondary voltage (output) will range from 7,000 to 15,000 VAC. When the secondary is opened to the igniter, the arc is generated across the igniter gap, located within the gas stream. The arc will be continuous as long as the connection is maintained. The energy level of the arc is not, generally, sufficient to ignite light or heavy oil.

References

1. Waibel, R. T., Athens, L., and Claxton, M., Effect of fuel composition on emissions from ultra low NOx burners, *American Flame Research Committee 1995 International Symposium*, Monterey, CA, October 15–18, 1995.
2. Chung, I. P., Chambers, J. S., Schnepper, C. A., Poe, R. L., Jayakaran, J. D., Waible, R. T., Claxton, M. G., Zimila, M., and Foote, D. R., Compact low NO_x gas burner apparatus and methods, U.S. Patent 6,695,609 B1, issued February 24, 2004.
3. Poe, R. L. and Claxton, M. G., Coanda gas burner apparatus and methods, U.S. Patent 7,878,798, issued February 1, 2011.
4. Claxton, M. G., Poe, R. L., and Wilkins, J., Coanda gas burner apparatus and methods, U.S. Patent 8,337,197, issued December 25, 2012.
5. Poe, R. L., Low NOx radiant wall burner, U.S. Patent 6,607,376, issued August 19, 2003.
6. Bussman, W. R., Chambers, J. S., Hayes, R. P., Poe, R. L., and Venizelos, D. T., High capacity/low NOx radiant wall burner, U.S. Patent 6,796,790, issued September 28, 2004.
7. Roe, P. L., Low NOx radiant wall burner, U.S. Patent 6,905,328, issued June 14, 2005.
8. Bussman, W. R., Baukal, C. E., Jr., Ruiz, R., Chung, I-P., Chellappan, S. G., and Waibel, R. T., Remote staged radiant wall furnace burner configurations and methods, U.S. Patent 7,025,590, issued April 11, 2006.
9. Chung, I. P. and Strupp, C., High efficiency fuel oil atomizer, U.S. Patent No. 6,478,239 B2, issued November 2002.
10. Chung, I. P. and Strupp, C., High efficiency method for atomizing a liquid fuel, U.S. Patent No. 6,691,928 B2, issued February 17, 2004.
11. Mullinger, P. J. and Chigier, N. A., The design and performance of internal mixing multijet twin fluid atomizers, *J. Inst. Fuel*, 47, 1947, 251–261.
12. Chung, I. P., Karan, J., and Strupp, C., High efficiency residual oil (HERO) atomizer for process heater burners, presented at *2002 American Flame Research Committee*, Spring Meeting, Ottawa, Canada.
13. Chung, I. P. and Colannino, J., High efficiency residual oil gun for liquid waste incineration, presented at *2007 Eastern Combustion Institute Fall Technical Meeting*, Charlottesville, VA, 2007.
14. Chung, I. P., Strupp, C., and Karan, J., New fuel oil atomizer for improved combustion performance and reduced emissions, Presented at *Sixth European Conference on Industrial Furnaces and Boilers*, Lisbon, Portugal, 2002.
15. Chung, I. P., Colannino, J., and Strupp, C., Low NOx apparatus and methods for burning liquid and gaseous fuels, U.S. Patent 6,422,858 B1, issued July 23, 2002.
16. Chung, I. P., Karan, J., and Strupp, C., New low NOx oil-gas-combination burner for process heater applications, Presented at *2002 American Flame Research Committee*, Spring Meeting, Ottawa, Ontario, Canada.
17. NFPA 85: Boiler and combustion systems hazard code, National Fire Protection Association, Quincy, MA, 2007.
18. NFPA 87: Recommended practice for fluid heaters, National Fire Protection Association, Quincy, MA, 2011.
19. ANSI/API Standard 560 (ISO 13705:2006): *Fired Heaters for General Refinery Service*, 4th edn., American Petroleum Institute, Washington, DC, August 2007.
20. API Recommended Practice 535: *Burners for Fired Heaters in General Refinery Service*, 2nd edn., American Petroleum Institute, Washington, DC, January 2006.
21. Chentronics Corporation, *The Motivator II,* Chentronics Motivator II Exciter—System Operation and Maintenance Manual MNL-07070707.

2

Oil Burners

I.-Ping Chung, Steve Londerville, Michael G. Claxton, and William Johnson

CONTENTS

2.1 Introduction

The efficient combustion of a liquid fuel, in bulk, is not possible. Intimate mixing of the fuel and air, or combustion oxygen source, is required. To achieve this intimate mixing requires atomizing the liquid into many small droplets to increase the surface area for vaporization and mixing with air. This is discussed in Volume 1, Chapter 10. In that chapter, the basic principle for oil gun design and its function is discussed. Here, the discussion is extended to practical applications and different types of oil guns and oil burners.

As covered in Volume 2, Chapter 6, an efficient combustion system must perform the following functions: (1) metering the fuel flow, (2) distributing the fuel into the air, (3) mixing the fuel and air, (4) metering the airflow, (5) maintaining stable combustion, (6) molding the flame, and more recently as environmental concerns have increased, (7) minimizing the pollutant emissions. For an oil burner, the first three functions are handled by the oil gun. An oil gun is a device to meter, atomize, and deliver the liquid fuel and assist with the mixing of the fuel with the air. The other three functions, metering air, stabilizing the flame, and molding the flame shape, are primarily the functions of burner design components. The last function, minimizing the pollutant emissions, is a combination of oil gun and burner designs.

Every burner design is comprised of multiple sizes. These “sizes” are based on different tile throat air flow capabilities. Selecting the proper burner size for correctly metering the air flow to suit the available air pressure loss, firing rate, heater radiant dimensions, and application is important. A burner that is too large makes the flame difficult to shape due to low air velocity, while a burner that is too small would restrict the airflow resulting in a shortage of O_2 for complete combustion. The function of metering the air in the oil burner is similar to that of the gas burner, which is discussed in detail in Volume 2, Chapter 6.

There are many methods used to stabilize oil flames. One method is to generate an aerodynamic bluff body by swirling the air. The atomized oil is then injected into the aerodynamic bluff body zone. This is typically utilized by boiler burners. Swirling air requires a fan or a blower to force airflow through the swirled blades. It takes a greater air pressure drop than typically available in natural draft applications. Therefore, a majority of the applications that use this method are forced-draft systems. The swirled flow enhances the air and fuel mixing resulting in the most common flame pattern from this stabilization method—a short and intense oil flame.

Another method is to use a mechanical bluff body stabilizer or refractory tile to generate a low-pressure zone for the stabilization of the flame. The benefit of using refractory tiles is the reradiation of heat back to the atomized oil to assist with vaporization. The use of refractory tiles is most common in process burner applications. Compared to boiler applications and swirl-stabilized burner systems, process burner air pressure drop is relatively small and the flame pattern achieved is longer and less intense.

Oil burners are generally not restricted to only liquid oil-firing applications. These burners usually have gas-firing capability, too. Therefore, burners with liquid-firing capability are, most commonly, gas and/or oil combination burners. This design feature allows firing either a liquid fuel, a gaseous fuel, or a combination of both liquid and gas. It is common for each of the fuel systems to be designed to deliver the total heat required. During combination firing, the total firing rate should not be allowed to be larger than the design of the burner. For example, if a burner is designed for 10 MMBtu/h (3 MW) for oil firing and 10 MMBtu/h (3 MW) for gas firing, the burner at combination firing should not be over 10 MMBtu/h (3 MW). The reason is that even though the fuel can be delivered up to 20 MMBtu/h (6 MW), the air is only designed for burning 10 MMBtu/h (3 MW) of fuel. In other words, there is not enough air to burn all the fuels.

2.2 Process Oil Burner

Process burners are primarily used in the hydrocarbon, petrochemical, and chemical industries. These burners usually have smaller fired duties than those found in power generation or steam generation boiler burners. Typically, process burners can be either natural draft or forced draft. The most common process oil burner will be designed with a single oil gun located in a central oil tile and burner throat tile. The oil gun is usually designed as an independent assembly for ease of maintenance. The flame stabilization mechanism is positioned around the oil gun.

2.2.1 Oil Gun Type

There are many different types of oil guns. Each of these types has features that make them more or less desirable for different applications. Typical oil guns are illustrated in Figure 2.1 and described in detail in Chapter 1, Section 1.5.

Process burner oil guns are mainly designed for highly viscous, residual fuel oils. These fuel oils are commonly called No. 6 and can widely range in heating value, specific gravity (SG), and viscosity. While these oil guns are most commonly designed for heavy oil applications,

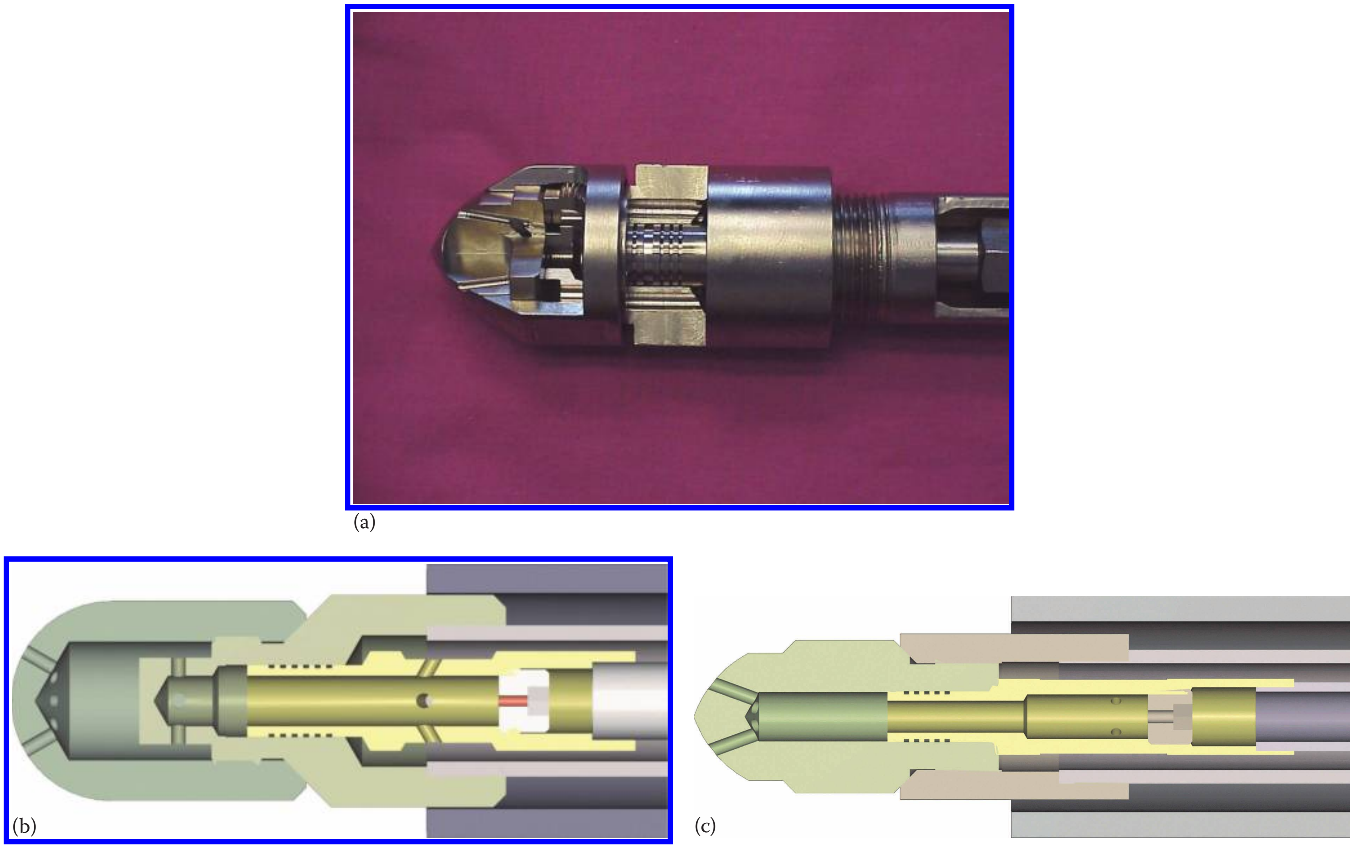


FIGURE 2.1
Typical liquid fuel atomizer-spray tip configurations: (a) HERO gun, (b) MEA oil gun, and (c) EA oil gun.

they can also be used for light oil applications. The oil gun capacity will vary between oils, especially when the oil characteristics change.

The heat content, lower heating value (LHV), or higher heating value (HHV) on a volume basis varies directly with the hydrogen to carbon ratio. Low-viscosity fluids require less pressure to flow the same volume. Heavy oils (more viscous oils) will require a temperature increase for viscosity reduction. Light oils can be operated at atmospheric temperature.

If steam is used as the atomizing medium, heat transfer between oil and steam will affect the oil gun capacity. Typically, for any specific oil gun, the capacity for light oil is higher than that for heavy oil. Figure 2.2 shows that at the same oil pressure, the oil gun delivers more heat release for light oil than for heavy oil.

As covered in Volume 1, Chapter 10, a twin-fluid oil gun can also use compressed air as the atomizing medium. Different atomizing media will result in different capacity curves. For a specific oil gun, its capacity using steam atomization is usually larger than that using air atomization as indicated in Figure 2.3. The figure illustrates that for the same fuel pressure, the oil gun delivers more fuel flow using steam atomization than that using air atomization. The reason is that air density is much smaller than steam density. In other words, air specific volume is much larger than steam specific volume. This results in air occupying more space than steam and delivering less liquid fuel.

Viscosity can be defined as resistance to separating liquid into droplets. Oil viscosity is a function of temperature. Generally, raising the oil temperature acts to lower its viscosity. When using compressed air to atomize heavy oil, heat transfer from the high-temperature oil to the low-temperature air can reduce the oil temperature. This results in increasing the oil viscosity, reducing the level of atomization of the oil, and degrading the flame quality. The preferred medium for heavy oil atomization is usually steam. For light oil, air atomization is less problematic since the viscosity of light oil at atmospheric temperature is low.

Selection of the oil gun type and the mechanical design of the atomizing system must take into account the type of atomizing medium. Using steam to atomize light oil or a highly volatile fuel may not be advisable. High-temperature steam can vaporize the volatile components in light fuel oils and cause vapor lock resulting in a pulsating oil flame.

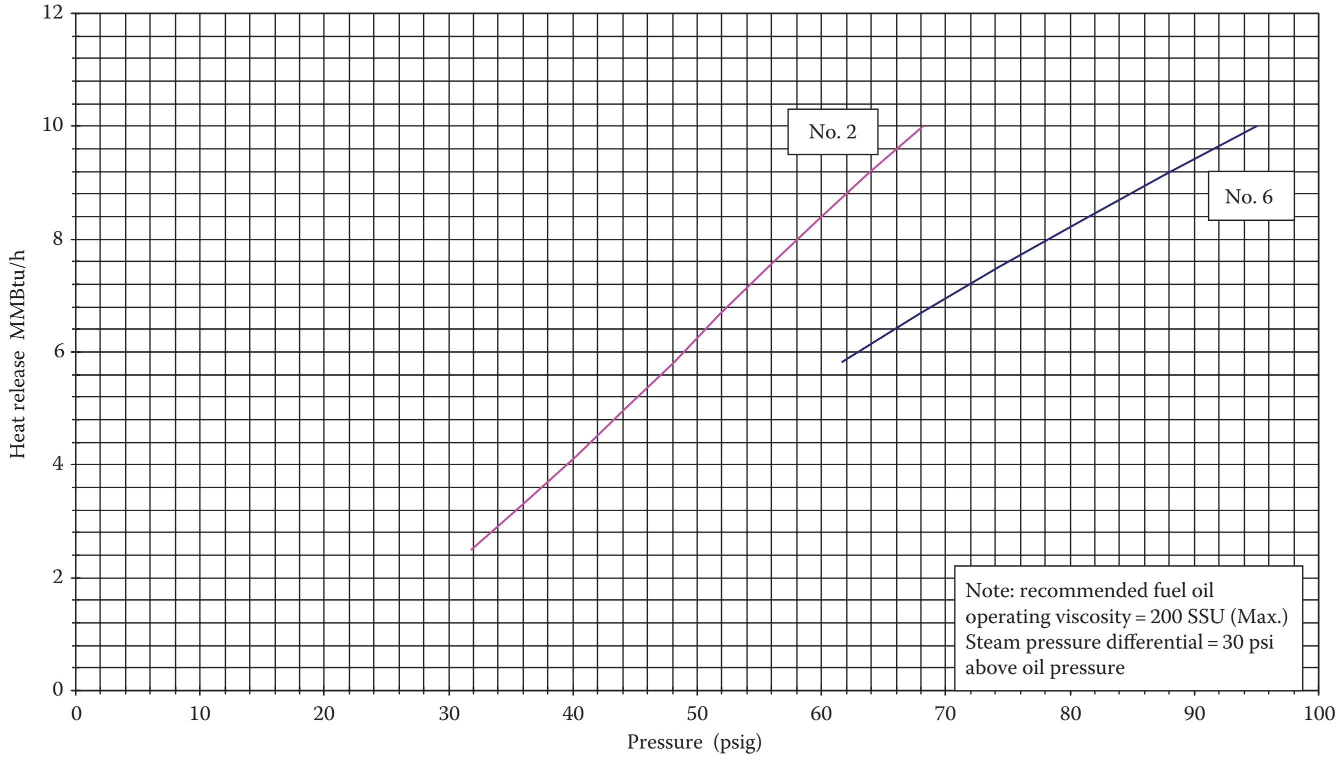


FIGURE 2.2
Oil gun capacity curves for heavy oil and light oil for one specific oil gun.

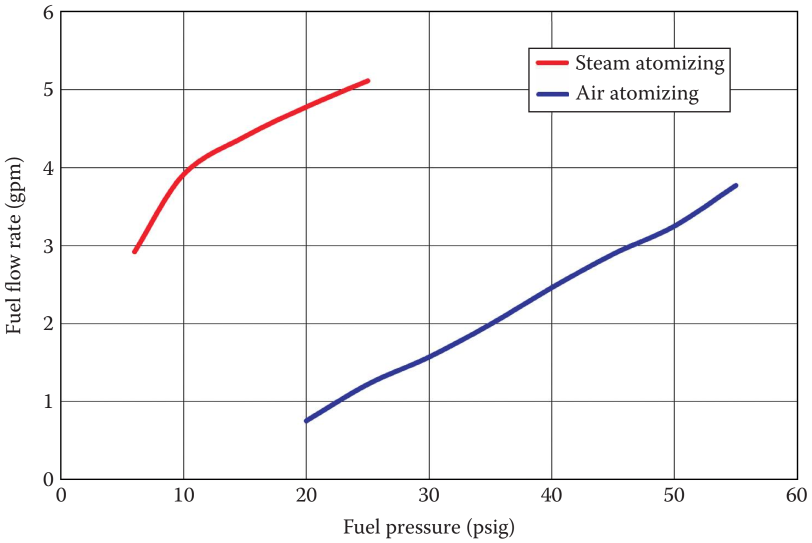


FIGURE 2.3
Oil gun capacity curves for steam atomizing and air atomizing for one specific oil gun.

2.2.2 Oil Flame Stabilization

Stabilization of the oil flame is a primary consideration when designing an oil burner. In low airside pressure drop burners, such as process heater burners, a mechanical stabilizer is used (see Figure 2.4). The function of a mechanical stabilizer is as a bluff body within the combustion airflow, creating recirculation zones. These recirculation zones bring combusting fuel/air mixtures and hot combustion products back to the exit ports of the oil gun dispersion tip. These hot gases aid in vaporizing the oil droplets and the combusting fuel/air mixture ignites the new vapors. This continually renewing

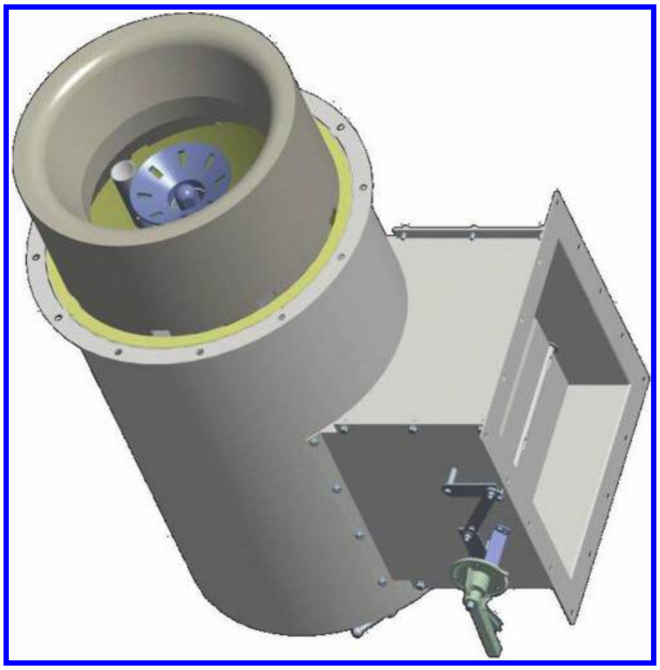


FIGURE 2.4
Oil flame cone-type stabilizer.

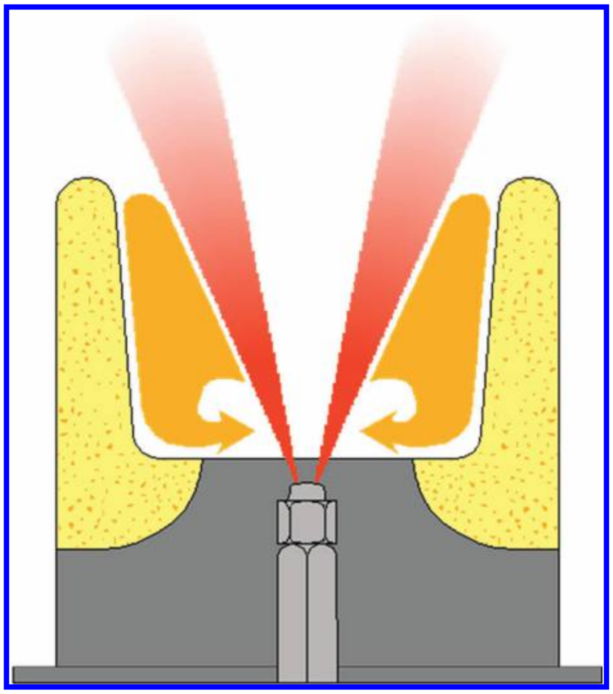


FIGURE 2.5
Oil tile to stabilize the oil flame.

circulation keeps combustion continuous and results in a stable oil flame. With a mechanical stabilizer, the burner tile throat is for air metering and flame shape molding. The tile throat opening size depends on the heat release and available furnace draft or burner pressure loss.

Another form of mechanical stabilizer is the "dual block" form, using an oil tile. In this form, a second burner tile is designed with a central ignition air source and a ledge. This design functions similarly to the mechanical bluff body discussed earlier. The low-pressure zone created by the ledge provides for the recirculation of the combusting fuel/air mixture. This design, using refractory in the bluff body, is a design enhancement for very heavy oils. The tile refractory provides high heat transfer back to the atomized oil (see Figure 2.5). Again, the burner tile throat is for air metering and flame shape molding, and the tile throat opening size depends on the heat release and available burner pressure loss.

2.2.3 Flame Shape

Due to the multitude of process heater designs, the flame shape required from process burners can vary greatly. In general, flame shapes are categorized as round or flat, but this is truly an oversimplification. Round flames can vary from long and narrow, pencil-like, to short and wide, basket-like. Flat flames can vary from wide to narrow fans. In addition, there are considerations for both flames when the burner is mounted either vertically (up or down fired) or horizontally. Horizontal mounting of the burner takes on another dimension when the flame pattern is flat. In that case, it is necessary to know if the "fan" of the flat flame is in the horizontal plane or the vertical.

In low airside pressure drop burners, like process burners, the oil flame shape is directly affected by the design of the final dispersion tip. For round flames, the number of exit ports, the spacing between ports, and the spray angle are as important to flame shape as having a round tile (see Figure 2.6). For flat flames, the best tile arrangement is rectangular (see Figure 2.7). In addition, the exit ports of the oil gun should be arranged in a linear pattern with little or no spread in the "narrow" direction and with a spread in the "fan" direction that represents the required flame shape. In the case of flat flame drillings, the number, size, and proximity of the ports are also of importance.

2.2.4 Emission Considerations

Since the 1970s, environmental concerns have been added to the functions that need to be addressed by combustion systems (see Volume 1, Chapter 14). In addition to the aforementioned functions of flame stabilization, air metering, and flame molding, burners must also meet emissions regulations. This includes oil-fired burners. This has resulted in a change in burner designs from high-efficiency fuel/air mixers to staged

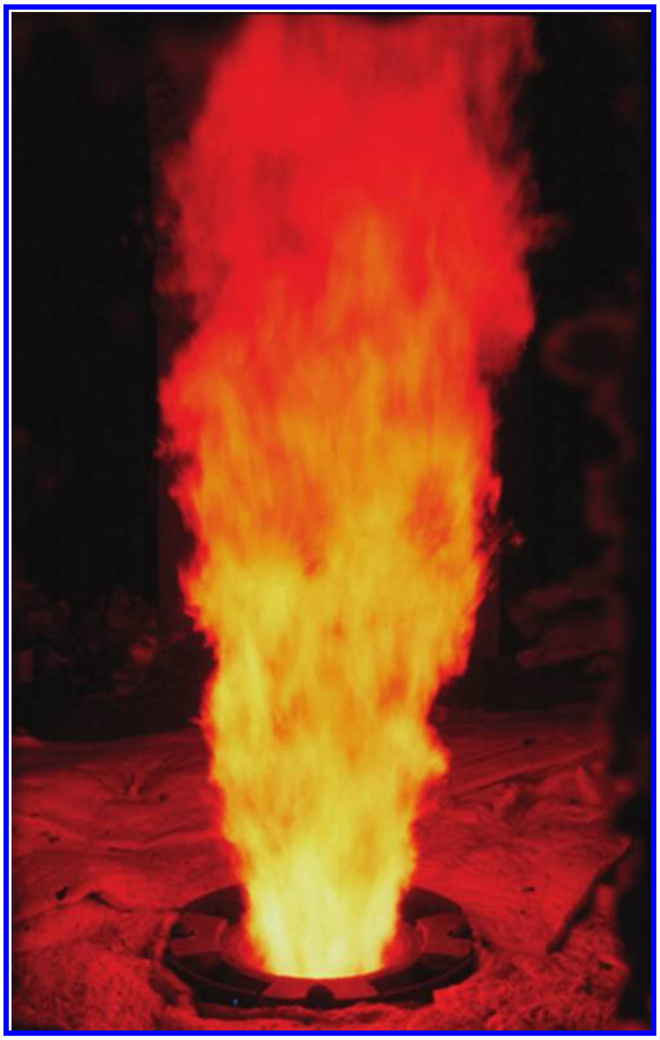


FIGURE 2.6
Round burner tile provides round oil flame.

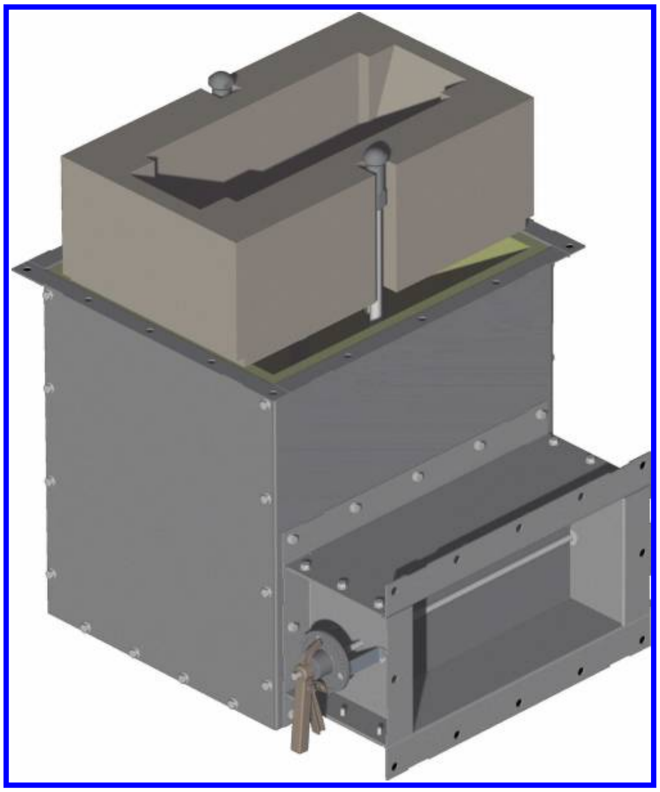


FIGURE 2.7
Rectangular burner tile provides flat oil flame.

combustion mixers. As discussed in Chapter 1, for gas fuel burners, both staged-air and staged-fuel combustion systems have been developed. For combination burners, the design selection becomes more challenging.

The pollutants presently being regulated most closely for process burners are NOx, CO, and particulates. The first two, NOx and CO, are common to both oil firing and gas firing and can be reduced through efficient staged combustion burner designs. For oil or combination firing, particulates can be a challenge. Because emissions have to be met for both gas firing and oil firing, not only NOx and CO, but also particulate emissions control must be designed.

The NOx source contains thermal NOx, prompt NOx, and fuel NOx (see Volume 1, Chapter 15). Thermal and prompt NOx are generated from high-temperature combustion reactions, while fuel NOx comes from the fuel content, that is, fuel-bound nitrogen. The oil fuel sometimes contains high amount of fuel-bound nitrogen, which contributes a significant amount of fuel NOx to the total NOx emission. In other words, NOx emissions for oil firing or combination firing also depend on the fuel property, that is, nitrogen content.

There are many ways to reduce burner NOx. The popular technologies are air-staging, fuel-staging, and flue gas entrainment. In the combination burner, gas firing can use any of the aforementioned technologies. However, for oil firing, since oil fuel is delivered by an oil gun, it is difficult to employ fuel-staging technology on the burner. Hence, most oil burners use air-staging technology to lower the NOx emissions. The patented DEEP*star* burner[1,2] uses staged-fuel/staged-air and flue gas entrainment to reduce NOx emissions when firing gas, and staged-air and flue gas entrainment to abate NOx emissions during oil firing. The detailed description of the DEEP*star* burner design is covered in Chapter 1.

Particulate emissions are controlled by the oil gun and burner designs. An oil gun with good atomization produces low particulates. This is discussed in Volume 1, Chapter 10. Poor air distribution in the burner design will also increase particulate emissions. Poor air distribution may create fuel-rich pockets, which enhance soot generation. Soot is a small solid carbon particle and is generated under fuel-rich environments. Air-staging technology can lower oil NOx emissions, but has a negative impact on particulate emissions. This is a common situation, where a technology that lowers NOx emissions creates a negative impact on CO and/or particulate emissions. Burner design engineers are required to pay extra attention on burner designs to achieve an optimum balance between NOx, CO, and particulate emissions.

2.3 Boiler Oil Burner

Boiler burners are similar to process burners in basic ways but very much different in application. The basics are the same: an oil atomizer (generally in the center); a flame holder, shield, spinner, etc.; and an air delivery body. The heat delivered is generally much larger ranging from 3 to 130 MW (10–450 MMBtu/h). The air delivery body is often referred to as the burner with the atomizer as a subpart of the burner.

As opposed to process burners, boiler burners need to have high heat intensity as they fire into smaller volumes. The typical range for what is known as “space heat release rate” is 30,000–150,000 Btu/h/ft^3 (310–1550 kW/m^3). This requires small droplets from the atomizer and uniform air distribution to the atomizer spray. It may also require swirling the air and/or specially shaped exit throats as well as shaped oil sprays. Turndown is usually 8:1, but can be enhanced. For multiburner boiler applications, the burners may be fitted with an isolation shutoff damper in order to operate with fewer burners, from a damaged burner, or for emission control.

2.3.1 Oil Gun Types

Smaller boilers in the range of 50–1000 K PPH (lb/h) steam generally use a variety of twin-fluid internal mix atomizers or tip mix such as a Y-jet[3] as discussed in detail in Volume 1, Chapter 10. Utility-sized boilers may use mechanical atomizers such as the simplex wide-range return flow (WRRF) design due to the enormous cost of atomizing steam and because the furnace residence time is long. Some utility units may use twin-fluid atomizers utilizing low steam consumption, typically 0.05–0.1 mass ratios. The decision is usually based on furnace size and emission requirements.

It is common to design special cap drilling patterns for boiler applications. For package boilers, the tip drilling may be elliptical to produce elliptical flames to fit the aspect ratio of the furnace. Similar tip features can be utilized on tangential fired utility units to prevent flame impingement. The drilling patterns can vary by producing staged combustion for NOx reductions, carbon loss, and opacity. Internally mixed atomizer are used rather than Y-jets due to their complex drilling pattern.

Boiler burners are usually designed to burn both gas and liquid fuels and sometimes multiple fuels. For multiple liquid fuels, the oil atomizer can be changed out to different kinds or sizes. When burning multiple liquid fuels simultaneously is required, special oil atomizers are designed to produce full cone sprays and partial cone sprays so that the sprays do not interact. For these applications, it is not uncommon to employ one to five oil atomizers in a single burner.

2.3.2 Oil Flame Stabilization

For boiler applications, atomized liquid flames can be stabilized in the internal cone of the spray or by the use of a flame stabilizer similar to a process burner.

The use of spray cone stabilization is usually limited to warm up burners for utility and special applications. Ordinary boiler burners use a stabilizer that reduces axial velocity and may impart swirl to the local flow to enhance recirculation of hot combustion products to establish continuous ignition. Much care must be taken in the design of these stabilizers to prevent droplet deposition and overheating. Material selection will depend on fuel liquid properties, air preheat, and boiler thermal environment.

2.3.3 Flame Shaping

One of the most unique features of boiler burners is the ability to shape the flame. In many cases, this is a requirement due to the furnace shape. This is accomplished with nonsymmetrical atomizer exit orifice drilling (see Figure 2.8), the aerodynamics of the burner, air, and the exit throat shape. In the industry, this is known as flame shaping.

Flame shaping is used for many reasons. One is to fill a nonuniform cross section of the furnace to increase droplet residence time and shorten the flame length. Other reasons may include CO control or steam superheat control.

FIGURE 2.8
Coen Co. elliptical cap with slots for low NOx.

2.3.4 Emission Considerations

Almost every boiler burner will have to meet some level of emissions. These emissions will be very dependent on boiler size, boiler condition, fuel properties, and boiler construction.

Nitrogen oxide emissions (NOx) will primarily come from a portion of the elemental nitrogen compounds in the fuel and thermal NOx. Prompt NOx is small and not very controllable in liquid-fired burners.

Thermal NOx is formed at high temperatures due to the dissociation of molecular nitrogen and oxygen. The details of formation and reduction are covered elsewhere in Volume 1, Chapter 15.

Specific to liquid fuel firing is the conversion of elemental nitrogen compounds in the fuel. The key to reducing this conversion is flame operation at fuel-rich conditions until the elemental nitrogen can form molecular nitrogen (N_2). The balance of the air can then be mixed with the rich products.

This can be done with atomizer modifications or the use of staged air, where a portion of air is removed from the burner and added later through ports into the flame outside the burner exit. The degree of this staging will yield fuel-bound NOx reduction of up to 60% at the expense of added flame length.

Carbon monoxide (CO), unburned hydrocarbons (UBHC), and volatile organic compounds (VOCs) are all forms of gaseous unburned fuel gases and vapors. These can be formed due to reacting flames close to cool walls (narrow furnaces), long flames (short furnaces), leaky division walls, low excess air, staged flames, large atomizer droplets, and all forms of air/fuel imbalance especially in multiburner applications. For liquid-fired burners, these gaseous/vapor emissions are generally low because particulate and opacity will generally be formed in excessive amounts before the gaseous/vapor limits are reached.

The process of liquid droplet combustion involves droplet heat up, vaporization, gaseous oxidation, and residual char oxidation. The residual char will contain carbon, ash, and some sulfur compounds that will contribute to particulate emissions and opacity.

The size of the residual char will be a function of the original oil droplet size distribution and other fuel properties. These char particles can be hard, and carbon-like particles, or partially hollow ceno-spheres.

Generally, one cannot control the ash or sulfur compound composition that contributes to particulate emission via burner design or boiler design. Reducing the carbon content is controllable with even air/fuel balance and low-liquid droplet sizes, flame shaping, and larger boiler sizes. Carbon loss should be lower than 0.05 lb/10^6 Btu (75 mg/N-m^3) as a goal.

Opacity is a direct result of particulate emissions and is due to light scatter in the stack exhaust of the fine particles. Small stacks, high particulate emissions, and small particulate sizes increase opacity. Opacity from liquid-fired boiler burners should be below 10%.

2.4 Oil Burner Performance

The combination burner performances can be evaluated by flame dimensions (length and width), turndown ratio, NOx emissions, particulate emissions, and noise level.

2.4.1 Flame Dimensions

The flame dimensions for oil or gas firings depend on applications. Some applications would like short flames and some prefer long flames. The flame length is mainly determined by the burner design. During operation, flame length would vary with heat releases and O_2 levels. Generally, flame length is estimated by the length per unit heat release. However, different O_2 levels would affect this estimate. The lower the O_2 level, the longer the flame length. For gas firing, the burner usually is operated at 3% O_2 (dry), and for oil firing it is typically at 4% O_2 (dry). Oil firing requires more excess air to prevent smoke.

2.4.2 Turndown Ratio

The burner turndown ratio is defined by the maximum heat release divided by the minimum heat release. The turndown ratio for oil firing is mainly determined by the oil gun design. A good atomization oil gun achieves a high turndown ratio. The oil gun turndown ratio is discussed in Volume 1, Chapter 10. Sometimes, reducing cold air quenching by partially closing the burner damper at low heat release helps increase the turndown ratio.

The turndown ratio for gas firing is determined by the burner design. The available fuel pressure is the main control parameter. The higher the fuel pressure, the better the turndown ratio. Typically, the burner turndown ratio is around three. High turndown ratio is favored by operators due to ease of managing the operation.

2.4.3 NOx Emissions

The NOx emission for oil firing is a function of furnace temperature, combustion air temperature, O_2 level, and oil gun operation conditions. Furnace temperature, combustion air temperature, and O_2 level, have direct effects on the NOx level. This means an increase in those parameters raises NOx emissions. For example, an air-preheated case always has higher NOx than the ambient condition. The effect of oil gun operation conditions on NOx emissions is discussed in Volume 1, Chapter 10.

2.4.4 Particulate Emissions

Generally, high particulate emissions are only generated from No. 6 heavy oil firing causing an environmental impact. Particulate emissions are a function of furnace temperature, combustion air temperature, O_2 level, and oil gun operation conditions. The effects are opposite to those for NOx emissions. For example, an air-preheated case has lower particulate emissions than those for the ambient case. The effect of oil gun operating conditions on particulate emissions is discussed in Volume 1, Chapter 10.

2.4.5 Noise Level

For gas firing, the burner noise level is a function of heat release and fuel compositions. The larger the heat release and the hydrogen content in the fuel, the higher the noise level. For oil firing, noise level is a function of heat releases and oil gun design. For example, the John Zink HERO® gun has much lower noise levels than that for the EA or MEA oil guns. To meet industrial noise limitations, an appropriate muffler design is required to reduce the noise level (see Volume 1, Chapter 16).

2.5 Burner Performance Predictions

Burner performance can be divided into two parameters, basic burner efficiency features and meeting emissions at low capital and operational costs.

Boiler burner and process burner emissions are closely coupled with the furnace environment and fuel properties. A complete description of the furnace is required including dimensions, wall surface description, tube location and spacing, refractory locations, burner(s) location and spacing, and combustion air temperature. A complete fuel analysis is required including ash, sulfur, nitrogen, hydrogen, carbon, SG or API gravity, viscosity, Conradson carbon, asphaltene content, heating value, and sometimes a distillation curve.

Other performance parameters such as turndown, excess air, and noise are burner/site specific and are incorporated in the design. For example, low excess air requires uniform air distribution to the burner from the wind box. Low draft loss is desirable to reduce horsepower (HP) costs, but may not be achievable due to constraints to even air distribution or high turndown. Minimal parasitic HP costs associated with external flue gas recirculating (FGR) fans are desirable.

2.5.1 Droplet and Carbon Burnout

As discussed earlier, the process of droplet burnout is heat up, evaporation, gaseous oxidation, and char burnout. Extensive analysis has been done on droplet heat up and evaporation, for example, Chin and Lefebvre.[4] Most of this work has been related to gas turbines where the rate of combustion is limited by evaporation rates. In furnaces, the evaporation rate is small compared to the available residence time.

A much more simple approach is to use the effective evaporation constant that includes heat up and evaporation as discussed by Lefebvre.[5] It is defined as

$$\lambda_{\text{eff}} = \frac{D_o^2}{t_e} \tag{2.1}$$

where

D_o is the initial droplet diameter in μm
t_e is the time in seconds
λ_{eff} is the evaporation constant

For injection of droplets into hot flame zones, λ_{eff} is about 0.8 $\mu m^2/\mu s$.

For usual liquid fuels used in furnaces, the total heat up and evaporation time ranges from 10 to 20 ms and is not significant compared to mixing and char burnout.

For oxidation of the gasified droplet gases, it is customary to use two-step kinetics where the hydrocarbon fuel is assumed to oxidize to CO and CO oxidizes to final products. In practice, the limiting chemistry is the final CO oxidation. Further large-scale eddy mixing is partially responsible for the mixing oxidizer and fuel. For HC oxidation, several sources are available such as Ref. [6].

In general,

$$\frac{d(C_aH_b)}{dt} = -5.52\times10^8(P^{-0.815})(T)e^{(12,200/T)}(C_aH_b)^{0.5}(O_2)\ \text{mol/cm}^2\ \text{s} \qquad (2.2)$$

For CO destruction, several kinetic data are available such as Ref. [7]:

$$\frac{d[CO]}{dt} = -1.8\times10^7 e^{(-25,000/RT)}(CO)(O_2)^{.5}(H_2O)^{.5}\left(\frac{P}{RT}\right)^2 \qquad (2.3)$$

Almost all published CO rates involve H_2O because CO destruction requires the $(OH)^{-1}$ radical to produce the reaction.

After, and partially during, gaseous oxidation, the original droplet becomes void of hydrocarbons and becomes a char "particle." The particle can be a solid sphere or a hollow ceno-sphere depending on the original droplet size and fuel properties such as Conradson carbon or asphaltene content. In either case, the resultant char size and mass has shown to be related to fuel Conradson carbon and asphaltene content via the coke formation index or CFI[8] (see Figure 2.9).

For hollow ceno-spheres, the CFI computes the residual char size from the original oil droplet as follows:

$$\text{CFI} = 6\frac{D_c^2}{D_o^3}\frac{\rho_{cs}}{\rho_o}\delta \qquad (2.4)$$

And for a solid char the relationship is

$$\text{CFI} = \frac{D_c^3}{D_o^3}\frac{\rho_c}{\rho_o} \qquad (2.5)$$

where

D_c is the char diameter
D_o is the droplet diameter
ρ_{cs} is the density of ceno-sphere char surface
ρ_o is the density of oil droplet
ρ_c is the density of char particle
δ is the ceno-sphere shell thickness

Smaller droplets will form solid char and larger droplets will form ceno-spheres. After determining the char size and structure, a char burnout model can be selected. The char will then oxidize by diffusion of oxygen to the surface and surface reactivity. Two models

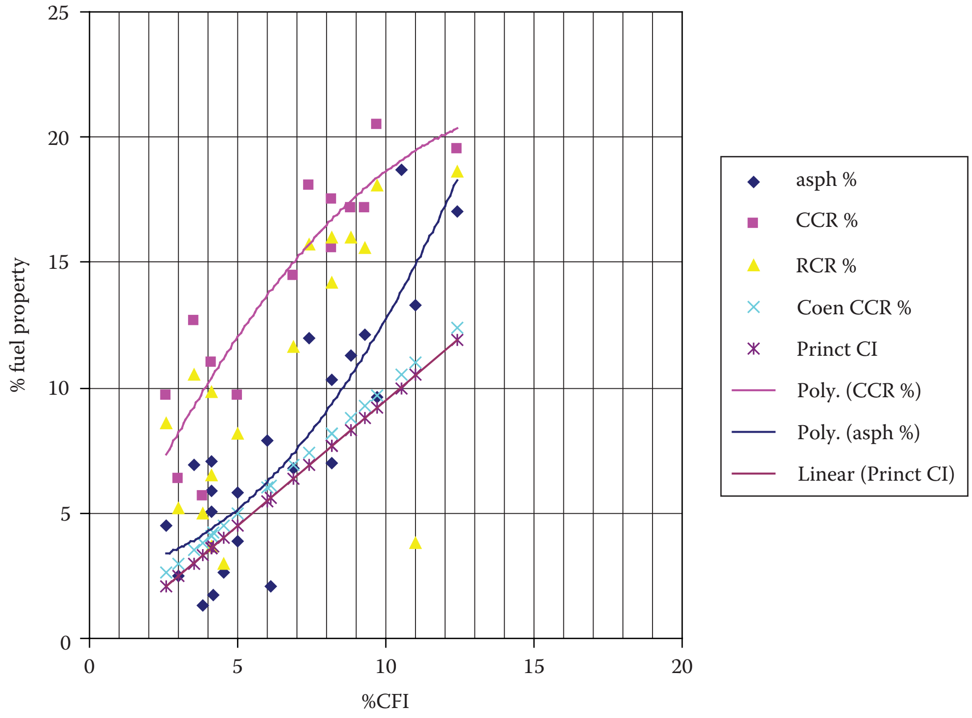


FIGURE 2.9
CFI versus CCR, RCR, and asphaltenes.

can be used depending on the initial atomizer droplet size. One model is shrinking diameter constant density for solid chars, and the other is constant diameter shrinking density for ceno-spheres. In either case, the oxidation is defined as follows[9]:

$$\frac{dm}{dt} = \frac{-12C_{\text{og}}A_{\text{p}}}{(1/K_{\text{m}}) + (1/K_{\text{r}})} \tag{2.6}$$

where

dm/dt is the grams carbon/second consumed
C_{og} is the molar density of O_2 gmol/cm^3
A_{p} is the area of initial char in cm^2
K_{r} is the surface reaction coefficient in cm/s
K_{m} is the diffusion coefficient in cm/s

and

$$K_{\text{r}} = \frac{K_{\text{p}}\bar{R}T_{\text{k}}}{32} \text{ cm/s} \tag{2.7}$$

where

$$K_{\text{p}} = Ae^{(-E/T_kR')} \text{ is the reaction rate char data in } \text{g/cm}^2 \cdot \text{s} \cdot \text{atm } O_2 \tag{2.8}$$

where

E is the activation energy in cal/g-mol
R' is 1.98 cal/g-mol K
$\underline{A}$ is the preexponential factor g/cm^2 · s · atm O_2
$\bar{R}$ is 82.057 cm^3 atm/(gmol °K)
T_{k} is the temperature in °K

and

$$K_{\text{m}} = \frac{2D_{O_2N_2}}{D_{\text{c}}} \text{ cm/s} \tag{2.9}$$

where[10]

$$D_{O_2N_2} \cong 0.00026T_{\text{k}}^3 \sqrt{\left(\frac{1}{MWO_2} + \frac{1}{MWN_2}\right)} \text{ diffusivity in cm}^2\text{/s} \tag{2.10}$$

and MW = molecular weight O_2 or N_2.

Combining Equations 2.6 through 2.10 yields a final char loss formula:

$$\frac{dm}{dt} = \frac{-12C_{\text{og}}A_{\text{p}}}{(D_{\text{c}}/(0.000116)T_{\text{k}}^{1.5} + 32/K_{\text{p}}\bar{R}T_{\text{k}})} \text{ g C/s} \tag{2.11}$$

Combining the preceding equations, dm/dt can be integrated numerically or directly for the case of constant diameter/shrinking density with constant O_2 and temperature. Since this is rarely the case, the normal solution will involve droplet size distribution, reducing O_2, changing temperature, and a simple "marching" solution that is simple enough to do in a spreadsheet. Either the constant diameter or constant density model can be used after defining the char size distribution. Further, this procedure will yield not only the carbon particulate, but the size distribution as well. From this distribution and mass loading, opacity can be predicted using simple light scattering equations.

2.5.2 Emission Formation and Prediction

In oil burner applications, similar emission predictions are required as in gas fuel burners, carbon monoxide, UBHC, VOCs, and NOx. Kinetic rate equations such as (2.2) and (2.3) can be utilized when the flow and temperature field is known to predict reduction in emissions. The equations for thermal NOx formation are:

$$\frac{d(\text{NO})}{dt} = 2Ae^{(-E/RT)}(O_2)_{\text{eq}}(N_2) \tag{2.12}$$

and

$$(O_2)_{\text{eq}} = \frac{k_{\text{o}}}{(RT)^{0.5}}(O_2)_{\text{eq}}^{0.5} \tag{2.13}$$

One generally accepted practice is to assume (O_2) in equilibrium with (O) and (O_2) concentration using the Westenberg[11] results for k_{o} for (O_2) equilibrium and Zeldovich constants, A and E, as measured by Bowman.[12]

Then in all cases, one can post-process thermal map data in some discrete volume form and/or insert into a computational fluid dynamics (CFD) code using the Rayleigh flux theorem as follows:

$$\frac{\partial}{\partial t}\int^{Cv} n\rho dv = \int^{Cs} n\rho(V \cdot da) \tag{2.14}$$

where

n is the chemical in mass units
t is the time
ρ is the density
v is the volume
a is the area
V is the velocity vector

where, described in words, the formation of (n) through the volume surface is equal to the integrated rate of formation over the control volume.

It is a simple extrapolation to extend this concept for even coarse volumes as follows:

$$\sum \frac{dn}{dt}\rho\Delta v = n\rho(V \cdot a) \tag{2.15}$$

In the case of NOx formation using liquid fuels, the amount of elemental fuel nitrogen is extremely important to the addition of total NOx. A portion of the elemental fuel nitrogen is converted to NOx generally during gaseous combustion. The range of this conversion is very dependent on the burner and furnace type as well as any NOx reduction techniques utilized. The range can vary from 15% to 80% and can be more or less than the thermal NOx contribution.

From a practical perspective, particulate and opacity can be calculated on every application as outlined with great accuracy using Equation 2.11. Very expensive or large applications will utilize CFD (see Volume 1, Chapter 13) to compute the total flow field and post-process emissions utilizing Equation 2.14 with the appropriate destruction or formation kinetics. The fuel contribution to NOx is a function of many factors and each burner manufacturer will have a set of algorithms used for prediction.

The oil-fired burner is a complex process involving atomization of droplets, evaporation, kinetics of destruction, kinetic formation, char oxidation, diffusion, two-phase flow, heat transfer, and special stabilization techniques. In many cases, the flames will need to be formed to fit the furnace. The right atomizer must be selected to produce the right droplet size and distribution together with the right burner for proper aerodynamics. Extreme care must be considered in material selection depending on the liquid properties.

2.6 Oil Burner Maintenance

2.6.1 Oil Burner

The typical oil burner is more complicated than the standard gas-only burner. Because the oil burner is normally supplied as a dual-fuel burner, the configuration contains both oil- and gas-firing capabilities (see Figure 2.10). This combination design utilizes a center-fired oil gun and multiple gas tips located around the oil tile. Some designs will have a concentric dual gun arrangement with a center oil gun and an outer gas gun.

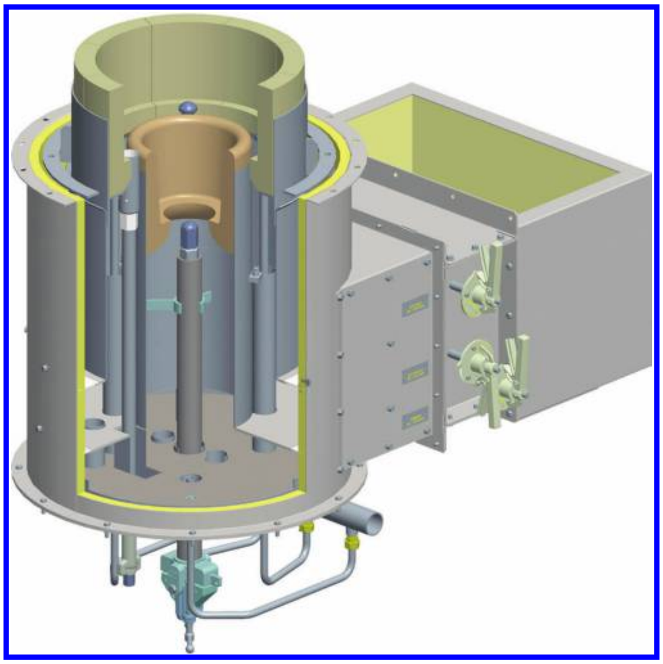


FIGURE 2.10
Combination oil and gas LoNOx burner.

The main components, of an oil burner, that require basic maintenance are

- Secondary tile
- Regen or primary oil tile
- Diffuser cone or swirler
- Air register or damper
- Oil gun assembly

The secondary tile and primary tile are cast from high-temperature refractory. Basic maintenance includes checking the dimensions for proper installation and repairing small cracks in the surface (see Figure 2.11). The secondary tile normally comes in several pieces that

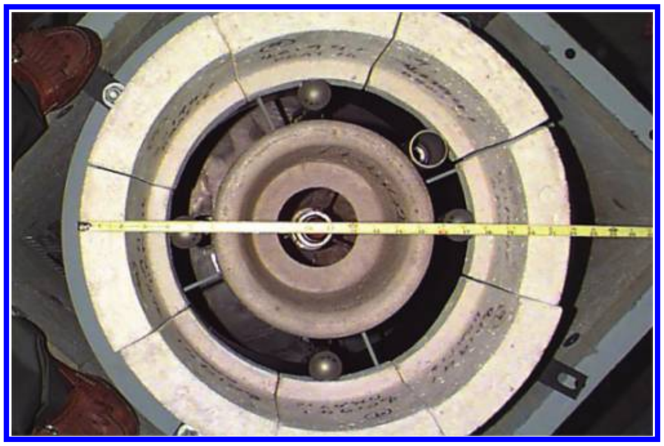


FIGURE 2.11
Secondary (outside) and primary (inside) tiles.

FIGURE 2.12
Regen tile and one section of secondary tile.

have to be installed as a single tile (see Figure 2.12). The joints between the individual pieces should be covered with a thin layer of high-temperature air-setting mortar.

The stability of the oil fire is accomplished by using an oil tile, a swirler, or a diffuser cone (see Figures 2.13 through 2.15). The type of burner and the physical characteristics of the oil determine which type of device is used. Diffuser cones and swirlers should be replaced if they become damaged or severely oxidized.

The combustion air control is achieved by using slotted registers, vanes, or dampers. Because there can be oil spillage and fouling when firing oil, the standard rotary air register must be given an upgrade to maintain its operability (see Figure 2.16). One way to achieve more reliable operation is to use a design that has bearings or rollers to help the movement of the inner and outer register cylinders (see Figure 2.17).

Another option is to use a multiple vane-type air register. This design provides very smooth and reliable operation. Since the vanes are normally curved, this design will add a degree of "spin" to the combustion air (see Figure 2.18).

A very good option, for oil firing, is to select a burner with an integral plenum box that uses a multiple vane-type damper to control the combustion air (see Figure 2.19).

Air registers and dampers can be lubricated with a high-temperature graphite- or silicone-type lubricant.

2.6.2 Oil Gun

The majority of oil guns currently in operation in the refining and petrochemical industry use steam or air as the atomization medium. The use of mechanical or high-pressure atomization is very rare and limited to specific types of equipment.

This section describes the design and maintenance for steam-atomized oil guns. The procedures and parts are basically the same for air-atomized oil guns.

The major components of an oil gun are the oil body, oil body receiver, oil tip, atomizer, and the orifice spud. Figures 2.20 through 2.24 show some oil gun components from several different oil gun types.

FIGURE 2.13
Regen oil tile with an oil gun in the center (secondary tile not shown).

FIGURE 2.14
Swirler for oil firing on forced draft.

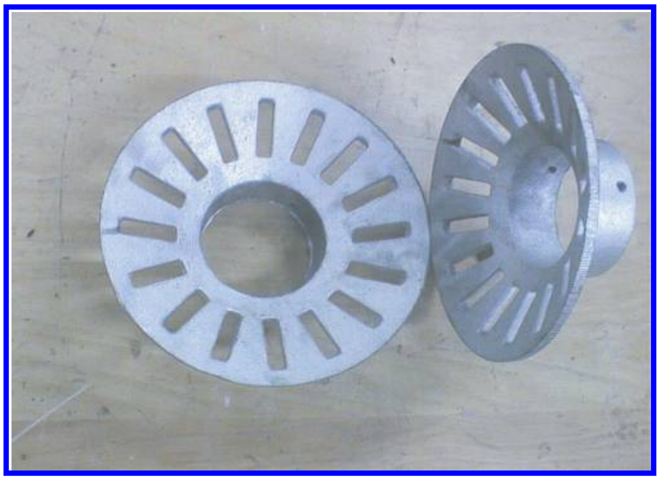


FIGURE 2.15
Diffuser cones for light oil firing.

Oil gun maintenance requires basic hand tools and the oil gun drawing. A normal list of tools would include

1. Cooling water bath
2. Stationary vise
3. Wire brush
4. 12 or 24 in. (305 or 610 mm) pipe wrench
5. 1 1/2 in. (38 mm) open end wrench
6. 1 5/8 in. (41 mm) open end wrench
7. 3/4 in. (19 mm) open end wrench
8. 1/4 in. (6 mm) Allen wrench (male hex drive)
9. Standard screw driver (slot type)
10. Copper-, bronze-, or nickel-containing high-temperature pipe thread lubricant with applicator brush
11. Hole gauges—contact the manufacturer for specific sizes

2.6.2.1 Oil Gun Insert Removal

Two types of oil gun assemblies are available from John Zink Company. These oil guns vary only in the method for supply connections.

One style, the Z-39-C oil body style, requires the oil and atomizing medium to be broken for oil gun insert removal, while the most common, the Z-56 oil body style, does not. In both cases, the oil gun guide tube, which provides the tip adjustment, does not have to be removed with the tip/atomizer assembly for cleaning.

In all fuel cases, when breaking the closed oil and atomizing medium systems, safety is of utmost importance. It is highly recommended that a full purge of the oil side be completed prior to insert removal. This will minimize the possibility of oil spills exterior to the burner.

2.6.2.2 Z-39-C

After purging the oil gun, disconnect the oil and steam piping making sure to shield any possible pressure spray from the connections due to oil gun blockage. Then, loosen the three set screws located at the back of the oil gun guide tube. The oil gun should then be completely retractable without disturbing the guide tube location.

2.6.2.3 Z-56 "Quick Change"

After purging the oil gun, loosen the oil gun clevis by use of the clevis handle making sure to shield any possible pressure spray from the interface sleeves due to oil gun blockage. Swing the clevis away and retract the insert.

2.6.2.4 Disassembly

1. Using the cooling water bath, quench any residual heat buildup in the tip area until the tip/sleeve may be gripped, in a bare hand, without discomfort.
2. Position the oil body in the stationary vice and tighten.
3. Using the 1 5/8 in. (41 mm) end wrench as a backup on the oil gun sleeve remove the oil tip using the 1½ in. (38 mm) wrench.
4. Using the pipe wrench, remove the steam tube/sleeve assembly from the oil body.
5. Using the pipe wrench as a backup on the oil tube remove the atomizer assembly using the 3/4 in. (19 mm) end wrench.
6. Holding the atomizer body with the 3/4 in. (19 mm) end wrench, remove the oil spud using either the Allen wrench or the slotted screw driver.

FIGURE 2.16
Typical rotary-type air registers.

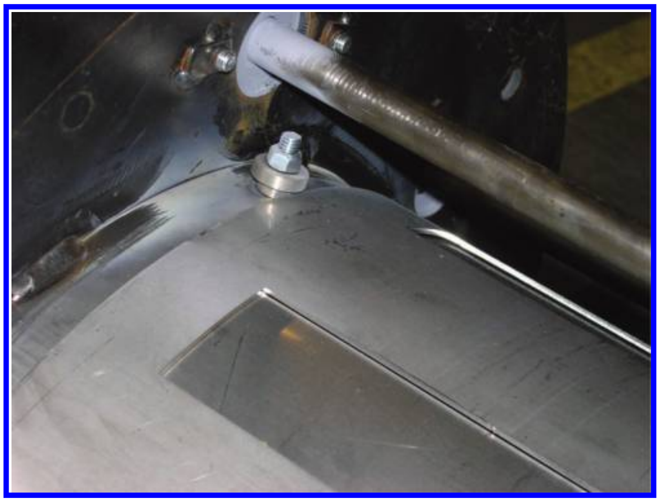


FIGURE 2.17
Air register with rollers for easy operation.

NOTE: Depending upon the materials of construction, the type of drive supplied on the oil spud may be either style. A quick inspection from the threaded end of the atomizer assembly will determine the proper tool.

2.6.2.5 Inspection

Oil tip—external inspection of the port region will reveal much about the condition of the oil gun. Visually, carbon or oil buildup around the exit ports is an indicator of three possible problems: the tip is inserted too far, low atomizing medium pressure or flow, and erosive or corrosive action on the tip.

After cleaning all foreign materials from the face of the oil tip, inspect the exit ports for wear. Visual erosion, egg-shaping of the exit ports, will cause disruption of the flame pattern and should be accepted as reason for replacement.

Atomizer—external inspection of steam ports should show them free of any foreign material and the exit port should be concentric and uneroded/corroded. The labyrinth seal should be clean and unscored in the longitudinal direction (see Figure 2.25).

Oil spud—oil spud should be free of foreign matter.

Tip/atomizer—atomizer labyrinth seal should insert into the oil tip with a minimum amount of tolerance. New atomizers often require a twisting motion to achieve insertion. If atomizer/tip fit is not "firm," steam bypassing of the atomizing chamber is possible. Check the atomizer in a new tip to determine if it is the tip or the atomizer that is worn.

2.6.2.6 Assembly

1. Lightly lubricate the threads of the oil spud and install into atomizer.
2. Lightly lubricate the pipe threads on the 3/8 in. (10 mm) SCH 40 oil tube, and using the pipe wrench for backup, screw the atomizer assembly firmly onto the oil tube.

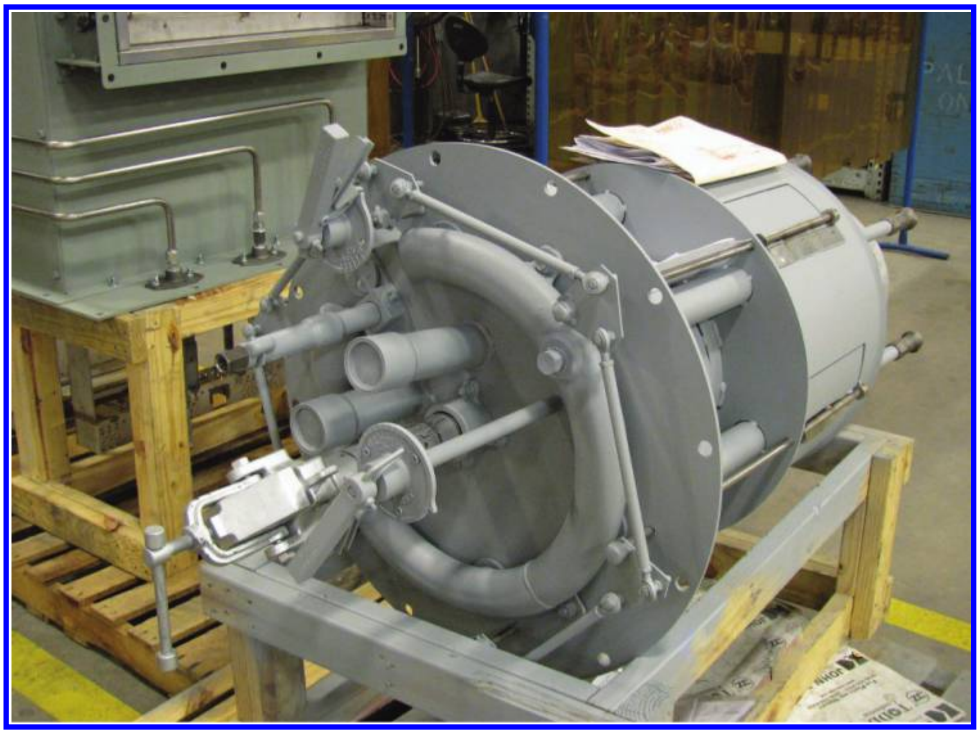


FIGURE 2.18
Vane-type air register.

FIGURE 2.19
Integral plenum box with inlet air damper and muffler.

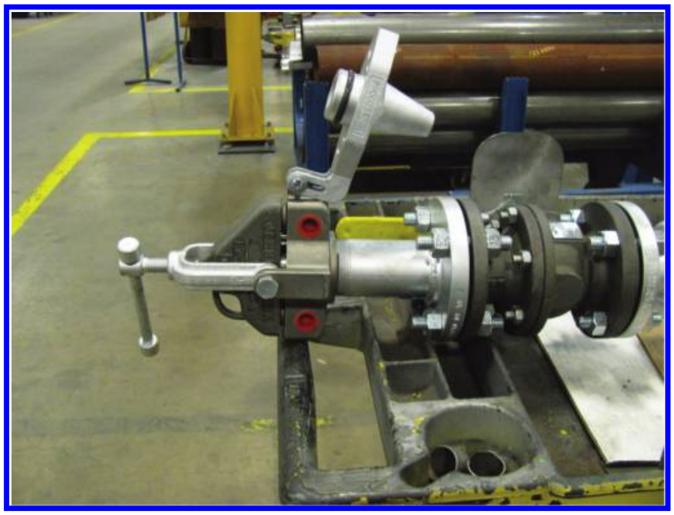


FIGURE 2.20
Oil gun insert and oil body receiver (with red caps).

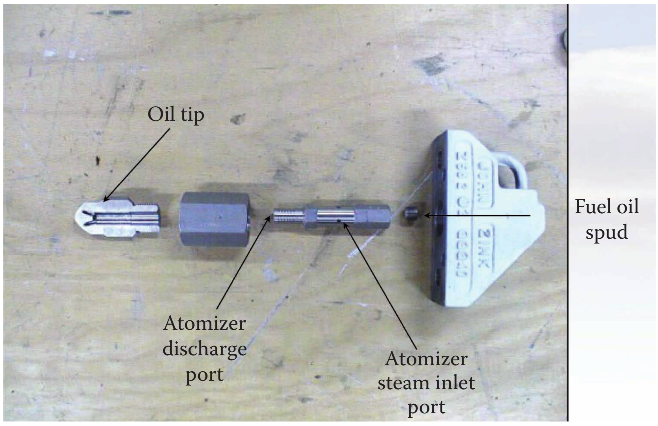


FIGURE 2.21
EA oil gun parts.

FIGURE 2.22
EA oil tip.

FIGURE 2.23
MEA oil tip.

FIGURE 2.24
MEA oil gun parts.

FIGURE 2.25
Atomizer with labyrinth seals and steam ports.

FIGURE 2.26
Checking atomizer location in the sleeve.

3. Lightly lubricate the labyrinth seal of the atomizer taking care to not load up labyrinths or fill exit port or steam ports.
4. Lightly lubricate the threads on the 1 in. (25 mm) SCH 40 steam tube and replace firmly onto the oil body. This procedure should result in the atomizer end being 1/4 ± 1/8 in. (6 ± 3 mm) inside of the sleeve (see Figure 2.26).
5. Lightly lubricate the threads on the oil tip and install onto the oil gun making sure that the beveled portion of the tip firmly seats on the beveled portion of the sleeve.

The oil gun is now ready to be reinstalled into the burner.

2.7 Troubleshooting the Oil Burner

Firing oil properly can be very difficult or relatively easy depending on the condition of the oil and steam. Basically, good oil firing with clean flames comes down to three key items:

1. Has the burner/oil gun been designed for the type of oil to be fired?
2. The oil must be at the right pressure and temperature at the burner.
3. The steam must be at the correct pressure and should be free of condensate.

2.7.1 Effect on Operations

The impact on operation, without the proper design and operating conditions, can result in serious and dangerous situations. Improper oil firing can result in many hazardous conditions including the following:

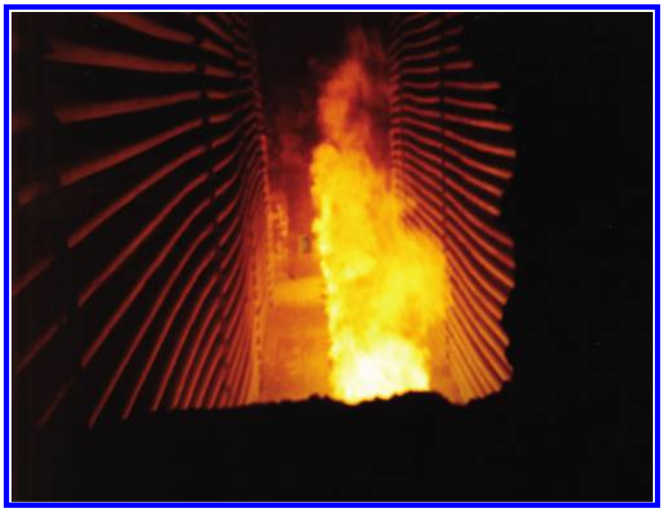


FIGURE 2.27
Flame impingement on tubes.

1. Oil spillage and dripping from the burners to grade
2. Incomplete combustion resulting in CO and UHC's in the flue gas
3. Smoke coming out of the stack
4. Flame impingement on the tubes (see Figure 2.27)
5. Unstable flames with possible liftoff or going out
6. Fouled and plugged oil guns (see Figure 2.28)

2.7.2 Corrective/Preventive Actions

The single most important item when firing heavy oil is the temperature of the oil at the burner. The temperature controls the viscosity of the oil. Having the right viscosity is required for proper atomization. The viscosity of a liquid is an indicator of the fluids resistance to internal shear. A higher viscosity makes it more difficult to atomize the oil properly. John Zink recommends that the viscosity of the oil be no more than 200 SSU (43 cSt) at the burner.

A liquid fuel does not readily burn like a gaseous fuel. To burn a liquid rapidly first requires that the liquid be changed into a vapor-like mist. This is accomplished by what is known as "atomization." When the proper amount of atomization is attained, the oil will be converted to a fine mist that can then be oxidized. This conversion of the oil to a mist increases the surface area of the fuel so that burning can take place. Compare this to a log burning in a fireplace. If one puts a match on one end of the log, it will take many attempts and several boxes of matches to ignite the log. However, if the log is ground into sawdust, then a single match would ignite the log easily. The same principle applies to atomization.

FIGURE 2.28
Fouled oil guns.

Steam is the best method to atomize heavy oil. Light oils such as No. 2 and naphtha can be atomized with air in some cases. For the majority of oils, steam is the best alternative for clean oil fires and proper atomization. The condition of the steam is very important to good atomization. This means that the steam must be dry and delivered to the burner at a higher temperature than the oil. Normally the standard 150 psig (11.4 barg) steam, available in most plants, will work for oil-firing applications. In some cases, such as a heavy pitch, the use of 400 psig (30 barg) steam may be required to ensure the steam is hotter than the oil temperature. We generally label oil as "pitch" if the API gravity is less than 10.

The steam has to be delivered to the burner at the right pressure for the oil gun to work as designed. Most John Zink guns are designed to operate with a constant differential pressure of 20–30 psig (2.4–3.1 barg). This means the steam pressure will be 20–30 psig (2.4–3.1 barg) higher than the oil pressure. In some cases, a higher differential may be required for proper atomization. Steam consumption varies with the type of oil gun. Normal rates are between 0.15 and 0.3 lb/lb.

Two of the most common problems, when using steam atomization, are the lack of insulation on the steam piping and faulty steam traps that allow condensate into the burner. The steam piping should be fully insulated all the way to the oil gun on the burner. Leaving just three to four feet of pipe uninsulated can cool the steam enough to produce condensate.

When a steam trap is working properly, there will be substantial difference between the temperature of the inlet and outlet piping to the trap. A difference of 50°F–70°F (10°C–21°C) usually indicates normal operation. A downstream temperature within a few degrees of the upstream temperature indicates that the trap may not be working as it should. The downstream temperature should always be lower than the inlet temperature.

Another "best operation" recommendation is to provide steam with a slight amount of superheat such as 40°F–50°F (22°C–33°C). This will keep the steam dry and free of condensate.

John Zink provides a very detailed "troubleshooting" guide in the "Installation and Operating Manual" provided with all of its oil-fired burners. The operators and maintenance people can use this to resolve many of the common problems associated with oil firing.

In general, poor oil firing will exhibit six problems to the field operator, either singly or in combination (see Figure 2.29):

1. Smoky flame
2. Fireflies or sparklers in the firebox
3. Lazy or uncontrollable flame pattern
4. Instability
5. Coke formation
6. Oil spills

2.7.3 Insufficient Air

Insufficient air in a process heater is often one of the most difficult problems to solve because it can be either the main problem, whose solution is simply to increase the air/fuel ratio, or it may be the result of any one or a combination of the other listed causes.

In the case of multiple burner operation, to address air deficiency as a cause, review the single burner operation with respect to the total heater operation. A single burner operating with insufficient air will still smoke even though the total firebox may exhibit sufficient excess air for completed combustion. This is due to the additive excess air from all other burners.

Specifically, an overall review of all burners, noting:

- Gas pressure on gas-fired burners
- Oil and atomization pressure on oil fired burners
- Position of hand block valves on
  - Gas
  - Oil
  - Atomization medium
- Register settings

Problem	Possible causes												
	A. Insufficient air	B. Lack of atomizing pressure	C. Lack of atomizing flow	D. Wet steam	E. Cold oil (heavy oil)	F. Tip/atomizer failure	G. Diffuser/regen failure	H. Low oil flow	I. Low oil pressure	J. High atomizing pressure	K. Leaking cross over valve	L. High steam temperature	M. Oil tip mispositioned
1. Smoky flame	X	X	X		X	X	X		X			X	X
2. Fire flies		X	X	X	X	X			X		X		X
3. Lazy flame	X	X	X		X	X	X		X				X
4. Instability	X			X		X		X	X	X	X	X	X
5. Coke	X	X	X		X	X	X		X				X
6. Oil spill	X	X	X		X	X		X	X				X

FIGURE 2.29
Oil-firing problems and possible causes.

should give a good indication of whether any particular burner is being operated at a capacity greater than would seem suitable for its air supply. Neglecting airflow distribution in any forced-draft system and minor tolerance differences in burner throat tile installations, all burners with equivalent fuel supplies should have approximately equivalent register settings. It should be noted that this superficial preliminary review is not valid unless all manual block valves are fully open.

If the aforementioned review yields no major variations in fuel/air supply systems and air shortage is still suspected, a physical inspection of primary and secondary (and tertiary where applicable) air throats should be made to ensure no foreign materials are obstructing proper airflow. This inspection can usually be made through sight ports and/or register assemblies; however, in some cases, a probe may be required. If no obstruction is evident in the tile throat(s), inspection and manipulation of the air register assembly(s) is indicated to rule out register blockage.

Given the aforementioned conditions are shown not to be a problem, the final step is to make a simple static pressure reading within the register assembly. This reading should entail no less than three locations, one of which should be made within the primary air supply zone, downstream from the primary air supply on a regen-style burner and within 6 in. (15 cm) of the backside of the diffuser in a diffuser-style burner. These static pressure readings are a valid indicator of plenum air distribution and/or furnace draft variations and should be compensated by register settings.

In some cases, spot checks of oxygen levels within the radiant box can also be used for register adjustment to achieve balanced multiple burner operation.

Once the air supply side of the burner(s) in question has been determined to be balanced, the fuel supply becomes questionable.

2.7.4 Lack of Atomizing Pressure

Check the manufacturer's capacity curve supplied with the burner. Each oil gun capacity curve should indicate both the type of oil used for design sizing and the suggested atomization pressure for that oil.

Generally, the low viscosity oils require lower atomizing pressures than the higher viscosity oils. If a change in viscosity has been experienced either through a change in oil from design or a change in operating temperature, it is possible that a revision in atomization pressure may be required.

Since the "EA" series gun uses an internal mixing chamber, this type of gun is subject to variations in oil flow at any single oil pressure. A small variation in atomizing medium supply pressure can significantly change the internal pressure of the mixing chamber. Therefore, low atomizing supply pressure will increase the net oil flow, increasing the firing rate, thereby causing that burner to be "overfired" at a pressure which it normally would have sufficient air.

2.7.5 Lack of Atomizing Flow

Plugging of atomizer steam ports with pipe scale, dirt, and/or particulates will cause a reduction in atomizing medium flow, effectively reducing the mixing chamber back pressure, resulting in the same effect as low atomizing pressure. A full discussion of disassembly and cleaning of the "EA" series oil gun is covered in the maintenance section.

2.7.6 Wet Steam

Specifically, on those burners designed to use steam as the atomizing medium, it should always be clean, dry steam. A suggestion of 20°F–40°F (11°C–22°C) superheat should ensure this situation. However, all atomizing steam systems should be trapped.

Since water is lower in energy than steam, atomization with wet steam is not as effective as with dry steam. This lower-energy atomization will result in a larger droplet size and some oil-coated water droplets being dispersed into the combustion zone. These larger droplets and oil-coated water droplets are slower burning and can often be seen floating on the internal firebox currents. These "fireflies" are a major source for ash and soot buildup on radiant and convective tubes.

A second effect is that since this lower-grade atomization is slower burning and accumulation (or sludging) of water is not combustible, burner stability may be severely reduced.

2.7.7 Cold Oil (Heavy Oil)

On those burners operating on heavy or viscous oils, the oil should be heated to a temperature sufficient to achieve 200–250 SSU. This suggested viscosity range is sufficient for proper atomization and any reduction in oil temperature will increase the oil's viscosity.

Viscosity is a measure of the oil's resistance to break up, and as the viscosity increases, the quality of atomization and combustion decreases.

2.7.8 Tip/Atomizer Failure

The "EA" series oil gun operation is dependent on a number of machined orifices, channels, and seals. These pieces are subject to high-velocity abrasive flows and corrosive action dependent on the type of oil fired.

Clearly these orifices, channels and seals are subject to some "normal" wear, making them a "maintenance item." Additionally, this condition is aggravated by the common, and some not so common, contaminants found in many oils. Coke or carbon particles, catalyst fines, and silica particles have a highly erosive action on metal parts when subjected to high-pressure, high-velocity metering, while sulfur, chloride compounds, and, in some cases, anhydrous acids will severely attack, through corrosion, the materials of the atomizer and dispersion nozzle.

The use of hardened tip and atomizer materials for erosive oils and 300 series stainless steel or higher for corrosive oils is suggested.

Some typical effects of tip and/or atomizer deterioration are as follows:

1. Enlargement of oil orifice—high oil flow, low atomizing medium ratio, poor atomization, and burner overfiring
2. Enlargement of atomizing orifices—high atomizing medium flow, low oil flow, reduction in oil gun capacity, and reduction in low fire stability
3. Enlargement of atomizer exit—lowered mixing chamber pressure, reduced atomization quality, and burner overfiring
4. Deterioration of atomizer labyrinth seal—steam bypassing of atomization chamber, poor atomization, instability, and unsymmetrical flame patterns
5. Deterioration of dispersion chamber—reduction of exit port L/D, deterioration of dispersion pattern, coking, and oil spills
6. Enlargement of exit ports—reduction of exit port L/D

See the maintenance section for details of disassembly, inspection, and cleaning of the oil gun.

2.7.9 Regen/Diffuser Failure

Commonly, a failure in either the refractory of the regen tile or the metallurgy of a diffuser cone is the result of some other oil-firing problem. However, it should be noted that these parts are integral and necessary to the proper function of their burner. Failure of these parts should be acted upon with replacement immediately.

As a secondary consideration, nonconcentricity of these parts with respect to secondary tile throat and oil tip will cause poor air distribution, nonuniform flame patterns, coking, and oil spills.

2.7.10 Low Oil Flow/Low Oil Pressure

Extreme reduction in oil-firing rate and/or plugging of the oil orifice from pipe scale or oil-borne contaminants can cause severe burner stability problems, while the lowered exit port velocities can cause dripping or internal oil spills. If, for any reason, the burner capacity requirements are reduced by any appreciable percentage, new reduced capacity oil guns are suggested.

2.7.11 High Atomizing Pressure

As discussed in Section 2.7.4, the converse is true. High atomizing medium pressure will increase the mixing chamber back pressure, thereby reducing the oil flow. In many cases, this raised atomizing medium/oil ratio can cause severe stability problems.

2.7.12 Leaking Crossover Valve

Since the "EA" series oil gun is commonly operated with the atomizing medium at a higher pressure than the oil, a leaking purge crossover valve can cause severe disruption in oil flow to the oil gun and be detrimental to the atomization of the oil supplied.

The bypassing of atomizing medium into the oil supply is typically characterized by what is commonly called "motor boating." This continuous disruption of oil flow is clearly audible, thus deriving its name from the similar sounds.

2.7.13 High Steam Temperature

High temperatures on the atomizing medium cause problems in two separate ways, but these problems can be directly tied to the medium temperature.

Light oils often can be adjusted to very clear, yellow fires; however, this same fuel will often exhibit instability, pulsation in flow, clear blue/bright yellow flame envelope, and haze at the flame boundary. These are all indicators of fuel oil vaporization within the oil gun. This two-phase or vapor flow, through orifices originally designed for liquid flow, will severely reduce the oil gun capacity and stability.

Heavy oils that contain residual or added light oil will exhibit these same problems, as the light oils flash, with the added problems of heavy oil slug flow and resulting in smoke and poor atomization of the heavy ends.

2.7.14 Incorrect Positioning of the Oil Tip

By far, the most common oil-firing problem and the most detrimental condition to oil firing is the incorrect positioning of the oil tip with respect to its air supply/stabilization source. Due to variations in oil, atomizing medium, oil temperature, atomizing medium quality, burner airside pressure, operating oil/atomizing medium pressures, and furnace requirements for flame pattern, all John Zink Company oil guns are supplied with oil tip adjustability.

While the tip position is located in the John Zink Company burner assembly drawing supplied with every job, the final position is a field operator adjustment for optimum operation.

References

1. Chung, I. P., Colannino, J., and Strupp, C., Low NOx apparatus and methods for burning liquid and gaseous fuels, U.S. Patent 6,422,858 B1, issued July 23, 2002.
2. Chung, I. P., Karan, J., and Strupp, C., New low NOx oil-gas-combination burner for process heater applications, Presented at *2002 American Flame Research Committee, Spring Meeting*, Ottawa, Ontario, Canada.
3. Mullinger, P. J. and Chigier, N. A., The design and performance of internal mixing multijet twin fluid atomizers, *J. Inst. Fuel*, 47, 251–261, 1947.
4. Chin, J. S. and Lefebvre, A. H. The role of the heat-up period in fuel drop evaporation, *Int. J. Turbo Jet Eng*, 2, 315–325, 1985.
5. Lefebvre, A. H. *Gas Turbine Combustion*, 2nd edn., Taylor & Francis, New York, pp 47–48, 1999.
6. Barnes, R. H., Saxton, M. H., Barrett, R. E., and Levy, A., *Chemical Aspects of Afterburner Systems, EPA report EPA-600/7-79-096*, NTIS PB298465, p. 21, April 1979.
7. Williams, G. C., H. C. Hottel, and A. C. Morgan., The combustion of methane in a jet-mixed reactor, *12th Symposium (International) on Combustion*, The Combustion Institute, Pittsburgh, PA, p. 913, 1969.
8. Urban, D. L., Huey, S. P. C., and Dryer, F. L., Evaluation of the coke formation potential of residual fuels, *24th Symposium (International)*, The Combustion Institute, Pittsburgh, PA, pp. 1357–1364, 1992.
9. Smoot, L. D. and Smith, P. J., *Coal Combustion and Gasification*, Plenum Press, New York, pp. 81–90, 1985.
10. Bird, R. B., Stewart, W. E., and Lightfoot, E. N., *Transport Phenomena*, John Wiley & Sons, New York, 1960.
11. Westenberg, A. E., Kinetics of NO and CO in lean, premixed hydrocarbon-air flames, *Combustion Sci. Technol.*, 4, 59–64, 1971.
12. Bowman, C. T., Kinetics of pollution formation and destruction in combustion, *Progr. Energy Combust. Sci.*, 1, 33–45, 1975.

3

Burners and Combustion Systems for Industrial and Utility Boilers

Vladimir Lifshits

CONTENTS

3.1 Introduction

Burners for industrial and utility boilers form a large class of combustion devices designed to efficiently convert the chemical energy of the fuel into heat within the space provided by a boiler radiant section that is also referred to as the furnace. Further heat absorption occurs in the convection section. The combination of the radiant section and convection section substantially defines the boiler. Most modern boilers with burners as a specific part of the system were developed predominantly over the last century, whereas earlier boilers were firing fuels such as coal or wood on a grate placed under the steam- and water-carrying tubes in an enclosed space with some rudimentary means of combustion process intensification. The best legendary book on boilers with over 40 editions is published by the Babcock & Wilcox Company and is considered a bible for the boiler industry.[1] This book gives an excellent overview of boilers with an emphasis on utility boilers, but goes very little into the more specific field of burners and especially the modern gas- and oil-fired burners that are a focus of this chapter. Another very diverse book on the subject of industrial combustion published in 2001 gives a good overview of combustion technologies for boilers at that time.[2]

Two main factors that have driven the development of new industrial burners over the last few decades were the worldwide increase in the cost of fuel and the requirement and legislation for cleaner air. These modern standards are already applied over a substantial period of time required close to absolute completeness of the combustion process, minimum losses of heat within the exhaust flow, and minimum supporting energy requirements. Now more attention is being focused on the effective utilization of by-products and lower grade fuels in addition to the main conventional fuels like coal, oil, and natural gas. However, most of all, the need for reducing air pollution resulting from combustion was the main factor, driving the development of new burners. In many instances, these new requirements resulted in levels of pollutants and specifically NOx emissions entering the atmosphere that were approaching close to the theoretical limits of a particular process.

As boilers vary considerably, so must the burners. The description of various burner types in this chapter will more or less follow the variety of boiler types used by the modern industries as well as specific requirements to their operation. The following will be considered:

- Burners for package boilers generating process steam and high-pressure superheated steam
- Burners for field-erected boilers equipped with several burners
- Burners for wall-fired utility boilers
- Burners for tangentially fired (T-fired) boilers
- Burners used primarily for boiler warm-up

The combustion equipment for low NOx once through steam generators (OTSGs) used for enhanced oil recovery will be discussed. At the end of this chapter, examples of burners used in black and red liquor boilers of the paper industry and a new class of burners for reheating flue-gas upstream of selective catalytic reduction (SCR) equipment located in the boiler exhaust will be shown.

For many years, the main goal of a high-quality burner and especially an industrial size burner was to provide mixing between the fuel and the air that will be completed within the allocated space while providing ignition of the fuel/air mixture starting at an early stage in the mixing process smoothly propagating through the fuel/air mixture as it converts to products. This primary goal was to be accomplished without creating heat damage to the burner components and with typical preference to compact flames.

However, this classical approach often results in high levels of NOx formation. For example, using gaseous fuels, the oxidation process started from the fuel-rich limits of flammability generating high prompt NOx, followed by combustion with high flame temperatures close to those observed in intense stoichiometric flames generating high thermal NOx. For liquid fuels, the situation was somewhat similar except that the droplet evaporation phase preceded the bulk of the fuel oxidation. In order to reduce NOx formation without detrimental effects on the performance characteristics, the combustion process must be substantially modified. These modifications resulted in new low-NOx concepts applied to the industrial burner design and new low-NOx concepts (see Volume 1, Chapter 15) of the overall combustion process inside the furnace that, where possible, are applied together for achieving the desired performance. With some exceptions, these new burners and combustion systems overall became more complex and more demanding in terms of the required mechanical accuracy of all the components and sophistication of the combustion controls that were being developed concurrently with the new burners.

While low-NOx burners and combustion systems present a very substantial part of modern combustion equipment design, the field of industrial boiler burners is rather vast and has room for a variety of products not requiring any special means of NOx emission reduction. The examples of such equipment are auxiliary burners like warm-up burners that are not meant for extended continuous operation, burners firing fuels that naturally generate low NOx such as fuels with low heating values having a substantial quantity of non-oxidizing species, or systems where the focus of NOx control is shifted to post-combustion cleanup. This equipment can be highly specialized and may need to meet unique requirements other than the NOx performance parameters and will require considerable expertise to be properly designed and applied.

This chapter is structured to outline various combustion features of different boilers, goes into some details of applicable emission reduction techniques, and describes the design of different burners and associated performance characteristics.

3.2 Burner Design Fundamentals

As the capacity, fuels, and performance requirements for industrial combustion equipment vary widely, so must the burner designs. The only required common components of all industrial burners are those parts providing the desired air-flow shape and structure and fuel distribution components including multiple fuel injectors that may also be the same parts affecting the air flow. Refractory throats or quarls surrounding the base of the flame, while common for many burners, may not be a necessary attribute of some industrial burners. Many other components like the air plenum around the burner—the wind box—may be a component of the combustion system, but not a single burner. These modern burners are also equipped with many common components providing a means of flame ignition and monitoring.

The process of burner design is a complicated multistep and usually iterative process requiring deep understanding of different fundamental disciplines as explained in Refs. [3,4]. As for any burner, the design of an industrial burner starts from considerations of basic fluid dynamics principles applied for sizing the main burner components—air and fuel passages, geometrical characteristics of the flame-stabilization components like bluff bodies, and flow swirling registers or spinners. During this phase, the designer must make sure that the necessary quantities of air and fuel can be delivered to the combustion zone with the required distribution quality and target pressure losses on both the fuel and air sides of the burner. As a second step, the designer conceives the desired mixing patterns between the air and fuel and checks that the means of achieving them are adequate, and finally, he checks the patterns of flame propagation and stabilization. What often simplifies the process is when the designer can start with a known burner design and apply these principles by modifying the combustion device to the new conditions with respect, for example, to the changes in the fuel or design capacity or some other performance parameter.

Among the important characteristics to consider here would be some bulk characteristics like the swirl number for the flow and the ratio of momentums between the air flow and the fuel jets as described in several well-known references.[5,6] For an even deeper understanding of the issues, the designer may even need go into the field of chemical kinetics and turbulent mass transfer models and considering such criteria as Damköhler numbers defined as the ratio of the characteristic residence time in specific areas of the flame to the characteristic reaction time.

The combustion air to the burners is usually provided by a centrifugal fan that can be positioned locally or remotely to the burner. The combustion air is provided with enough pressure and mass flow to overcome the sum of the burner air-side pressure loss and the boiler convection section pressure loss, unless the boiler is of balanced draft type equipped with an induced draft fan. As the fan power (see Volume 2, Chapter 3) usage is a function of the burner differential pressure also sometimes called register draft loss (RDL), selecting a burner with a low RDL can produce a benefit by lowering fan horsepower. However, lower RDL burners can be physically bigger and may require bulkier fans and larger air ducts and plenums. Lower RDL burners generally also produce lower-intensity flames and have lower turndowns. Depending on the application, the burners are usually designed for a draft loss ranging at high fire from as little as 2 to as high as 12 in. W.C. (5–30 mbar) with the most common range being from 4 to 8 in. W.C. (10–20 mbar).

Most industrial sites have similar utilities that can be used by the burner. Gaseous fuels like natural gas are typically available with considerable pressure ranging from 10 psig (0.7 barg) to about 50 psig (~3.4 barg). Liquid fuels, if used, are delivered by pumps generating pressure as necessary for the type of selected atomizers: 100–200 psig (7–14 barg) when steam or compressed air-assist atomizers are used or to 300–1000 psig (20–70 barg) for mechanical atomization. The steam for atomization is usually available with pressures of 100–250 psig (7–17 barg). If compressed air is used, its typical available pressure range is about 15 psig (1.0 barg) or ~100 to 150 psig (7–10 barg). A good burner design would effectively use most of the available mechanical energy of the utilities to achieve the desired combustion results instead of excessively wasting energy in the control components.

The process of modern burner design or burner selection also requires in-depth understanding of the origination of different pollutant formations and the available means of its reduction described in this chapter. Modern modeling tools like computational fluid dynamics (CFD) software packages have reached a high level of sophistication and allow with some accuracy to simulate some simple combustion problems including detailed modeling of the burner performance as described in Volume 1, Chapter 13. The more advanced CFD packages also include some amount of chemical kinetics to approximate modeling of the formation of CO and NOx. Some companies have also developed their own codes for estimating NOx emissions in a postprocessor using the modeling results by commercial codes. The successful use of CFD and additional tools requires knowing their limitations and considerable background in engineering and physics without which the CFD results may deviate greatly from reality or miss important phenomena of the process.[3]

While predominantly operating on gaseous fuels due to its currently comparatively lower cost, many industrial boiler burners are designed for having oil firing capabilities as a backup. A few sites still operate on oil where gas is not available. Liquid fuels are burned in industrial burners using fuel atomizers described in Chapter 2. The combustion process of atomized fuel is very different from gas and difficult to accurately model even with modern CFD packages. The droplets are formed in different sizes, and the characteristics of the droplet size distribution are seldom well defined. Even with the use of advanced particle-size instrumentation, the design of the atomizer specific to a combustion process contains a substantial degree of proprietary know-how.

It is also important to understand that as the burner operates it interacts with several dynamic systems. These interactions take place on several levels. The first one is obvious—the flame behavior depends on the configuration of the space available for the flame and the temperature of the surrounding walls confining the flame. The second level is the configuration of the supply and the distribution of combustion air or combustion air mixed with some amount of flue gas. The third level is acoustical interactions with the furnace space, the convection part of the system, and the exhaust as well as the wind box and the air delivery system. These interactions may be rather complicated and difficult to anticipate. Overlooking the possibility of these interactions can result in difficult to solve problems discovered only during the burner commissioning and should be addressed as a complete system issue during the design phase of any project.

There are several types of pollutant emissions that are affected by the burners and not just by the type of fuel or minor species that may be present in the combustion air such as NOx emissions, carbon monoxide (CO) emissions, unburned hydrocarbons (UHC), and particulate matter (PM). These pollution emissions are discussed in Volume 1, Chapter 14. In general, well-designed combustion devices generating a flame inside an appropriately sized volume can easily achieve negligible amounts of CO and UHC emissions. The organic portion of PM related to the incomplete oxidation of liquid fuels can also be easily reduced to insignificant levels with good-quality fuel atomization. The control of NOx emissions can be difficult and will be an important parameter when considering different burner designs and combustion solutions in this chapter.

3.3 Techniques for Reducing NOx Formation

Two molecules, nitrogen oxide (NO) and nitrogen dioxide (NO_2) together constitute the definition of NOx. During the combustion process, most of the NOx formed will be NO. In the atmosphere, the NO is then oxidized to NO_2 in the presence of sunlight and certain hydrocarbons. When computing NOx emissions on a weight basis, all the NOx is considered as NO_2. When measured out of a combustion chamber, the instrument should measure both NO_2 and NO as parts per million dry volume (ppmdv).

As described in Volume 1, Chapter 15, there are three generally recognized significant sources of NOx emissions formed during the combustion process:

- Thermal NOx, formation which was first described by Zeldovich,[7] comes from oxidation of atmospheric nitrogen (N_2) to NO inside the areas of the flame where temperatures are developed in excess of about 2600°F (1400°C).
- Prompt NOx experimentally discovered first by Fenimore[8] is also the result of atmospheric nitrogen oxidation formed predominantly in the initial parts of the flame via reaction with fuel radicals where local stoichiometric conditions are fuel rich.
- NOx originated from oxidation of nitrogen present in the fuel in several forms other than N_2 is also called fuel bound nitrogen (FBN).

An approximate theoretical relation for thermal-formed NO as a function of temperature is shown in Figure 3.1. The chart takes into account only the forward-going reactions of nitrogen oxidation that in boilers are almost always the case. It shows about 10 times increase in the NO formation rate with about 220°F (~120°C) increase in temperature. However, this steep increase in the NO with temperature never directly translates to the relation between the boiler effluent NOx and the peak theoretical flame temperature, also called the adiabatic flame temperature (AFT). The explanation of this is that the temperatures close to the AFT occur in the flame in only small areas at low residence times due to heat transfer from the flame to the surrounding lower-temperature gas and combustion chamber walls and the rate of heat dissipation from these areas is greatly accelerated with the temperature. Furthermore, some experimental NOx data presented in Section 3.3.4 indicate a much lower dependence of NO formation of with temperature that is difficult to explain on the basis of known kinetic reactions.

If uncontrolled, the concentration of thermally produced NOx in medium-size boilers can reach the level of a few hundred parts per million (ppm) and higher if preheated air or oxygen enriched air is used. In large boilers, the uncontrolled NOx could be much higher reaching 500–1200 ppm level and more.[9,10]

When burning conventional fossil fuels, the production of prompt NOx is typically considered in the range of 10–15 ppm. This number, however, is very approximate as the formation of prompt NOx also depends on the peak flame temperature and there is no easy experimental way to separate it from the thermal NOx and produce data for validation of theoretical models. The control of prompt NOx is only considered after the

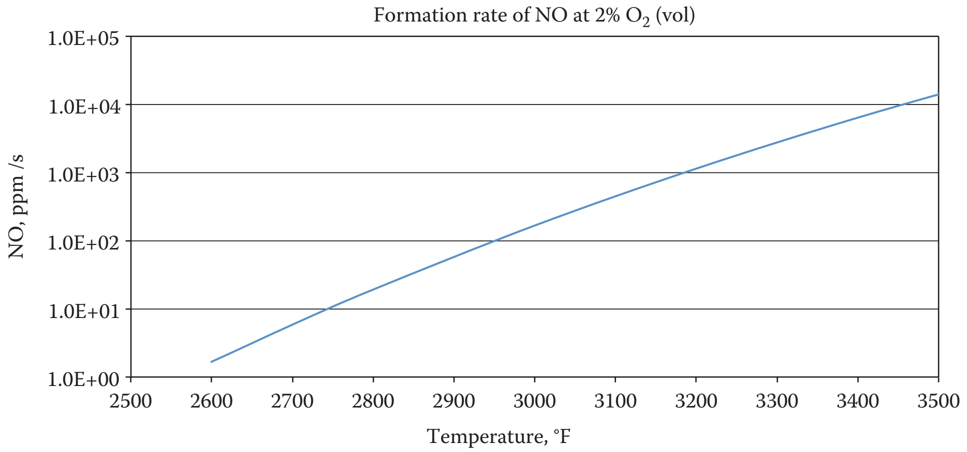


FIGURE 3.1
Approximate rate of thermal (Zeldovich) NO formation.

thermal NOx is already reduced to a very low level. Utilizing lean premixed burners accomplishes control of both thermal and prompt NOx.

FBN is mostly a concern in fuel oils and biofuels. Gaseous fuels on occasion may contain ammonia or other elemental nitrogen compounds that at medium temperatures readily oxidize to NOx. The percent conversion of FBN to NOx depends on many factors: specific molecules with nitrogen-carrying radicals, rate of release of species with nitrogen into the gas phase, and amount of FBN in the fuel. A conversion rate of 100% of FBN present in fuel oil in the amount of 0.1% by weight would result in about 130 ppm of NOx in a low-oxygen exhaust. Number 2 Oil typically contains from 0.01% to 0.05% of FBN[11] with the rate of conversion to NOx of 40%–75%. Heavy oil usually contains between 0.3% and 0.6% of FBN with a conversion to NOx in the range of 30%–50%. The operation of the burners with minimum overall excess air and/or using air staging explained in Section 3.3.2 reduces the conversion of FBN to NOx.

Extensive literature exists on the subject of NOx formation and control for those who want to study deeper into the subject.

All practical ways of reducing NOx formation during combustion that are described as follows complicate the combustion process and make it more susceptible to increases in other emissions (CO, UHC, PM). So the art of delivering lowest NOx combustion often becomes a balancing act of designing the process to lower NOx with minimum trade-offs to the increase of other pollutants.

3.3.1 Flue-Gas Recirculation and Injection of Steam into the Flame

The peak flame temperature and thus thermal NOx can be effectively reduced when the initial reactants—fuel or combustion air—are mixed with non-reacting gases like atmospheric nitrogen, carbon dioxide, or water vapor introduced into the combustion zone at temperatures substantially below the temperature of the flame. As all the aforementioned inerts for the combustion gases are present in the combustion products, the recirculation of cooled flue gas back to the burner (flue-gas recirculation [FGR]) becomes a powerful way of controlling thermal NOx.

The quantity of FGR is usually defined as the ratio of the mass of recirculating FGR—$M_{\text{flue gas}}$—coming from the same combustion process to the mass of combustion products exiting the stack M_{fgr}:

$$\text{FGR} = \frac{M_{\text{fgr}}}{M_{\text{flue gas}}} \tag{3.1}$$

A system designer needs to be careful in sizing the FGR delivery system if the outgoing flue gas is diluted with combustion air (e.g., leakages in the air heaters) or taken from another process that may produce flue gas with a different composition.

As a generic heat balance equations will show, the recirculation of a typical flue gas in the amount of 10% results in 6%–8% reduction in the theoretical peak flame temperature (AFT). Relations between NOx and FGR for several specific cases when firing fuel without FBN are shown on Figure 3.2.[12]

The same effect can also be achieved when steam or finely dispersed water is injected directly into the flame, or mixed with combustion air or fuel. As FGR is almost always available, the monetary penalty of using FGR for NOx control is mostly associated with the cost of mechanically moving FGR back to the burner and is usually insignificant compared to a change in any heat losses with the exhaust due to a slight increase in outgoing flue-gas temperature. For retrofit applications, the increase in stack temperature and thermal losses is on the order of 10°F (6°C) per 10% of FGR depending on the design of the boiler convection section. This rule of thumb was determined by observations and

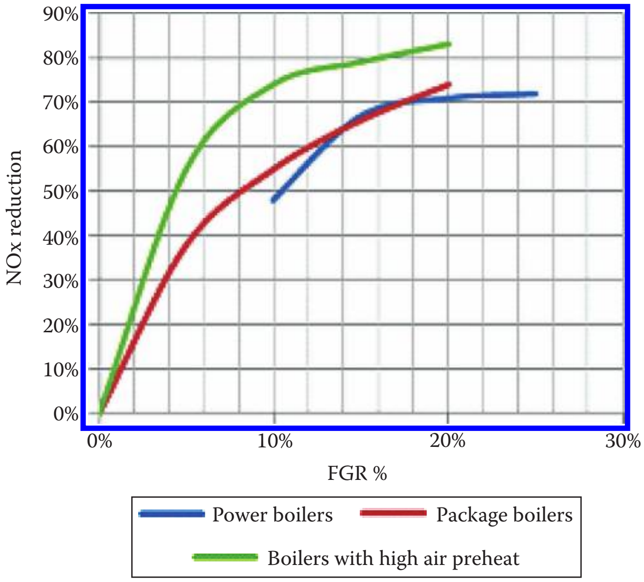


FIGURE 3.2
Approximate relation of NOx reduction with FGR for fuels without FBN. (Data for power boilers are per Rawdon, A.H. and Johnson, S.A., Application of NOx control technology to power boilers, Presented at the *American Power Conference*, May 10, 1973; other data are from several COEN applications.)

calculations of the heat transfer inside the furnace and the convection section. For comparison, a 10°F (6°C) increase in stack temperature reduces boiler thermal efficiency by about 0.25%.

The extent that thermal NOx can be controlled with FGR strongly depends on the ability of the burner to remain stable and provide sufficient completeness of combustion with the rate of combustion reactions reduced due to reduced temperatures. This issue will be discussed further in this chapter for specific types of burners.

The addition of FGR to the combustion air prior to the forced draft fan demands a substantial increase in the fan capacity. The increase depends on whether the boiler and the burner were originally designed for operation with FGR, Case 1, or the FGR is a retrofit addition to the system, Case 2.

Using fan laws (see Volume 2, Chapter 3), it is easy to estimate factors for the fan power increase prior to making the selection of the specific equipment or determining if the existing fans will work. In the first case, the power of the fan for the same boiler capacity needs to be increased at least about linearly with the amount of FGR as defined by the approximate ratio shown in Equation 3.2:

$$\frac{P_{\text{fan}}\text{FGR}}{P_{\text{fan}}0} = 1.1 + 1.08 * \left(\frac{\%\text{FGR}}{100\%}\right) * \frac{T_{\text{fgr}}}{T_{\text{amb}}} \tag{3.2}$$

In *Case 2*, the power increase factor with FGR is much more substantial:

$$\frac{P_{\text{fan}}\text{FGR}}{P_{\text{fan}}0} = 0.1 + \left[1 + 1.08 * \left(\frac{\%\text{FGR}}{100\%}\right) * \frac{T_{\text{fgr}}}{T_{\text{amb}}}\right]^{n} \tag{3.3}$$

where the power factor n varies between 2.3 and 2.8 depending on the relative hydraulic resistance of the burner to the boiler convection section with n being lower for boilers with convection section pressure drops much lower than the burner resistance. In the aforementioned equations, the temperatures of FGR and air passing through the fan—T_{fgr} and T_{amb}—are absolute temperatures.

The equations given above are rough references and approximately account for the increased pressure drop across the burner and the boiler convection section as well as additional pressure losses for inducing the flue gas. For any particular application, the requirements of the fan characteristics can be accurately determined based on fundamental considerations of the pressure drop characteristics of the boiler and selected burner.

FGR can also be added to the combustion air using a separate FGR fan. This "forced FGR" can be mixed with air with the use of a custom-designed mixing device or delivered to the burner exiting cross section through a plenum with ports. In the latter case, the FGR can be strategically directed to the parts of the flame where it is most effective. The method is also called selective FGR—the term that is contrary to the forced or induced FGR uniformly (or in bulk) mixed with combustion air.

There are pro and cons with either way of introducing the FGR. In retrofits, the forced FGR may allow a better chance of reusing the existing combustion air fan, but would require a more complicated FGR ducting arrangement and more complicated combustion controls (see also Volume 2, Chapter 2). Some burners, described later in this chapter, are also better suited for operation with forced FGR if delivered directly to the burner. Utilizing induced FGR, consideration must be given to additional water and sulfuric acid condensation and corrosion problems that are better contained when delivering forced FGR to the burner. The benefit of induced FGR is its simplicity.

In some cases, the flue gas can be mixed with a gaseous fuel. For a given percentage of FGR, this method potentially achieves somewhat better NOx reduction than the FGR mixed with combustion air. This can be explained by concentrating the FGR in the high-temperature combustion zone rather than spending it partially on the dilution of the excess combustion air initially bypassing the combustion zone. Except for a few specific cases, the technique is difficult to use.

It requires specially designed burners and also complicates the controls and safety system.

A small amount of FGR can be mixed inside the burner with a gaseous fuel delivered at high pressure using the fuel gas mechanical energy to aspirate the flue gases from the boiler exhaust and pass the fuel/FGR blend through the fuel injectors, also called fuel-induced recirculation (FIR). FIR works if the burner is designed for the high fuel supply pressure usually in excess of 20 psig (1.3 barg) and with burners designed for low-pressure injectors of the fuel–flue gas blend into the combustion zone. In other cases, an FGR fan with substantial static pressure is needed. The levels of NOx reduction with this technique may be quite substantial—up to about 80%–85%.

Contrary to the low operating cost of using flue gas to control NOx, the use of steam or water injection for NOx control carries substantial penalties for the efficiency of the boiler or combustion system. The amount of steam that is equivalent to 10% FGR would be equal to about 4.5% of the boiler steam output—an unacceptable loss of the system efficiency as well as capacity if the produced steam is used. In some cases, a low-pressure waste steam is available and can be used for NOx control with minimal capital costs and operating cost as waste steam has minimum impact on the efficiency. It may also be justifiable to use steam when it is reserved for some infrequent occasions of peak power production in a very tight regulatory environment. Some modern burners can utilize steam much more effectively by using selective injection points rather than just injecting it into the air with an increase of its effectiveness by a factor of two or more. However, this method usually involves modifications to the burner negating to some degree the advantage of its low capital costs.

Water injection into the flame is even more problematic than the injection of steam. First of all, water needs to be very finely atomized to below 100 μm diameter particles to be effective, and that requires using compressed air and special atomizing nozzles. Secondly, its injection has to be properly distributed throughout the flame body, but in a way not to impact flame stability. Lastly, all this water in the flame will be converted to water vapor again substantially degrading the overall efficiency of the process on the order of ~50% to 75% of the efficiency loss with the injection of produced steam.

The ultimate ability of mixing air with inerts for combustion gases depends on the burner. Conventional burners designed for the use of FGR can utilize up to 20% FGR when firing natural gas and a somewhat lesser amount up to ~15% when firing oil and achieve anywhere from 40% to 75% reduction in NOx when compared with the baseline (no FGR) level. As will be discussed further in this chapter, advanced burner designs can utilize much more FGR.

3.3.2 Air Staging

Thermal NOx may also be reduced by burning fuel under local conditions substantially different from stoichiometric such as lean and rich when combustion does not generate high NOx-forming temperatures.

Using conventional combustion equipment, the fuel burning typically starts after engaging only a partial amount of air well below the stoichiometric amount. The balance of the air is added to the combustion after the products of combustion lose a substantial amount of heat. Thus, the initial rich section of the flame and the lean final part of the flame are at lower temperatures due to prior heat transfer from the rich section. The result is a reduction in the peak flame temperature over the entire flame zone and a reduction in thermal NOx. When the burner is designed to enhance this described effect, the technique is often called air staging. For the concept to work, it is essential for the products of the initial rich flame to lose a substantial amount of heat before the rest of the air being introduced.

When firing fuels containing FBN, the use of air staging also helps to reduce conversion of FBN to NOx as the release of fuel nitrogen is taking place in a low-oxygen environment. The use of air-staged combustion usually generates flames with increased volume. Air-staging implementation and potential effectiveness depends on the space available for the fuel oxidation to be completed. Possible problems include increased CO and UHC emissions and potential flame stability problems as the burner becomes more susceptible to the deviations from optimum operating conditions.

Practically, air staging can be achieved with redistribution of the fuel within a burner by creating a very coarse mixing pattern using fewer fuel injectors, injecting fuel into the flame zone with low momentum or directing fuel to mix differently in specific flow streams. Multiple air passages and some means of controlling air distribution to those passages can also be used to achieve the desired effects of air staging. These techniques were the first embraced by early combustion technologies for controlling the NOx as it required relatively minor changes to the combustion systems. On the air side, the staging can be achieved by imposing minimum intensity swirling motion (generating flows with low swirl numbers as defined by Ref. [5]) or by introducing a portion of the air through ports spaced away from the burner.

Air staging alone can typically reduce the NOx by 20%–60%. Any deeper reduction may become problematic and not reliable. Air-staged burners are usually poor candidates for operation with FGR as the combination of fuel-rich conditions with reduced oxygen results in marginal flammability for most hydrocarbon fuels.

3.3.3 Fuel Staging

Fuel staging requires injecting a portion of the fuel away from or completely out of the flow of the combustion air. This portion of the fuel is termed staged fuel or a secondary fuel, while the other portion is referred to as primary fuel. For the fuel-staging technique to be effective, the secondary fuel needs to enter the combustion zone at some distance from the burner exit cross section after the flame created by the primary fuel loses a substantial amount of heat by radiation and convection to the boiler water walls. The staged fuel may also be injected in a way to bring into the combustion zone products of combustion from the furnace volume surrounding the flame. When this volume contains low-oxygen gas at a temperature much lower than the flame, the effect of the furnace gas re-entrainment into the flame on the NOx will be somewhat similar to the use of selective FGR.

Burners with fuel staging are quite different from conventional burners and burners with air staging as fuel-staged burners are equipped with fuel injectors external to the combustion air passage. Fuel-staged burners, if improperly designed, may be susceptible to partial loss of flame and combustion instabilities as the location of the flame front and ignition areas for the staged fuel may be shifting by the highly turbulent combustion process. Fuel-staged burners can reduce NOx emissions by up to 50% and can be designed to accept moderate amounts of FGR for a deeper NOx reduction up to ~80% overall.

3.3.4 Lean Premixed Combustion

Lean premixed combustion applied to natural gas can reduce prompt NOx to below 2 ppm. With high excess air or FGR, the technique is capable of reducing thermal NOx almost to the level of prompt NOx formation. Lean premixing technology was developed to reduce prompt NOx and operate with higher levels of FGR at the same time.

The simple delivery of combustion air to the burner in excess of the stoichiometric amount alone does not make the combustion process fuel lean. As the fuel is injected into the air flow, the mixing process always generates fuel-rich mixtures first, and that is where combustion easily starts with any source of ignition. In order to achieve fuel oxidation under lean conditions throughout the flame, the fuel and the air need to be premixed to a fuel lean blend prior to ignition. Premixing of the fuel and air upstream of the burner for industrial size burners would require large volumes of flammable mixture that would be very difficult to handle with adequate safety and would present significant issues with respect to controlling any leakages. So, in large burners, the premixing needs to be accomplished by the burner. To differentiate large premix burners from small burners where premixing may take place in the duct or a pipe upstream of the burner, large premix burners are designed differently and are sometimes called simulated premixed combustion burners.

In order to achieve the benefits of thermal NOx reduction by premixed combustion over the alternative ways of controlling NOx, the mixture of fuel and air requires substantial excess air and/or inert gas like FGR. When firing natural gas, for example, the substantial benefits of premixed combustion on the NOx are realized only if the excess air levels are over ~40%. In combination with bulk mixed air and FGR, the excess air may be reduced to "normal" levels of 10%–15% or even less with full benefits of the lean premixed combustion.

Turbulent premixed flames are prone to generating intense combustion instabilities with frequencies defined by the burner and the overall system. Only specially designed burners with some of them described further in this chapter, along with accurate controls, allow satisfactory operation on a large industrial scale with significant turndown. The key for reducing premix flames instabilities is diffusing—stretching the flame front either with the increased velocity and turbulence of the flow or with the reduction of the oxidation kinetics with high amounts of excess air or flue gas. Some other techniques incorporated into the burner design may help to improve operation with desynchronizing the combustion instabilities developed in different parts of the flame.[13]

Premixed burners are typically designed for operation with NOx emissions below 15 ppm (corr. 3% stack O_2, dry) with the most advanced boiler burners currently operating in selected equipment with sub 5 ppm NOx and low excess air. A typical relation between the rates of FGR, excess air, and NOx for a medium-size package boiler equipped with a premixed burner is shown on Figure 3.3[13] with a series of iso-NOx curves.

When lean premixed combustion is substantially diluted with FGR or excess air to the level below 10 ppm, the effect of the NOx formation beyond the flame zone becomes significant. Figure 3.4 shows NOx emissions measured when firing lean premixed burners with a substantial amount of FGR and about 20% excess air recorded during the commissioning of QLA burners in 1999.[14] The fuel in this case was natural gas and the combustion air was preheated to 425°F (218°C). The 60,000 lb/h (27,000 kg/h) CE boiler furnace was sized for about 1 s residence time and had ~50% refractory coverage of the flame surrounding walls. The two burners were generating relatively short flames not reaching the furnace back wall. NOx, CO, O_2, and combustible emissions were measured in proximity to the boiler back wall at locations about 11 ft (3 m) opposite to the burners and at the stack using well-calibrated and accurate instruments. The NOx was measured using a

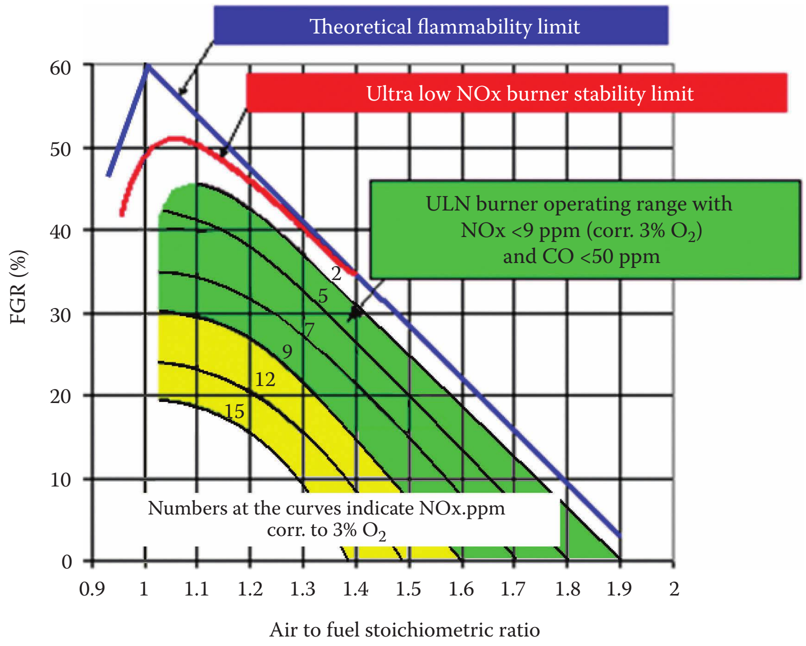


FIGURE 3.3
NOx performance of a typical Coen premix burner firing natural gas in a package boiler. The numbers on the chart are NOx emissions at 3% O_2 (dry). (From Lifshits, V., Ultra low NOx burner with widened stability limits, *AFRC/JFRC International Symposium*, Maui, Hawaii, 1998.)

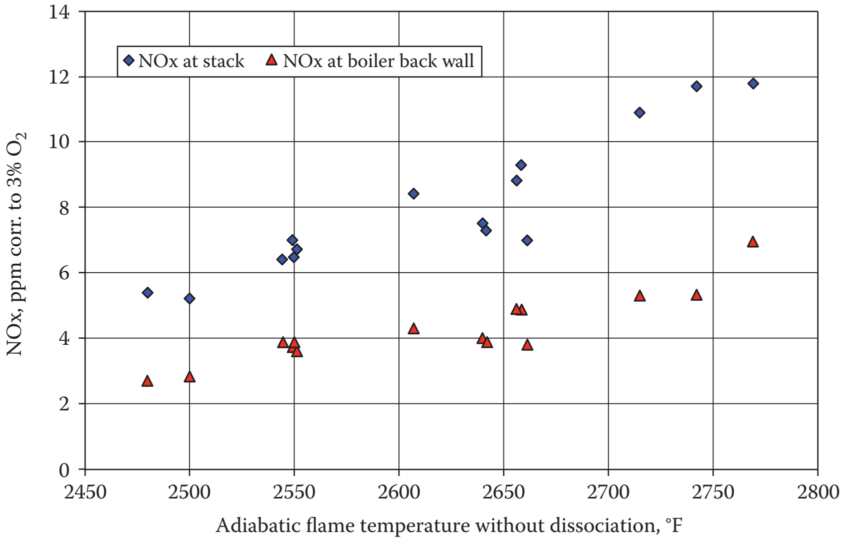


FIGURE 3.4
QLA burner performance at high fire.

chemiluminescent NOx analyzer. An interesting fact here is that almost 50% of the NOx ranging from 3 to 5 ppm was produced in the post-flame volume of the furnace after the combustion was complete as proven by very low-CO readings at the back of the furnace. Similar data of post-flame production of NOx on the order of 1.5–2 ppm were also recorded when testing lean premixed burners at the company test facility in a furnace with minimum refractory coverage. This level of post-flame NOx production is impossible to explain based only on the Zeldovich NOx rate approximately shown by Figure 3.1. One possible explanation to the observed effect suggested is the conversion of some amount of N_2O in the post-flame zone to NO. The formation of N_2O during the combustion tests is seldom measured as this species is not part of the NOx that is regulated.

No practical concepts exist yet for creating the equivalent of premixed combustion for fuels delivered to a burner in a liquid form.

3.3.5 Furnace Gas Circulation and Combining Different Techniques

This technique of NOx reduction uses the burner-created aerodynamic effects to return within the furnace a portion of combustion products that already lost some heat to the walls back into the combustion zone. The staged fuel jets as described earlier can create this effect. The desired patterns of furnace gas circulation (FGC) can also be promoted by the combustion air momentum. The technique is always used in combination with fuel staging achieving NOx reduction in some cases up to 50%–70% when firing natural gas and somewhat lesser amount ~20% when firing light oil with low nitrogen content. The difficulty of applying the FGC technique is the dependence of its effectiveness on the furnace geometry. Deeper NOx reductions can be achieved when burner technology effectively utilizes a combination of fuel staging, FGC, FGR, and elements of premixed combustion together approaching or exceeding the best performance of the premixed combustion devices.

3.4 Burners for Package Boilers

Package boilers by definition are substantially prefabricated and shop-assembled systems with a maximum size allowing shipment by some common means of transportation and with a minimum field installation cost. With rare exceptions, these boilers are designed for a single burner. The thermal capacity of burners varies from about 30×10^6 to 350×10^6 Btu/h (~9 to 100 MW) that corresponds to about 25,000 to 250,000 lb/h (11 to 113 metric-ton/h) of steam production. There are several package boiler design configurations with respect to the layout of the drums, tubes, and radiant and convection section of the boiler. Package boilers may produce a relatively low-pressure saturated steam at 150–250 psig (10–17 barg) pressure or high-pressure superheated steam of about 600–1500 psig (~40 to 100 barg) for driving steam turbines.

These boilers are typically designed for burning different gaseous fuels, oil, and on occasion some fine coal or biomass. The radiant part of the boiler has a close to parallelepiped shape with the width varying from 5.5 ft to about 12 ft (1.5 to 3 m), the height varying from 7 to 12 ft (2 to 3.5 m), and the length of 15 to 40 ft (4.5 to 12 m). In modern boilers, the furnace walls and their tops are usually formed by sheets of tubes each with two fins welded together forming a continuous membrane wall. Older types of boilers were designed with walls formed by the tubes placed side by side without gaps and not connected by welding—tangent tubes walls.

The furnace space heat release at capacity varies from ~40,000 to ~120,000 Btu/(h-ft^3) (400 to 1200 kW/m^3) with the lower numbers more characteristic for small boilers. These numbers can be approximately converted to an average residence time for the species passing through the furnace in the range from 0.5 to 1.0 s when fired at capacity. However, the actual residence time of a substantial portion of the flow moving through the furnace will be much higher than average velocities resulting in lower residence times for this portion. The heat release per the unit of furnace cross section varies from 0.65 to 3.0×10^6 Btu/(h-ft^2) (about 2–6 MW/m^2). Boiler furnaces usually operate at positive pressure defined by the hydraulic resistance of the boiler convection section including the economizer. The range of furnace pressure at peak loads varies from as little as 3 in. W.C. (0.75 kPa) to as high as 1 psig (7 kPa).

Package boilers are also widely used in marine applications. Marine package boilers are usually more compact than stationary boilers, fire only oil, and require very compact flames. The aforementioned considerations are important when determining the characteristic residence time of species inside the flame affecting the maximum allowable size of the evaporating fuel droplets.

Knowing the residence time and furnace pressure is also important for estimating the kinetic/mixing energy necessary to be transferred by the burner to the reacting gases entering the combustion zone.

Knowing the type of furnace walls, membrane or tangent type, has some relevance to the type of burner to be selected and regime of its operation. Tangent wall boilers are prone to short circuiting—the effects of a small portion of furnace gas surrounding the flame to pass between the tangent tubes into the convection section carrying with it a small amount of unburned fuel (UHC) and a CO. Controlling the UHC and CO emissions from tangent tube boilers often requires operation with elevated levels of excess air helping to reduce the concentration of UHC and CO reaching the tangent tube walls and selection of burners generating more compact flames—typically not very low-NOx burners unless of the premixed combustion type.

Firing into the relatively limited space of a package boiler often raises concerns regarding actual or perceived flame impingement on the boiler walls. The issue is important here because it is somewhat subjective and often a source of debate when something goes wrong. The impingement is really defined only by its consequences.

The main concern of flame impingement is creating such a high heat flux onto the tube surface that the boiling process inside the tube starts developing a film of steam on the internal tube surface impeding tube cooling with otherwise two-phase steam-water flow. In boilers with properly designed water circulation and good water quality, however, a soft impingement of visible flame tails onto the tubes does not create a strong enough heat flux that causes any problems.

The other and more often occurring result of the flame being in contact with the tubes is soot deposition on the walls. The soot can be formed when products of incomplete combustion containing carbon atoms or aggregates get in contact with the relatively low-temperature tubes. The buildup of soot on the surfaces is usually limited as soot has a low thermal conductivity, so the conduction heat transfer is low and that the surface of a thick soot layer acquires a higher temperature that causes the soot to oxidize. The effect, however, is still unwanted as it will interfere with the overall heat transfer from the flame to water the tubes and may create an accumulation of soot that will fall off onto the furnace floor.

The situation is different when tubes of the superheater are experiencing impingement. The superheated steam does not have the same cooling intensity as the two-phase flow even when it moves with high velocity. The superheater tubes may actually overheat even without flame impingement by fully oxidized high-temperature combustion products that still emit substantial thermal radiation that visually can be mistaken as flame impingement. It is more often that burners making longer flames or burners operating with FGR (higher mass flow flames) will generate higher temperatures in the area of the superheater at the back of the furnace. Burner suppliers need to be aware if there are some specific requirements for the temperature in the area of the superheater and whether the boiler is equipped with the proper means of controlling it. Problems like this usually happen when an old burner needs to be replaced with a new low-NOx burner using FGR or when a new burner generates a more transparent (lower luminosity) flame.

3.4.1 Conventional Round Burners and Burners with Air Staging

Round burners are the most common burners historically used in package boilers. The term refers to the burners with a predominantly round shape of the combustion air passage or several concentric air passages for combustion air through the burner. The burners are typically equipped with one or two sets of adjustable or fixed register louvers placed at the air inlet into the burner for imposing a swirling motion to the combustion air and a bluff body or another swirler/spinner placed at the burner centerline close to its exiting plane. With all these features, the air-flow pattern through the burner can be described predominantly as 2D, with all velocity vectors a function of only radial and axial coordinates, but not the angular/circumference coordinate. The presence of the louvers or spinner blades, or slots in the flame-stabilizing shields, creates some patterns of periodic disturbance over the flow circumference, but these patterns are usually weak, fast dissipating, and not always essential for the burner operation.

The very first round burner designs had a predominantly uniform distribution of fuel over the burner circumference and were designed for high stability and flame intensity. The burners trace their lineage back to the days of ship-mounted boilers, where boilers had to be kept as small and light as possible, thus making the requirement for short flames very important. Without these boilers in operation, a ship could be left dead in the water resulting in the need for the burners to reliably operate regardless of changes to the fuel or air supply. The maneuvering demands of the ship also require that these burners be able to operate across a wide range of firing rates and to be able to operate within a wide range of air to fuel ratios essential for maintaining good flame stability even with a very crude setting of single-point positioning combustion controls and deviations in fuel pressure, etc., explained in Volume 2, Chapter 2.

To accomplish all of these goals, the design resulted in burners that mix air and fuel and ignite the mixture as rapidly as possible to create a highly stable and compact flame. A strong swirling motion imposed on the flow facilitated the mixing and compactness of the flame. For operation on gaseous fuels, the fuel could be injected through a gas ring surrounding the burner flow exiting cross section at the entrance to the flared refractory throat. The gas rings were drilled with a series of holes grouped in small clusters injecting fuel in the direction of about 45°–60° to the burner centerline. Other gas injection methods were also used. For oil firing, the burner was equipped with an atomizer placed through the center of the flame-stabilizing shield.

Register-type swirling devices and some spinners often had a tendency to produce an overly excessive swirling motion at the burner centerline and starve this region on air or create backflows. This effect was undesirable, especially for oil firing when fuel was injected from the point close to the burner centerline causing oil gun coking and other problems. To avoid the effects of over-swirling, some burners were designed with a small air zone surrounding the atomizer that did not have a swirling motion.

The overall swirling and burning of the fuel–air mixture typically exits the burner through a conical refractory throat with a length of about 50% of the burner diameter. This divergent throat helps to further facilitate the development of a large-size recirculation zone in the wake of the spinner delivering hot combustion products to the area of ignition at a location close to the fuel injection. The glowing parts of the throat also provided a source of ignition for the peripheral parts of the flow.

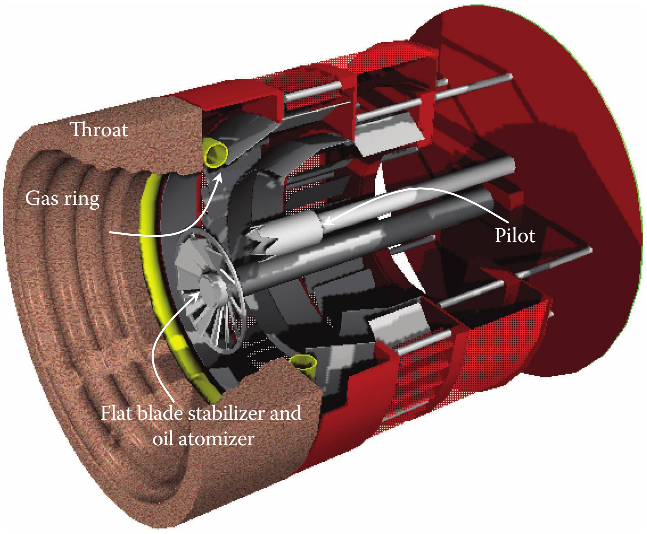


FIGURE 3.5
Coen *DAZ*™ burner.

The Coen *DAZ*™ burner—dual air zones burner schematically shown on Figure 3.5—is an example of such a burner. The burner has two sets of adjustable register-type louvers typically set for spinning the flow in opposite directions, achieving very intense stable flames of the necessary shape, operating with a wide range of air–fuel ratios, and making it a perfect burner at the time for many applications including marine and package boilers. When the burner was not in use, both sets of louvers could be closed to reduce the draft through the burner.

Applying the burner to stationary boilers firing natural gas with reduced NOx requirements was found to be difficult as the burner generates relatively high NOx emissions of about 70–160 ppm (ref. 3% O_2 dry) on natural gas and its ability to tolerate FGR was very poor. The problem stemmed from the pattern of gas injection from the periphery of the air flow.

A new class of low-NOx round burners appeared then as a first response to the demand of simplifying the burner design that also allowed operation with reduced NOx emissions on gas and oil in boilers. These burners were equipped with multiple fuel gas injectors positioned around the spinner or a bluff body providing patterns of fuel gas injection beneficial for operation with FGR or patterns delivering delayed mixing with air—air staging. The following are a few examples of modern air-staged low-NOx burners for package boilers designed to fire both gaseous and liquid fuels.

A *Variflame*™ burner shown in Figure 3.6 features a venturi-shaped passage for the air flow, providing a well-balanced air-flow pattern just upstream of the combustion zone and two sets of gas fuel injectors connected to the same fuel plenum. The first smaller set delivers fuel directly into the wake of a large spinner developing where the overall flame is stabilized. The second main set placed adjacent to the spinner provides fast initial mixing of fuel and air at each injection point and delayed mixing further in the main body of the flame. The number of main fuel injectors varies from 3 to 8 depending on the application. Another feature of this burner is that a small portion of combustion air can be supplied around the venturi passage. The amount of this flow can be adjusted. This flow diversion around the venturi has a strong effect on the burner draft loss and the overall shape of the flame. The burner is simple and efficient and can be optimized for operation with FGR.

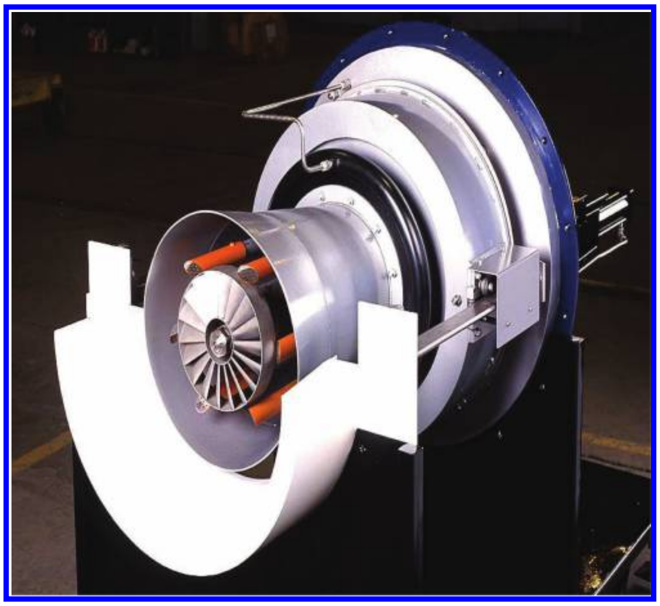


FIGURE 3.6
Coen *Variflame*™ burner.

Figure 3.7 shows another air-staged burner with both types of fuel injectors placed around the spinner—the

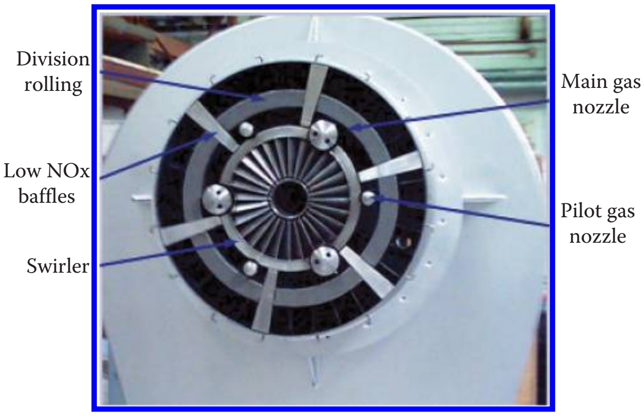


FIGURE 3.7
Hamworthy *DFL*® burner.

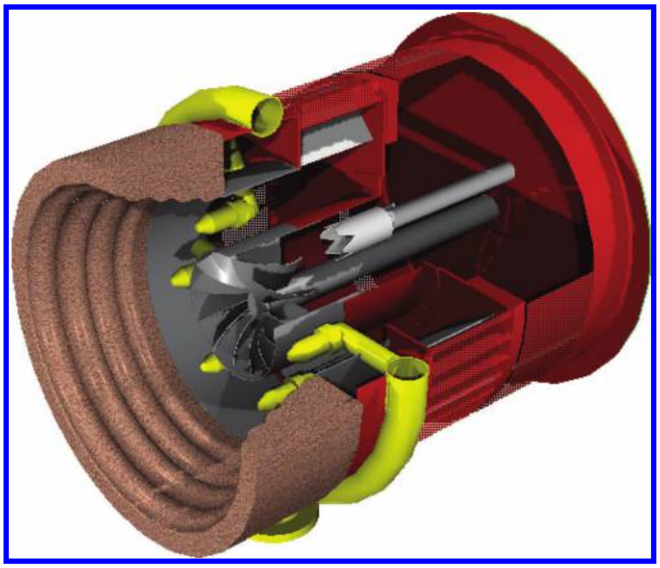


FIGURE 3.8
Coen *DAF*™ burner.

DFL® burner. On the air side, the burner design includes multiple concentric passages some with the fixed turning vanes and a concentric bluff body filled with ceramic. When properly applied, the burner generates low baseline NOx and is capable of operating with low excess air and low CO emissions.

Figure 3.8 shows a picture of one of the most versatile air-staged burners for a package boiler—the Coen *DAF*™ burner. The burner is mechanically more complicated than the Variflame burner. It has two major air passages and the ability for external adjustments to the air distribution between the passages. It has a set of externally adjustable louvers—register—that control the swirling motion imposed onto the air flow in either the clockwise or the counterclockwise direction. A set of fuel injectors around the spinner is connected to the external or internal gas ring header. If the burner needs to fire multiple gas fuels separately or in combination, additional sets of fuel injectors and ring gas headers are added. The fuel injectors deliver a small portion of fuel into the wake of the spinner while most of the fuel is delivered in several lobes to gradually mix with the combustion air. The drilling pattern of all fuel injectors is usually customized for each application. The available substantial adjustments to the air flow allow effective additional control over the flame pattern as well as the ability to detune the operation from some unwanted acoustical interactions with the system. The burner can also operate with good stability with the FGR rates on natural gas up to 20% with the NOx emissions reduced to 25–30 ppm level. When firing oil, the maximum amount of FGR is limited to 15% with a corresponding reduction in thermal NOx of about 65%.

3.4.2 Burners with Fuel Staging

Figures 3.9 and 3.10 provide examples of fuel-staged burners: the Coen *Delta-NOx*™ burner and the Hamworthy *ECOjet*® burner. In both burners, the combustion air is delivered through a single venturi-shaped passage, and the staged fuel is injected into the flame from a series of fuel injectors placed outside of the diverging refractory throat.

In the *Delta-NOx* burner, the primary fuel is delivered through a series of small gas headers—spuds, placed in radial directions around a centrally located spinner, or

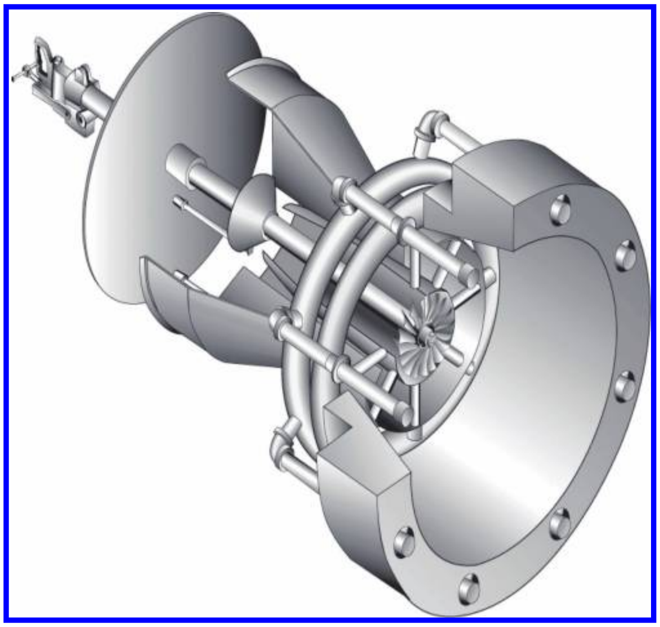


FIGURE 3.9
Coen *Delta-NOx*™ burner.

FIGURE 3.10
ECOjet® gas-only burners.

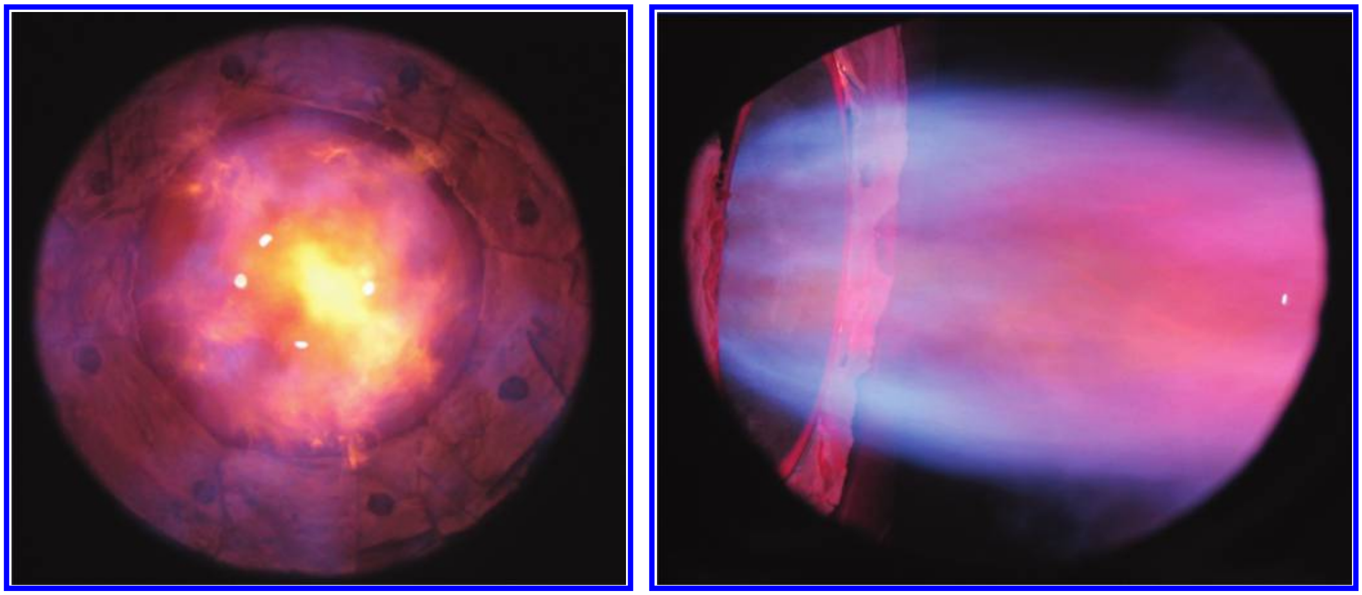

(a) (b)

FIGURE 3.11
(a) Hamworthy *ECOjet*® flame. (b) Natural gas firing at 30 MW (100 × 10^6 Btu/h) at Hamworthy test facility.

a combination of a bluff body and a spinner. Fuel injection from each spud is distributed along the radius. The *Delta-NOx* burner ignition points are stabilized by the action of the spinner or the bluff body and by the fuel jet arrangement that generates flame patterns sufficient to ignite the staged fuel at some distance from the burner refractory throat. When firing natural gas without FGR, the burner delivers NOx performance in the range of 45–60 ppm depending on the relative furnace to the heat input size. With about 10% FGR, the emissions can be typically reduced by about 45%.

In the *ECOjet* burner, all of the primary fuel is delivered through a center-fired gas gun. The primary fuel forms a highly stable primary flame zone. The flame is stabilized by the action of the spinner. The primary zone is large enough to provide ignition of the staged fuel combustion zone. Figure 3.11 shows the burner flame when operating on natural gas without the FGR.

Both burner designs are very simple and are usually designed with an external means of adjusting the air distribution between the zones. The NOx performance of the *ECOjet* burner is slightly better than that of the *Delta-NOx* burner but less forgiving to the deviations in the excess air ratio. In some cases, the *ECOjet* burner performance on natural gas can achieve 10–12 ppm NOx.

3.4.3 Burners with Partial Lean Premixed Combustion

A description of burners utilizing partial lean premixed combustion is given here using the Coen *QLN*™ burner as a classic example. Partial lean premixed combustion is fundamentally different than partial premixed fuel-rich combustion, the most common example of which is utilized in gas burners for all cooking ranges. The burner is schematically shown in Figure 3.12. The burner was the first in the industry to generate simulated lean premixed combustion in a substantial part of the flame. The burner was developed in the early 1990s[15–17] with a goal to efficiently fire natural gas in package boilers with less than 30 ppm NOx without using FGR. It had several new revolutionary features. Visually, the main unique feature is an air distribution plate or a number of bluff bodies shaping the

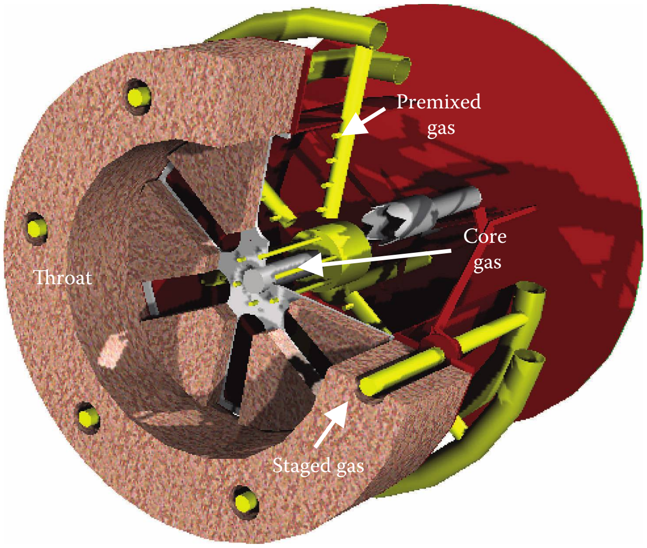


FIGURE 3.12
Schematic of a Coen *QLN*™ burner.

combustion air flow in a series of radially aligned passages separated by bluff bodies and spaced uniformly around another centrally located bluff body, whereas in all prior burners for package boilers, the air flow exiting the burner was predominantly circumferentially uniform. Another departure from convention at the time was the injection of fuel into the middle of the air-flow streams at a substantial distance from the possible area of fuel ignition to form a uniform very fuel lean mixture prior to its possible sources of ignition. It was known at the time that such premixing would result in a high-intensity acoustic response; however, the design of the *QLN* burner solved the problem with desynchronizing these instabilities and reducing them to well below the would-be critical levels.

The burner has three zones of fuel delivery: the flame-stabilization zone that burns a small portion of fuel in the wake of the bluff bodies, the premixed zone, and the staged fuel zone that gets the majority of its fuel from injectors placed through the ports of the short straight cylindrical refractory throat. The oxidation of the premixed fuel and staged fuel takes place gradually as the premixed fuel and air mix with hot combustion products from the flame-stabilization zone. The oxidation of the staged fuel also takes place very gradually as it is first mixed with medium-temperature low-oxygen combustion products around the flame (FGC effect described in Section 3.3.5) and then mixes with the remaining oxygen of the products of premixed combustion. In the *QLN* flame, most of the fuel burns at locally fuel lean or close to stoichiometric conditions minimizing the production of prompt NOx. The thermal NOx is also substantially reduced due to very fuel lean conditions for burning the premixed fuel and FGC effect created by the staged fuel.

The NOx emissions with *QLN* burners depend on the furnace size and the burner peak heat input. In small boilers rated to less than 75,000 lb/h (34,000 kg/h) of steam production (~32 ton/h) firing natural gas, *QLN* burners achieve typically less than 25 ppm of NOx with low excess air and no FGR. *QLN* burners in larger boilers with a capacity of up to 100,000 lb/h (45 m-t/h or 45,000 kg/h) of steam emit higher NOx up to about 40–45 ppm.

A small amount of induced FGR can be used in *QLN* burners to reduce NOx emissions by about 20%–25%. Better results, down to 15 ppm NOx, can be achieved with forced FGR delivered to the plenum around the burner and injected into the furnace around the staged fuel injectors. A typical appearance of a *QLN* flame is shown in Figure 3.13.

QLN burners can also fire oil. The NOx emissions on oil are also reduced due to the effects of FGC enhanced by the star-shaped pattern of air injection. However, NOx reduction with the atomizer design by creating the air-staging effects is reduced as only a narrow range of atomizer designs can be applied with the burner.

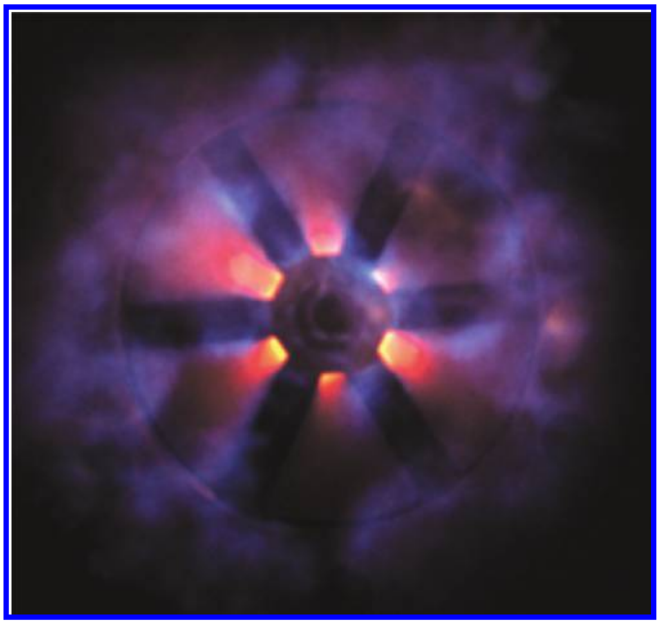


FIGURE 3.13
Coen QLN burner flame with 20 ppm NOx firing natural gas without FGR.

Over almost 20 years of its use to date, the *QLN* was successfully adapted for applications with multiple fuels firing simultaneously or one at a time proving to be a very versatile and highly tunable to the specifics of the application. The product always delivered the expected low NOx performance. *QLN* burners utilizing small amounts of forced FGR can operate with NOx emissions down to 15 ppm level in small- or medium-size package boilers and in larger package boilers with up to 250,000 lb/h (113,000 kg/h) steam production if the furnace of the boiler is slightly oversized.

3.4.4 Premixed Burners

Burners with all premixed and predominantly premixed combustion were developed in middle of the 1990s. The first burner of this kind utilized rapid mixing of filtered fuel gas passing through hundreds of tiny laser-punched orifices uniformly distributed through the flow of combustion air mixed with flue gas. The burner name, the *RMB*™ burner, is an abbreviation for *rapid mix burner*. The fuel distribution in the *RMB* was accomplished using a multitude of radially placed airfoil-shaped miniature fuel gas risers imposing a swirling action onto the combined flow of air fuel and flue gas.[18] The flame was stabilized by a strong recirculating flow of hot combustion products developed as the result of swirling flow in combination with a diverging refractory throat and another refractory bluff body at the burner centerline. Figure 3.14 shows an overall view

FIGURE 3.14
Coen RMB™ burner equipped with an air isolation sliding barrel damper.

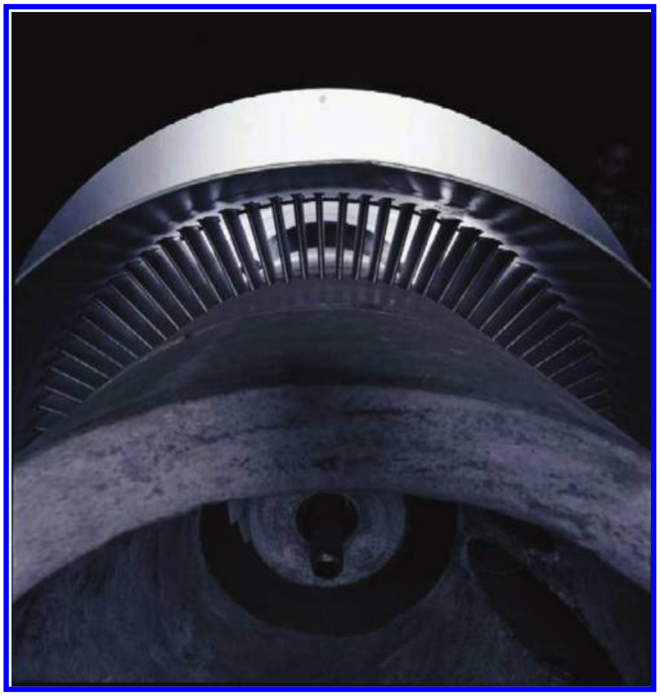


FIGURE 3.15
Fuel risers of a Coen RMB burner.

of the burner. Figure 3.15 shows multiple fuel risers. Figure 3.16 shows enhanced photographs of the flame.

The burner produces a relatively compact and short flame. The burner operation can be tuned for low excess air and NOx as low as 5 ppm. With low excess air, the burner uses substantial amounts of FGR. During normal operation, the burner does not generate any appreciable CO. The burner is also capable of firing liquid fuels.

With the aforementioned described design, the burner was developed in several sizes ranging in thermal capacity from 50 to 100 × 10^6 Btu/h (15 to 30 MW). For higher capacities up to 300 × 10^6 Btu/h (88 MW), the *RMB* concept was expanded with a second similarly premixed fuel zone placed around the primary *RMB* burner together making *D-RMB*™ burner—dual RBM (Figure 3.17). In order to limit the flame diameter, the second zone was designed for a very low swirl number. The outer premix zone is ignited from the primary *RMB* burner. The *D-RMB* burners were able to control the NOx emissions down to 7–9 ppm. The turndown of both burners was in the range of up to about 6–1. Figure 3.18 shows an example of a large *D-RMB* performance.[2]

Another premixed burner, the *QLA* burner, utilizes primarily fuel gas jets to generate a weak swirling motion of the combined fuel, air, and FGR blend.[13,19] This burner is designed with a variable geometry feature—the area of the burner where the premixed blend enters the combustion zone is modulated with the firing rate. The primary means of flame stabilization is a massive bluff body with a specially designed spinner on its periphery that desynchronizes combustion instabilities. The burner relies on an increased mixing distance between the fuel jets and the air making it possible to keep minimum fuel-orifice sizes relatively large, in excess of 1/8 in. (3 mm) in diameter. At reduced loads, the action of the spinner enhances as a higher portion of the flow passes through the spinner blades due to the variable geometry. The variable geometry and presence of smaller auxiliary fuel zones, one in the wake of the bluff body and one in a staged fuel zone, allow better control over the combustion instabilities at reduced

FIGURE 3.16
Enhanced images of the gas-fired RMB flame.

loads and overall better turndown of up to 10–1 and higher in some cases. The burner can typically deliver NOx performance down to 6–8 ppm NOx with capacities as high as 320 × 10^6 Btu/h (94 MW). The burner can operate with preheated air making NOx emissions as low as 5 ppm[14] and operate with ambient temperature or preheated air. Design modifications of the burner were incorporated to fire light oil as a backup fuel. Figures 3.19 through 3.22 show a schematic of the burner, the pictures of the burner assembly, and a close-up photograph of the flame, respectively.

FIGURE 3.17
Coen D-RMB® burner mounted inside the wind box.

3.4.5 Burners for Low Heating Value Gaseous Fuels and Fuels with Highly Variable Composition

Firing fuels with low heating value (low-Btu gases) presents some difficulties due to the high percentage of inerts in the combustion gases, high moisture content, etc., making the gas mixture poorly flammable or not flammable at all. These fuels are usually by-products of some processes and come to the burner with very low pressure ranging from a few inches of water column to a few psig (10–100 mbarg) and vary in composition. Such fuels can be effectively utilized in boilers with the help of small amounts of more flammable fuels like natural gas or oil and a scroll burner. Figure 3.23 shows a picture of one of such burners designed for firing blast furnace gas (BFG).

Low-Btu gases naturally produce very low NOx emissions. Firing low-Btu gases that may change in composition to a more flammable gas presents additional challenges that when combined with low NOx emissions requirements are difficult to resolve. When such fuels are introduced into conventional burners with fixed fuel nozzles, the changing composition would not create consistent mixing patterns. These, along with inconsistent flammability, the characteristics of the mixture can easily result in unreliable flame stabilization and combustion instabilities. The problem can be solved with burners mixing fuel with flue gas in quantities making the overall blend of fuel and FGR more consistent from the combustion standpoint. Figure 3.24 shows a picture of such a burner. In this particular case, the burners were even more complicated to accommodate two separate fuel streams varying in proportion to each other. The burner was able to reliably operate with less

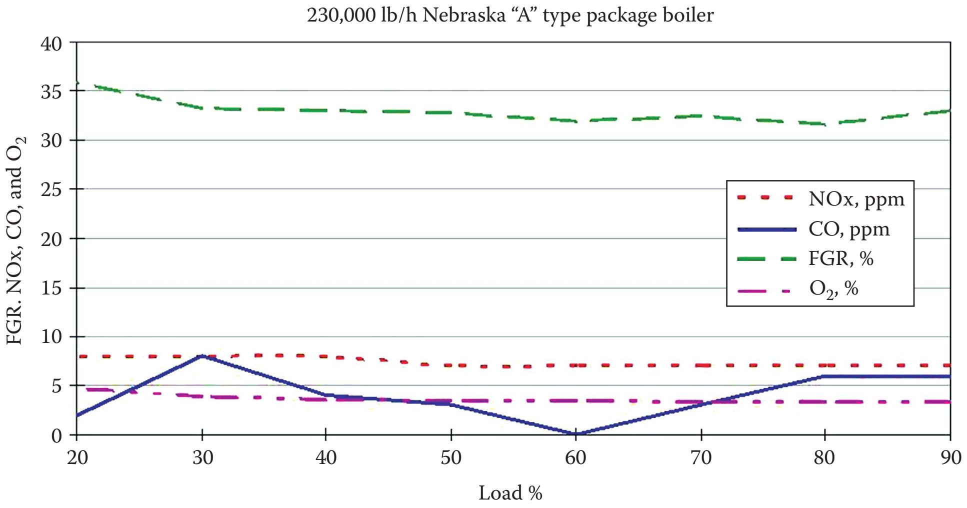


FIGURE 3.18
Coen D-RMB™ burner performance in a large package boiler.

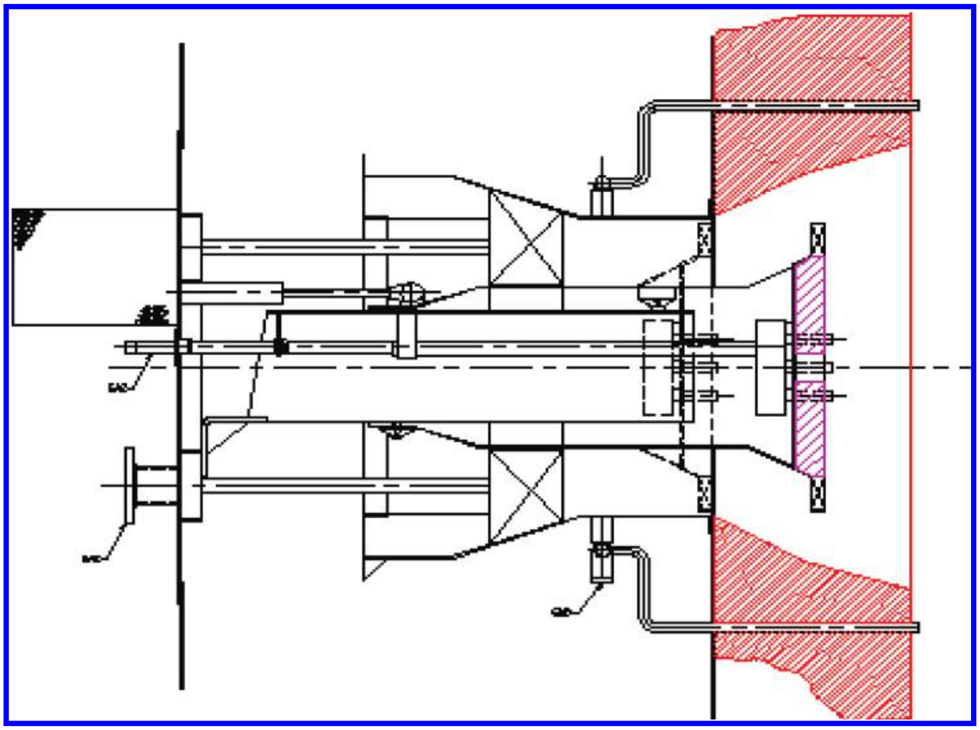


FIGURE 3.19
Coen QLA burner schematic.

than 15 ppm NOx (31 mg/N-m^3), referenced to 3% O_2 (vol, dry) firing a wide variety of fuels.

Mixing FGR with fuel is usually more effective than mixing FGR with air in terms of the NOx reduction for a given amount of FGR especially at lower FGR levels as shown in Figure 3.25.

The higher effectiveness of fuel mixed with FGR by of about 20% can be explained by the fact that the full amount of flue gas is engaged in the combustion at the very moment of fuel ignition rather than being spread over the entire flow including the excess air. The rest of the difference is due to a more favorable fuel and air mixing pattern with fuel-mixed FGR achieved with a specific burner for lower NOx control. For another burner, a further increase in the FGR rate may cause the mixing pattern to deviate from the optimum and slow down the NOx reduction with fuel-mixed FGR. This is also shown in Figure 3.25 (see data taken when firing a small Coen *CLN* burner).

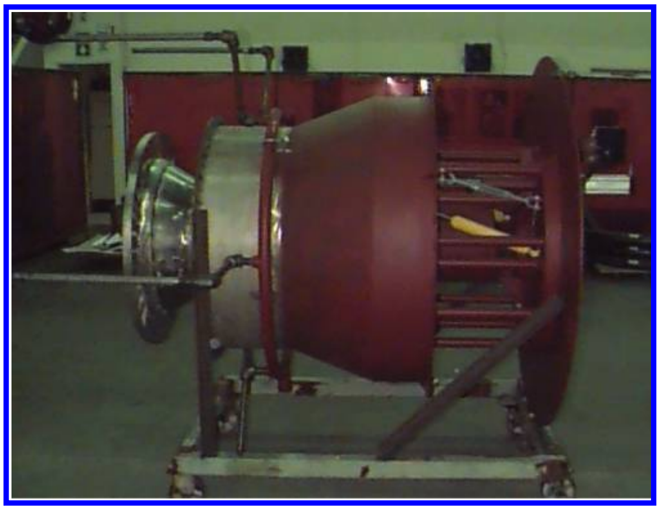


FIGURE 3.20
Assembled Coen QLA burner (side).

FIGURE 3.21
Assembled Coen QLA burner (front).

FIGURE 3.22
Coen QLA burner flame with 7 ppm NOx (natural gas firing).

FIGURE 3.23
Peabody LVC™ burner for firing BFG.

FIGURE 3.24
Modified Coen LCF burner for simultaneous low-NOx firing of multiple fuels of variable composition.

3.4.6 Oil Atomizers for Package Boilers

For firing liquid fuels, burners for package boilers are usually equipped with the internal-mix steam-assist atomizers generating fine droplets with a Sauter mean diameter (SMD) of 40–120 µm. In the industry, oil atomizers are also called oil guns. The definition of SMD, the description of the internal-mix atomizer concepts, and the examples of industrial atomizer designs were reviewed earlier in Volume 1, Chapter 10. The same chapter also describes two types of internally mixed atomizers: one with a common internal chamber for

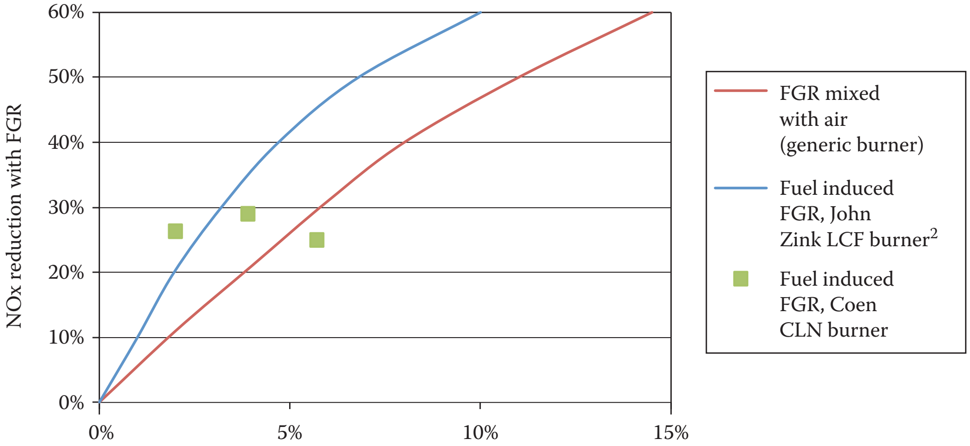


FIGURE 3.25
NOx reduction with FGR mixed with combustion air or fuel.

the primary stage of atomization and the other type with a single-stage atomization between steam and oil taking place in multiple passages where steam and oil combine. An example of the first type is the Coen *MVI* atomizer. An example of the second type is the Y-jet atomizer. For many package boiler applications, the preferred atomizer type is similar to an *MVI*® atomizer. The selection of this type of atomizer for many boilers is preferred as it allows substantial latitude in making the desired pattern of fuel injection as well as easy modifications to the flame pattern and performance by replacing a relatively inexpensive atomizer tip. Often problems that can be resolved with the cap drilling are carbon buildup on the caps that may interfere with the oil spray, collapsing of the flame in the middle of the oil spray preventing its adequate mixing with the air; oil impingement on the burner throat causing deterioration of the throat refractory; and impingement on the boiler side walls causing carbon buildup on the boiler tubes.

The amount of atomizing steam and the atomizer design details needs to be accurately tailored to achieve the desired atomization quality while minimizing the usage of these utilities. At high-fire operation, the atomizers in package boilers usually consume steam in the amount of 10%–18% of the mass of fuel oil. Internal-mix atomizers typically require 100–200 psig (7–14 bar) pressure of oil at high fire. The steam-pressure requirements are usually slightly higher than of oil by about 15 psi (1 bar). Having the steam pressure higher than the oil pressure assures that oil cannot penetrate into the steam passages even in cases of passages plugging. Increasing the amount of atomizing steam reduces droplets SMD and shortens the flame.

As the steam adds substantially to the combined momentum of steam-fuel jets, varying the amount of steam becomes a powerful tool for optimizing the mixing patterns between the fuel and air and the overall burner operation and flame shape. Often, however, there are trade-offs between the fineness of atomization and the NOx emissions as increased fineness of atomization may reduce the air-staging effects and increase the NOx. When designing internal-mix atomizers, it is important to understand effects of interactions between different jets in an overall usually very complex pattern. In many cases, even a few degrees change in the drilling angles and patterns creates a substantial change in the flame appearance and the burner/atomizer performance.

3.5 Burners for Enhanced Oil Recovery Boilers

The extraction of heavy oil in some oil fields requires in situ heating of bituminous oil deposits at the source deep underground in order to make the oil flow. The heating is done by injection of high-pressure wet steam produced by special boilers. The amount of steam used for enhancing the oil recovery is massive ranging from 2 to 3.5 times the amount of oil produced. This is the equivalent of using 10%–18% of the thermal energy of natural gas for oil reservoirs heating relative to the thermal energy of mined oil.

These oil fields usually do not have large supplies of freshwater making it necessary to recycle all the water that comes back out with the oil. This return water carries a high concentration of dissolved solids that is not practical to completely remove before using it again as boiler feedwater.

The steam for enhanced oil recovery is mostly generated by specially designed OTSGs, drum-less boilers, which are better suited for operating with such water. These generators are also often called steam flood generators. As a two-phase steam-water flow develops inside the generator tubes, the concentration of solids in the water phase increases in reversed proportion to the remaining percentage of water phase making it important to closely control this parameter. For the process effectiveness, the produced steam quality (steam content in a two-phase flow) needs to be rather high—typically in the range of 75%–80%. Furthermore, due to its multi-water pass design of these boilers, it is necessary to transfer heat from the flame to the tubes with very good uniformity and relatively low peak heat fluxes. Excessive heat flux on a tube already carrying only a small amount of liquid phase may cause local deposition of solids on the internal tube surface subsequently causing tube overheating and failure. With this danger in mind, the radiant section of the generators is sized substantially larger than package boilers with comparable heat input. The OTSGs are used by the industry in several regions throughout the world. The vast majority of units not coupled with gas turbines for electric power generation have thermal capacities from 50 to 350×10^6 Btu/h (15 to 100 MW) with corresponding furnace sizes varying from 7 to 21 ft (2.1 to 6.4 m) in diameter and 25 to 65 ft (7.5 to 20 m) in length.

The massive amount of fuel consumed by these generators as well as generator placement in environmentally sensitive areas has prompted regulatory requirements to operate the boilers with low and ultralow NOx emissions. Among the number of low-NOx technologies described earlier in this chapter are burners with partial premixed combustion and fuel staging that have demonstrated the best performance and necessary reliability for operation in mostly unattended environments. For example, the natural gas-fired *QLN* burners operating without the use of FGR and with low excess air can be tuned to produce NOx emissions in the range from 14–15 ppm to 20–25 ppm NOx depending on the generator size and the level of combustion air preheat used in cold climates. For lower NOx emission, down to 9–15 ppm, the same burners can be set to operate with some FGR.

Many burners in the field are designed to operate with natural gas or a combination of natural gas blended with "produced" gas. The produced gas is essentially a portion of recycled natural gas that is injected into the oil reservoir and then being absorbed by the oil to reduce its viscosity. The produced gas carries some moderate amounts of CO_2 (~30%–35% vol) and small amounts of other species. It also contains trace amounts of H_2S making it more difficult to apply FGR due to corrosion problems. So, QLN burners delivering low NOx emissions without the use of FGR are especially attractive to industry. Figure 3.26 shows the largest *QLN* burners built to date for OTSGs. As a reference, the round wind box on the photograph is 12 ft (3.6 m) in diameter.

FIGURE 3.26
Large Coen QLN burners mounted inside wind boxes.

In the most environmentally sensitive areas, regulations further require operation of steam flood generator with even lower NOx emissions. For this market, an ultralow-NOx burner, *QLN*™-II burner was developed that is capable of operating with sub-5 ppm NOx.[20,21] The burner concept is an extension of the original *QLN* burner technology modified specifically for operation with FGR. The burner requires FGR for operation. The amount of FGR depends on the desired NOx level. For sub-5 ppm NOx on natural gas, less than 30%, FGR is needed. Figure 3.27 shows a view of the burner from inside the furnace. With some modifications, the burner can be configured to fire light oil without any impact on its performance on gas. Figure 3.28 shows a row of operating steam flood units all equipped with *QLN-II* burners. The versatility of this burner allows addition of secondary fuel streams of a different calorific value

FIGURE 3.27
Coen *QLN-II* burner inside a furnace.

that are sometimes available as by-products of oil recovery process. Figure 3.29 shows the relation between NOx emissions and FGR for *QLN-II* burners sized for 50–100 × 10^6 Btu/h (15–29 MW) heat input operating with low excess air. With high FGR and NOx of about 3 ppm, the fuel oxidation visually becomes substantially flameless with very complete combustion (low CO).

In some areas, the steam generators are powered by syngas produced in the process of bitumen oil mining. This syngas is mostly composed of CO (~60% vol) and H_2 (~35% vol) and hydrocarbon gases, primarily CH_4, as the balance. For low NOx, burning of this syngas, along with natural gas, a version of a *DAF* burner operating with FGR is used. Figure 3.30 shows photographs of the discharge end of the burner. Figure 3.31 shows photographs of the burner flames when firing natural gas and syngas inside a large steam generator.

3.6 Burners for Multiple Burner Process Steam Field-Erected Boilers

When the capacity of a boiler needs to exceed the capacity of available package boilers or when the available footprint cannot accommodate a large package boiler, the boiler is built for firing with multiple burners. There are also older existing multiple burner boilers designed at the time when larger burners were not available. Process steam boilers with multiple burners are typically equipped with two, four, or six burners placed on one of the furnace walls. For higher efficiency, many field-erected boilers are equipped with combustion air heaters delivering air with 300°F–650°F (150°C–340°C) temperature.

Typically all burners in service are controlled by a single fuel control valve and therefore operate at the same heat input and with a common air supply. For multiple burner applications, each burner can be brought in and out of service independently, allowing greater flexibility in operating turndown and availability of the boiler when any of the burners needs to be serviced. From a burner operation standpoint, the online change in the number of burners in service translates into the requirement to reliably operate during disturbances in the air to fuel ratio created during the changes and the ability to light off the burners at substantial load and with some flow of FGR if used. Burners for multi-burner boilers are usually equipped with air isolation dampers of a sliding barrel type designed to pass only a small amount of combustion air in the closed position for cooling.

When selecting burners for multiple burner boilers, it is important to consider the possibility of heat damage to the burners out of service from the heat generated by adjacent burners. Other aspects include the need to

FIGURE 3.28
Row of boilers equipped with Coen *QLN-II*™ burners rated to 63 and 90 × 10^6 Btu/h (18.5–26 MWt).

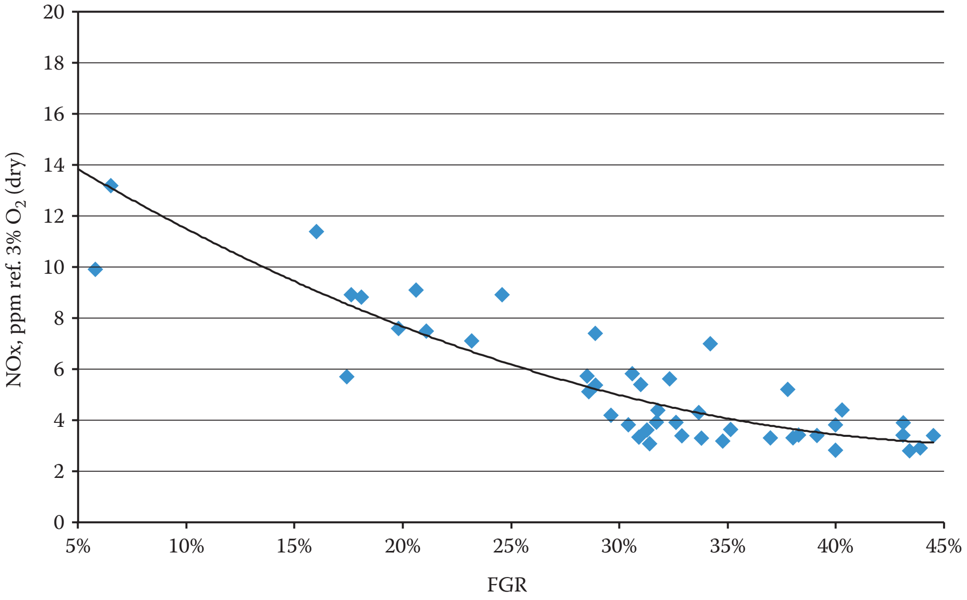


FIGURE 3.29
Typical NOx performance of *QLN-II*™ burner with FGR.

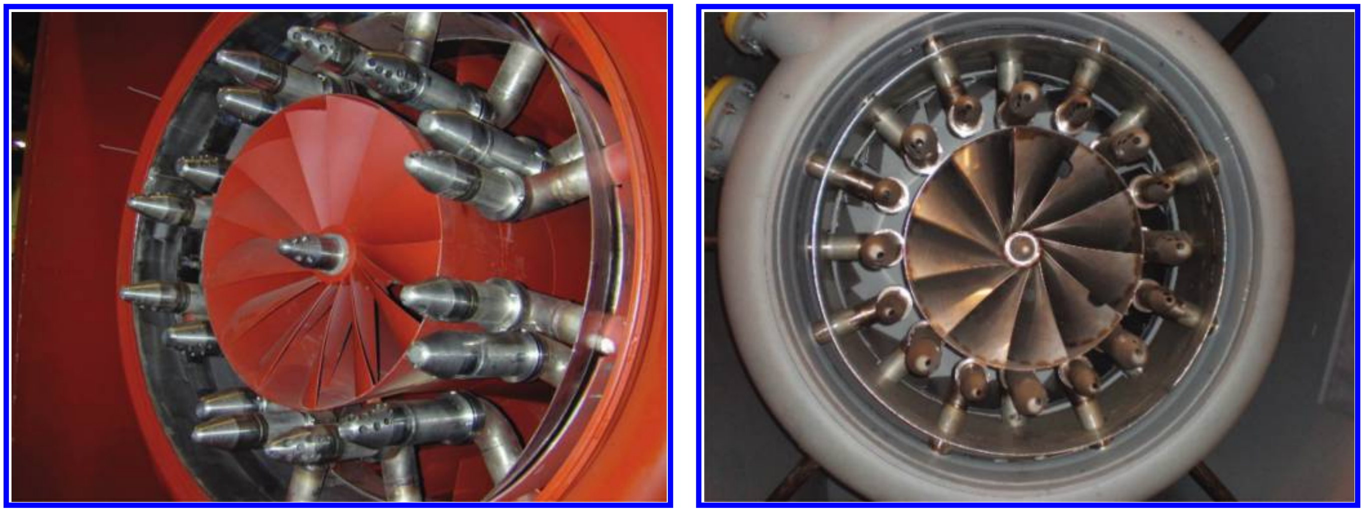

(a) (b)

FIGURE 3.30
Large 350 × 10^6 Btu/h (103 MW) Coen *DAF*™ burner for firing (a) syngas and (b) natural gas.

deliver fuel gas to the burner through a single line so that fuel to the individual burner can be easily cut off.

From a performance standpoint, it is important to know that the emissions generated by the boiler will depend on the number of burners in operation since heat transfer between adjacent flames typically increases the NOx. The effect is strongly dependent on the burner spacing.

Due to typically lower overall volumetric heat release in field-erected boilers than in package boilers, the flame-shaping requirements to the burners are less stringent. This makes application of a simple *Variflame* burner described earlier in this chapter or another burner like the *Delta Power*™ burner shown schematically in Figure 3.32, which is a common choice. Both burners are single air-zone venturi type designed for reduced NOx (air staging) and low excess air.

Delta Power burners are also designed for a low pressure drop making them well suited for retrofits with higher air preheat temperature or the addition of FGR. The burner is also equipped with an integrated air-flow measuring device allowing troubleshooting and adjustments of the operation in case of some problems with the air distribution. The burner is equipped with an external gas header with cane type gas spuds (gas tips). The external gas header makes it possible to replace or service spuds without dismantling the burner.

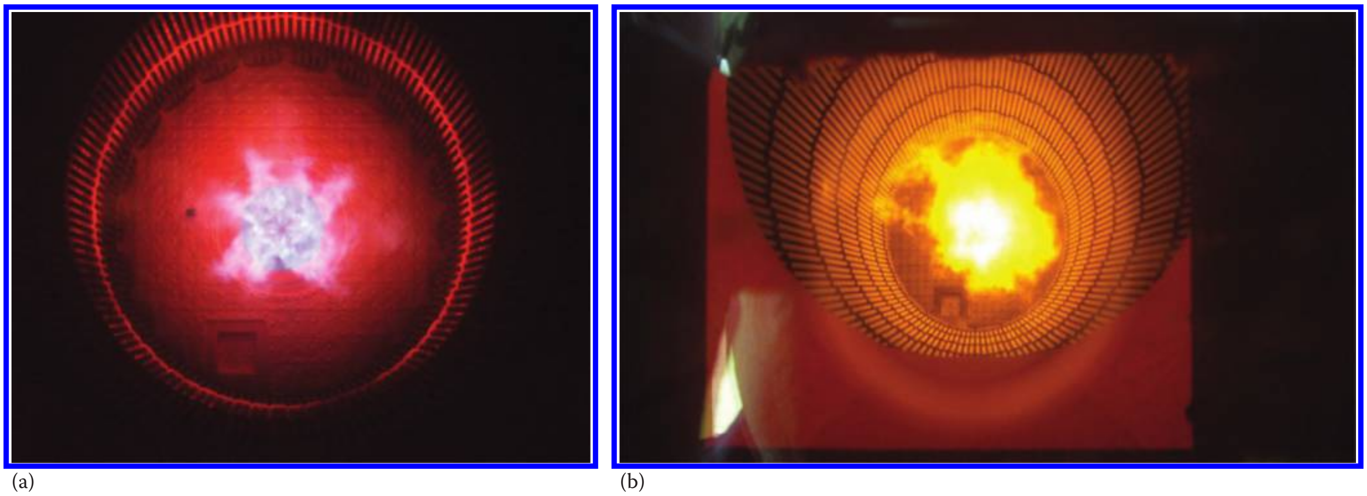

(a) (b)

FIGURE 3.31
Flames of Coen *DAF*™ burner firing (a) natural gas and (b) syngas (baking soda was added to the airflow for better visualization of syngas flame).

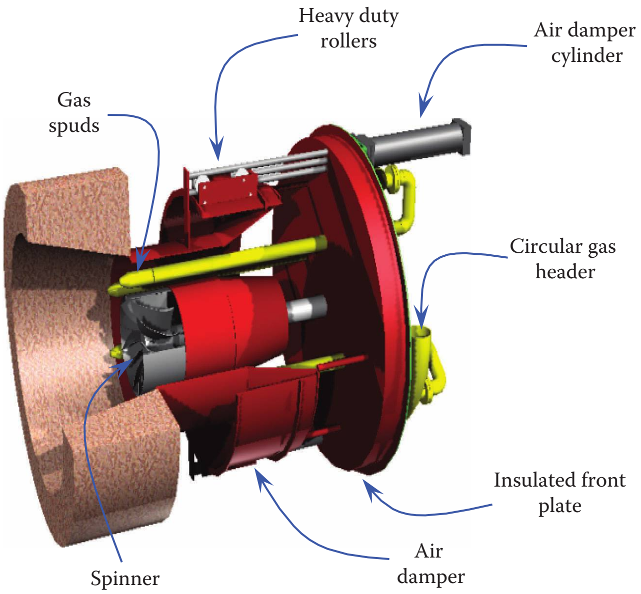


FIGURE 3.32
Schematic of Coen Delta Power™ burner.

More sophisticated lower NOx *QLN* burners have also been successfully applied on many field-erected boilers operating with and without preheated air firing natural gas, refinery fuels, and oil (as a backup). When using *QLN* burners, some special procedures in the combustion controls were implemented to reduce the disturbances when changing the number of burners in service.

There are a few field-erected boilers equipped with two premixed burners. The two burners in these boilers operate as one burner firing only simultaneously. Two burners placed close together on the furnace wall may form a cell with the added effect of enhancing flame stability of each other due to the flame interactions. This produces lower NOx emissions than with a single burner—as low as 5 ppm in some cases.

3.7 Gas and Oil Burners for Utility Boilers

Steam cycles of fossil fuel fired utilities have much lower efficiency of converting thermal power to electric than combined cycle plants using large stationary gas turbines. Practically all oil and gas utility boilers are either older boilers or coal-fired boilers converted to gas and oil. The conversions of coal-fired boilers to gas were prompted by concerns of global warming, more stringent emission requirements, and the increasing availability of natural gas. Oil firing capabilities are still considered as an important backup alternative. The two most common types of utility boilers are wall-fired boilers and tangentially fired boilers that use two quite different combustion concepts.

3.7.1 Combustion Systems for Oil and Gas Wall-Fired Utility Boilers

The wall-fired utility boilers are usually equipped with a number of burners ranging from 6 to 36 that are placed on one or two opposing walls of the boiler. The furnaces are generously sized with a significant part of the furnace volume above the burner levels. A typical average residence time of the combustion products in the furnace at high fire is about 1–1.5 s. All boilers operate with air preheated to 450°F–650°F (230°C–340°C).

The operation of utility boilers is continuously closely monitored by the operators overseeing the performance.

When necessary, the operators can add biases to setpoints of different controlled parameters, one of which is excess oxygen at the stack that normally is set in the range of 1%–2%. The boiler controls also monitor multiple parameters and notify operators if anything deviates outside of the normal range.

Most of the wall-fired utility boilers were designed originally for firing coal—a fuel producing high-temperature very luminous flames. Firing faster burning fuels like natural gas creates a different temperature pattern and more transparent flames with a typically higher relative rate of heat transfer to the steam tubes in the area of the superheater and reheater and less in the saturated steam tubes zone. For the efficiency of the steam turbine cycle, it is important to maintain the design temperature of the steam coming out of the superheater and reheater. Excessive temperature in the superheater can be controlled with water injection—also called attemperation. However, attemperation in the reheater is very undesirable as it changes the ratio of flows in different stages of the turbine and reduces the cycle efficiency. The conversion of utility boiler operation to gas shifts the heat distribution to the superheater and the reheater requiring more attemperation and loss of efficiency or the inability to operate at full load due to tube overheating.

The use of forced FGR recirculation back into the furnace through a hopper on the bottom is also a common feature of the boiler serving to control the loss of steam temperature coming out of the superheater and reheater at partial boiler loads.

When evaluating a multi-burner combustion system, it is important to understand that burners are only one of many elements affecting the system performance. The overall performance of the combustion system substantially depends on the design of the fuel distribution and air distribution to the burners. The difficulties of designing a uniform air and fuel distribution system are quite substantial. The relative magnitude of the buoyancy effect on the air distribution and the hydrostatic effects on the pressure of liquid fuel delivered to the burners vary with load and often are not properly accounted for. Addressing potential air distribution problems with compartmentalizing the wind box and measuring and controlling the air flow to individual burners may look attractive at a glance. However, it often creates more problems than it solves as the complexity of the system is increased and the reliability reduced. The potential severity of oil maldistribution between the burners depends on the piping design and on the type of atomizers and available margins of the supply pressure. Typically during retrofits, the wind box is modeled and, if necessary, modified for uniform air distribution.

Gas- and oil-fired utility burners are usually sized for the available draft loss typically in the range from 4 to 6 in. W.C. (10–15 mbar). Sizing burners for lower differential air pressure is helpful for reducing the NOx emissions and creating more luminous flames, but may also result in longer flames, impingement problems, as well as additional difficulties with achieving the desired quality of the air distribution.

3.7.1.1 NOx Reduction Techniques in Wall-Fired Utility Boilers

Air staging and FGR are the main NOx reduction techniques used in wall-fired utility boilers. The selection of the means of NOx reduction must be considered in combination with an evaluation of its effect on the boiler operation in addition to NOx reduction. The air-staging techniques tend to increase the luminosity of the flame and may reduce the mostly unwanted shift in the heat distribution toward the superheater when converting boilers to fire gas. The addition of FGR may increase this shift toward the superheater. FGR introduced through the burners for the purpose of NOx control has a strong effect on the thermal NOx and a strong effect on the heat transfer in the furnace and superheater. By comparison, FGR introduced through the boiler hopper has a relatively small effect on the effluent NOx, but practically the same effect on the superheater. The quantitative evaluation of the effects, checking adequacy of the available attemperating flows, etc., often requires creation of complex CFD models of the boiler.

Air staging using low-NOx utility burners can be done to a larger extent than in package and small field-erected boilers. This is due to the substantially higher residence time in utility furnaces, higher combustion air temperatures, and more intense mixing taking place in utility furnaces. The techniques are effective for both oil- and gas-fired boilers.

The leading modern technique of air staging when firing heavy oil was originally developed by a series of Electric Power Research Institute (EPRI) funded projects in the early 1990s. The technology is called REACH. The first early reports on REACH are summarized in Ref. [22]. The essence of the refined REACH technology resulting from those studies is the creation of a two-lobed pattern of fuel injection by the atomizer in conjunction with a simple venturi-type burner equipped with a refined medium-size spinner. The exact pattern of fuel injection and the atomizer components are tailored to the specifics of the application. For heavy oil firing, a typical level of NOx reduction with air-staged burners is ~40%. Similar staging techniques were developed for firing gas with 25%–50% effective NOx reduction from the original uncontrolled levels (see Figure 3.33).

When firing large arrays of burners, the effects of air staging can also be achieved or enhanced with uneven fuel distribution between the burner levels with some levels firing fuel rich and other levels firing fuel lean. Obviously

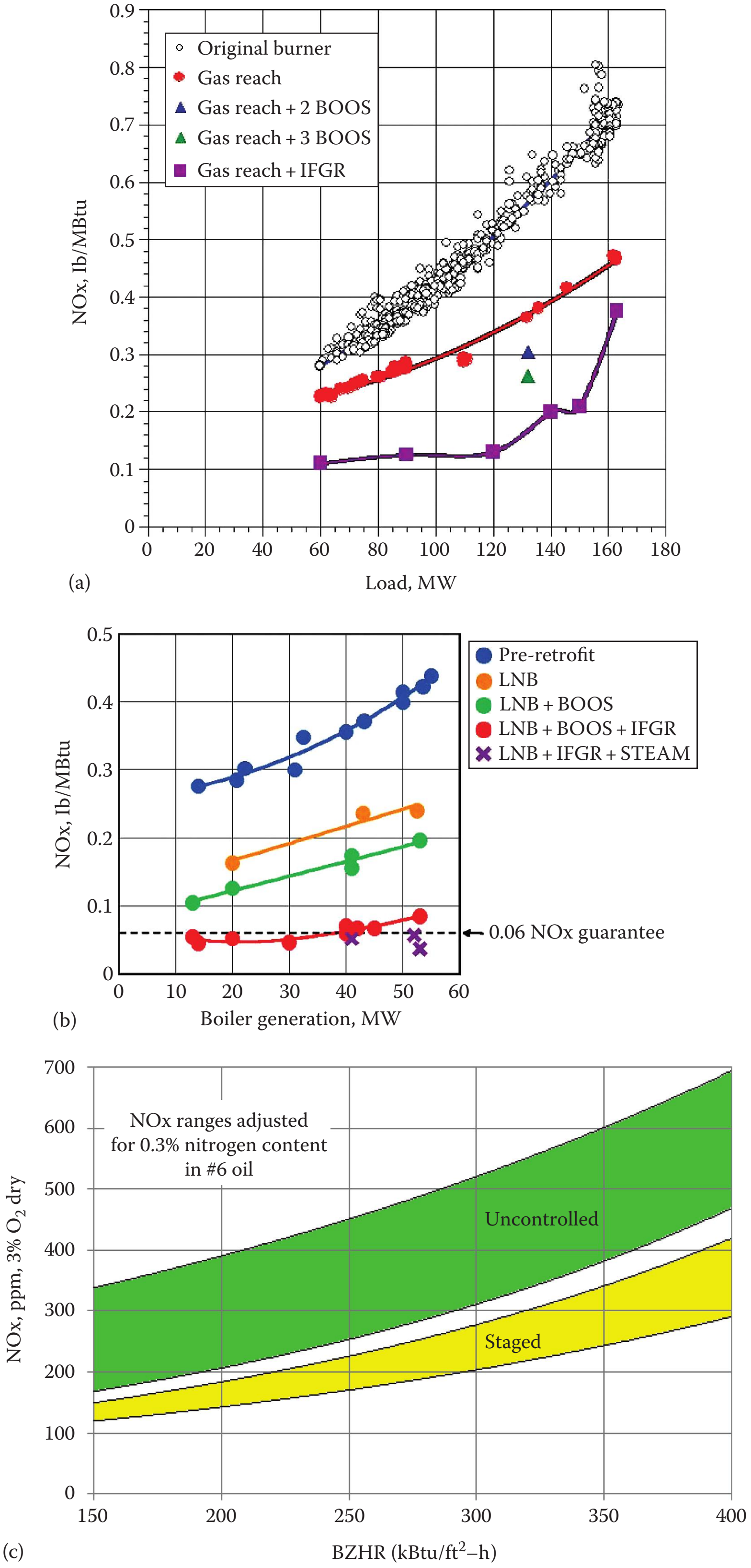


FIGURE 3.33
Examples of NOx reduction with air staging and FGR in utility boilers when firing natural gas (various boilers). (a) and (b) illustrate NOx reduction firing natural gas; (c) illustrates NOx reduction when firing #6 oil in different boilers.

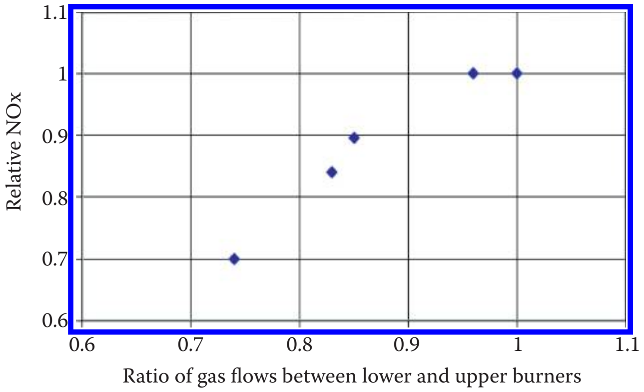


FIGURE 3.34
Effect of fuel biasing on the NOx. 250 MWe CE boiler with 16 wall-fired burner; natural gas firing. (From Lifshits V. and Crovato G., Experience with high efficiency, low emission burners to improve plant operation, *Latin America Power'98, Conference Papers*, Buenos Aires, Argentina, 1998.)

such shifts can be done only up to the extent of maintaining combustion stability at each burner. A field study on fuel biasing performed on a 250 MWe (850 × 10^6 Btu/h) boiler with 16 burners arranged in four rows[23] illustrates the effectiveness of fuel biasing when firing natural gas. With 13% higher than average gas flow to the lower level and similar reduction in the gas flow to the upper level of burners, the NOx was reduced by 30% in comparison with the unbiased operation (see Figure 3.34). The biasing also helped to reduce attemperation in the superheater so the boiler could operate up to its full capacity.

For stronger staging effects, the boiler may also operate with the fuel to some burners cutoff, but without the burner isolation on the air side. These relatively simple techniques are called fuel biasing and burners out of service (BOOS). The effectiveness of BOOS on NOx reduction is very boiler dependent. The optimal pattern of burners with shutoff fuel is usually established experimentally. The optimum pattern may also depend on the boiler firing rate and availability of the burners and the BOOS effect on the temperature at the superheater. The BOOS technique can be applied in combination with staged burners. The effectiveness of the described NOx reduction techniques is illustrated by Figures 3.33 and 3.34.

The cumulative NOx reduction from the uncontrolled level will always be less than the sum of reductions of each technique applied separately as illustrated by Figure 3.33. The NOx emissions data shown on Figure 3.33c are shown in relation to the burner zone heat release rate (BZHR). The parameter is defined as the total heat input coming into the furnace from fuel relative to the surface area of a parallelepiped matching the boiler cross-sectional dimensions and having a height from the slopes of the hopper to 4 ft (1.2 m) above the upper level of firing burners. The BZHR at high-fire operation is a boiler design parameter to which many different boiler emission data are correlated with reasonable statistical accuracy.

The initial success of these techniques for a moderate NOx control in 1970s and 1980s prompted changes to the boiler designs with the addition of a row of ports above the upper burner level for injection in the furnace of a portion of the combustion air that otherwise would flow through the burners. The technique was called overfired air (OFA). NOx control with OFA is more effective than with BOOS, but requires substantial boiler modifications as the OFA ports are not part of the original boiler. In some cases, the whole row of upper burners was converted to OFA ports. Typically up to 20% of the total combustion air can be diverted to the OFA ports leaving a uniformly sub-stoichiometric amount of air to the burners. OFA should not be combined with BOOS as it is always more effective to increase the OFA than to shut off fuel to some of the burners.

For maximum effectiveness, the OFA ports need to be properly designed to achieve adequate mixing between the OFA air and the upcoming combustion products and leave sufficient time to complete the fuel oxidation process before the furnace gas gets cooled in the superheater and the convection pass to temperatures when the oxidation of combustibles—primarily the CO is stopped. OFA is usually controlled separately from the rest of the combustion air. Each port can have an isolation damper and resemble a smaller burner without the fuel parts. When properly designed, OFA allows lower excess air operation than with the BOOS. CFD modeling of the boiler combustion process is often used for designing the OFA system.

Further reduction in NOx emissions can be achieved with FGR injected into the air stream downside of the air heater. This is usually done in lieu of or in parallel with the injection of the FGR through the furnace hopper. When the boiler is equipped with OFA, the FGR is injected only into the portion of the combustion air going to the burners. The effect of FGR on NOx reduction is very strong in gas-fired boilers operating with highly preheated air. An addition of 10% FGR typically reduces NOx by 50%–55% as illustrated by Figure 3.35.

One of the curves in Figure 3.33a shows another example of the FGR effect on NOx. The sharp increase in NOx shown by the curve at high fire was due to limited fan capacity of the system causing a substantial drop in the FGR at high fire.

The FGR effect on NOx is much lower when firing #6 oil as a substantial amount of the formed NOx comes from fuel nitrogen. With a high nitrogen content of ~0.5% to 0.6%, the effect of the FGR may not be significant if the thermal NOx component is low as illustrated by the test data in Figure 3.36. These old data were taken when firing #6 oil with 0.54% FBN with preheated air to

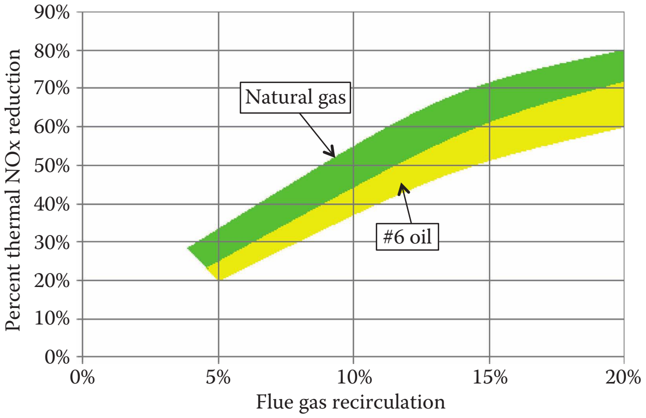


FIGURE 3.35
Effect of FGR on the thermal portion of NOx in different utility boilers.

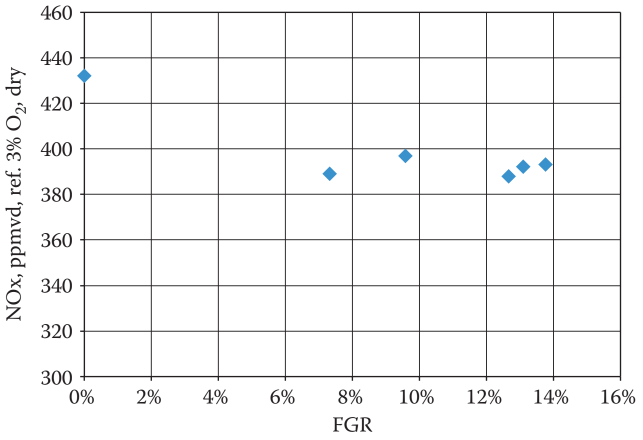


FIGURE 3.36
NOx emissions firing #6 oil with 0.54% FBN (COEN test data with CPF burner, not a low-NOx design).

~360°F (180°C) in an 8 ft (2.4 m) diameter furnace. The heat input was ~27×10^6 Btu/h (8 MW). The burner was not a low-NOx design.

The reduction of thermal NOx with FGR when firing #6 oil is somewhat substantial in boilers with high BZHR not equipped with low-NOx burners. In combination with low-NOx burners and air staging, the use of FGR becomes less effective. The likely explanation of this effect is that lower-temperature less luminous flames with FGR dissipate less heat from the fuel-rich combustion zone. Furthermore, FGR shifts the fuel-rich flammability limit closer to stoichiometric conditions making it more difficult to apply deep air-staging NOx reduction techniques. When considering the addition of FGR, one also needs to consider that FGR is usually the only NOx reduction technique that may cause an increase in the heat transfer in the superheater. Corrosion issues are also a problem when firing with FGR when applied to fuels like #6 oil containing sulfur.

The combined effects of multiple NOx reduction techniques are always less than the simple multiplication of different NOx reduction factors. Applying additional NOx reduction techniques often requires easing or taking away other means of NOx reduction in order to maintain flame stability.

3.7.1.2 Burners for Wall-Fired Utility Boilers

The type of burners used in wall-fired utility furnaces is similar to those used in field-erected boilers. The available round opening between the tightly packed tubes is always all that is available for mounting the burner. The burners for utilities are built heavy duty with thick, often stainless materials. For reliable operation and light off at any load level, the burners are equipped with high-capacity pilots and flame scanners with good discrimination between the monitored burner flame and the flame of adjacent or opposite wall burners. The air isolation dampers are usually part of the burner, except for rare cases when each burner has a dedicated wind box with an air isolation damper being part of the wind box. Other requirements for the wall-fired utility burners stem from the NOx reduction techniques utilized by the system.

The array of burners that operate on oil and gas in large wall-fired utilities and incorporate the necessary fuel-staging techniques includes the already described *Delta Power* and *Variflame* burners. Another product for wall-fired utilities is a *Dynaswirl*™ burner that is a version of the *Variflame* burner that permits fitting a larger-capacity burner into the limited opening between the steam-water tubes. The utility versions of these burners are built to withstand high levels of air preheat and thermal radiation from the furnace and usually include air-flow measurement devices added to the venturi passages.

Fuel-staged burners, like *QLN*, *Delta-NOx*, or *ECOjet* burners, are not suitable for wall-fired utility retrofit applications due to the high costs of making additional ports in tube walls for placing staged fuel injectors and likely insufficient benefits over the air-staging techniques.

3.7.2 Combustion Systems for Corner-Fired (T-Fired) Boilers

The combustion concept of a T-fired boiler is based on creating a single large vertical flame vortex in the middle of an elongated parallelepiped-shaped furnace. Originally, the very crudely distributed mix of some of the combustion air and pulverized coal was delivered into the vortex through a series of corner compartments placed on several levels. Additional preheated combustion air was delivered through vertically elongated compartments above and below the fuel injection points.

The process of fuel oxidation in the vortex is very heavily air staged. Most of the flame stabilization is done effectively by the vortex. Reduced temperature combustion products from the areas of proximity to the furnace walls and the corners readily reenter the vortex by the aspirating effects of flat air jets coming from the corners. All of the aforementioned features of the combustion process deliver lower NOx emissions than wall-fired boilers.

As the boiler load modulates, the relative intensity of the heat transfer in the superheater changes. In T-fired boilers, control of the superheater and reheater temperatures is done usually by means of "tilting buckets," adding an up or down vertical component to the direction of air flow and fuel injection coming from the corners. FGR injection into the furnace through the hopper is also used for the same purpose. The FGR that can be added for additional control of the NOx emissions is also a factor to be considered in the boiler superheater and reheater designs.

As the mixing and the combustion substantially take place in the vortex, the requirement for uniform air distribution to individual corners and in the vertical direction in T-fired boilers is more relaxed if compared with wall-fired furnaces. This reduces the burner differential air pressure at high fire to about 3–4 in. W.C. (7.5–10 mbar). However, the stratification of species concentration in the radial direction usually makes it necessary to operate the boilers overall with excess air slightly higher than for wall-fired boilers. For a typical T-fired boiler, the stack O_2 is maintained in the 0.8%–1.5% range when firing gas and 1.5%–3% when firing oil.

A potential problem that needs to be addressed with the burner design is the ability to operate throughout the load range without generating combustion instabilities while meeting the emission guarantees. The instabilities happen when some of the flames get detached by a substantial distance from the fuel injection points and reattached again. The conditions are more likely to occur at partial loads and during warm-ups when the overall firebox temperature is relatively low.

An oil and gas or gas-only burner for a T-fired boiler may be defined as a vertically stretched part of one corner containing the following three major components: a few air distribution buckets with a common tilting mechanism if the burner is designed for tilting, a flame-stabilizing spinner or shield, and a few fuel nozzles positioned in the vicinity of the flame stabilizer. This definition is similar to what originally constituted a coal-firing burner.

When a T-fired boiler is retrofitted with new lower NOx burners or burners firing a new fuel, a common scope of burner replacement usually includes modifications to the openings from the corners into the furnace and some stationary turning vanes for improved air distribution through the burner. Extensive modeling of the air distribution to the burners and through the burners usually provides the basis for the design. When the new operation needs tilting burners—usually the case when the original boiler was designed with that feature—some components of the existing tilting system can be preserved. The buckets are usually replaced and to some extent modified. The fuel injectors and flame stabilizers are always new and custom engineered for the application.

The new burners for T-fired boilers usually come with stringent emission requirements especially for NOx. The key NOx reduction concepts used in T-fired boilers are similar to those used in wall-fired boilers: low-NOx burners (levels out of service—not firing levels with only the air flowing through the buckets), OFA (upper level of buckets with the fuel, but not air cutoff), and FGR. The OFA in T-fired boilers can be of close-coupled type (CCOFA) (when an upper part of the air compartment above the upper point of fuel injection passes more air than the part below the fuel injection level) or of separated type (SOFA) (when the upper level of air compartments is substantially remote from the upper level of fired burners). The effects of FGR on the NOx reduction in T-fired boilers are quite similar to the wall-fired boilers as illustrated by Figure 3.35. One of the main burner design goals is the need to operate throughout the boiler load range without generating combustion instabilities while making the emission guarantees. The instabilities happen when some of the flames get detached by a substantial distance from the fuel injection points and reattached again. Insufficient means of flame stabilization incorporated into the burner design, excessive or improper staging, and use of FGR—most of the same techniques that make the system low NOx—create difficulties with achieving operation without excessive instabilities. The best burner designs successfully navigate through these difficulties and deliver NOx emissions in the range of 40–60 ppm when firing natural gas and 150–230 ppm NOx when firing #6 oil. The actual numbers depend on such factors as preheated air temperature, boiler size, and nitrogen content in the oil. Figure 3.37 shows a natural gas flame of a scaled-down

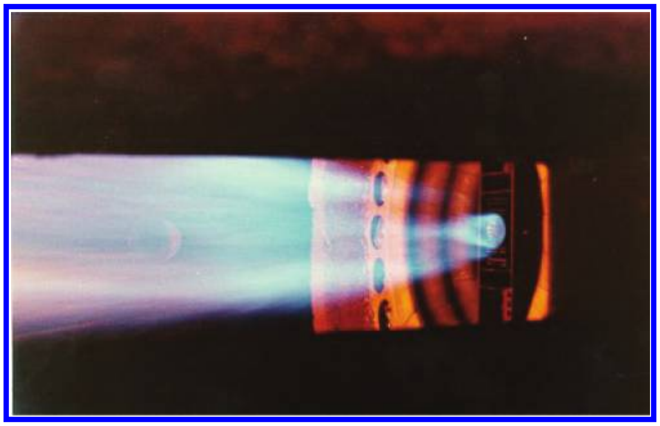


FIGURE 3.37
Low-NOx natural gas flame. T-fired burner prototype operating with FGR.

FIGURE 3.38
Main components of a fixed geometry burner for a gas and oil T-fired boiler.

prototype of a T-fired burner operating over a wide range of excess air with a very stable flame anchored at the spinner. The flame in this example was quiet and without any signs of possible instabilities.

There are two types of burners currently offered for T-fired boiler retrofits: burners without tilting—fixed geometry burners—and burners designed to be integrated with the tilting buckets. The fixed geometry burners are usually custom designed and include airflow conditioning plates and turning vanes. Other parts include a spinner with an oil gun in the middle and a series of lance-type gas fuel injectors placed around the spinner. Some areas of the air passages where air enters the furnace may be blocked with refractory-lined plates. The pattern of fuel injection by the lances and the oil gun is customized to the remaining air of the air-flow passages and the needs to achieve the desired air-staging and flame-stabilization effects.

The design of tilting burners is more complex. It includes flexible oil atomizers placed in the middle of tilting spinners or shields connected to the bucket in the middle of the burner and means to change directionality of the gas injection. Figures 3.38 through 3.41 show overall and close-up views of different burners and some of their parts. Figure 3.41 shows some distortion of the bucket structure caused by the furnace heat sustained during operation with insufficient air flow through some compartments.

Flame monitoring in T-fired boilers involves monitoring the overall flame in the vortex and monitoring the attachment of the flames to the burners. The logic of generating alarms and burner shutdowns in a T-fired boiler is different from the wall-fired boilers. At high loads, any flame detachment from the burners is not likely to generate instability problems so that the operation of the boiler and the burner can continue as long as the main flame vortex is present. This mode of operation may be necessary for operation with some additional FGR needed to control the NOx at high fire. This mode of operation is called *fireball logic*. When the boiler load reduces along with a reduction in the FGR rates, the flame can become reattached to the burners by itself or with the help of burner pilots that can be brought back into operation for the transition.

FIGURE 3.39
Spinners for tilting burners.

3.7.3 Oil Atomizers for Utility Burners

Due to very high heat inputs in utility boilers, on the order of several GWt, utility boiler operation is much more concerned with minimizing the amount of steam used for liquid fuel atomization.

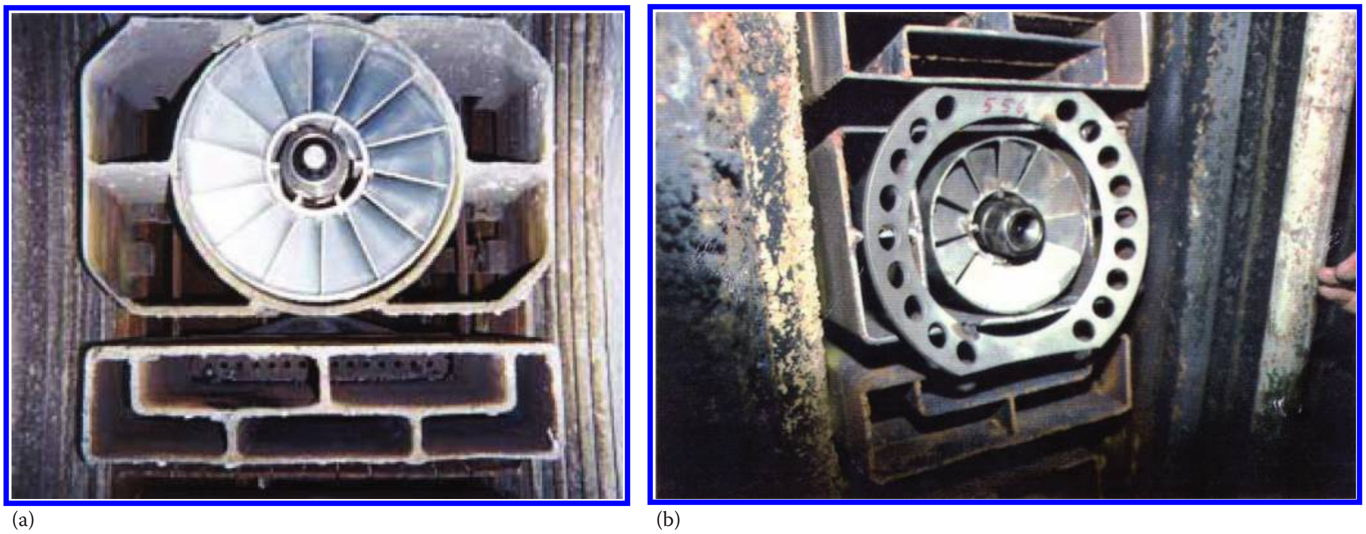

(a) (b)

FIGURE 3.40
Flame stabilizers and buckets of tilting (a) gas-fired burner and (b) oil-fired burner.

FIGURE 3.41
Corner of a T-fired boiler with tilting burners with some heat damage.

Mechanical atomizers do not use steam and from that standpoint are the most efficient. The mechanical atomizers need higher-pressure pumps, delivering fuel with up to ~1000 psig (70 barg) pressure to the burners. They may have low turndown and are more susceptible to erosion. However, the main drawbacks are symmetrical fuel injection, poor control over the atomization quality, and minimal ability to make changes to the fuel injection pattern. For those reasons, existing mechanical atomizers usually get replaced with low steam consumption internal-mix atomizers producing sprays with SMD of about 80–120 μm.

Both types of internal-mix atomizers described in Volume 1, Chapter 10, are common in utility boilers. The single-stage Y-jet atomizers may have a very slight advantage with respect to the amount of steam they use; however, the two-stage internal-mix atomizers like *MVI* allow much larger latitude in the design of the drilling patterns and are able to achieve low NOx emissions. The steam consumption of these atomizers at high fire varies from 5% to 10% of the amount of oil. More details on the principles of fuel atomization can be found in Volume 1, Chapter 10.

3.8 Specialized Burners

3.8.1 Warm-Up Burners

Many boiler applications require dedicated warm-up burners. One group of such boilers is black and red liquor recovery boilers in the paper industry. Black and red liquor boilers are used for the recovery and recycling of chemical components as well as recovery of thermal energy contained in the by-products of paper production. The liquors are sprayed on the boiler walls covered with a thin refractory layer. Volatile organic compounds evaporate and burn engaging oxygen in the furnace volume. The heaviest residue of the liquor flows along the walls to the bottom of the boiler burning in the process using furnace oxygen. The whole process generates a substantial amount of PM. For the whole process to start the recovery, boilers need warm-up burners that use conventional fuels like gas or oil. The warm-up burners may also be tasked to carry a substantial load to compliment or substitute the firing of liquors for the purpose of power generation. The main requirement for the warm-up burners is to have small openings into the furnace as smaller openings would require a minimum amount of purge air to fend off burner contamination with PM that can easily form aggregates plugging the throats. The second requirement for the burners is to make them tolerant of the intense thermal radiation from the refractory-lined furnace walls and flame. The third requirement is simplicity and reliability of operation. The combustion air going to the burners is initially not preheated, but then reaches temperatures up 600°F (320°C). At high-fire operation, the air is delivered to the burners in the amounts of 0.7–0.8 times of stoichiometric. The rest of the air for combustion comes from tramp air in the furnace. There are typically no strict NOx requirements for the warm-up burner performance as the liquids contain a substantial amount of nitrogen that easily converts to NOx. So these boilers rely on post-combustion cleanup of the exhaust including the removal of NOx.

Another group of boilers that uses warm-up burners with similar requirements is fluidized bed boilers. Warm-up burners for these boilers are installed 5–7 ft (1.5–2 m) above the fluidized bed of finely dispersed coal or biomass at an angle toward the bed to allow intense heat transfer from the flame to the bed material. Using compact burners here is beneficial for creating higher-momentum intense flames capable of delivering heat to the bed without being quenched/extinguished by the bed particles. The available air pressure drop for the burners is usually high—over 8 in. W.C. (20 mbar) as the combustion air comes from the same source as the air for bed fluidization that is substantially higher.

For both types of warm-up applications, the same type of burners can be applied. Figure 3.42 shows examples of warm-up burners designed for high-intensity combustion. All warm-up burners have minimally flared throats or not flared at all and cantilevered center-fired guns that also support flame stabilizers. The stabilizer and the fuel guns are often equipped with retraction mechanisms. Flame scanning of the warm-up burner flames often presents a challenge due to substantial blocking of the flame view by the flame stabilizer and high dust load in the flame area.

3.8.2 Flue-Gas Reheat Burners

Reheating boiler flue gas may be required for temperature conditioning purposes upstream of the emission-reducing catalysts. An example may be an SCR process

FIGURE 3.42
Coen warm-up gas burners.

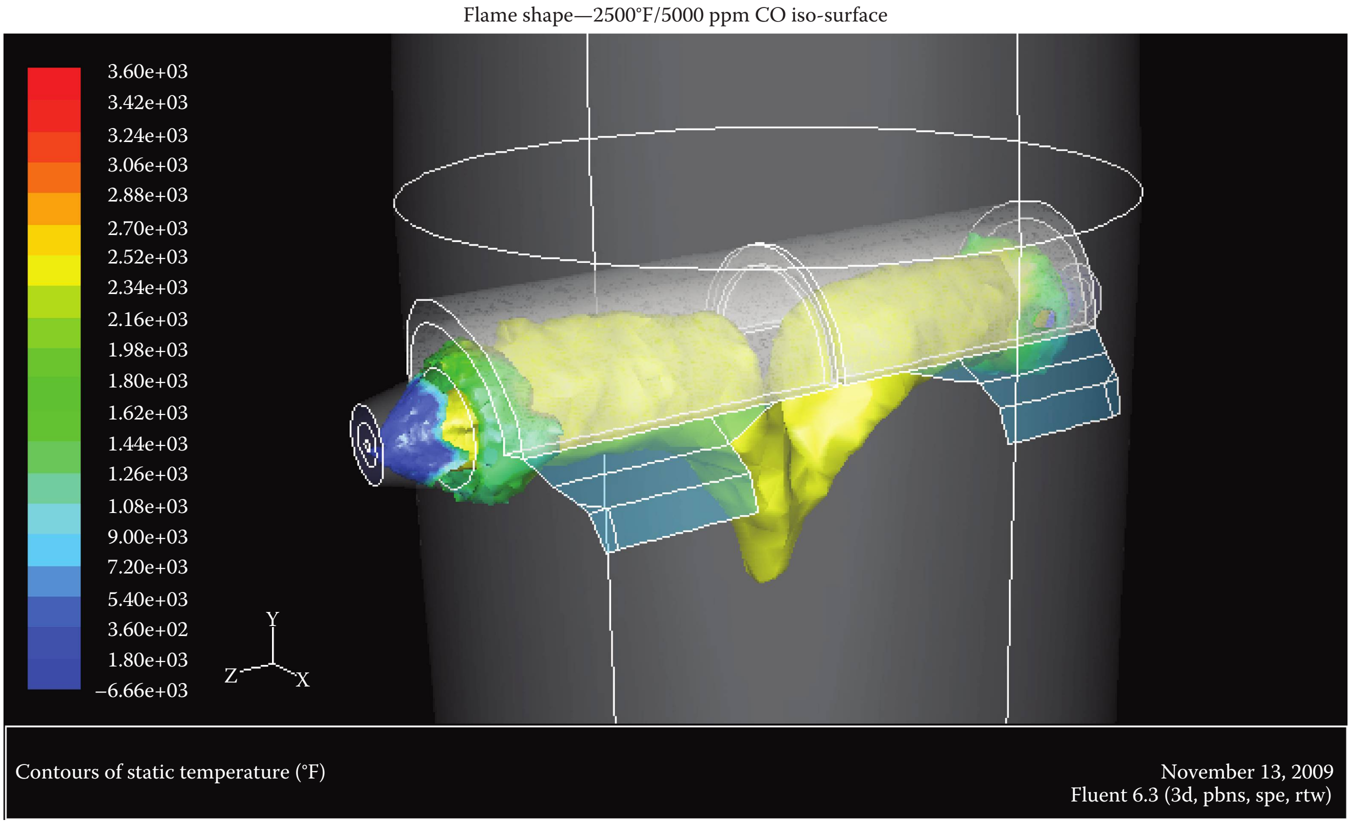


FIGURE 3.43
Conceptual design of low-CO flue-gas reheat system for refinery gas firing.

(see Chapter 10) to convert NOx to atmospheric nitrogen requiring temperature of flue gas in the range of about 650°F–750°F (340°C–400°C) for the most efficient operation.

As the boiler load modulates, so does the temperature of the exhaust making it necessary during some or all of the regimes to add heat to the flow. A convenient way of adding heat would be with special duct burners that generate flames using their own source of combustion air as the oxygen content in the surrounding flue-gas flow is typically very low. The difficulty of this approach is flame quenching by the flue gas that may cause incomplete combustion or even flame blowout. Using external burners with combustion chambers outside on the other hand is also inconvenient due to difficulties handling high-temperature combustion products and distributing it across the flow of flue gas. The applications of this type need to be carefully custom engineered. For minimizing the stack losses, it is important that reheat systems operate with minimum amounts of combustion air and a uniform temperatures downstream. To achieve these goals, CFD modeling (see Volume 1, Chapter 13) is usually utilized. Figure 3.43 shows one of the CFD modeling plots of the reheat process, showing an example of such a solution with side-fired burners and refractory-lined flame shields positioned across the duct designed for very low CO emissions generated by the reheat burners. The key feature of the design is a sufficient delay in mixing of the burner combustion products with surrounding flue gas under different operating scenarios.

Another burner concept that can be applied for such application would be a *ProLine*™ burner originally designed as a forced air for low-NOx and low-CO duct burner for air-heating applications. The burner utilizes simulated premixed lean combustion concepts. Figure 3.44 shows the appearance of the burner flame when firing natural gas.

3.9 Summary

The variety of boiler burners described in this chapter is still a small part of the burners used in industrial boilers. The diversity of the equipment, fuels, geographical regions, and environmental standards makes the need to custom engineer the majority of combustion systems even when previously developed burner concepts are

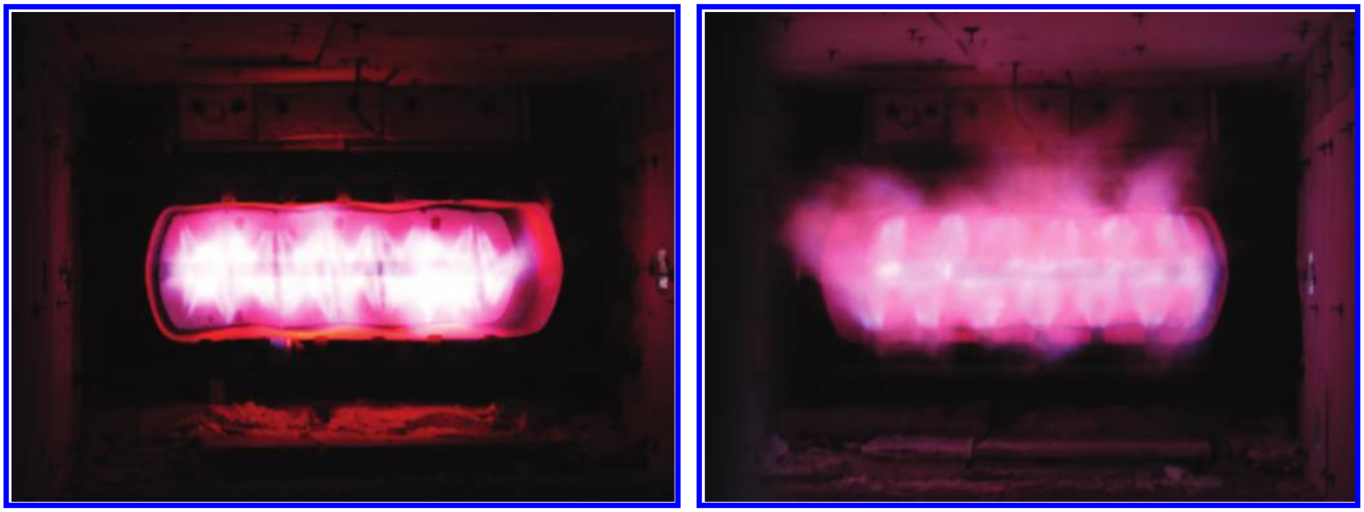


FIGURE 3.44
Coen *ProLine*™ burner flames at low- (left) and high-fire (right) operation.

applied. The customization starts with selecting the most suitable burner type based on the available fuels, capacity demand, and environmental requirements. The next typical step is balancing between economics and performance. During that process, some compromises to the equipment specification can be made. In retrofit applications, the variety of initial options with respect to what to reuse, refurbish, modify, or replace often includes both steps to jointly become an iterative optimization process. In some cases, none of the prior-developed burner types are sufficiently suited for the application, prompting development of the new concepts.

From this perspective, the issues discussed in this chapter and also in Volume 2, Chapter 2, on combustion controls should help end users and boiler designers to better understand the issues and options that combustion engineers face when selecting and designing burners and combustion systems. The days have long passed when a burner and a combustion system were simple and separate.

Changes in burner designs and system applications have been driven primarily by emissions since the passage of the Clean Air Act in the early 1970s. Of necessity, burner designs have become more complex requiring a much deeper technical understanding of the system interactions and required controls.

NOx emissions are typically about 10 times lower, and CO emissions are on the order of 100 times lower than in the 1970s. Each application of a burner has to be evaluated based on the details of a particular furnace design. The performance of boilers and especially their superheaters and reheaters is heavily impacted by the utilized low-NOx techniques. This chapter discusses custom-engineered solutions. In many ways, every application of a burner is a custom-engineered solution. The only question will be the degree of custom engineering that is required. Every burner type will perform differently in different furnaces. Each application has technical and economic limitations, especially retrofit projects.

References

1. S. C. Stultz and J. B. Kitto, *Steam: Its Generation and Use*, 40th edn., Babcock & Wilcox—A McDermott Company, Barberton, OH, 1992.
2. C. E. Baukal, *The John Zink Combustion Handbook*, John Zink Co LLC, CRC Press, Boca Raton, FL, 2001.
3. C. E. Baukal, V. Y. Gershtein, and X. Li, *Computational Fluid Dynamics in Industrial Combustion*, CRC Press, Boca Raton, FL, 2001.
4. C. E. Baukal, *Industrial Combustion Testing*, CRC Press, Boca Raton, FL, 2011.
5. J. M. Beer and N. A. Chigier, *Combustion Aerodynamics*, Robert E. Krigier Publishing Company, Malabar, FL, 1983.
6. A. K. Gupta, *Swirl Flows*, Abacus Press, Tunbridge Wells, U.K., 1984.
7. Y. B. Zeldovich, P. Y. Sadonikov, and D. A. Frank-Kamenetskii, *Oxidation of Nitrogen in Combustion*, Academy of Science, USSR, Institute of Chemical Physics, Moscow-Leningrad, Russia, 1947.
8. C. P. Fenimore, Formation of nitric oxide in premixed hydrocarbon flames, in *Thirteenth Symposium on Combustion*, The Combustion Institute, Pittsburgh, PA, pp. 373–380, 1971.
9. W. Bartok, A. R. Crawford, H. J. Hall, E. H. Manny, and A. Skopp, Stationary sources and control of nitrogen oxide emissions, in *Proceedings of the 2nd International Clean Air Congress*, Academic Press, New York, pp. 80–90, 1971.
10. NOx controls for utility boilers, in *Proceedings of the Electric Power Research Institute*, July 7–9, Cambridge, MA, 1992.

11. Turner, D. Al., R. L. Andrews, and C. W. Siegmund, Influence of combustion modifications and fuel nitrogen oxides emission from fuel oil combustion, *AIChE Symp. Ser.*, 68, 55, 1972.
12. A. H. Rawdon and S. A. Johnson, Application of NOx control technology to power boilers, in presented at the *American Power Conference*, May 10, 1973.
13. V. Lifshits and S. Drennan, *Development of an Ultra Low NOx Burner with widened Stability Limits*, American Flame Research Committee, Maui, Hawaii, 1998.
14. *Ultra Low NOx Gas-Fired Burner with Air Preheat*, CARB Contract Number 94–354, Final Report, Prepared for California Air Resources Board, California Environmental Protection Agency, November 2000.
15. V. Lifshits and S. Londerville, Vibration resistant low NOx burner, U.S. Patent # 5,310,337, issued May 10, 1994.
16. V. Lifshits and S. Londerville, Vibration resistant low NOx burner, U.S. Patent No. 5,460,512, issued October 24, 1995.
17. V. Lifshits, Development of a high performance versatile low NOx burner, in presented at *AFRC International Symposium*, Baltimore, MD, September 1996.
18. S. J. Bortz, Apparatus and method for reducing NOx, CO and hydrocarbon emissions when burning gaseous fuels, U.S. Patent No. 5,407,347, issued April 18, 1995.
19. V. Lifshits, Low NOx fuel gas burner, U.S. Patent No: 6,027,330, issued February 22, 2000.
20. V. Lifshits, Energy efficient low NOx burner and method of operating same, U.S. Patent No: 7,422,427, issued September 9, 2008.
21. V. Lifshits, Energy efficient ultra low NOx burner with reduced flue gas recirculation, in *P16th IFRF Members' Conference Symposium*, Boston, MA, June 2009.
22. D. Giovanni, Atomization and Swirler design techniques for improved NOx emissions, NOx Controls for Utility Boilers, in *EPRI Meeting*, Cambridge, MA, July 7–9, 1992.
23. V. Lifshits and G. Crovato, Experience with high efficiency, low emission burners to improve plant operation, in *Latin America Power'98, Conference Papers*, Buenos Aires, Argentina, pp. 592–601, 1998.

4

Duct Burners

Peter F. Barry, Stephen L. Somers, and Steve Londerville

CONTENTS

4.1 Introduction

Linear and in-duct burners were used for many years to heat air in drying operations before their general use in cogeneration systems. Some of the earliest systems premixed fuel and air in an often complicated configuration that fired into a re-circulating process air stream. The first uses were in high-temperature oxygen-depleted streams downstream of gas turbines in the early 1960s were to provide additional steam for process use in industrial applications and for electrical peaking plants operating steam turbines. As gas turbines have become larger and more efficient, duct burner supplemental heat input has increased correspondingly.

Linear burners are applied where it is desired to spread heat uniformly across a duct, whether in ambient air or oxygen-depleted streams. In-duct designs are more commonly used in fluidized bed boilers and small cogeneration systems.

4.2 Applications

4.2.1 Cogeneration

Cogeneration implies simultaneous production of two or more forms of energy, most commonly electrical (electric power), thermal (steam, heat transfer fluid, or hot water), and pressure (compressor). The basic process involves combustion of a fossil fuel in an engine (reciprocating or turbine) that drives an electric generator, coupled with a recovery device that converts heat from the engine exhaust into a usable energy form. Production of recovered energy can be increased independently of the engine through supplementary firing provided by a special burner type known as a duct burner. Most modern systems will also include flue gas emission control devices. A typical plant schematic is shown in Figure 4.1. Aerial views of typical combined cycle electric power plants are shown in Figures 4.2 and 4.3.

Reciprocating engines (typically diesel cycle) are used in smaller systems (10 MW = 34 × 10^6 Btu/h and lower) and offer the advantage of lower capital and maintenance costs, but produce relatively high levels of pollutants. Turbine engines are used in both small and large systems (3 MW = 10 × 10^6 Btu/h and above) and, although more expensive, generally emit lower levels of air pollutants.

Fossil fuels used in cogeneration systems can consist of almost any liquid or gaseous hydrocarbon, although natural gas and various commercial-grade fuel oils are most commonly used. Mixtures of hydrocarbon gases and hydrogen found in plant fuel systems are often used in refining and petrochemical applications. Duct burners are capable of firing all fuels suitable for the engine/turbine, as well as many that are not, including heavy oils and waste gases.

Heat recovery for large systems is usually accomplished by convective heat transfer in a boiler (commonly referred to as a heat recovery steam generator, also known by the acronym HRSG). Smaller systems utilize either a steam or hot water boiler, or, alternatively, some type of air-to-air heat exchanger or direct transfer to a process.

Supplementary firing is often incorporated into the boiler/HRSG design as it allows increased production of steam as demanded by the process. The device that provides the supplementary firing is a duct burner, so called because it is installed in the duct connecting the engine/turbine exhaust to the heat recovery device, or just downstream of a section of the HRSG superheater (see Figures 4.4 and 4.5). Oxygen required for the combustion process is provided by the turbine exhaust gas (TEG).

4.2.2 Combined Cycle

Combined cycle systems incorporate all components of the simple cycle configuration with the addition of a steam turbine/generator set powered by the HRSG. This arrangement is attractive when the plant cannot be located near an economically viable steam user. Also, when used in conjunction with a duct burner, the steam turbine/generator can provide additional power during periods of high or "peak" demand.

4.2.3 Air Heating

Duct burners are suitable for a wide variety of direct-fired air heating applications where the physical arrangement requires mounting inside a duct, and particularly for processes where the combustion air is at an elevated temperature and/or contains less than 21% oxygen. Examples include the following:

- *Fluidized bed boilers* (see Figure 4.6): Where burners are installed in combustion air ducts and used only to provide heat to the bed during startup. At cold conditions, the burner is fired at maximum capacity with fresh ambient air, but as combustion develops in the bed, cross-exchange with hot stack gas increases the air temperature and velocity. Burners are shut off when the desired air preheat is reached and the bed can sustain combustion unaided.

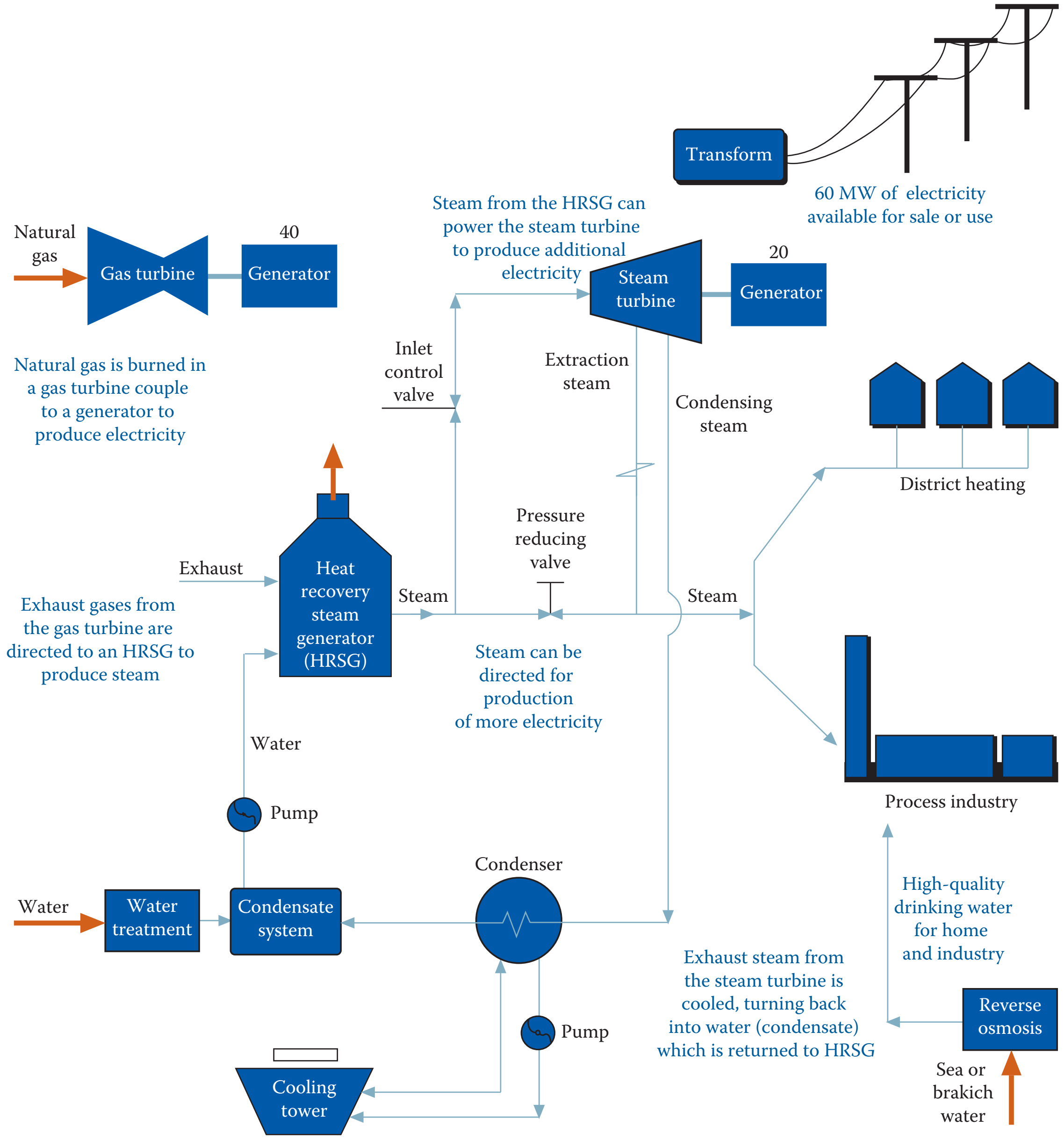


FIGURE 4.1
Typical cogeneration plant schematic.

- *Combustion air blower inlet preheat*: Where burners are mounted upstream of a blower inlet to protect against thermal shock caused by ambient air in extremely cold climates (–40°F/°C and below). This arrangement is only suitable when the air will be used in a combustion process as it will contain combustion products from the duct burner.
- *Drying applications:* Where isolation of combustion products from the work material is not required, such as certain paper and wallboard manufacturing operations.

4.2.4 Fume Incineration

Burners are mounted inside ducts or stacks carrying exhaust streams primarily composed of air with varying concentrations of organic contaminants. Undesirable components are destroyed, both by an increase in the gas stream bulk temperature and through contact with localized high temperatures created in the flame envelope. Particular advantages of the duct burner include higher thermal efficiency as no outside air is used, lower operating cost as no blower is required, and improved destruction efficiency resulting from distribution of the flame across the duct section with grid-type design.

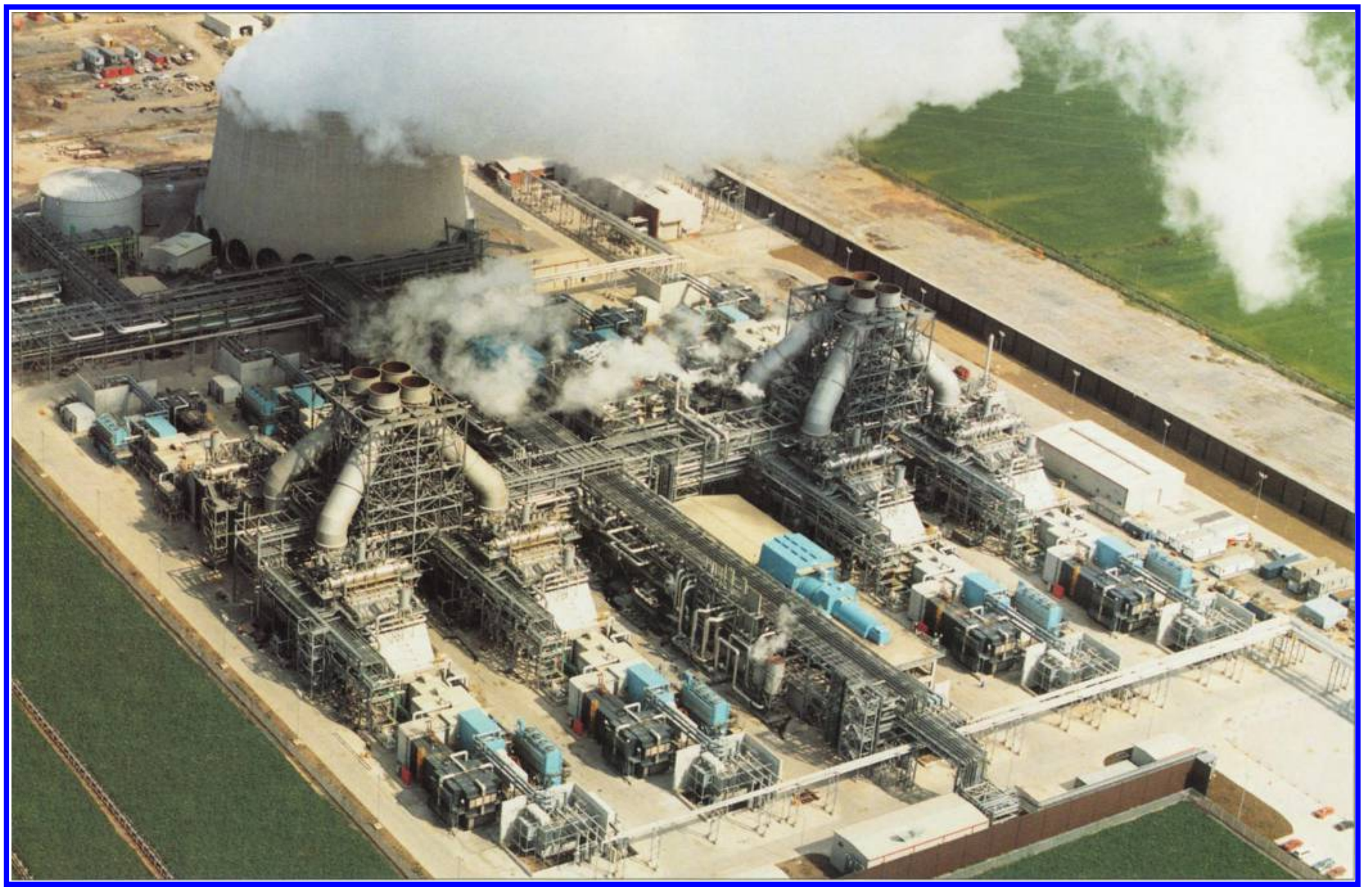


FIGURE 4.2
Cogeneration plant at Teesside, England. (Courtesy of Nooter/Eriksen, Fenton, MO.)

4.2.5 Stack Gas Reheat

Mounted at or near the base of a stack, heat added by a duct burner will increase natural draft, possibly eliminating a need for induced draft or eductor fans. In streams containing a large concentration of water vapor, the additional heat can also reduce or eliminate potentially corrosive condensation inside the stack. A source of ambient augmenting combustion air is often added if the stack gas oxygen concentration is low. This arrangement may also provide a corollary emissions reduction benefit (see Section 4.7). A discussion of testing duct burner performance is given in Reference [1].

4.3 Burner Technology

4.3.1 In-Duct or Inline Configuration

Register or axial flow burner designs are adapted for installation inside a duct. The burner head is oriented such that the flame will be parallel to and co-flow with the air or TEG stream, and the fuel supply piping is fed through the duct side wall, turning 90° as it enters the burner (see Figure 4.7). Depending on the total firing rate and duct size, one burner may be sufficient, or several may be arrayed across the duct cross section. Inline burners typically require more air/TEG pressure drop, produce longer flames, and offer a less uniform heat distribution than grid-type. On the other hand, they are more flexible in burning liquid fuels, can be more easily modified to incorporate augmenting air, and sometimes represent a less expensive option for high firing rates in small ducts without sufficient room for grid elements.

4.3.2 Grid Configuration (Gas Firing)

A series of linear burner elements that span the duct width are spaced at vertical intervals to form a grid. Each element is comprised of a fuel manifold pipe fitted with a series of flame holders (or wings) along its length. Fuel is fed into one end of the manifold pipe and discharged through discrete multi-port tips attached at intervals along its length, or through holes drilled

FIGURE 4.3
Combination (oil and gas)-fired duct burners at Dahbol, India. (Courtesy of Enron, Houston, TX.)

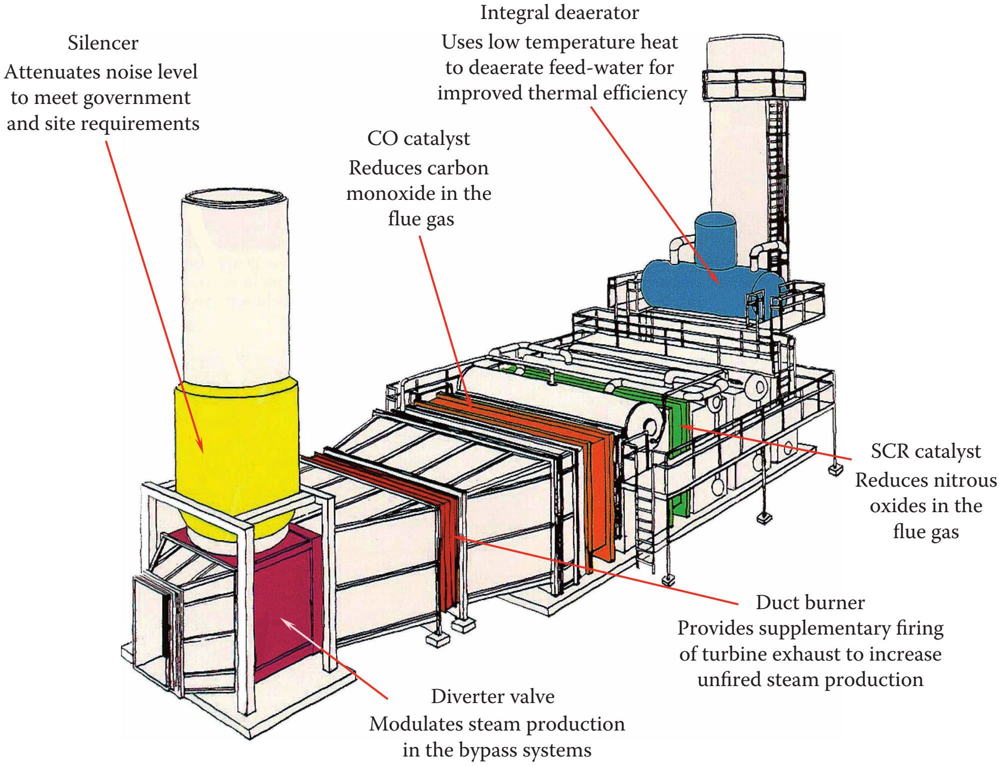


FIGURE 4.4
Typical location of duct burners in an HRSG. (Courtesy of Deltak, Plymouth, MN.)

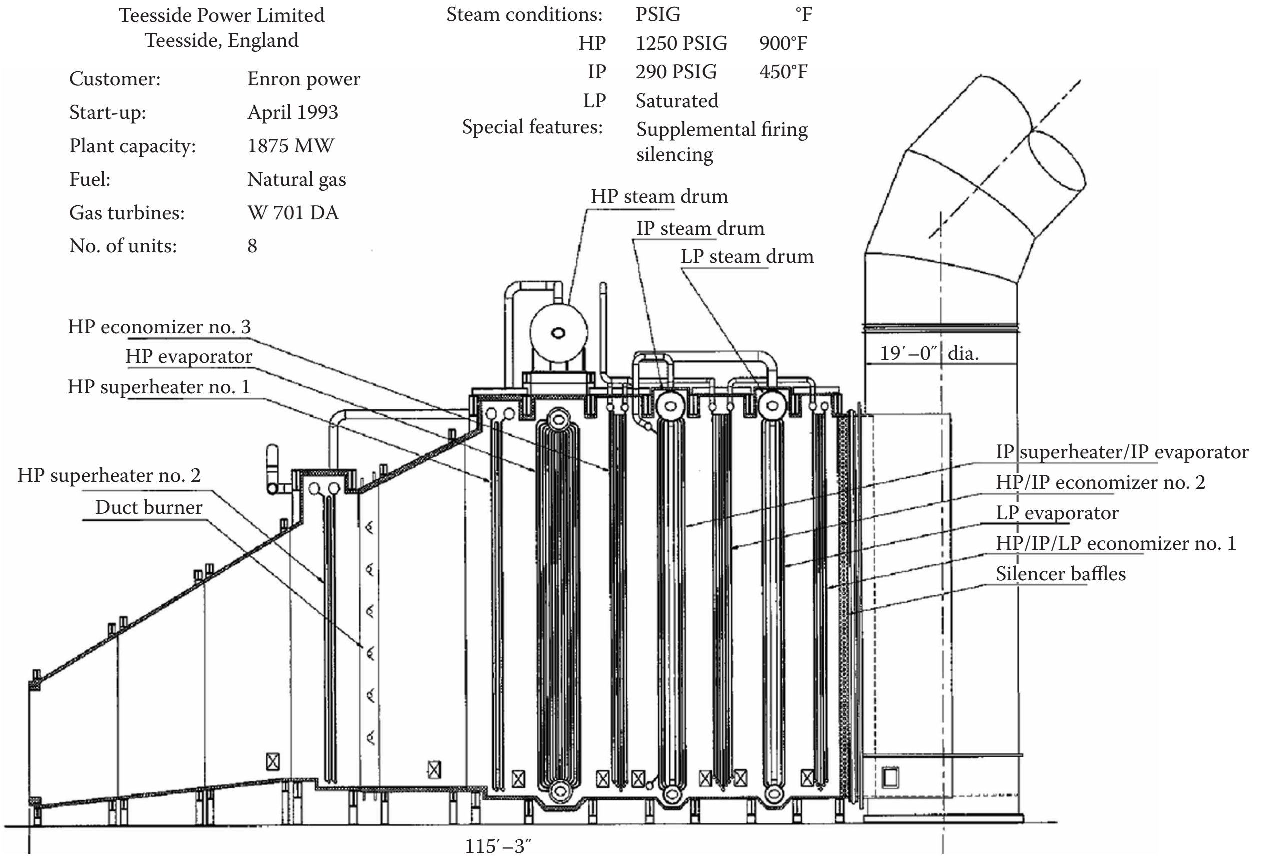


FIGURE 4.5
Schematic of HRSG at Teesside, England. (Courtesy of Nooter/Eriksen, Fenton, MO.)

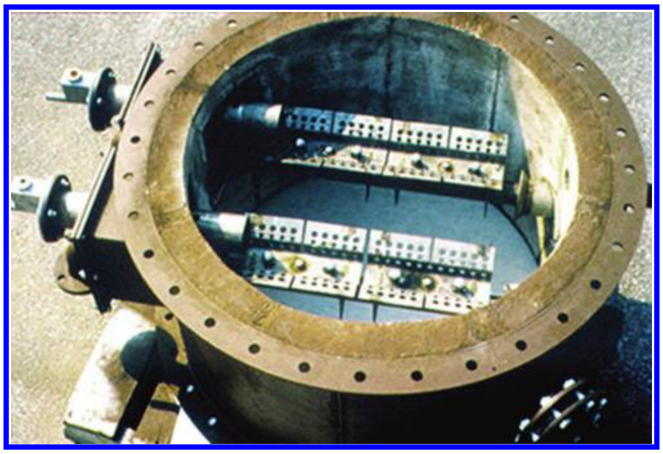


FIGURE 4.6
Fluidized bed startup duct burner.

directly into the pipe. Gas ports are positioned such that fuel is injected in co-flow with the TEG. The wings meter the TEG or airflow into the flame zone, thus developing eddy currents that anchor ignition. They also shield the flame in order to maintain suitably high flame temperatures, thereby preventing excessive flame cooling that might cause high emissions. Parts exposed to TEG and the flame zone are typically of high-temperature alloy construction (see Figures 4.8 and 4.9).

4.3.3 Grid Configuration (Liquid Firing)

As with the gas-fired arrangement, a series of linear burner elements comprised of a pipe and flame holders (wings) span the duct width. However, instead of multiple discharge points along the pipe length, liquid fuel is injected downstream of the element through the duct sidewall, and directed parallel to the flame holders (cross-flow to the TEG). This configuration utilizes the duct cross section for containment of the flame length, thus allowing a shorter distance between the burner and downstream boiler tubes (see Figure 4.10). The injection device, referred to as a side-fired oil gun, utilizes a mechanical nozzle supplemented by low-pressure air (2–8 psi) (14–55 kPa) to break the liquid fuel into small droplets (atomization) that will vaporize and readily burn. Although most commonly used for light fuels, this arrangement is also suitable for some heavier fuels,

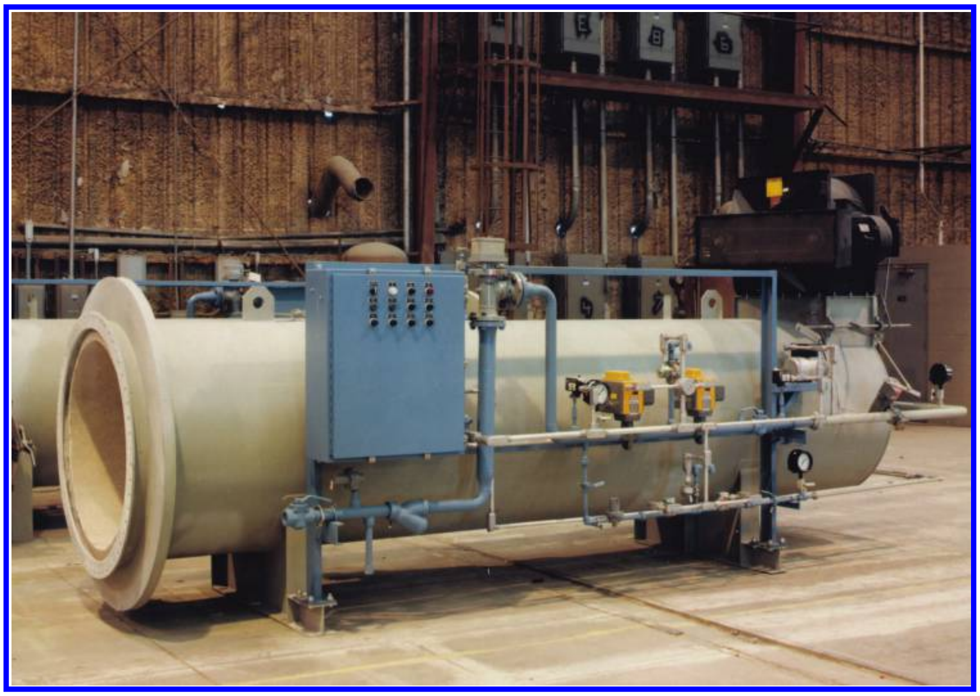


FIGURE 4.7
An inline burner.

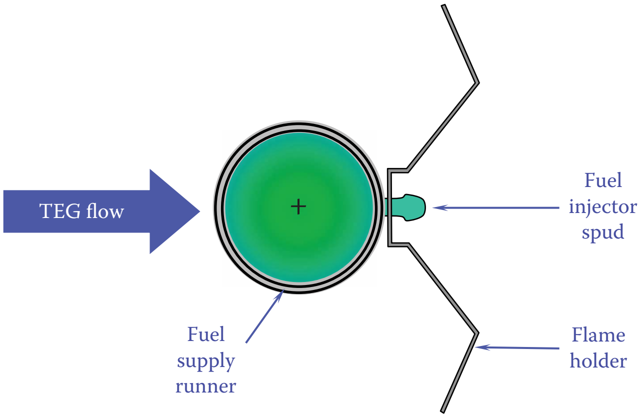


FIGURE 4.8
Linear burner elements.

FIGURE 4.9
Gas flame from a grid burner.

where the viscosity can be lowered by heating. In some cases, high pressure steam may be required, instead of low-pressure air, for adequate atomization of heavy fuels.

4.4 Fuels

4.4.1 Natural Gas

Natural gas is, by far, the most commonly used fuel because it is readily available in large volumes throughout much of the industrialized world. Because of its ubiquity, its combustion characteristics are well understood, and most burner designs are developed for this fuel.

4.4.1.1 Refinery/Chemical Plant Fuels

Refineries and chemical plants are large consumers of both electrical and steam power, which makes them ideal candidates for cogeneration. In addition, these plants maintain extensive fuel systems to supply the various direct- and indirect-fired processes, as well as to make the most economical use of residual products.

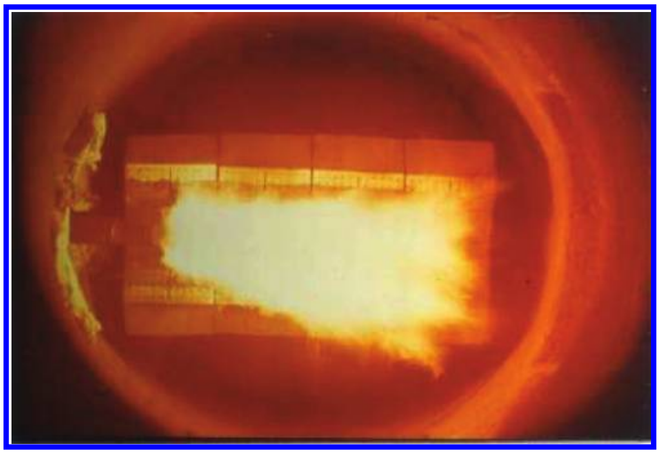


FIGURE 4.10
Oil flame from a side-fired oil gun.

This latter purpose presents special challenges for duct burners because the available fuels often contain high concentrations of unsaturated hydrocarbons with a tendency to condense and/or decompose inside burner piping. The location of burner elements inside the TEG duct, surrounded by high-temperature gases, exacerbates the problem. Plugging and failure of injection nozzles can occur, with a corresponding decrease in online availability and an increase in maintenance costs.

With appropriate modifications, however, duct burners can function reliably with most hydrocarbon-based gaseous fuels. Design techniques include insulation of burner element manifolds, insulation and heat tracing of external headers and pipe trains, and fuel/steam blending. Steam can also be used to periodically purge the burner elements of solid deposits before plugging occurs.

4.4.1.2 *Low Heating Value*

By-product gases produced in various industrial processes such as blast furnaces, coke ovens, and flexicokers, or from mature landfills, contain combustible compounds along with significant concentrations of inert components, thus resulting in relatively low heating values (range of 50–500 Btu/scf = 1.9–19 MJ/m^3). These fuels burn more slowly and at lower temperatures than conventional fuels, and thus require special design considerations. Fuel pressure is reduced to match its velocity to flame speed, and some form of shield or "canister" is employed to provide a protected flame zone with sufficient residence time to promote complete combustion before the flame is exposed to the quenching effects of TEG.

Other considerations that must be taken into account are moisture content and particulate loading. High moisture concentration results in condensation within the fuel supply system, which, in turn, produces corrosion and plugging. Pilots and igniters are particularly susceptible to the effects of moisture because of small fuel port sizes, small igniter gap tolerance, and the insulation integrity required to prevent "shorting" of electrical components. A well-designed system might include a knockout drum to remove liquids and solids, insulation and heat tracing of piping to prevent or minimize condensation, and low-point drains to remove condensed liquids. Problems are usually most evident after a prolonged period of shutdown.

Solid particulates can cause plugging in gas tip ports or other fuel system components and should therefore be removed to the maximum practical extent. In general, particle size should be no greater than 25% of the smallest port, and overall loading should be no greater than 5 ppm by volume.

4.4.1.3 *Liquid Fuels*

In cogeneration applications, duct burners are commonly fired with the same fuel as the turbine, which is typically limited to light oils such as No. 2 or naphtha. For other applications, specially modified side-fired guns or an inline design can be employed to burn heavier oils such as No. 6 and some waste fuels.

4.5 Combustion Air and Turbine Exhaust Gas

4.5.1 Temperature and Composition

When used for supplementary firing in HRSG cogeneration applications, the oxygen required for the combustion reaction is provided by the residual in the TEG instead of from a new, external source of air. Because this gas is already at an elevated temperature, duct burner thermal efficiency can exceed 90% as very little heat is required to raise the combustion products' temperature to the final fired temperature. TEG contains less oxygen than fresh air, typically between 11% and 16% by volume, which, in conjunction with the TEG temperature, will have a significant effect on the combustion process. As the oxygen concentration and TEG temperature become lower, emissions of CO and unburned hydrocarbons (UHCs) occur more readily, eventually progressing to combustion instability. The effect of low oxygen concentration can be partially offset by higher temperatures; conversely, higher oxygen concentrations will partially offset the

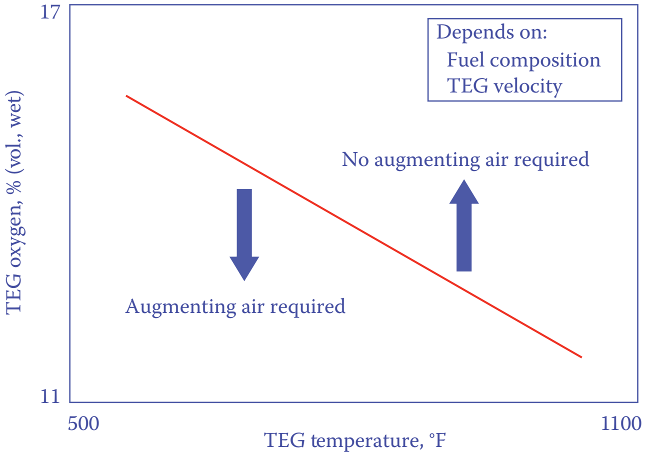


FIGURE 4.11
Approximate requirement for augmenting air.

detrimental effects of low TEG temperatures. This relationship is depicted graphically in Figure 4.11. Duct burner emissions are discussed in more detail elsewhere in this chapter.

4.5.2 Turbine Power Augmentation

During periods of high electrical demand, various techniques are employed to increase power output, and most will increase the concentration of water vapor in the TEG. The corresponding effect is a reduction in TEG oxygen concentration and temperature with consequent effects on duct burner combustion. Depending on the amount of water vapor used, CO emissions may simply rise, or in extreme cases the flame may become unstable. The former effect can be addressed with an allowance in the facility operating permit or by increasing the amount of CO catalyst in systems so equipped. The latter requires air augmentation, a process whereby fresh air is injected at a rate sufficient to raise the TEG oxygen concentration to a suitable level.

4.5.3 Velocity and Distribution

Regardless of whether TEG or fresh air is used, velocity across flame stabilizers must be sufficient to promote mixing of the fuel and oxygen, but not so great as to prevent the flame from anchoring to the burner. Grid-type configurations can generally operate at velocities ranging from 20 to 90 ft/s or 6 to 27 m/s and pressure drops of less than 0.5 in. water column. Inline or register burners typically require velocities of 100–150 ft/s (31–46 m/s) with a pressure drop of 2–6 in. water column (5–15 mbar).

Grid burners are designed to distribute heat uniformly across the HRSG or boiler tube bank, and thus require a reasonably uniform distribution of the TEG or air to supply the fuel with oxygen. Inadequate distribution causes localized areas of low velocity, resulting in poor flame definition along with high emissions of CO and UHCs. Turbine exhaust flow patterns, combined with rapidly diverging downstream duct geometry, will almost always produce an unsatisfactory result that must be corrected by means of a straightening device. Likewise, the manner in which ambient air is introduced into a duct can also result in flow maldistribution, requiring some level of correction. Selection and design of flow-straightening devices are discussed elsewhere in this chapter (see Figure 4.12).

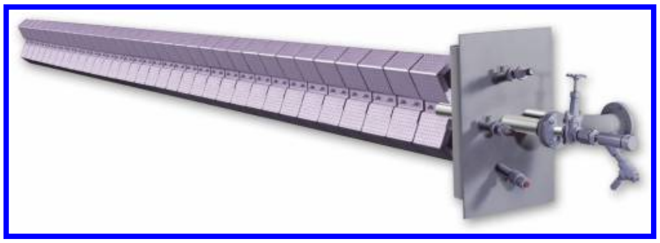


FIGURE 4.12
Drawing of a duct burner arrangement.

In instances where the bulk TEG or air velocity is lower than required for proper burner operation, flow straightening alone is not sufficient and it becomes necessary to restrict a portion of the duct cross section at or near the plane of the burner elements, thereby increasing the “local” velocity across flame holders. This restriction, also referred to as blockage, commonly consists of unfired runners or similar shapes uniformly distributed between the firing runners to reduce the open flow area.

Inline or register burners inject fuel in only a few (or possibly only one) positions inside the duct and can therefore be positioned in an area of favorable flow conditions, assuming the flow profile is known. On the other hand, downstream heat distribution is less uniform than with grid designs, and flames may be longer. As with grid-type burners, in some cases, it may be necessary to block portions of the duct at or just upstream of the burners to force a sufficient quantity of TEG or air through the burner.

4.5.4 Ambient Air Firing (Air-Only Systems and HRSG Backup)

Velocity and distribution requirements for air systems are similar to those for TEG, although the inlet temperature is not a concern because of the relatively higher oxygen concentration. As with TEG applications, the burner elements are exposed to the products of combustion, so material selection must take into account the maximum expected fired temperature.

Ambient (or fresh) air backup for HRSGs presents special design challenges. Because of the temperature difference between ambient air and TEG, designing for the same mass flow and fired temperature will result in a velocity across the burner approximately one-third that of the TEG case. If the cold condition velocity is outside the acceptable range, it will be necessary to add blockage, as described earlier. Fuel input capacity must also be increased to provide the heat required to raise the air from ambient to the design firing temperature. By far, the most difficult challenge is related to flow distribution. Regardless of the manner in which backup air is fed into the duct, a flow profile different from that produced by the TEG is virtually certain. Flow-straightening devices can therefore not be optimized for either case, but instead require a compromise design that provides acceptable results for both. If the two flow patterns are radically different, it may ultimately be necessary to alter the air injection arrangement independently of the TEG duct-straightening device.

4.5.5 Augmenting Air

As turbines have become more efficient and more work is extracted in the form of, for example, electricity, the oxygen level available in the TEG continues to get lower. To some extent, a correspondingly higher TEG temperature provides some relief for duct burner operation.

In some applications, however, an additional oxygen source may be required to augment that available in the TEG when the oxygen content in the TEG is not sufficient for combustion at the available TEG temperature. If the mixture adiabatic flame temperature is not high enough to sustain a robust flame in the highly turbulent stream, the flame may become unstable.

The problem can be exacerbated when the turbine manufacturer adds large quantities of steam or water for NOx control and power augmentation. A corresponding drop in the TEG temperature and oxygen concentration occurs because of dilution. The TEG temperature is also reduced in installations where the HRSG manufacturer splits the steam superheater and places tubes upstream of the duct burner.

With their research and development facilities, manufacturers have defined the oxygen requirement with respect to TEG temperature and fuel composition and are able to quantify the amount of augmenting air required under most conditions likely to be encountered. It is usually not practical to add enough air to the turbine exhaust to increase the oxygen content to an adequate level. Specially designed runners are therefore used to increase the local oxygen concentration. In cases where augmenting air is required, the flow may be substantial: from 30% to 100% of the theoretical air required for the supplemental fuel.

The augmenting air runner of one manufacturer consists of a graduated air delivery tube designed to ensure a constant velocity across the length of the tube. Equal distribution of augmenting air across the face of the tube is imperative. The augmenting air is discharged from the tube into a plenum then passes through a second distribution grid to further equalize flow. The air passes through perforations in the flame holder, where it is intimately mixed with the fuel in the primary combustion zone. This intimate mixing ensures corresponding low CO and UHC emissions under most conditions likely to be encountered. Once the decision has been made to supply augmenting air to a burner, it is an inevitable result of the design that the augmenting air will be part of the normal operating regime of the combustion runner.

4.5.6 Equipment Configuration and TEG/Combustion Airflow Straightening

The turbine exhaust gas/combustion air velocity profile at the duct burner plane must be within certain limits to ensure good combustion efficiency; in cogeneration applications, this is rarely achieved without flow-straightening devices. Even in non-fired configurations, it may be necessary to alter the velocity distribution to make efficient use of boiler heat transfer surface. Figure 4.13 shows a comparison of flow variation with and without flow straightening.

Duct burners are commonly mounted in the TEG duct upstream of the first bank of heat transfer tubes, or they

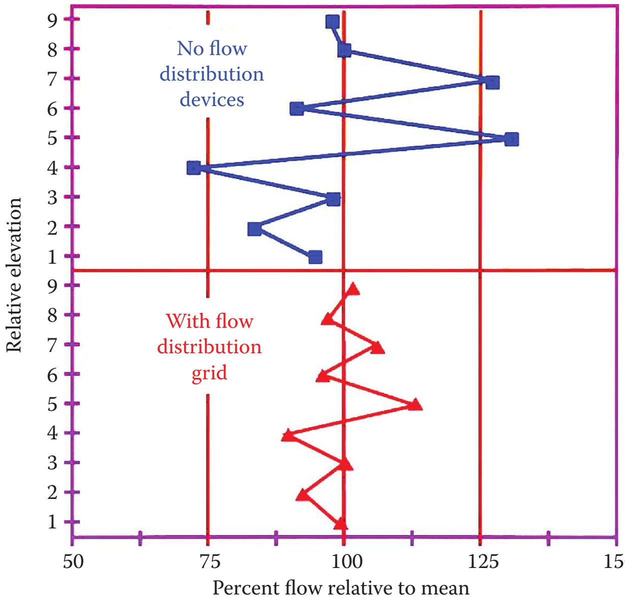


FIGURE 4.13
Comparison of flow variation with and without straightening device.

may be nested in the boiler superheater between banks of tubes. In the former case, a straightening device would be mounted just upstream of the burner, while in the latter it is mounted either upstream of the first tube bank or between the first tube bank and (upstream of) the burner. Although not very common, some HRSG design configurations utilize two stages of duct burners with heat transfer tube banks in between and a flow-straightening device upstream of the first burner. Such an arrangement is, however, problematic because the TEG downstream of the first-stage burner may not have the required combination of oxygen and temperature properties required for proper operation of the second-stage burner.

Perforated plates that extend across the entire duct cross section are most commonly used for flow straightening because experience has shown they are less prone to mechanical failure than vane-type devices, even though they require a relatively high pressure drop. The pattern and size of perforations can be varied to achieve the desired distribution. Vanes can produce comparable results with significantly less pressure loss, but require substantial structural reinforcement to withstand the flow-induced vibration inherent in HRSG systems. Regardless of the method used, flow pattern complexity—particularly in TEG applications—usually dictates the use of either physical or computational fluid dynamic (CFD) modeling for design optimization.

4.6 Physical Modeling

TEG/airflow patterns are determined by inlet flow characteristics and duct geometry and are subject to both position and time variation. Design of an efficient (low pressure loss) flow-straightening device is therefore not a trivial exercise, and manual computational methods are impractical. For this reason, physical models (see Volume 1, Chapter 11), commonly 1:6 or 1:10 scale, are constructed, and flow characteristics are analyzed by flowing air with smoke tracers or water with polymer beads through the model (see Figure 4.14). Although this method produces reliable results, tests conducted at ambient conditions (known as "cold flow") are not capable of simulating the buoyant effects that may occur at elevated temperatures.

4.6.1 CFD Modeling

Flow modeling with CFD (see Volume 1, Chapter 13), using a computer-generated drawing of the inlet duct geometry, is capable of predicting flow patterns and pressure drop in the turbine exhaust flow path. The model can account for swirl flow in three dimensions, accurately predict pressure drop, and subsequently help design a suitable device to provide uniform flow. The CFD model must be quite detailed to calculate flow patterns incident and through a perforated grid or

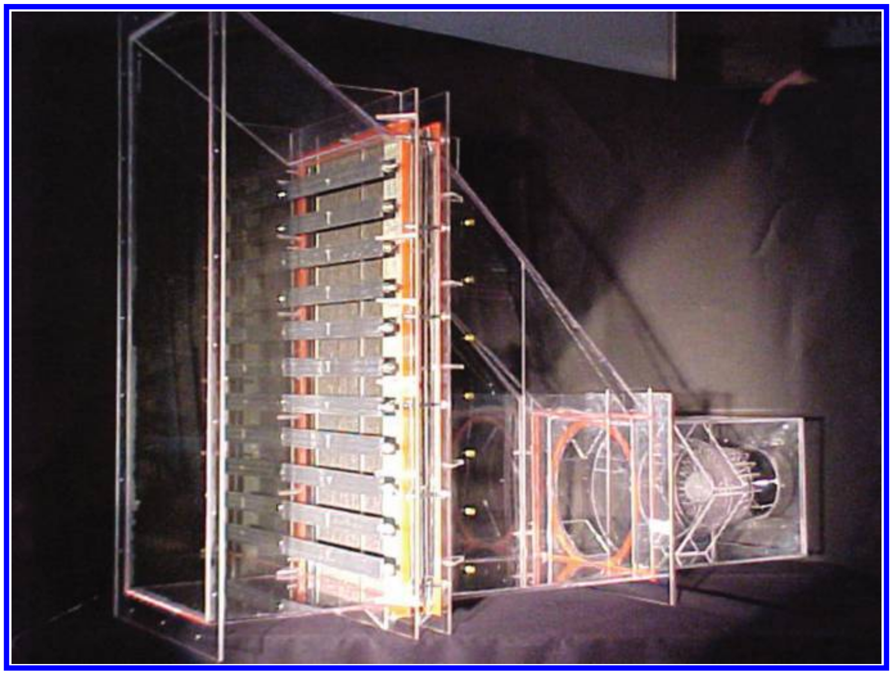


FIGURE 4.14
Physical model of a duct burner array.

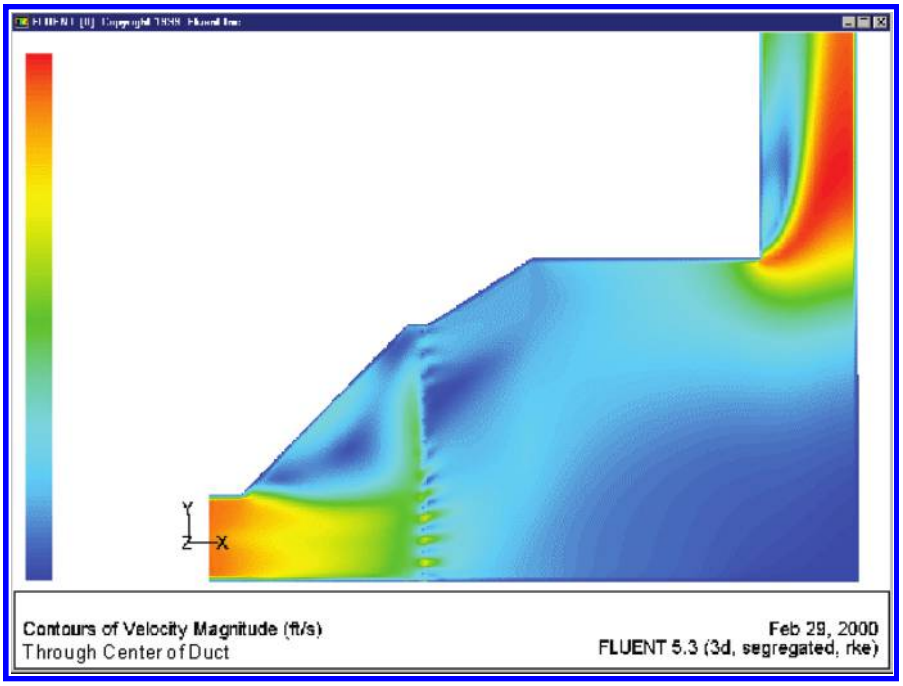


FIGURE 4.15
Sample result of CFD modeling performed on an HRSG inlet duct.

tube bank while also keeping the overall model solution within reasonable computation time. Combustion effects can be included in the calculations at the cost of increased computation time.

CFD simulation has the capability to provide complete information provided the aforementioned is true. The issue of validity has been a hot topic for years. A Department of Energy report[2] has cited CFD to be capable of

1. Predicting catastrophic failure
2. Qualitative trends and parametric analysis
3. Visualization
4. Predicting non-reacting gaseous flows
5. Quantitative analysis of gas velocity and temperature patterns
6. Qualitative analysis of radiation heat transfer
7. Flame dynamics and shape
8. Effects geometry changes
9. Models of temperature and heat release patterns and qualitative trends associated with major species
10. Detailed burner codes with heating process

For combustion systems, CFD is the only general-purpose simulation model capable of modeling reacting flows in order to predict emissions, heat transfer, and other furnace parameters. Figure 4.15 shows a sample result of CFD modeling performed on an HRSG inlet duct.

4.6.1.1 *Wing Geometry: Variations*

4.6.1.1.1 *Flameholders*

Design of the flame stabilizer, or flameholder, is critical to the success of supplementary firing. Effective emission control requires that the TEG be metered into the flame zone in the required ratio to create a combustible mixture and ensure that the combustion products do not escape before the reactions are completed. In response to new turbine and HRSG design requirements, each duct burner manufacturer has proprietary designs developed to provide the desired results.

4.6.1.1.2 *Basic Flameholder*

In its basic form, a fuel injection system and a zone for mixing with oxidant are all that is required for combustion. For application to supplemental firing, the simple design shown in Figure 4.16 consists of an internal manifold or "runner," usually an alloy pipe with fuel injection orifices spaced along the length. A bluff body plate, with or without perforations, is attached to the pipe to protect the flame zone from the turbulence in the exhaust gas duct. The low-pressure zone pulls the flame back onto the manifold. This low-cost runner may overheat the manifold, causing distortion of the metallic parts.

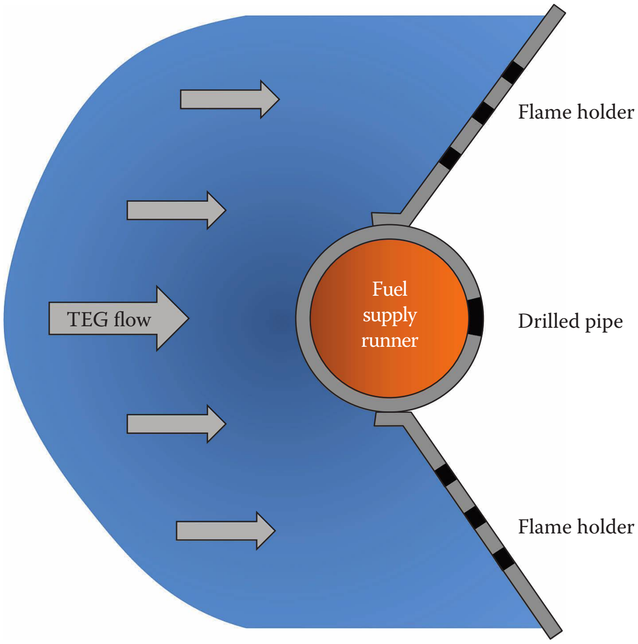


FIGURE 4.16
Drilled pipe duct burner.

Emissions are unpredictable with changing geometry, and CO is usually much higher than the current typically permitted levels of under 0.1 lb/MMBtu.

4.6.1.1.3 Low-Emission Design

Modifications to the design for lower-emission performance generally have a larger cross section in the plane normal to the exhaust flow. The increased blocked area protects the fuel injection zone and increases residence time. The NOx is reduced by the oxygen-depleted TEG and the CO/UHC is reduced by the delayed quenching. The correct flow rate of TEG is metered through the orifices in the flameholder, and the fuel injection velocity and direction are designed to enhance combustion efficiency. The flame zone is pushed away from the internal manifold ("runner" pipe), creating space for cooling TEG to bathe the runner and flameholder and enhance equipment life.

Each manufacturer approaches the geometry somewhat differently. One manufacturer uses cast alloy pieces welded together to provide the required blockage. These standard pieces often add significant weight and are difficult to customize to specific applications. Hot burning fuels, such as hydrogen, may not receive the cooling needed to protect the metal from oxidation. Alternately, fuels subject to cracking, such as propylene, may not have the oxygen needed to minimize coke buildup.

Another manufacturer supplies custom designs to accommodate velocity extremes, while maintaining low emissions. In the design shown in Figure 4.17, the flameholder is optimized with CFD and research experimentation to enhance mixing and recirculation rate. Special construction materials are easily accommodated. This supplier also uses removable fuel tips with multiple orifices, which can be customized to counteract any unexpected TEG flow distribution discovered after commercial operation. Figure 4.18 depicts the flow patterns of air/TEG and fuel in relation to the duct burner flameholder.

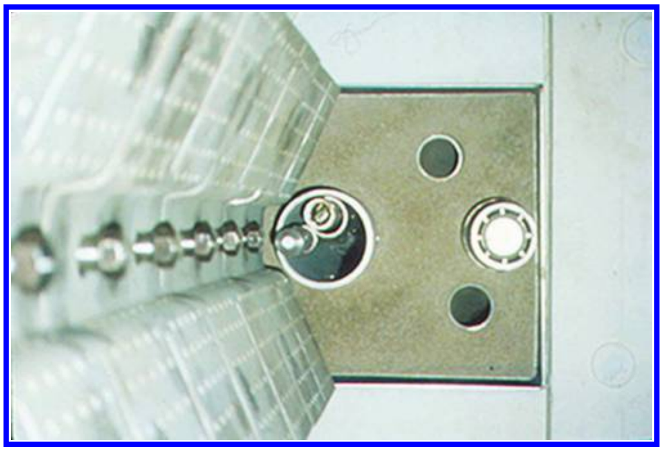


FIGURE 4.17
Low-emission duct burner.

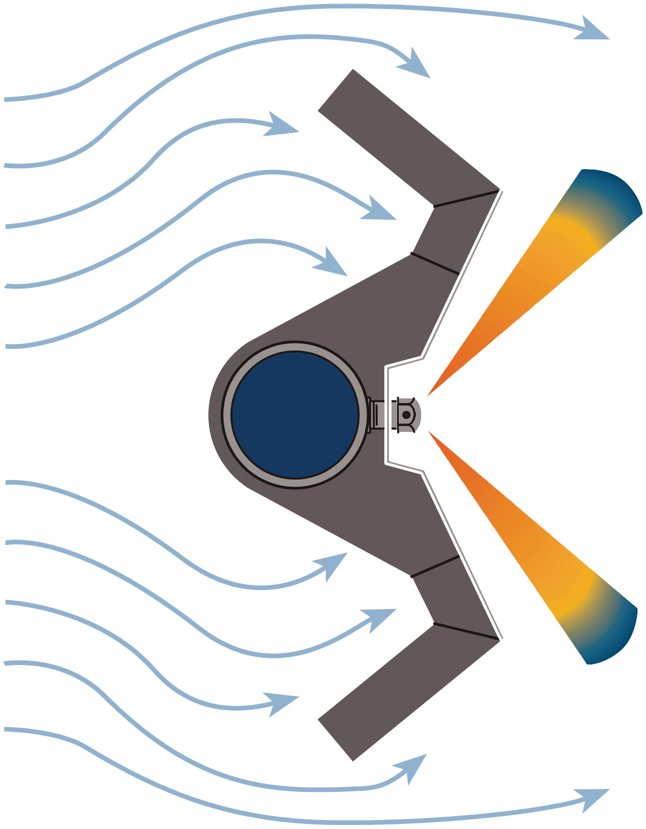


FIGURE 4.18
Flow patterns around flame stabilizer.

4.7 Emissions

Duct burner systems can either increase or reduce emissions from the generally large volume of mass flow at the input. Generally this flow includes particulates, NOx, CO, and a variety of HCs including a subset of HCs defined as volatile organic compounds (VOCs), where VOCs are defined by EPA (40 CFR 51.100, February 3, 1992) as "any compound of carbon, excluding carbon monoxide, carbon dioxide, carbonic acid, metallic carbides or ammonium carbonate, which participates in atmospheric chemical reaction." Other compounds are also exempt such as methane, ethane, methylene chloride, methyl chloroform, and other minor chemicals.

In order to accurately predict emissions, kinetic equations are created using first-order equations for oxidation in the general form of

$$\frac{d(\text{Chemical})}{dt} = -K[O_2][\text{Chemical}] \tag{4.1}$$

where

$$K = Ae^{\left[\frac{-E}{RT}\right]} \tag{4.2}$$

and

A is the pre-exponential factor/frequency factor in appropriate units
R is the universal gas constant in appropriate units
T is the absolute temperature in Kelvin
E is the activation energy, usually listed in kcal/mol

4.7.1 Visible Plumes

Stack plumes are caused by moisture and impurities in the exhaust. Emitted NO is colorless and odorless, and NO_2 is brownish in color. If the NO_2 level in the flue gas exceeds about 15–20 ppm, the plume will take on a brownish haze. NOx also reacts with water vapor to form nitrous and nitric acids. Sulfur in the fuel may oxidize to SO_3 and condense in the stack effluent, causing a more persistent white plume.

4.7.2 NOx and NO vs. NO_2

Formation of NO and NO_2 is the subject of ongoing research to understand the complex reactions. Potentially, several oxides of nitrogen (NOx) can be formed during the combustion process, but only nitric oxide (NO) and nitrogen dioxide (NO_2) occur in significant quantities (see Volume 1, Chapter 15).

In the elevated temperatures found in the flame zone in a typical HRSG turbine exhaust duct, NO formation is favored almost exclusively over NO_2 formation. Turbine exhaust NOx is typically 95% NO and 5% NO_2. In the high-temperature zone, NO_2 dissociates to NO by the mechanism of

$$NO_2 + O + \text{Heat} \rightarrow NO + O_2$$

However, after the TEG exits the hot zone and enters the cooling zone at the boiler tubes, the reaction slows and the NO_2 is essentially fixed. At the stack outlet, the entrained NO is slowly oxidized to NO_2 through a complex photochemical reaction with atmospheric oxygen. The plume will be colorless unless the NO_2 increases to about 15 ppm, at which time a yellowish tint is visible. Care must be taken in duct burner design because NO can also be oxidized to NO_2 in the immediate post-flame region by reactions with hydroperoxyl radicals:

$$NO + HO_2 \rightarrow NO_2 + OH$$

if the flame is rapidly quenched. This quenching can occur because of the large quantity of excess TEG commonly present in duct burner applications. Conversion to NO_2 may be even higher at fuel turndown conditions where the flame is smaller and colder. NO_2 formed in this manner can contribute to "brown plume" problems and may even convert some of the turbine exhaust NO to NO_2.

Two of the principle mechanisms in which nitrogen oxides are formed:

1. *Thermal NOx:* The primary method is thermal oxidation of atmospheric nitrogen in the TEG. NOx formed in this way is called thermal NOx. As the temperature increases in the combustion zone and surrounding environment, increased amounts of N_2 from the TEG are converted to NO. Thermal NOx formation is most predominant in the peak temperature zones of the flame.
2. *Fuel-bound nitrogen NOx:* A secondary method utilized to form NOx is the reaction of oxygen with chemically bound nitrogen compounds contained in the fuel. NOx formed in this manner is called fuel NOx. Large amounts of NOx can be formed by fuels that contain molecularly bound nitrogen (e.g., amines and mercaptans). If a gaseous fuel such as natural gas contains diluent N_2, it simply behaves as atmospheric nitrogen and will form NOx only if it disassociates in the high-temperature areas. However, if the gaseous fuel contains, for example, ammonia (NH_3), this nitrogen is considered bound. In the low concentrations typically found in gaseous fuels, the conversion to NOx is close to 100% and can have a major impact on NOx emissions.

Bound nitrogen in liquid fuel is contained in the long carbon-chain molecules. Distillate oil is the most common oil fired in duct burners as a liquid fuel. The fuel-bound nitrogen content is usually low, in the range of 0.05 wt.%. Conversion to NOx is believed to be 80%–90%. For No. 6 oil, containing 0.30 wt.% nitrogen, the conversion rate to NOx would be about 50%. Other heavy waste oils or waste gases with high concentrations of various nitrogen compounds may add relatively high emissions. Consequently, fuel NOx can be a major source of nitrogen oxides and may predominate over thermal NOx.

The impact of temperature on NOx production in duct burners is not as pronounced as in, for example, fired heaters or package boilers. One reason is that both the bulk fired temperature and the adiabatic flame temperature are lower than in fired process equipment.

In the formation of NOx, the equations are similar to formation of thermal NOx and are presented as such:

$$\frac{d(\mathrm{NO})}{dt} = 2Ae^{\left(-\frac{E}{RT}\right)}(\mathrm{O}_2)_{eq}(\mathrm{N}_2) \tag{4.3}$$

and

$$(\mathrm{O}_2)_{eq} = \frac{k_o}{(RT)^{0.5}}(\mathrm{O}_2)_{eq}^{0.5} \tag{4.4}$$

One generally accepted practice is to assume (O_2) in equilibrium with (O) and (O_2) concentration using the Westenberg[3] results for k_o for (O_2) equilibrium and Zeldovich constants, A and E, as measured by Bowman.[4]

When used to provide supplementary firing of turbine exhaust, duct burners are generally considered to be "low NOx" burners. Because the turbine exhaust contains reduced oxygen, the peak flame temperature is reduced and the reaction speed for O_2 and N+ to form NOx is thus lowered. The burners also fire into much lower average bulk temperatures—usually less than 1600°F (870°C)—than process burners or fired boilers. The high-temperature zones in the duct burner flames are smaller due to large amounts of flame quenching by the excess TEG. Finally, mixing is rapid and therefore retention time in the high-temperature zone is very brief.

The same duct burner, when used to heat atmospheric air, is no longer considered "low NOx," because the peak flame temperature approaches the adiabatic flame temperature in air.

Clearly, operating conditions have a major impact on NO formation during combustion. To properly assess NOx production levels, the overall operating regime must be considered, including TEG composition, fuel composition, duct-firing temperature, and TEG flow distribution.

4.7.3 CO, UBHC, SOx, and Particulates

A general discussion of pollutant emissions is given in Volume 1, Chapter 14.

4.7.3.1 Carbon Monoxide

Carbon monoxide (CO), a product of incomplete combustion, has become a major permitting concern in gas turbine-based cogeneration plants. Generally, CO emissions from modern industrial and aero-derivative gas turbines are very low, in the range of a few parts per million (ppm). There are occasional situations in which CO emissions from the turbine increase due to high rates of water injection for NOx control or operation at partial load, but the primary concern is the sometimes large CO contribution from supplementary firing. The same low-temperature combustion environment that suppresses NOx formation is obviously unfavorable for complete oxidation of CO to CO_2. Increased CO is produced when fuels are combusted under fuel-rich conditions or when a flame is quenched before complete burnout. These conditions (see Figure 4.19) can occur if there is poor distribution of TEG to the duct burner, which causes some burner elements to fire fuel-rich and others to fire fuel-lean, depending on the efficiency of the TEG distribution device. The factors affecting CO emissions include

- TEG distribution
- Low TEG approach temperature
- Low TEG oxygen content
- Flame quench on "cold" screen tubes
- Improperly designed flame holders that allow flame quench by relatively cold TEG
- Steam or water injection

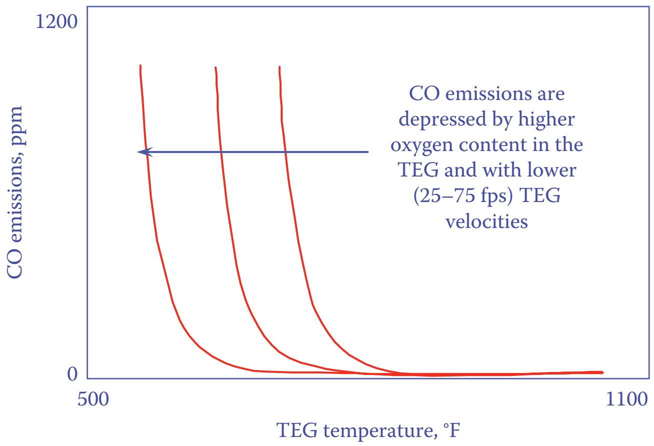


FIGURE 4.19
Effect of conditions on CO formation.

For utilization and performance prediction, kinetic data can be utilized from the literature. For instance, for CO destruction, several kinetic data are available such as[5]

$$\frac{d[\mathrm{CO}]}{dt} = -1.810^{7}\, e\left(-\frac{25{,}000}{RT}\right)(\mathrm{CO})(\mathrm{O_2})^{0.5}(\mathrm{H_2O})^{0.5}\left(\frac{P}{RT}\right)^{2} \tag{4.5}$$

Most published CO rates involve H_2O because CO destruction requires the $(OH)^{-1}$ radical to produce the reaction.

4.7.3.2 UHCs

In the same fashion as carbon monoxide generation, UHCs are formed in the exhaust gas when fuel is burned without sufficient oxygen or if the flame is quenched before combustion is complete. UHCs can consist of hydrocarbons (defined as any carbon–hydrogen molecule) of one carbon or multiple carbon atoms. The multiple carbon molecules are often referred to as long-chain hydrocarbons. UHCs are generally classified in two groups:

1. UHCs as methane
2. Non-methane hydrocarbons or VOCs

The reason for the distinction and greater concern for VOCs is that longer-chain hydrocarbons play a greater role in the formation of photochemical smog. VOCs are usually defined as molecules of two carbons or greater and are sometimes considered to be three carbons or greater. These definitions are set by local air quality control boards and vary across the United States.

UHCs can only be eliminated by correct combustion of the fuel. However, hydrocarbon compounds will always be present in trace quantities, regardless of how the HRSG system is operated.

For HC and VOC incineration, several sources are available such as Barnes et al.[6]

In general,

$$\frac{d(\mathrm{C_aH_b})}{dt} = -5.52(10^{8})P^{-0.815}\mathrm{T}e^{\frac{(12{,}200)}{T}}(\mathrm{C_aH_b})^{0.5}(\mathrm{O_2})\,\mathrm{mol/cm^3 s} \tag{4.6}$$

4.7.3.3 Sulfur Dioxide

Sulfur dioxide (SO_2) is a colorless gas that has a characteristic smell in concentrations as low as 1 ppm. SO_2 is formed when sulfur (S) in the fuel combines with oxygen (O_2) in the TEG. If oxygen is present (from excess of combustion) and the temperature is correct, the sulfur will further combine and be converted to sulfur trioxide (SO_3). These oxides of sulfur are collectively known as SOx.

Except for sulfur compounds present in the incoming particulate matter (PM), all of the sulfur contained in the fuel is converted to SO_2 or SO_3. Sulfur dioxide will pass through the boiler system to eventually form the familiar "acid rain" unless a gas-side scrubbing plant is installed. Sulfur trioxide can, in the cooler stages of the gas path, combine with moisture in the exhaust gas to form sulfuric acid (H_2SO_4), which is highly corrosive and will be deposited in ducts and the economizer if the exhaust gas is below condensing temperatures. Natural gas fuels are fortunately very low in sulfur and do not usually cause a problem. However, some oil fuels and plant gases can be troublesome in this respect.

4.7.3.4 PM

Particulate emissions are formed from three main sources: ash contained in liquid fuels, unburned carbon in gas or oil, and SO_3. The total amount of particulate is often called TSP (total suspended particulate). There is concern for the smaller-sized portion of the TSP, as this stays suspended in air for a longer period of time. The PM-10 is the portion of the total PM that is less than 10 μm (10×10^{-6} m) in size. Particles smaller than PM-10 are on the order of smoke. Typical NOx and CO emissions for various fuels are shown in Table 4.1.

For particulate oxidation, an equation can be developed from fundamental principles utilizing a combination of diffusion of oxygen and surface reactivity as follows:

$$\frac{dm}{dt} = \frac{(12C_{\mathrm{og}}A_{\mathrm{p}})}{\left(\dfrac{1}{K_m}+\dfrac{1}{K_r}\right)} \tag{4.7}$$

where
m is the mass of particle
t is time
C is the molar density
A is the surface area
K_m is the diffusion coefficient of oxygen in nitrogen
K_r is the reaction coefficient of the form $Ae^{-E/RT}$
A is the frequency factor
E is the activation energy
R is the universal gas constant
T is the temperature

The equation can be integrated for constant density particles and using particle tracking in time steps with constant or varying oxygen and temperature. An excellent source of char rate data is available by Smith and Smoot[7].

Then, in all cases, one can post-process thermal map data in some discrete volume form and/or insert into a CFD code using the Rayleigh flux theorem as follows:

$$\frac{\partial}{\partial t}\int^{cv} n\rho\, dv = \int^{cs} n\rho(V\cdot da) \tag{4.8}$$

TABLE 4.1

Typical NOx and CO Emissions from Duct Burners

Gas	NOx (lb/10^6 Btu Fired)	CO (lb/10^6 Btu Fired)
Natural gas	0.1	0.08
Hydrogen gas	0.15	0.00
Refinery gas	0.1–0.15	0.03–0.08
Plant gas	0.11	0.04–0.01
Flexicoker gas	0.08	0.01
Blast furnace gas	0.03–0.05	0.12
Producer gas	0.05–0.1	0.08
Syn fuels	0.08–0.12	0.08
Propane	0.14	0.14
Butane	0.14	0.14

Note: NOx emissions from butane and propane can be modified by direct steam injection into a gas or burner flame. CO emissions are highly dependent on TEG approach temperature and HRSG-fired temperature.

where

n is the chemical in mass units
t is the time
ρ is the density
v is the volume
a is the area
V is the velocity vector

where described in words, the formation of (n) through the volume surface is equal to the integrated rate of formation over the control volume.

It is a simple extrapolation to extend this concept for even coarse volumes as follows:

$$\sum \frac{dn}{dt}\rho\Delta v = n\rho(V\cdot a) \tag{4.9}$$

This method can be very useful for fully mixed downstream products even with coarse volumes. But one must be careful with coarse volumes to be sure that the temperature and concentrations are uniform.

4.8 Maintenance

1. *Normal wear and tear*: If nothing has been replaced in the past 5 years and the burner (or turbine/HRSG set) is operated fairly continuously, it is likely that some tips and wings may require replacement.
2. *Damage due to misuse, system upsets, or poor maintenance practices*: Older systems designed without sufficient safety interlocks (TEG trip, high temperature) sometimes expose parts to excessively high temperatures, which results in wing warpage and oxidation failure.
3. *Fuel quality/composition*: Some refinery fuels or waste fuels contain unsaturated components and/or liquid carryover. Eventually, these compounds will form solids in the runner pipes or directly in tips, which results in plugging.

The following are some items to consider when operational problems are encountered:

- *Plugged gas ports*: Which are evidenced by gaps in the flame or high fuel pressure: Gas ports may simply consist of holes drilled into the element manifold pipe, or they may be located in individual removable tips. Designs of the former type may be redrilled or else the entire manifold pipe must be replaced. Discrete tips can be replaced individually as required.
- *Warped flame holders (wings)*: Some warping is normal and will not affect flame quality, but excessive deformation such as "curling" around the gas ports will degrade the combustion and emissions performance. Most grid-type burner designs permit replacement of individual flameholder segments.
- *Oxidation of flame holders (wings) or portions of flame holders*: If more than one-third of the flameholder is missing, it is a good candidate for replacement. Fabricated and cast designs are equally prone to oxidation over time. Most grid-type burner designs permit replacement of individual flameholder segments.
- *Severe sagging of runner pipes (grid design only)*: If the manifold pipe is no longer supported at both ends, it should be replaced. Beyond that relatively extreme condition, sagging at midspan in excess of approximately 2–3 in. (5–7 cm) should be corrected by runner replacement and/or installation of an auxiliary support.

4.8.1 Accessories

4.8.1.1 Burner Management System

All fuel-burning systems should incorporate controls that provide for safe manual light-off and shutdown, as well as automatic emergency shutdown upon detection of critical failures (see Volume 2, Chapter 2). Control logic may reside in a packaged flame safeguard module, a series of electromechanical relays, a programmable logic controller (PLC), or a distributed control system (DCS). At a minimum, the duct burner management system should include the following:

- Flame supervision for each burner element.
- Proof of completed purge and TEG/combustion airflow before ignition can be initiated.
- Proof of pilot flame before main fuel can be activated.
- Automatic fuel cutoff upon detection of flame failure, loss of TEG/combustion air, and high or low fuel pressure.

Other interlocks designed to protect downstream equipment can also be included, such as high boiler tube temperature or loss of feedwater.

4.8.1.2 Fuel Train

Fuel flow to the burners is controlled by a series of valves, safety devices, and interconnecting piping mounted on a structural steel rack or skid. A properly designed fuel train will include, at a minimum, the following:

- At least one manual block valve
- Two automatic block valves in series
- One vent valve between the automatic block valves (gas firing only)
- Flow control valve
- High and low fuel pressure switches
- Two pressure gauges, one each at the fuel inlet and outlet

Depending on the custom and operating requirements at a particular plant, pressure regulation, flow measurement devices, and pressure transmitters can also be incorporated. See Figures 4.20 through 4.27 for typical duct burner fuel system piping arrangements.

4.9 Design Guidelines and Codes

4.9.1 NFPA 8506 (National Fire Protection Association)

First issued in 1995, this standard has become the *de facto* guideline for HRSGs in the United States and many other countries that have not developed their own national standards. Specific requirements for burner safety systems are included, but as stated in the Foreword, NFPA 8506 does not encompass specific hardware applications, nor should it be considered a “cookbook” for the design of a safe system. Prior to the issuance of NFPA 8506, designers often adapted NFPA boiler standards to HRSGs, which resulted in design inconsistencies.

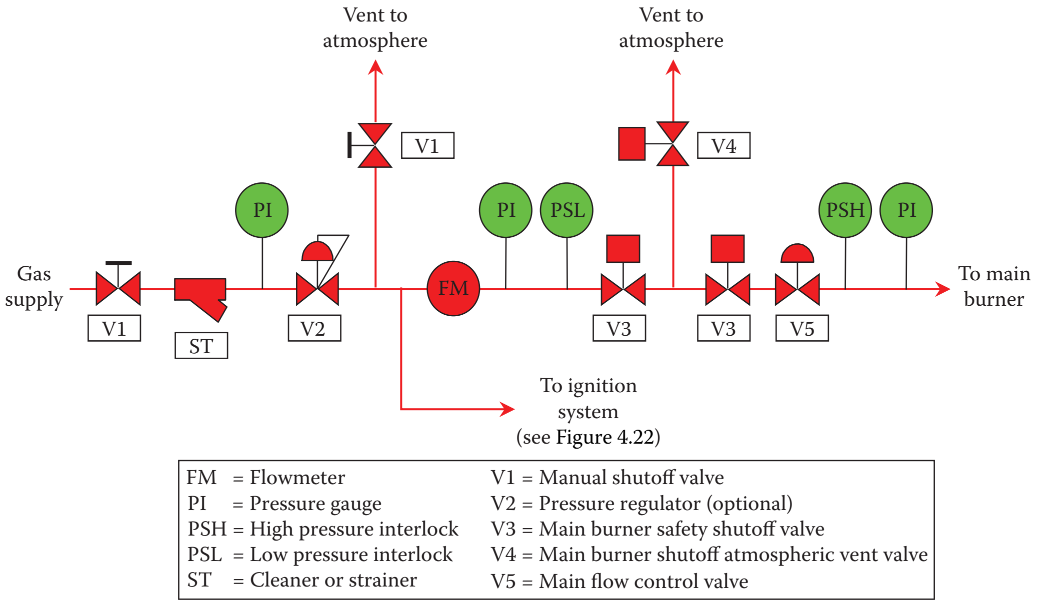
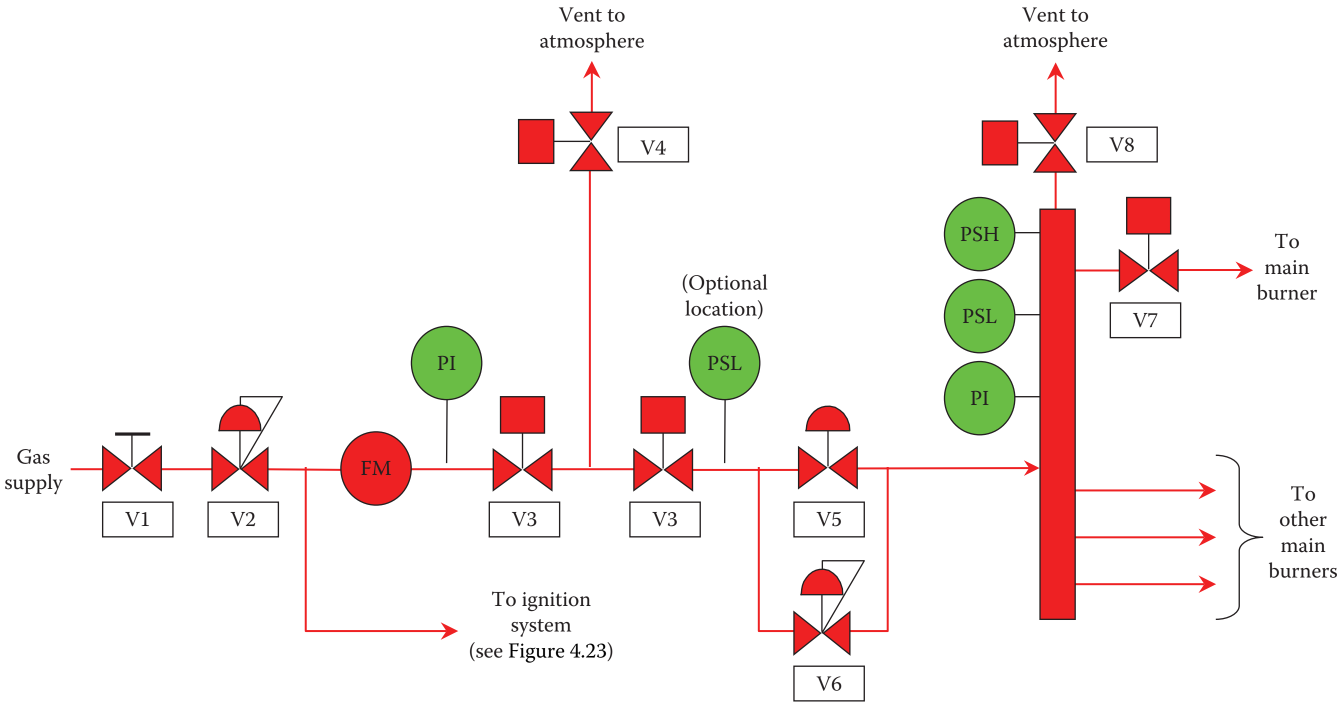
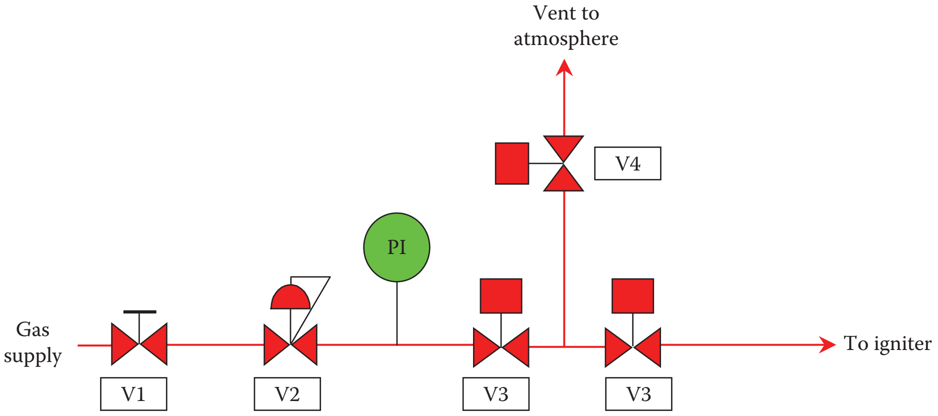


FIGURE 4.20
Typical main gas fuel train: single element or multiple elements firing simultaneously.



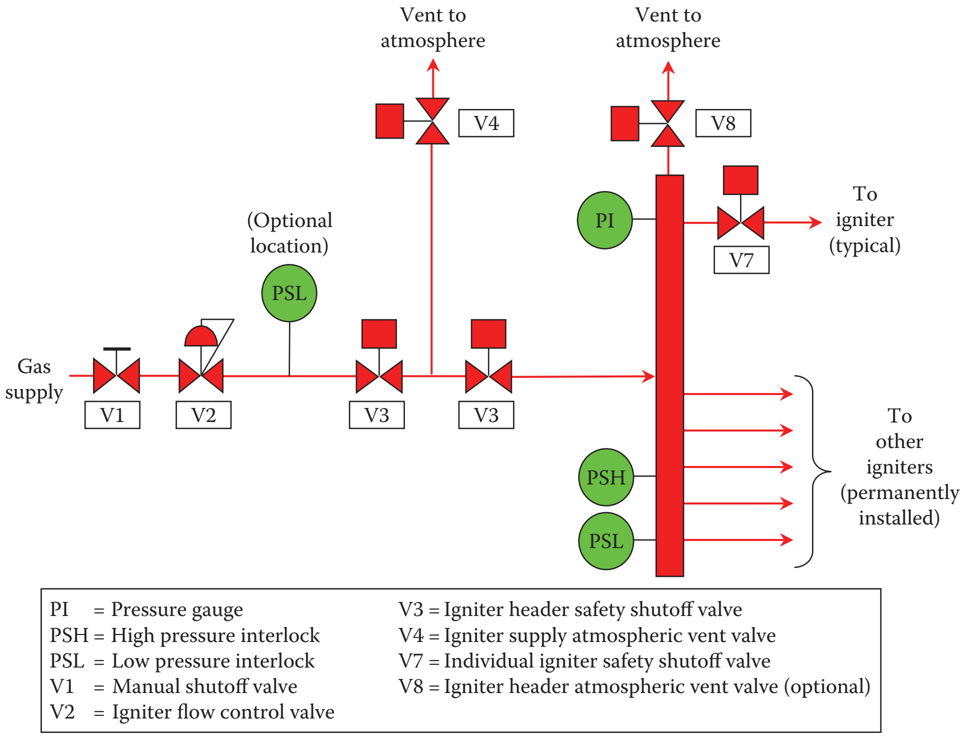
FM = Flowmeter
PI = Pressure gauge
PSH = High pressure interlock
PSL = Low pressure interlock
V1 = Manual shutoff valve
V2 = Pressure regulator (optional)
V3 = Main safety shutoff valve
V4 = Main burner header shutoff atmospheric vent valve
V5 = Main flow control valve
V6 = Main flow bypass control valve (optional)
V7 = Individual burner safety shutoff valve
V8 = Main burner header charging atmospheric vent valve (optional)

FIGURE 4.21
Typical main gas fuel train: multiple elements with individual firing capability.



PI = Pressure gauge
V1 = Manual shutoff valve
V2 = Igniter flow control valve
V3 = Igniter safety shutoff valve
V4 = Igniter shutoff atmospheric vent valve

FIGURE 4.22
Typical pilot gas train: single element or multiple elements firing simultaneously.



PI = Pressure gauge
PSH = High pressure interlock
PSL = Low pressure interlock
V1 = Manual shutoff valve
V2 = Igniter flow control valve
V3 = Igniter header safety shutoff valve
V4 = Igniter supply atmospheric vent valve
V7 = Individual igniter safety shutoff valve
V8 = Igniter header atmospheric vent valve (optional)

FIGURE 4.23
Typical pilot gas train: multiple elements with individual firing capability.

4.9.2 Factory Mutual (FM)

An insurance underwriter that publishes guidelines on combustion system design, Factory Mutual (FM), also "approves" specific components such as valves, pressure switches, and flame safeguard equipment that meet specific design and performance standards. Manufacturers are given permission to display the FM symbol on approved devices. Although FM approval may be required for an entire combustion control system, it is more common for designers to simply specify the use of FM-approved components.

4.9.3 Underwriters Laboratories (UL)

Well known in the United States for its certification of a broad range of consumer and industrial electrical devices, Underwriters Laboratories (UL) authorizes manufacturers to display their label on specific items that have demonstrated compliance with UL standards. Combustion system designers will frequently require the use of UL-approved components in burner management systems and fuel trains. Approval can also be obtained for custom-designed control systems, although this requirement generally applies only to a few large cities and a few regions in the United States.

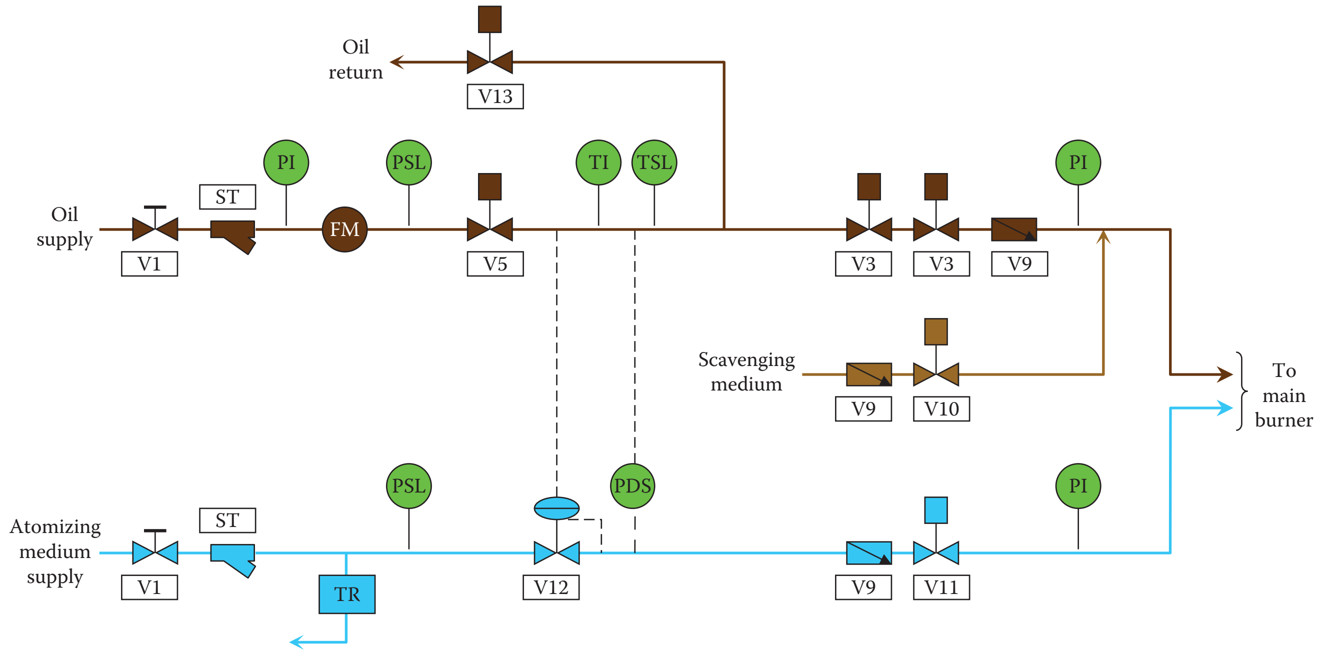
4.9.4 American National Standards Institute (ANSI) B31.1 and B31.3

These codes address piping design and construction. B31.1 is incorporated in the NFPA 8506 guideline, while B31.3 is generally used only for refining/petrochemical applications.

4.9.5 Others

The following may also apply to duct burner system designs, depending on the country where equipment will be operated:

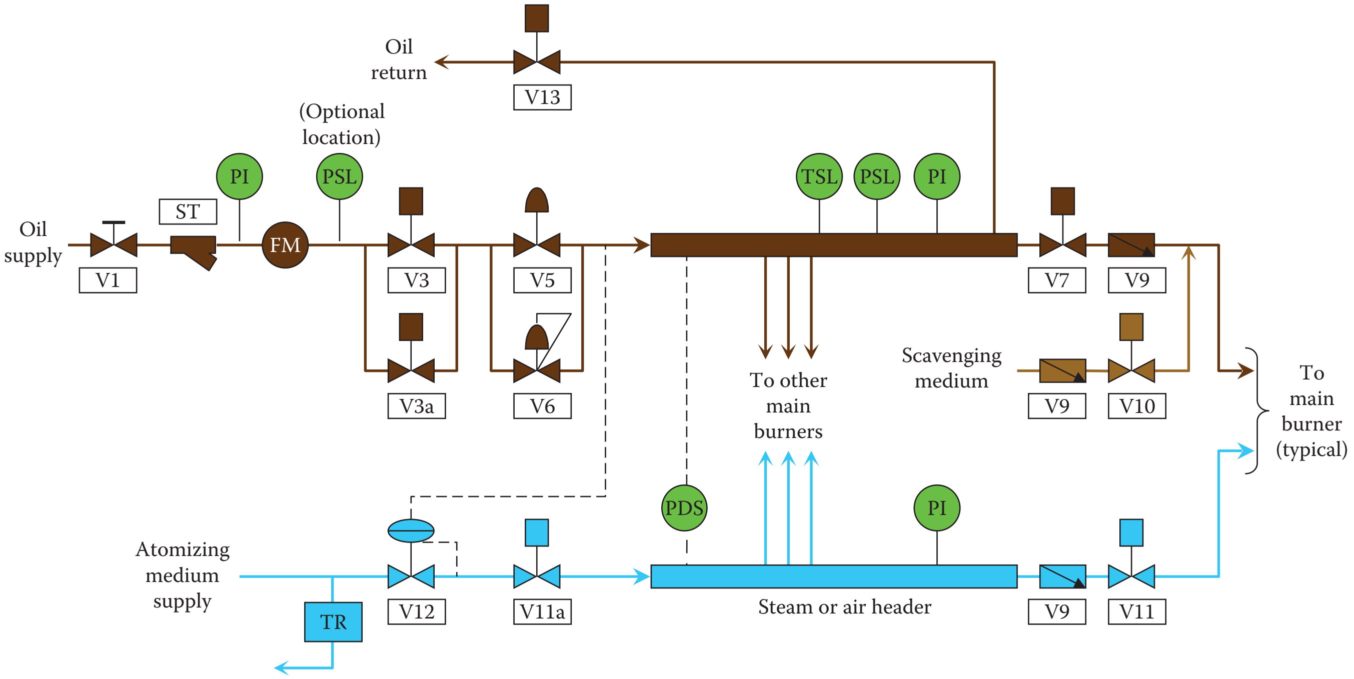
- National Electrical Code (NEC)
- Canadian Standards Association (CSA)
- International Electrotechnical Commission (IEC)
- European Committee for Electrotechnical Standardization (CENELEC)



FM = Flowmeter
PI = Pressure gauge
PDS = Differential pressure alarm and trip interlock
PSH = High pressure interlock
PSL = Low pressure interlock
TI = Temperature gauge
(optional for unheated oil)
TSL = Low temperature or high viscosity alarm
(optional for unheated oil)
ST = Cleaner or strainer
TR = Trap

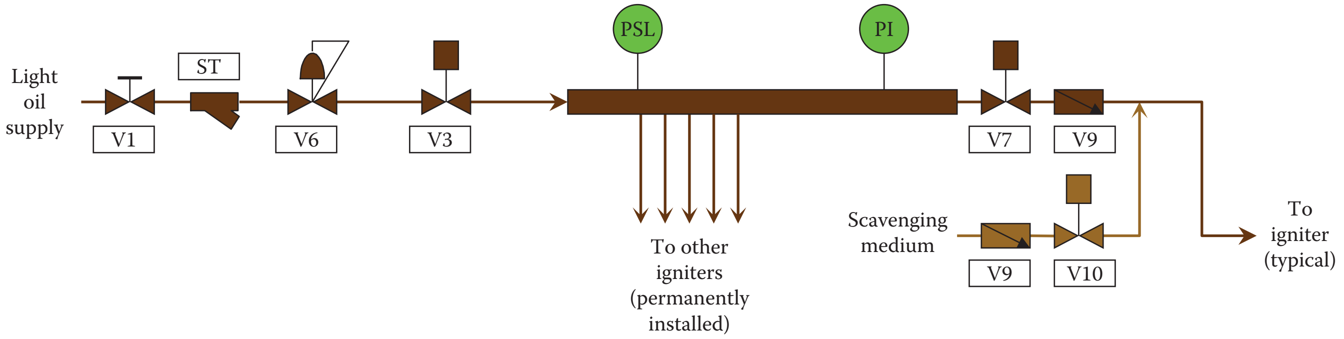
V1 = Manual shutoff valve
V3 = Main burner safety shutoff valve
V5 = Main flow control valve
V9 = Check valve
V10 = Scavenging valve
V11 = Atomizing medium individual burner
shutoff valve, automatic
V12 = Differential pressure control valve
V13 = Re-circulating valve
(optional for unheated oil)

FIGURE 4.24
Typical main oil fuel train: single element.



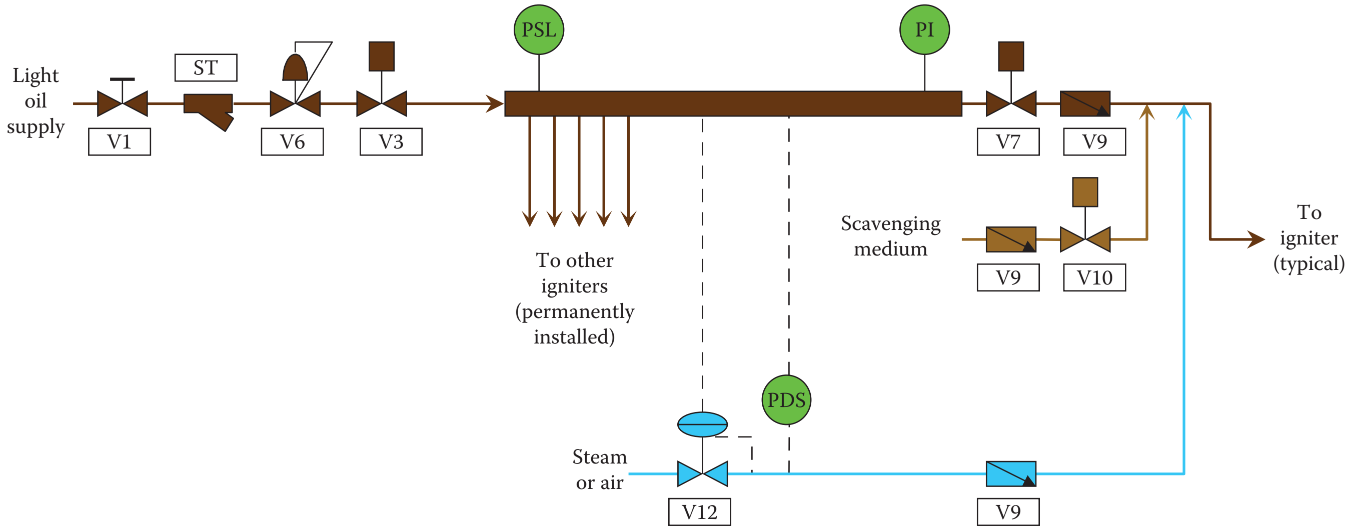
FM = Flowmeter
PI = Pressure gauge
PDS = Differential pressure alarm and trip interlock
PSL = Low pressure interlock
TSL = Low temperature or high viscosity alarm (optional for unheated oil)
ST = Cleaner or strainer
TR = Trap
V1 = Manual shutoff valve
V3 = Main safety shutoff valve
V3a = Circulating valve (optional for unheated oil)
V5 = Main flow control valve
V6 = Main flow by-pass control valve (optional)
V7 = Individual burner safety shutoff valve
V9 = Check valve
V10 = Scavenging valve
V11 = Atomizing medium individual burner shutoff valve, automatic
V11a = Atomizing medium header shutoff valve, automatic (alternate to V11)
V12 = Differential pressure control valve
V13 = Re-circulating valve (optional for unheated oil)

FIGURE 4.25
Typical main oil fuel train: multiple elements.



PI = Pressure gauge
PSL = Low pressure interlock
ST = Cleaner or strainer
V1 = Manual shutoff valve
V3 = Igniter safety shutoff valve
V6 = Igniter flow control valve
V7 = Individual igniter safety shutoff valve
V9 = Check valve
V10 = Scavenging valve

FIGURE 4.26
Typical pilot oil train: single element.



PI = Pressure gauge
PDS = Differential pressure alarm and trip interlock
PSL = Low pressure interlock
ST = Cleaner or strainer
V1 = Manual shutoff valve
V3 = Igniter safety shutoff valve
V6 = Igniter flow control valve
V7 = Individual igniter safety shutoff valve
V9 = Check valve
V10 = Scavenging valve
V12 = Differential pressure control valve

FIGURE 4.27
Typical pilot oil train: multiple elements.

References

1. S. Londerville, Performance prediction of duct burner systems via modeling and testing, in *Industrial Combustion Testing*, C.E. Baukal (ed.), CRC Press, Boca Raton, FL, 2011, Chapter 26.
2. Department of Energy, *Improving Industrial Burner Designs with Computational Fluid Dynamic Tools: Progress, Needs and R&D Priorities*, Workshop Report, September, 2002.
3. A.E. Westenberg, Kinetics of NO and CO in lean, premixed hydrocarbon-air flames, *Combustion Science and Technology*, 4, 59–64, 1971.
4. C.T. Bowman, Kinetics of pollution formation and destruction in combustion, *Progressing Energy and Combustion Science*, 1, 33–45, 1975.
5. G.C. Williams, H.C. Hottel, and A.C. Morgan, The combustion of methane in a jet-mixed reactor, *12th Symposium (International) on Combustion*, p. 913, The Combustion Institute, Pittsburgh, PA, 1969.
6. R.H. Barnes, M.H. Saxton, R.E. Barrett, and A. Levy, *Chemical Aspects of Afterburner Systems*, EPA report EPA-600/7-79-096, NTIS PB298465, p. 21, April 1979.
7. D.L. Smoot and P. Smith, *Coal Combustion and Gasification*, Plenum Press, New York, pp. 81–88, 1985.

5

Marine and Offshore Applications

Richard Price

CONTENTS

5.1 Introduction

Since the first steam plant was installed on a boat circa 1803, all, but a few ships, have had a fired boiler of one type or another. The earliest steam-propelled ships used coal as their fuel, and it was not until the early 1900s that the steam boilers used fuel oil, and the importance of the register burner established. Oil-fired, superheated, main propulsion boilers rated at 60–63 barg (880–930 psig) were applied on ships until the mid-1970s, when the diesel engine took over the main propulsion role, primarily due to higher fuel efficiency. These early vessels raised steam in boilers with the propulsion by screw (propeller) or paddle, driven by reciprocating steam engines and later by steam turbines.

Diesel engine ships (motor ships) still, however, have a steam requirement, albeit on a much smaller duty. The steam is needed for heating duties, so auxiliary boilers of duty 1–5 metric-ton/h (1.1–5.5 U.S. ton/h) are common on container ships, bulk carriers, general cargo ships, and ferries. The larger auxiliary boilers of the range 10–15 metric-ton/h (11–16.5 U.S. ton/h) are installed on large cruise ships and on tankers of the range 20–50 metric-ton/h (22–55 U.S. ton/h) as the cargo (crude oil) pumps are steam turbine driven, using saturated steam at 16–20 barg (240–290 psig). Marine and offshore applications are wide ranging in application depending on the total steam demand that is composed of turbine drives (pumps, gas compressors, and turbogenerators) and heating requirements.

All ships are governed and regulated by the International Maritime Organization (IMO) that is responsible for the safety and security of shipping and the prevention of marine pollution, and classification societies (such as the American Bureau of Shipping [ABS], Det Norske Veritas [DNV], Lloyds Register of Shipping [LRS]) which are responsible for verifying the ship's design, construction, and maintenance.

5.2 Fuels

5.2.1 Fuel Oils

The stock marine fuel for boilers (and ships' main engines) has for many years been residual heavy no. 6 fuel oil, often referred to as "bunker C," with viscosity 180–600 cSt at 50°C (122°F). Marine diesel oil is, however, always available onboard as the cold-start or back-up fuel, with viscosity of 2–11 cSt at 40°C (100°F). The grade and quality of marine fuel oils are governed globally by the ISO standard ISO 8217, which defines limits for viscosities, densities, sulfur, water, ash and flash point, and other properties. The heavy fuel oil is maintained at a temperature of 50°C (120°F) in the ship's fuel tanks, from where it is pumped and preheated to the appropriate temperature to achieve the required viscosity of the main engines or burner fitted to the steam boilers.

5.2.2 Low-Sulfur Marine Gas Oil

New marine legislation from 2007 relating to sulfur dioxide (SO_2) emissions from ships has had a significant effect on marine fuel oils and ships' engine room systems. This new legislation started in the European Union (EU) to reduce SO_2 emissions from ships that are a prime contributor to premature deaths and hospital admissions. This legislation has resulted in parts of the seas and main trade routes becoming emission control areas (ECAs), where the ships must use low-sulfur fuel (or have SO_2 emissions equivalent to using low-sulfur fuels). At the time of writing, the main ECAs are the English Channel (the busiest waterway in the world), the North Sea, the Baltic Sea, and 200 nautical miles around the coasts of the United States. The ship owner now has to use low-sulfur marine gas oil (LSMGO) that is ~50% more expensive than heavy fuel oil and convert the engine room plant and fuel tanks to suit.

5.2.3 LNG Boil-Off Gas

LNG, or liquefied natural gas, first began being transported as a liquid cargo at –163°C (–261°F) by ships in the 1960s. Since the cargo tank insulation is not 100% perfect and the tank pressure being near ambient, a small amount of the LNG evaporates, which has resulted in LNG boil-off gas (BOG) as a fuel. This fuel stream, being almost 100% methane, was first used on the main propulsion boilers of LNG carriers in the early 1970s and gave rise to the first dual-fuel (DF) register burners on ships.

5.2.4 Produced Fuel Gas and Crude Oil

Remote offshore installations, particularly floating production, storage, and off-loading (FPSO) and floating storage and off-loading (FSO) vessels, where regular fuel-oil supplies are not readily available, use or produce their own fuel from their main process or cargo they are handling.

On FPSOs, the crude oil taken from the oil field beneath the sea floor is stabilized and reduced in pressure to a condition where it can be stored in the cargo tanks. This process results in the light-end gases (methane, ethane, etc.) separating from the crude oil and, thus, providing a suitable fuel for the vessel's power and heating requirements. The produced gas is typically heavier than natural gas having ethane, propane, and butane of 10%–30% by volume and on certain installations carbon dioxide up to 50%.

Where produced gas quantities are low, normally on heavy crude-oil fields, it is common for the produced crude oil itself to be fired in steam boilers, as this represents the lowest cost fuel option.

Fuel gases and fuel oils with flash points lower than 60°C (140°F) (most crude oils) are defined as hazardous fuels by marine classification societies. When these fuels are taken into safe areas like the engine room, they have to be contained within double gas pipes and ventilated enclosures and use certified hazardous area equipment and instrumentation.

5.3 Auxiliary Boiler Applications

5.3.1 Small "Donkey" Auxiliary Boilers

Modern small marine auxiliary boilers for duties of range 1–5 metric-ton/h (1.1–5.5 U.S. ton/h) are typically cylindrical vertical, operating at 7 barg (100 psig), and side fired with the burner firing across the diameter of the circular furnace. Thus, there is a need for the flame shape of the heavy oil firing burner to be short and wide. The furnace construction is a water-tube wall with a refractory floor.

Other marine auxiliary boiler types are conventional shell/fire-tube boilers, two or three passes, in either horizontal configuration or vertical configuration, thus minimizing the boiler footprint.

The majority of burners on small auxiliary boilers utilize simple pressure-jet atomizers in the range of 1–4 MW (3.3–14 × 10^6 Btu/h), with the HFO preheated to ~120°C (250°F) to achieve an oil viscosity of 15–20 cSt and pumped to a pressure of 20–30 barg (290–440 psig). The burner construction and operation is simple, with often just on–off or high–low operation. Larger pressure-jet burners may be fully modulating with a turndown of just 3 or 4 to 1. The ignition of the heavy fuel oil is by direct spark of the main heavy fuel-oil spray from the pressure-jet nozzle, with a simple high-tension (HT) spark.

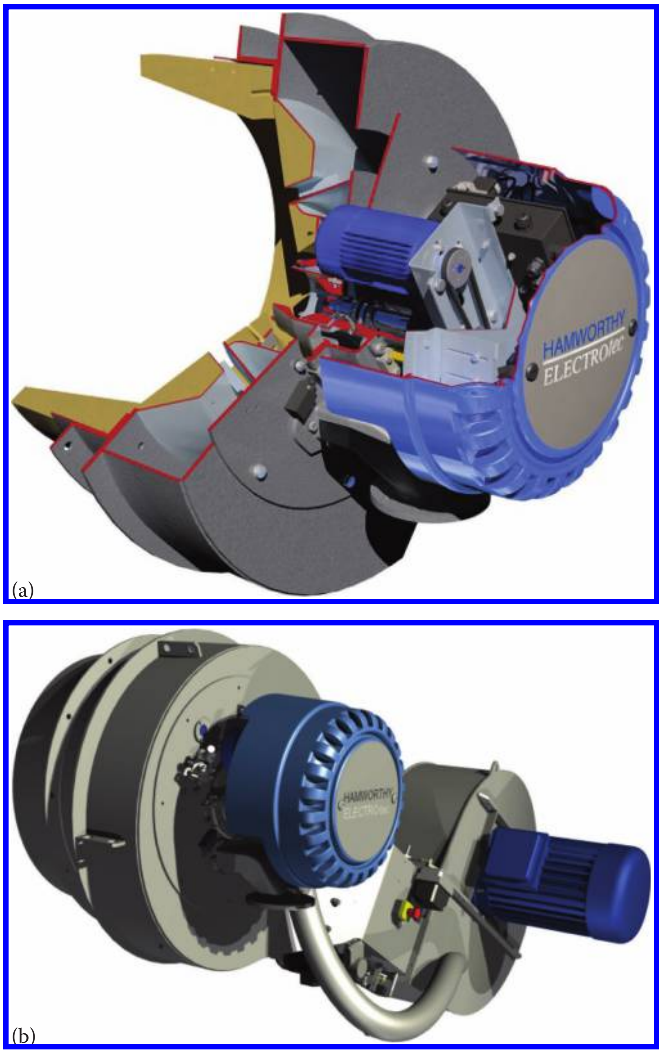


FIGURE 5.1
Hamworthy Combustion Electro*Tec*® rotary-cup burner. (a) Burner machine sectional view and (b) complete assembly with FD fan.

Some small auxiliary marine boilers and larger boilers up to 10 metric-ton/h (11 U.S. ton/h), which is the equivalent of approximately 8 MW (27 × 10^6 Btu/h) heat input, have applied rotary-cup burners, such as the Hamworthy Electro*Tec*® burner (see Figure 5.1).

Rotary-cup burners are capable of burning heavy and dirty oils without the use of an atomizing fluid or high-pressure fuel supply. Also the required heavy fuel-oil temperature is lower at ~90°C (190°F) to achieve 55–70 cSt.

The rotary cup is driven by a constant speed motor at typically 5500 rpm. Oil is delivered to the rear of the cup at low modulated pressure (from a separate pump) and runs along the inside of the cup as a thin film, picking up speed. When the oil spills off the lip of the cup, it is in droplet form with a high radial velocity and so is mechanically atomized. The combustion air fan is directly driven by another motor to blow air through dampers to supply primary, secondary, and tertiary air to the combustion zone immediately in front of the rotary cup. This gives a well-controlled flame shape and turndowns of up to 10:1 with capabilities of adjusting the air/fuel ratio throughout the range of the burner.

The rotary-cup burner has a separate diesel-oil igniter that is used to ignite the main heavy fuel-oil flame.

5.3.2 Water-Tube Auxiliary Boilers

Crude oil and product tankers require larger boilers with a typical range of 20–50 metric-ton/h (22–55 U.S. ton/h) operating at 16–20 barg (240–290 psig), for the main purpose of providing steam for the turbine drives of cargo pumps, used to off-load the cargo. It is common for two boilers to be fitted to each tanker to provide redundancy of steam supply, although some tankers are fitted with just one boiler, with capacities up to 100 metric-ton/h (110 U.S. ton/h) on the largest of very large crude carriers (VLCCs).

Modern marine boilers of this type are either vertical cylindrical water tube, with the burner firing downward along the axis of a cylindrical furnace, or D-type water tube, with the burner firing downward into a rectangular box–shaped furnace. A single steam-atomized register burner, such as the Hamworthy Combustion's type DF burner (see Figure 5.2), is applied for boilers up to 50 metric-ton/h (55 U.S. ton/h), and two burners with isolation air-door registers are applied for larger boilers 50–100 metric-ton/h (55–110 U.S. ton/h).

Marine boilers are very compact and highly rated compared with equivalent land boilers of the same duty. Furnace heat release rates (defined as the thermal heat input divided by the furnace volume) can be as high as 1.4 MW/m^3 (135 K Btu/h ft^3), which is about two times that of land boilers. The reasons for this are

- Space in the engine room is limited, so the smallest physical size is preferred.
- There is no NOx limitation or legislation for marine boilers on ships (unlike the main propulsion engines that have strict NOx limits).

Hence, the burner is required to produce a relatively small and compact flame, and this is achieved by good fuel-oil atomization, effective air mixing, and high mixing energy provided by burner draft losses as high as 30–35 mbar (12–15 in. W.G.).

Heavy fuel-oil and atomizing steam are delivered to a fuel-oil sprayer, which is usually of concentric tube construction, with the steam helping to keep the fuel oil up to temperature all the way to the Y-type atomizer tip (see Figure 5.3). The fuel-oil pressure is typically 16–18 barg (240–260 psig) at the burner maximum

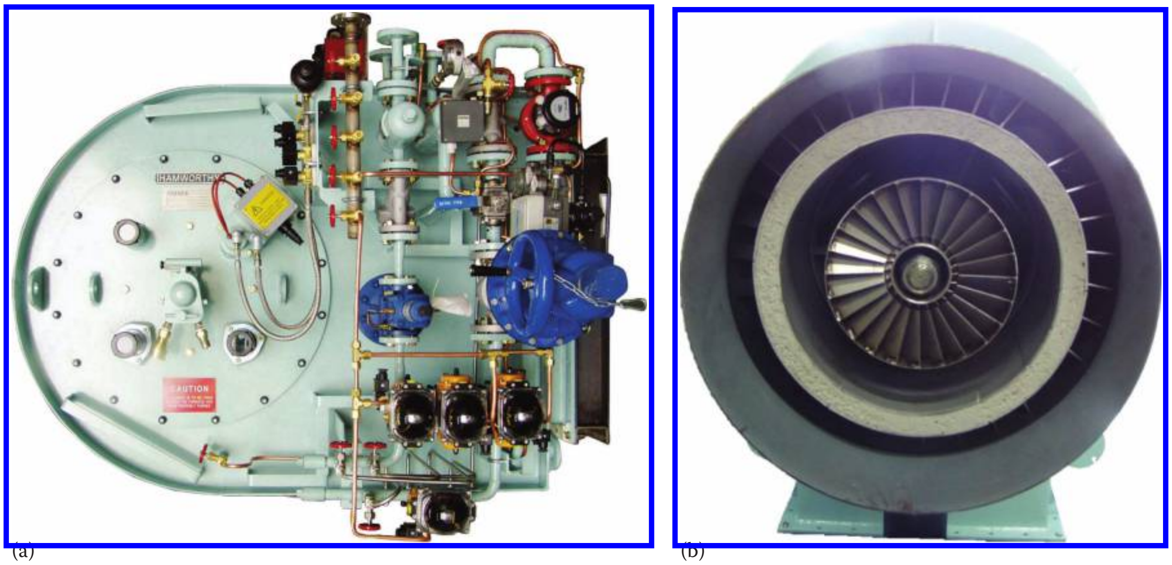


FIGURE 5.2
Hamworthy Combustion DF register burner. (a) Windbox view of fuel valve train and (b) burner throat view.

FIGURE 5.3
Heavy fuel-oil sprayer and twin-fluid Y-jet atomizer.

combustion rating (MCR), reducing to ~2 barg (30 psig) at the burner turndown that is of the order of 8–10 to 1.

The atomizing steam is saturated, regulated to a constant pressure of between 5 and 10 barg (70 and 150 psig). Cold start is by firing marine diesel oil, with compressed air as the atomizing medium at ~5 barg (70 psig). The burner includes a pressure-jet diesel-oil igniter, which uses a simple HT spark.

Combustion air is provided by an electric motor–driven forced-draft fan, with air-flow regulation by either a twin-bladed damper on the burner wind box inlet or a multi-vane inlet damper on the fan inlet. The burner is fired with an excess air at MCR of typically 15% (providing ~3% oxygen in the boiler exhaust) so as to ensure complete and clean combustion of the residual heavy fuel oil.

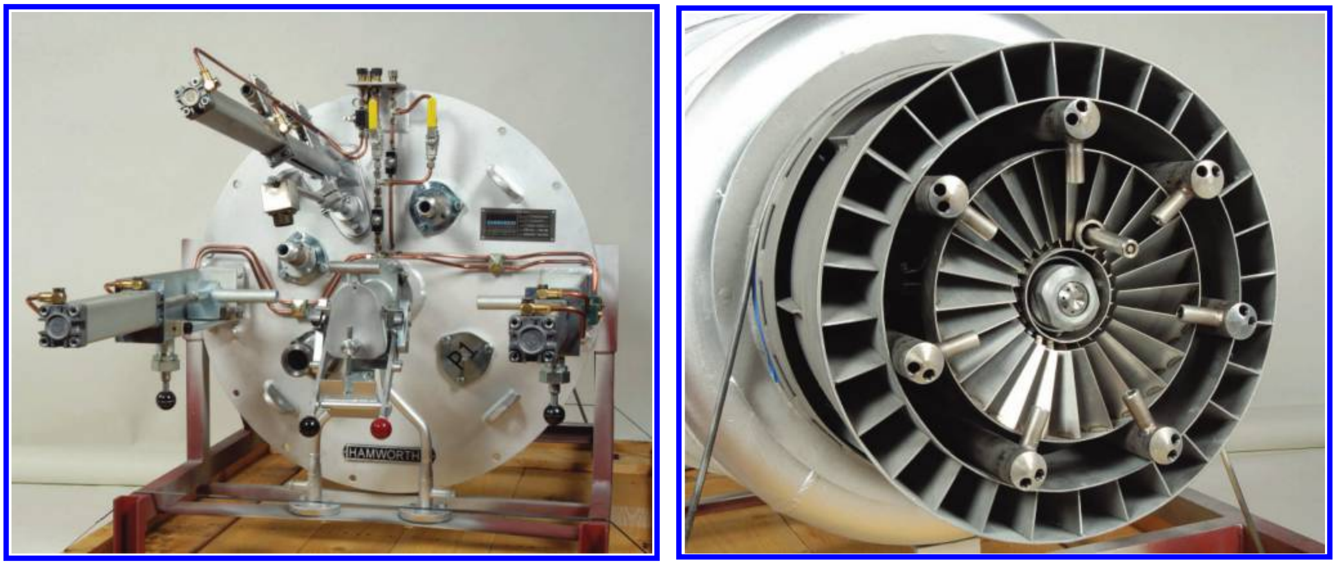
On tankers, the boiler exhaust gases are commonly used as an inert gas for the cargo tanks. The exhaust gases are extracted from the boiler uptakes and cooled and scrubbed by an inert gas system, which blows the gases into the cargo tanks, as the crude oil is unloaded. The oxygen content in the inert gas (and hence, boiler exhaust gas) must be below 5%, and so the excess air level of the burner must be accurately controlled through the burner load range.

5.4 LNG Carriers

As described in Section 5.2.3 earlier, LNG carriers produce BOG from the LNG cargo which can be used as an alternative to heavy fuel oil, thus providing a high fuel cost saving for the ship's operation. The first DF LNG carrier was built in the early 1970s with the main propulsion boilers providing high-pressure superheated steam to a steam turbine directly coupled to the ship's propeller via a gearbox. Over 200 steam LNG carriers with DF boilers were built up to circa 2005, when a fundamental change to DF medium speed and standard heavy fuel-oil-fired engines, which provided a significant efficiency improvement, was realized. To date, over 120 LNG carriers have since been ordered with engine propulsion.

5.4.1 LNG-Carrier Main Propulsion Boilers

Two such boilers are fitted for propulsion purposes on LNG carriers, with each having 2 or 3 DF register burners in downward firing orientation, such as the Hamworthy Combustion's type HXG burner (Figure 5.4).



(a)

(b)

FIGURE 5.4
Hamworthy Combustion HXG dual fuel register burner. (a) Burner frontplate view and (b) burner throat view.

Each burner incorporates an air door for open/closed operation for when the burner is in/out of service, actuated by twin air cylinders. The heavy fuel-oil sprayer and atomizer use the same design and principles as that described for the burners for auxiliary water-tube boilers in Section 5.3.2, although the atomizing steam is often superheated to ~300°C (570°F).

The needs of the LNG carrier to maneuver at very slow speeds demand a high turndown of the propulsion boilers. Thus, individual burner turndowns when fuel-oil firing of 15 to 1 are common.

The LNG BOG is compressed to a nominal 1 barg (15 psig) and heated to an ambient temperature. Thus, the fuel-gas pressure at the burners at MCR is ~0.6 barg (9 psig), with a maximum burner turndown of 7 to 1. The fuel gas is injected into the air stream via multiple gas nozzles, which include staged injection so as to provide stable combustion across all operating conditions and boiler loads.

The burners can also fire heavy fuel oil and BOG in combination, should the BOG availability not be sufficient to meet the boiler requirements.

As with auxiliary boilers, the boilers are very compact compared to equivalent land boilers, with furnace heat release rates of 1–1.2 MW/m^3 (120–140 K Btu/(h ft^3)). Flames are therefore very compact and vigorous when viewed, with burner draft losses between 25 and 30 mbar (10 and 12 in. W.G.) at MCR.

Ignition is usually by direct sparking of the main heavy fuel-oil atomized spray by a high-energy spark rod, typically of type Chentronics® (see Figure 5.5) that delivers 20 sparks per second. The igniter rod is advanced and retracted in and out of the fuel-oil spray zone by a pneumatic air cylinder.

FIGURE 5.5
Chentronics® high-energy igniter.

5.4.2 LNG-Carrier Gas Combustion Units

With the change in LNG-carrier propulsion type from steam turbine to engines, a new type of combustion application resulted. As the BOG from the cargo tanks is continuous, there always needs to be a method of safely disposing of the gas, since simply venting the


(a) (b)

FIGURE 5.6
Hamworthy Combustion AMOxsafe® GCU. (a) GCU system arrangement of aft deck and (b) GCU body.

methane-rich stream is not allowed as part of normal operations by the International Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk (the IGC Code) that governs the design, construction, and operation of LNG carriers.

When there is more BOG than required by the propulsion need of a steam LNG carrier, the total BOG is fired in the boilers and the excess steam produced "dumped" to a seawater-cooled condenser. LNG carriers with engines however, cannot burn this excess BOG without engine load, so a requirement for an alternative safe method of safely burning the gas is needed. So, the need for a gas combustion unit, or GCU, evolved, such as the Hamworthy Combustion's AMOxsafe® GCU (see Figure 5.6).

The prime function of the GCU is to safely incinerate the BOG while maintaining an exhaust temperature below 535°C (995°F), a figure selected taking into account the autoignition temperature of methane with a 50°C (90°F) margin. The GCU is sized for the maximum BOG rate of the cargo tanks, which is based on the guarantee value of the tank design. This is typically 0.15% of the tank volume per day, and an LNG carrier of capacity 175,000 m^3 (5.8×10^6 ft^3) would have a maximum BOG rate of ~4.5 metric-ton (5 U.S. ton/h). This BOG stream is near 100% methane, although just after ship's loading, it will contain a high level of nitrogen that quickly reduces. Thus, 4.5 metric-ton (5 U.S. ton/h) equates to a thermal rating of the GCU of 70 MW (240×10^6 Btu/h).

To achieve an exhaust temperature of less than 535°C (995°F), and an industry standard bulk average of 450°C (840°F), at MCR, the products of combustion are mixed and cooled by large volumes of dilution air.

In the case of the AMOxsafe GCU, the Hamworthy Combustion DF burner is used in an up-fired configuration, with combustion air supplied by centrifugal fans. The products of combustion are then mixed with dilution air from axial fans, which is also used to cool the double lining of the GCU body, in the same way as a twin-shell air heater. The flow schematic of the AMOxsafe GCU is shown in Figure 5.7, and the temperature profile is taken from computational fluid dynamic (CFD) simulations in Figure 5.8. The BOG supply to the GCU uses the same compressors that supply the LNG carrier's main engines, and hence, the pressure at the burner is ~2 barg (29 psig). The ignition of the AMOxsafe GCU is by a small diesel-oil igniter using a pressure jet, which can also be used as a permanent pilot when the nitrogen content of the BOG stream is high after cargo tank loading, so as to ensure complete and safe combustion of the methane.

5.5 Offshore Applications

5.5.1 FPSOs

Floating production, storage, and off-loading (FPSO) vessels can be either the conversions of existing crude-oil tankers or purpose built as new. Most have the benefit of produced fuel gas being available for heating and power requirements, as described in Section 5.2.4.

5.5.1.1 Tanker Converted FPSOs

The majority of FPSOs have been tanker conversions, which have existing steam boilers in the engine room, with these being either main high-pressure propulsion boilers (if the tanker was propelled by a steam turbine) or lower-pressure auxiliary boilers.

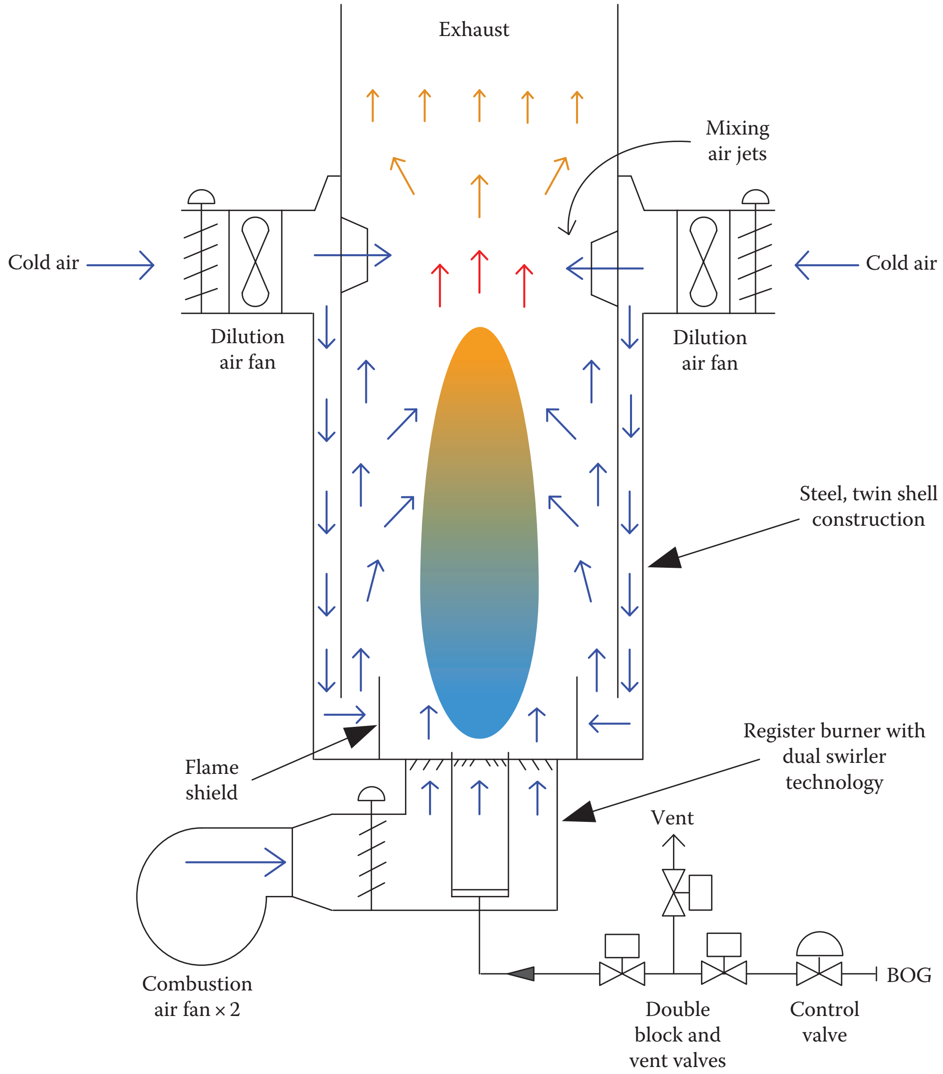


FIGURE 5.7
AMOxsafe® GCU flow schematic.

Since the original main boilers were only designed to fire fuel oil, their performance on fuel gas must be calculated to determine the effect on the original boiler design and materials used. As the fuel-gas flames provide less radiant heat transfer to the furnace water-tube walls compared to fuel-oil firing, the temperature at the furnace exit, and in particular at the superheater, is increased. Thus, the temperature of the superheater material must be considered, and to keep the original boiler rating, it is common for the superheater to be replaced with material of a high-temperature rating.

Since the exhaust gases at the superheater are hotter when gas fires, the attemperator (which controls the steam delivery temperature usually by a tube coil in the boiler's lower water drum) and the attemperator control valve must also be verified as their operating duty will have changed. The reduction in radiant heat transfer in the boiler furnace results in an increase in the boiler exhaust temperature and an overall reduction in boiler efficiency of typically 3%–4%.

For main boiler conversions, the original register burners that fired fuel oil are replaced with register burners with DF capability, being very similar to those for LNG-carrier main boiler (described in Section 5.4.1), such as Hamworthy Combustion's type DF burner, as shown in Figure 5.9.

Since the fuel gas is considered to be a hazardous gas, when it is taken into the engine room, pipelines must be double skinned with an outer pipe either pressurized with an inert gas (nitrogen) or ventilated under negative pressure by extraction air fans at a minimum rate of normally 30 air changes per hour. All potential leak sources must be fully enclosed in the ventilated valve

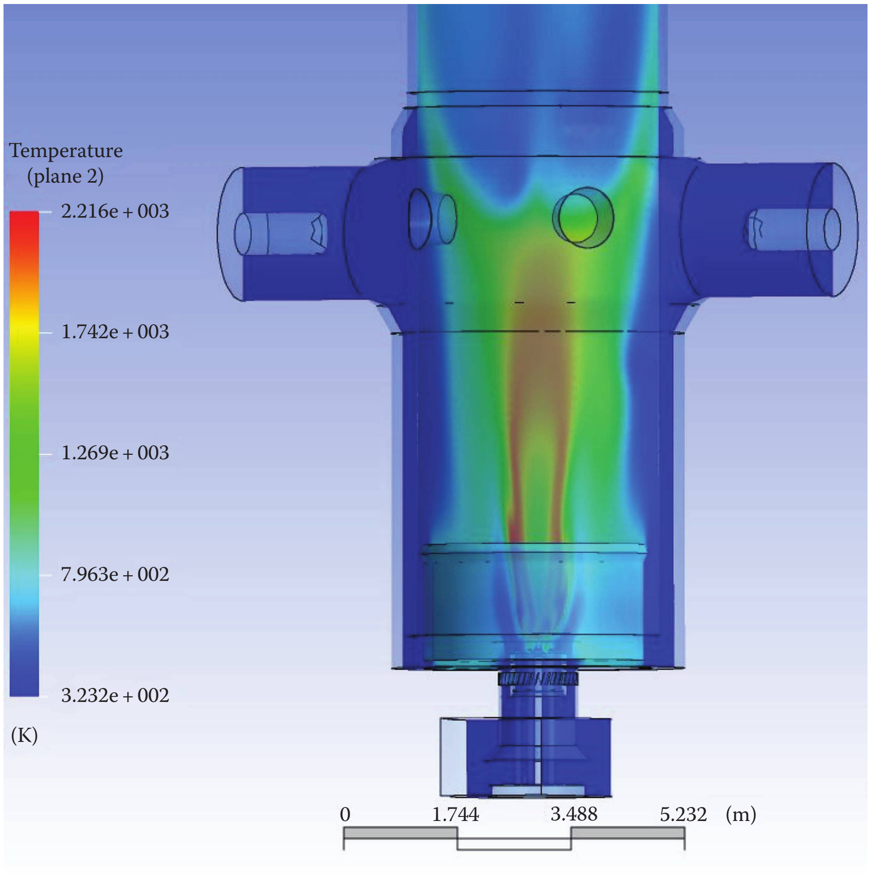


FIGURE 5.8
AMOxsafe® GCU temperature profile as predicted by CFD modeling.

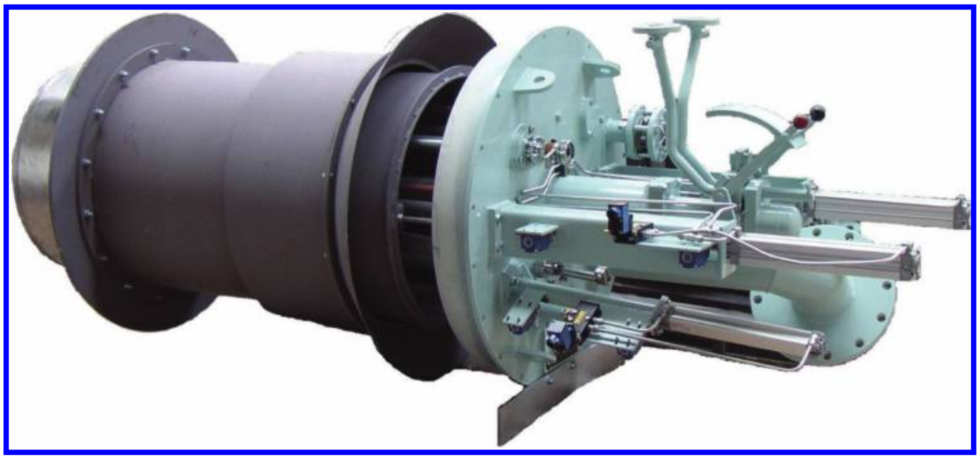


FIGURE 5.9
Hamworthy Combustion DF register burner.

boxes (like the one shown in Figure 5.10) or "boiler hood rooms," with all electrical items rated for hazardous area applications.

Typical fuel-gas supply pressures from the crude-oil process are 6–8 barg (90–120 psig), and this pressure is regulated and controlled to give about 2 barg (30 psig) at the burner gas nozzles at the boiler maximum rating.

Fuel gas is the main fuel consumed as this is the lowest cost and cleanest available. There is usually a "backup" second fuel, primarily being marine diesel oil, which must be automatically switched to in the event of interruption of the fuel-gas supply, so as to keep the boilers online at all times. Burner ignition is commonly by direct high-energy spark of the main fuel, in the

FIGURE 5.10
Fuel-gas valve enclosure for FPSO engine room boiler.

same way as the register burners for the main boilers of LNG carriers, described in Section 5.4.1.

The conversion to gas firing of the auxiliary boilers on motor tankers as part of their conversion to FPSOs is simpler than that of main boilers in that there is no steam superheater or feedwater economizer and there is commonly just one burner on each boiler. A boiler performance study to assess the suitability of the boiler to fire fuel gas and calculate the drop in efficiency is, however, still necessary.

5.5.1.2 *FPSO New Boilers*

Often the existing boilers of tankers are not sufficient in capacity for the new duty of that of an FPSO, and so new additional boilers are required. There are several possible reasons: the power generation and/or machinery drives that are to be used by new steam turbines, the high crude-oil process heating requirements, or the original tanker only had a single auxiliary boiler and the redundancy of steam supply is required. As the engine rooms of tankers have no spare space for such additional boilers, the new boilers are placed on the aft main deck or on the cargo (process) main deck. In order to reduce the time in the tanker conversion shipyard, it is a common practice for the new boilers to be supplied as complete modules, as shown in Figure 5.11.

Superheated, high-pressure steam boilers and associated turbines provide higher efficiency (reduced fuel consumption) and hence reduced emission solutions for power generation than lower-pressure, saturated steam boilers. Where there is insufficient produced fuel gas available, as is often the case later in the FPSO operation term as the oil-field reserves deplete, then crude oil is fired as the prime fuel. Hence, plant efficiency is an important economic consideration. A main deck boiler module for 2 × 71 metric-ton/h (2 × 78 U.S. ton/h) steam boilers, providing steam at 63 barg (930 psig) and 515°C (959°F), to fire fuel gas, crude oil, and/or diesel oil, is shown in 3D CAD format in Figure 5.12.

5.5.2 FSOs

FSOs are tankers that are permanently moored to receive crude oil as it is processed from a neighboring production platform, with the crude oil arriving on the FSO via a subsea pipe. Steam is required from auxiliary boilers on the FSO for crude-oil heating and for the cargo pumps that off-load the crude oil periodically to shuttle

FIGURE 5.11
Hamworthy Combustion's triple 120 metric-ton/h (132 U.S. ton/h) steam boiler module for FPSO.

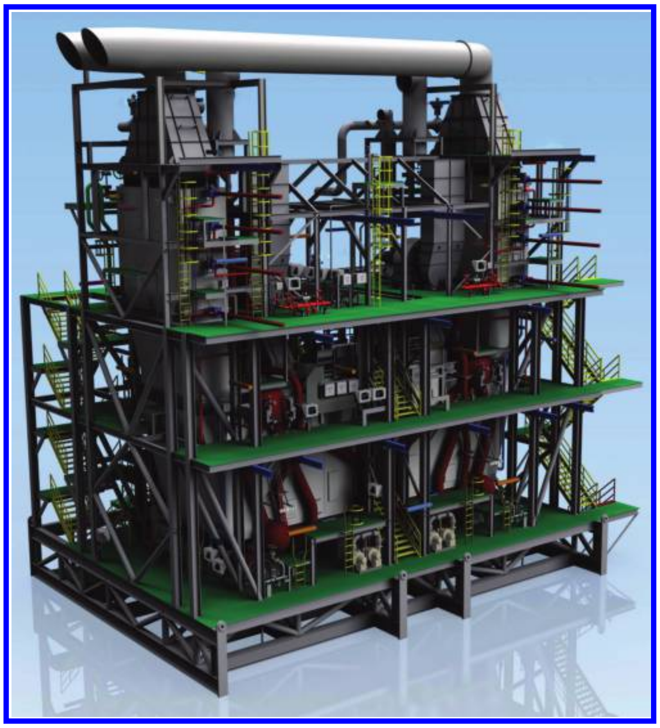


FIGURE 5.12
Hamworthy Combustion's high-pressure steam boiler module for power generation.

tankers, which then take the crude oil to refineries. The produced gas is often available at the production platform, and this is commonly supplied to the FSO for the boilers, via a second subsea pipe.

The firing of the fuel gas on the auxiliary boilers is the same as for an FPSO; however, as the fuel gas is cooled (due to the supply lying on the sea floor), any condensed liquids (water and hydrocarbons) must be removed before use in the boilers. Hence, fuel-gas "knockout" pots similar to the one shown in Figure 5.13 are common additions, with the recovered liquids drained to a suitable slop tank.

5.5.3 Floating LNG

Floating LNG, or FLNG, is a relatively new market, covering the application of permanently moored LNG carriers for storage or re-gasification duties or the offshore production of LNG over a gas field.

5.5.3.1 LNG Re-Gasification Vessels

With the demand for natural gas continuing to rise due to low prices and environmental benefits in terms of emissions, an alternative to installing long pipelines to

FIGURE 5.13
Hamworthy Combustion's fuel-gas knockout pot on an FSO.

carry the gas from the source to the users has evolved by way of LNG tankers called re-gasification vessels or floating storage and re-gasification units (FSRUs).

The FSRUs, whether new purpose built or converted from existing LNG carriers, are permanently moored at the shore side, usually close to the natural gas demand. The ships are fitted with LNG pumps and re-gasification systems, so it can deliver the LNG in its gaseous state (natural gas) directly into the natural gas pipe network. The LNG tanks are then regularly refilled from conventional LNG tankers that tie up alongside, and a ship-to-ship transfer of the LNG is carried out.

The re-gasification process requires a considerable amount of heat, usually by way of steam, to convert the LNG from a liquid state at –163°C (–261°F) to a gaseous state at an ambient temperature. Hence, a demand for low-pressure steam has to be available on the FSRU from its own boilers. The fuel for the boilers is LNG BOG from the cargo tanks or the re-gassed LNG should the BOG quantity not be sufficient to meet the demand.

Since the FSRU is permanently moored at the shore, it is common for the steam boilers to become subject to the same exhaust gas emission limits as the local legislation onshore. The natural gas firing provides very clean combustion with practically zero stack solids, carbon monoxide, and SO_2. However, since NOx levels of marine boilers are high (due to the very high-furnace heat release ratings), low-NOx burners such as Hamworthy Combustion's type DFL burner (see Figure 5.14) can be fitted. Depending on the NOx limit imposed, which varies greatly around the globe, other NOx reduction combustion techniques such as flue gas recirculation or NOx removal by SCR (selective catalytic reduction) may be necessary (see Volume 1, Chapter 15).

FIGURE 5.14
Hamworthy Combustion's DFL low-NOx register burner.

5.5.3.2 *Floating LNG Production Vessels*

As onshore and shallow-water gas supplies reduce, the move to deeper offshore gas production has also evolved. For deep water, a floating facility is required, and also since these gas fields are far offshore, pipelines to shore are impractical. Hence, the facility must not only produce sales quality natural gas, but also refrigerate and store it at −163°C (−261°F) as LNG. This is the FLNG FPSO.

The power demands of FLNG FPSOs are much larger, about—three to four times, than that of an equivalent oil producing FPSO, with one of the main demands being the large LNG liquefaction compressors, which are often steam turbine driven. Hence, another offshore steam boiler requirement, where the boilers would fire the produced fuel gas or LNG BOG, but also with the capability to fire liquid condensate that is produced, is a by-product of the raw gas separation and treatment process.

6

Process Heaters

Erwin Platvoet, David Brown, and Rasik Patel

CONTENTS

6.1 Introduction

A fired process heater is a piece of equipment with a fluid flowing inside tubes, which is heated by means of radiative and convective heat transfer from hot combustion gases (“flue gas”) to the tubes. Fuels that produce the hot combustion gases may be gas, oil, or a combination of both.

Process heaters are widely used in the process industries for heating, vaporization, and chemical reaction. The liberation duty can vary from 5 MMBtu/h (1.5 MW) for a small hot oil heater to 1500 MMBtu/h (440 MW) for large reforming and cracking furnaces. Volume 1, Chapter 2 describes the major chemical and refining processes and the types of heaters that can be found there.

This chapter discusses the typical process heater features and basic design procedures. There are many degrees of freedom when designing fired heaters for process applications.[1] The firebox can be rectangular or cylindrical; tubes can be installed horizontally or vertically, in the center of the firebox or against walls; burners can be fired horizontally or vertically, upward or downward, freestanding or against a wall, and so forth. Typical heater types and typical burner arrangements are given in API 560.[2] Despite this myriad of possible variations, the basic furnace design in each of the main petrochemical processes has become relatively standardized. This chapter will highlight the furnace designs in some of the main petrochemical and refining processes and show how these designs have evolved as a function of technological developments and the process requirements.

6.2 Furnace Components

The process heater is typically divided into a radiant section and a convection section. The radiant section is a refractory lined chamber, also called the firebox, in which the burners provide the required process heat. The process tubes are located both in the firebox (“radiant tubes”) and in the convection section (“convection tubes”). Air to the burners is often provided by natural draft, but can also be provided by a forced draft fan. In that case, it is also possible to preheat the air in order to maximize the furnace thermal efficiency. The negative pressure in the firebox is achieved using the draft developed from a stack or an induced draft fan connected to the convection section.

6.2.1 Firebox

The firebox of a heater is a steel enclosure (“casing”), which is lined with refractory. It can be cylindrical or rectangular, often depending on the total duty. Rectangular heaters are either box or cabin type. Cylindrical heaters are cheaper to build and take up less plot space. Rectangular heaters are typically used for large duties (>100 MMBtu/h or 29 MW).

6.2.2 Refractory

The firebox refractory lining is typically 4–12 in. (10–31 cm) thick and can be firebrick, cast refractory, ceramic blankets, or a combination of layers of these components. More details on refractory types can be found in Volume 2, Chapter 5.

6.2.3 Radiant Tubes

Radiant tubes are typically located on the walls and roof of the firebox. They are spaced at a certain distance from each other and from the refractory lining in order to permit some reradiation from the refractory to the backside of the tubes. Tubes may also be installed in the center of a rectangular firebox to enable them to be directly heated from both sides by burner flames. This gives a more even heat flux distribution. Large heaters may have both wall and center tubes with multiple lanes of burners.

In box- or cabin-type heaters, tubes can be oriented vertically or horizontally. In cylindrical heaters, the tubes are generally vertical, except for very small heaters where a helical coil is sometimes applied.

6.2.4 Burners

The burners can be located in the floor, the sidewalls, the end walls, or even the roof. Burners can be fired vertically upward (“up fired,” or “upshot”) or downward (“down fired,” or “downshot”) or horizontally. The overall burner and tube arrangement in the heater is strongly dependent on the process requirements. The vertical cylindrical upshot heater is the most widely used type of heater in the process industries.

In radiant wall heaters, the burners are arranged to fire onto the refractory surface, which then reradiates that heat to the radiant coils.

In most cases, the air to the burners is supplied by the negative pressure inside the heater, which is the “natural” draft resulting from the column of hot firebox gas. The typical natural draft burner pressure loss is in the range of 0.25–0.7 inH_2O (0.62–1.7 mbar). The pressure drop across a burner is a measure for the mixing intensity of the air and fuel. A low pressure drop, therefore, means less mixing, which tends to lead to relatively longer and wispier flames. Due to the long flames, natural draft burners are usually limited to 25 MMBtu/h (7.3 MW). Air distribution over the burners is harder to control with natural draft systems, so the minimum required excess air for combustion is typically 15% or more.

Alternatively the air can be supplied by a forced draft fan at higher positive pressure (2–6 inH_2O = 5–15 mbar) in order to create a more compact flame. This also allows the air to be preheated in order to lower the required fuel consumption. Forced draft gives better control over the airflow to the burners, which means that the minimum required excess air is typically around 10%.

6.2.5 Convection Section

The thermal efficiency of the radiant section is typically in the range of 40%–60%. This means that flue gas leaves the radiant section at a temperature between 1000°F and 2300°F (540°C and 1260°C). The significant amount of energy that is still contained in the flue gas can be recovered in the convection section. Depending on the lowest process inlet temperatures, the overall thermal efficiency can be as high as 92% when a convection section is installed. The tubes in the convection bank are installed horizontally with the flue gas flowing vertically upward. Tubes are generally spaced equilaterally in a square or triangular pattern. The convection bank is normally located above the radiant section, and one convection bank can be shared between multiple fireboxes.

Convection bank tubes typically have extended surfaces installed on them in order to increase the heat transfer area except for the first three rows (or "shock tubes"), which see direct radiation from the firebox.

Solid fins are the most common form of extended surface, but serrated fins and studs are also used. Fin spacing depends on the available pressure loss across the banks, the required heat transfer rate, and the degree of fouling expected. Fins are generally welded to the tube. Studs are typically used if oil firing is expected as they are easier to clean.

6.2.6 Fans, Stacks, and Dampers

Fans can be used to supply air to the burners (forced draft) or extract flue gas from the heaters (induced draft) or both (balanced draft). Process heater fireboxes are run at a pressure slightly below atmospheric. This prevents hot flue gas leaking out, which could damage the structure of the firebox and be hazardous to operators. The difference between the atmospheric pressure outside the heater and the pressure inside the heater is referred to as draft. When speaking about draft, positive numbers reflect negative pressures. The highest absolute pressure in a heater typically occurs at the flue gas exit of the radiant section, an area called the "arch" or "bridge wall." Draft at the arch is normally maintained at a minimum level of 0.1 inH_2O (0.25 mbar) (i.e., 0.1 inH_2O below atmospheric pressure). Too much draft (too much negative pressure in the heater) will increase the amount of leakage or "tramp" air into the firebox through sight doors, instrument connections, tube seals, etc. This "tramp air" reduces furnace efficiency as energy is absorbed in heating it to the final exhaust gas temperature (see Volume 1, Chapter 12). The draft in the firebox, and the suction required to overcome the gas pressure drop through the convection bank, is provided either by the chimney effect of the stack or by an induced draft fan. It is controlled by means of a damper located in the ducting leading to the stack or fan. In a forced or balanced draft system, dampers or inlet guide vanes will also be used to control the air rate through a forced draft fan and to balance the distribution of air to the various burners. The stack discharges the flue gases to a safe location such that they cannot endanger personnel or nearby equipment and that local pollution requirements are met. In the absence of an induced draft fan, it also provides the necessary draft to overcome the pressure loss across the convection bank, damper, and ducting and provides adequate draft at the top of the radiant section.

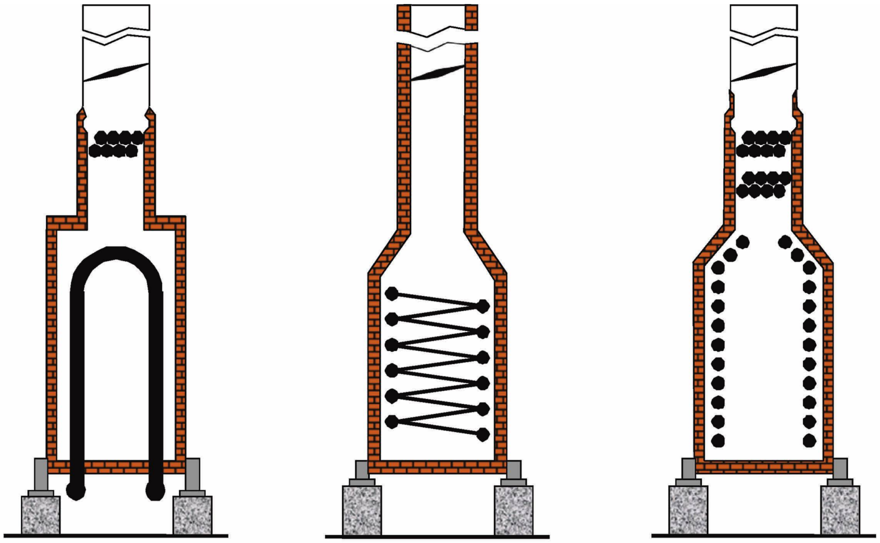
6.3 Common Heater Types

6.3.1 General

Heaters come in various shapes in order to best fit the required radiant and convection tube arrangements. The most common types are

- Box type
- Cabin type
- Vertical cylindrical
- Arbor/wicket
- Helical coil

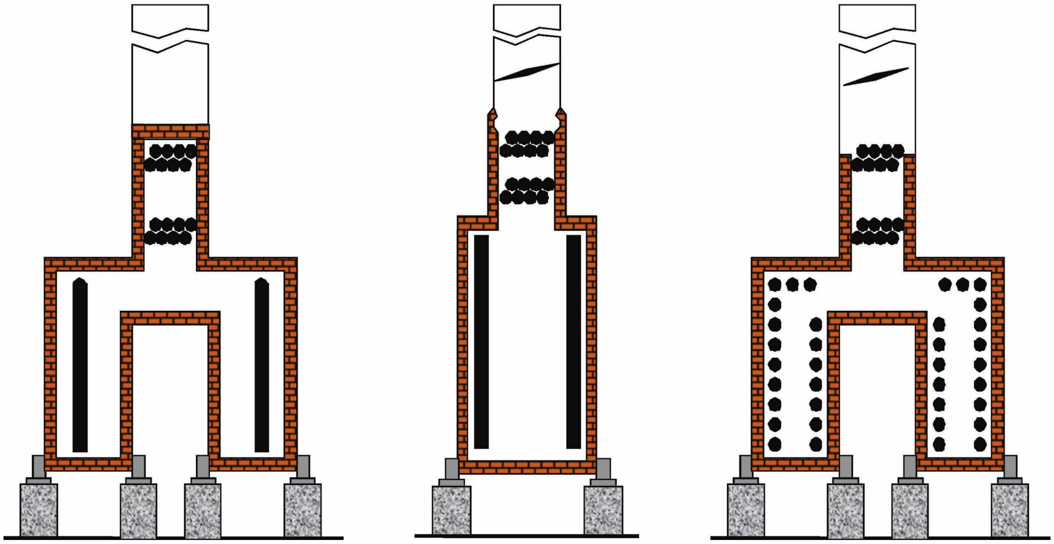
See Figure 6.1 for an overview.



Type A—
Box heater with
arbor coil

Type B—
Cylindrical heater
with helical coil

Type C—
Cabin heater with
horizontal tube coil



Type D—
Box heater with
vertical tube coil

Type E—
Cylindrical heater
with vertical coil

Type F—
Box heater with
horizontal tube coil

FIGURE 6.1
Typical heater types. (The Type designation from ANSI/API Standard 560, *Fired Heaters for General Refinery Service*, 4th edn., August 2007, Section 4.3, pp. 12–13.)

6.3.2 Refining Heaters

6.3.2.1 Refining Heater Design

The design of a refinery furnace is typically determined by the most economical way of arranging a certain heat transfer area within a firebox. The required heat transfer area is set by limits on the average heat flux by the process. For example, a crude heater is typically limited by a maximum average heat flux of 12,000 Btu/h/ft^2 (38 kW/m^2), while a vacuum heater limit is typically 10,000 Btu/h/ft^2 (32 kW/m^2). Combining these limits with the absorbed duty requirement will yield the required heat transfer area. The optimum radiant tube diameter and length can then be determined

FIGURE 6.2
Vertical cylindrical furnace arrangement.

knowing the pressure drop and velocity limitations as well as the desired vaporization profile and two-phase flow regime.

The required radiant tube metallurgy is a function of the peak heat flux, which itself is a function of the chosen tube arrangement and burner design.

Due to the relatively mild heat flux and temperature demands and the smaller footprint per heater duty, a large portion of newer refinery furnaces are vertical cylindrical with vertical tubes against the wall and burners arranged in a circle in the center of the firebox (see Figures 6.2 and 6.3).

From the point of view of tube supporting, plot requirements, number of burners, structural steel, refractory cost, and so forth, this is usually the most economical way to build a heater. However, since the radiant process tubes are heated only from one side, the heat flux is unevenly distributed around the tube circumference and along the length of the tube. The large temperature differences that can result means that such an arrangement can only be applied in services where the peak metal and film temperatures are low compared to their respective limits.

Box- or cabin-type heaters, shown in Figures 6.4 through 6.6, with horizontal tubes are often selected for maintenance reasons. This arrangement allows for easy access to the tubes for inspection, cleaning, and replacement. For this reason, it is often used to heat process streams that foul easily, such as heavy gas oils and vacuum residue.

FIGURE 6.3
Two vertical cylindrical fireboxes with a common convection section.

When peak metal temperature and process film temperature become the limiting factor, the furnace designer often has to use double-fired tubes. In that case, the tubes are located in the center of the firebox with burners on each side of the coils. This results in a much better tube circumferential heat distribution compared to tubes that are heated from one side only.[3] As a result, the peak temperature is lower for a given average heat flux, which allows for a significant reduction in radiant

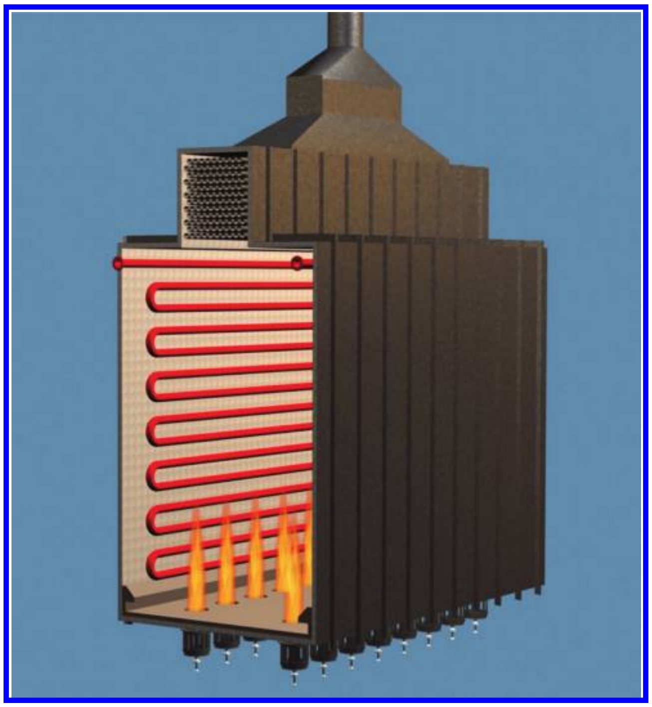


FIGURE 6.4
Box-type heater with horizontal tubes.

FIGURE 6.5
Box-type heater with horizontal tubes. (From McAdams, J.D. and Waibel, R., Methanol furnace combustion troubleshooting, *Proceedings of International Methanol Technology and Operators Forum (IMTOF)*, June 2007.)

coil surface area. Another benefit of this arrangement is that burner spacing and flame length can be matched with a desired temperature profile along the tubes. Even more control on the radiant heat transfer can be obtained by firing the burners against the sidewalls and using the walls as radiating planes.

Tube coils and burners in larger heater designs can be arranged in multiple lanes or cruciforms to optimize the firebox volume.

6.3.2.2 Burners for Refining Processes

Due to the large variation in refinery furnace designs, there also exists a wide variety in burner designs (see Chapter 1). Burners can be round or rectangular, firing against a wall or freestanding, installed horizontally or vertically, firing upward or downward, oil fired or gas fired, forced draft or natural draft, premix or raw gas, fuel staged or air staged, and so forth. As such, it is difficult to describe a typical refinery burner. However, they do all face the same typical challenges.

Some common challenges specific to refinery applications are relatively low furnace operating temperature, significant variation in fuels, and retrofitting existing equipment to utilize modern low or ultralow emission burners. Relative to reforming and petrochemical applications, most refinery heaters operate at relatively low temperatures, 1400°F–1600°F (760°C–870°C). Depending on the type of fuel, type of burner, turndown requirements, and emission requirements, low furnace temperatures can present a challenge for maintaining low CO and unburned hydrocarbon emissions. Maintaining stable burner operation, especially with ultralow emission burners, is also more difficult at lower furnace temperatures. Burners in refinery applications typically burn fuels referred to as refinery fuel gas (RFG), which are a collection of products or by-products of many processes throughout the plant and may be supplemented with natural gas makeup (see Volume 1, Chapter 3). Due to the large number of processes that exist in a typical refinery, the composition of the available fuel gas varies significantly from plant to plant, from heater to heater, and even over time on a given heater. Variation in fuel properties can significantly impact burner performance parameters such as stability, flame dimensions, and emissions.

Due to the age of many refineries, many process burners in refinery applications are being replaced either because they have reached their reliable life span or because regulatory mandates are requiring the installation of modern low emission burners. Replacing burners in an old heater with ultralow emission burners can present several unique challenges. Due to the necessary design approach, ultralow NOx

FIGURE 6.6
Cabin heater.

burners generally produce longer and wider flames than conventional raw gas or premix burners. The burners also generally have a larger footprint than conventional burners. The performance of ultralow NOx burners is also more sensitive to burner spacing and flame-to-flame interaction. Because many refinery heaters were originally equipped with conventional raw gas or premixed burners, replacing them with low emission burners can result in flames that are taller than the maximum 66% of radiant section height recommended by API 560.[4] It can also be difficult to fit the new burners in the original mounting locations due to the larger size and spacing requirements of low emission burners. If the low emission burners are installed too close together, the flames may interact, resulting in increased emissions and flame length.

6.3.3 Delayed Coker Heaters

6.3.3.1 Coker Heater Design

The delayed coking process is used to convert vacuum residues to lighter hydrocarbon fractions such as LPG, naphtha, light gas oil, and heavy gas oil. See Volume 1, Chapter 2 for more information on the coking process. Like any other typical process heater, a coker heater consists of a convection section and a radiant section cell, but often several radiant cells (typically up to four) are connected to a single convection section. A typical coker furnace has horizontal tubes arranged in the center of the firebox, with burners firing on both sides of the coils (Figure 6.7). The liquid inside the tubes is gradually heated and vaporized as it flows from the top to the bottom tube.

Double wide arrangements have two horizontal coils with burners in three parallel lanes (Figure 6.8). The center burners then typically have double the capacity of the outer burners.

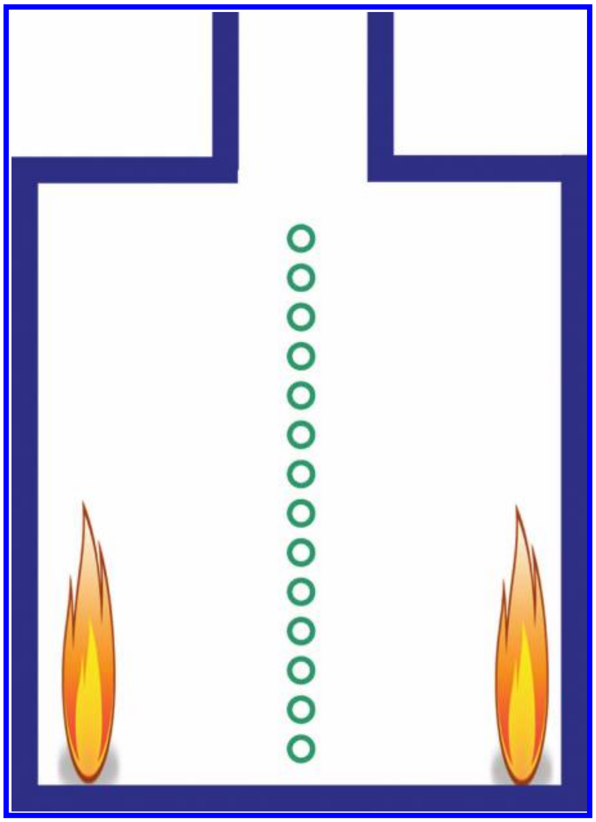


FIGURE 6.7
Typical coker furnace.

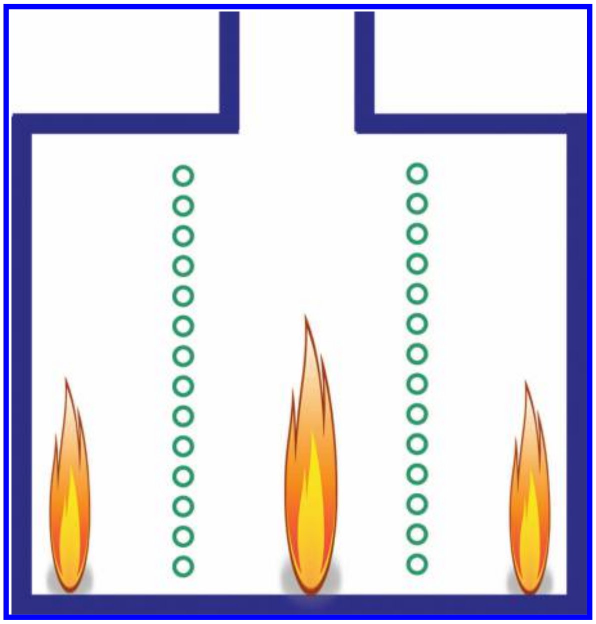


FIGURE 6.8
Double wide coker furnace.

6.3.3.2 Burners for Delayed Coker Heaters

Burners in coker furnaces are usually fired at relatively low duties, 1.2–2.5 MMBtu/h (0.35–0.73 MW). They are normally up-fired floor burners that can be fired in a freestanding mode or against a fired wall. Alternatively, the burners can be mounted horizontally in the wall and firing up the refractory wall. Because cracking and film boiling are to be avoided, the process fluid residence time in the tubes is short and the heat flux is high. In order to achieve a gradual and constantly increasing process temperature profile, the burner heat flux profile needs to be precisely controlled. This translates to a tight specification of flame length and width. Combined with air preheat and high firebox temperature, these specifications can be challenging for a designer to achieve ultralow NOx guarantees. See Chapter 1 for the various process burner options.

6.3.4 Reforming Heaters

6.3.4.1 Reforming Furnace Design

The heat for the endothermic reactions is provided under controlled conditions in a combustion chamber, wherein hot flue gas provides heat to the process gas mixture passing through high alloy, catalyst-filled tubes. The key elements in a steam reformer design are

- *Firing arrangement*. The tubes and burners are arranged to allow for adjustment and control of temperature profile along the length of the tubes. A double-fired arrangement (i.e., tubes fired equally from both sides) is used to achieve good circumferential heat flux distribution on the tubes.
- *Thermal expansion*. The furnace operates at elevated temperatures, and therefore, the inlet feed distribution system and outlet collection system connected to the catalyst tubes are designed to allow thermal expansion without risk for over stressing. Equally important is good flow distribution to each catalyst tube, and to help ensure this, piping design should be symmetrical.
- *Mechanical design*. The materials of construction are selected suitable for service to ensure prolonged life. For the inlet system, up to about 1050°F (566°C) operating temperature, the use of low alloy material such as 2¼Cr-1Mo is common, while for higher temperatures, stainless steel would be used. For the catalyst tubes, most modern steam reformers use either 25Cr-35Ni alloy modified with niobium or a microalloy version of the alloy, which contains trace elements such as titanium for added stress to rupture strength. The tubes are designed for a minimum life of 100,000 h. The outlet collection headers are typically made of cast 20Cr-32Ni material.

Commercially there are four types of steam reformers:

1. Top fired
2. Side fired
3. Terrace wall fired
4. Bottom fired

The top-fired reformer (see Figure 6.9) consists of a firebox containing anywhere from one to several rows of tubes. The process gas flows downward through the catalyst-filled tubes. The burners are located in between the tube rows in the top of the firebox and as such are located at one level, albeit above grade. The tubes are heated by radiation from the flames, flue gas, and some convective effects. The process gas and flue gas flow is cocurrent with both exiting at the bottom of the firebox. This arrangement allows for rapid heating of the hydrocarbon steam mixture. Additional heat recovery takes place in the convection section, which is typically located at grade. This type of furnace arrangement can accommodate small to very large capacities within a single firebox.

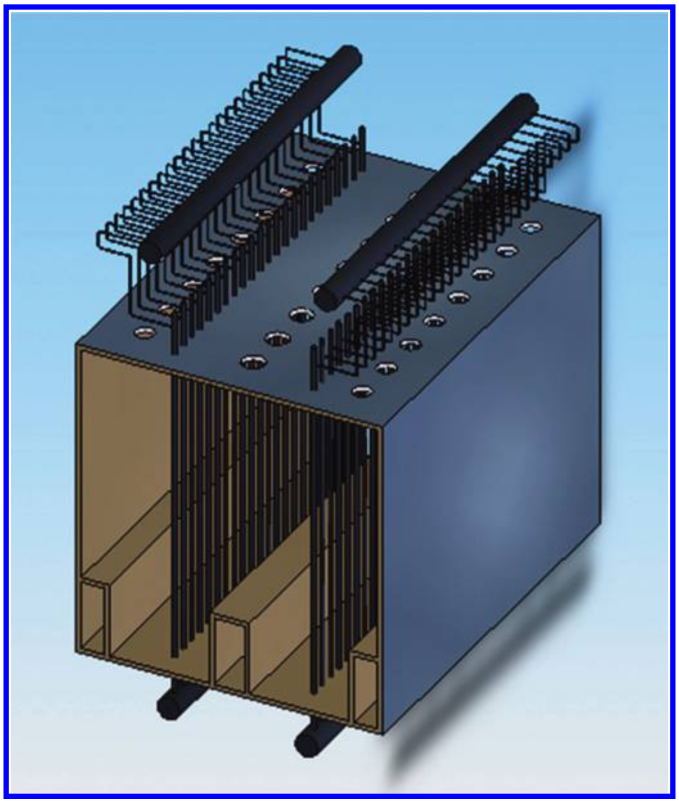


FIGURE 6.9
Top-fired reformer.

The side-fired reformer (see Figure 6.10) consists of catalyst tubes that are positioned in a single row along the centerline of the firebox. Due to the layout, this type of furnace arrangement results in a long narrow firebox. The burners are located at multiple levels on the two sidewalls of the firebox and evenly distributed over the entire wall. The tubes are heated by radiation from the walls, flue gas, and, to a small extent, convective effects. The process gas flow through the catalyst tubes is downward, while the flue gas flow is upward. For large capacities, it is common to have multiple fireboxes located side by side sharing a common inlet distribution and outlet collection system and waste heat recovery section.

The terrace wall-fired reformer (see Figure 6.11) is a refinement of the sidewall-fired reformer in that many of the salient features described earlier are similar but with two major differences: the number and type of burners used. This type of arrangement typically uses two levels of firing and therefore results in fewer burners for a given capacity. The primary objective of either

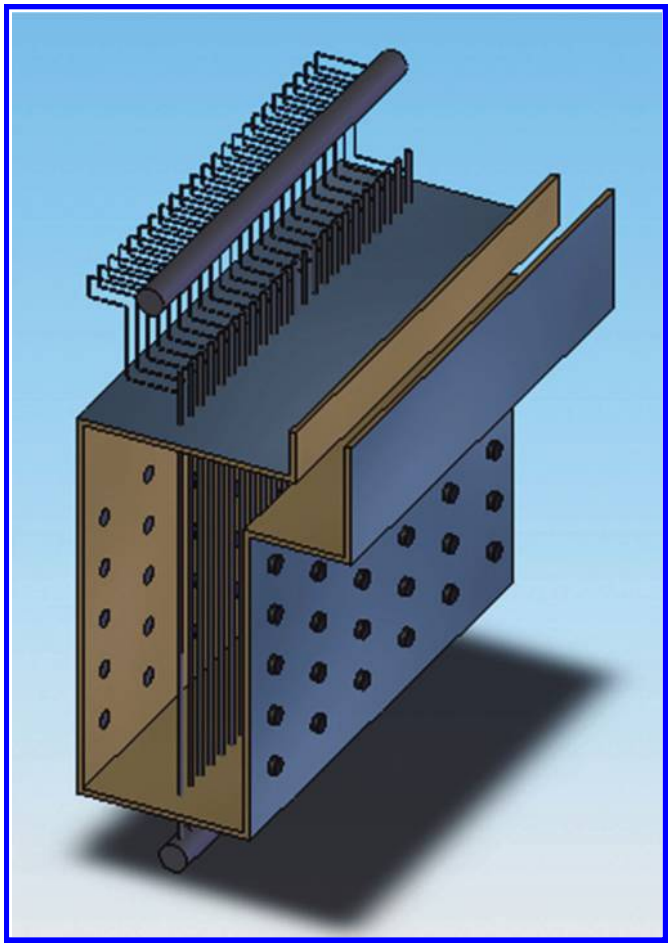


FIGURE 6.10
Side-fired reformer.

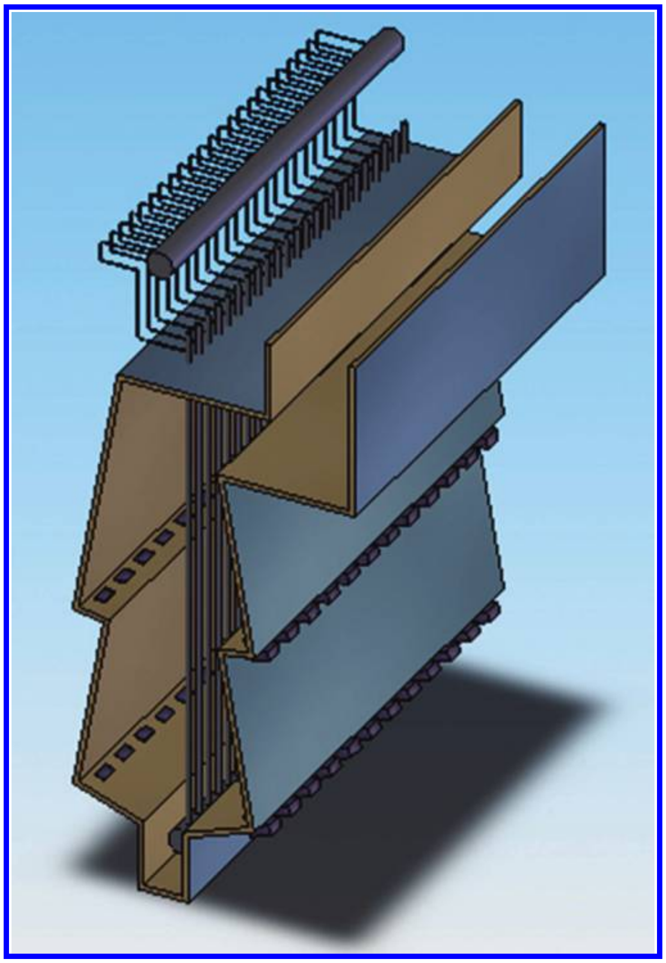


FIGURE 6.11
Terrace wall-fired reformer.

of the side-fired arrangements is to allow a controlled temperature profile along the length of the tube.

The bottom-fired reformer (see Figure 6.12) can be designed as a box type or a cylindrical type depending on the size of the unit. Process flow in the box-type reformer is down through the tubes. Heat input is accomplished via burners located at the floor of the furnace. Process gas and flue gas flows are countercurrent in this instance. In this arrangement, since the heat liberation is occurring near the process outlet, special attention needs to be applied in the burner selection and flame characteristics. Consequently, in most instances, additional margin should be applied to the tube design temperature so as not to be limiting during operation.

In contrast, the cylindrical reformer is single fired and has process flow up through the catalyst tubes. Due to this arrangement, the pressure drop across the tubes is relatively low so as to prevent catalyst lifting. This type of reformer is typically used in very-small-capacity plants.

6.3.4.2 Burners for Reforming Heaters

Typical fuels for steam reformers consist of a waste gas stream generated in the process supplemented with makeup fuel such as natural gas or refinery gas or sometimes even liquid naphtha. The composition and quantity of waste gas available will depend on the application. For instance, in a hydrogen plant with a pressure swing adsorption (PSA) unit, the waste gas is low heating value gas due to the relative high content of carbon dioxide (~50% to 60%) and is available at low pressure (between 2 and 3 psig = 0.14 and 0.20 barg). This requires secondary fuel piping and burner fuel distributors to handle the higher volumetric flow rate and lower pressures. The relative contribution in terms of heat release is dependent on feedstock and whether air preheat is used or not. Typical splits are shown in Table 6.1.

The burner design is further complicated in that during initial start-up of the plant, PSA gas is not available and, hence, burners need to be sized to fire on makeup gas only. It is good practice in this instance to size the

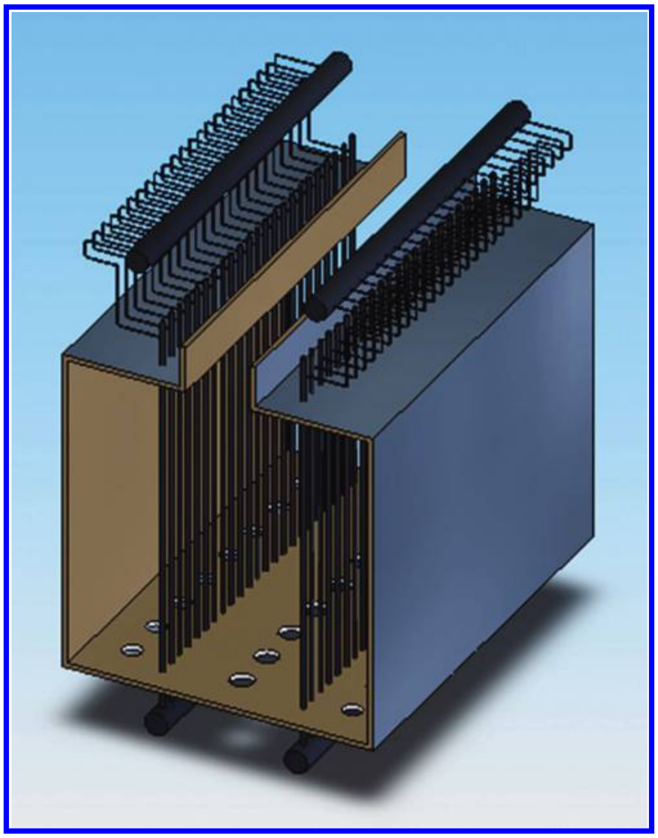


FIGURE 6.12
Bottom-fired reformer.

TABLE 6.1
Typical Firing Splits

Application	Firing Split (PSA: Makeup Gas)
Ambient air	40%:60%
Preheated air	80%:20%

burners for approximately 50%–60% of the design heat release on makeup gas only. That way, once the PSA gas is introduced, the turndown on makeup gas does not become a limiting factor.

On the other hand, methanol plant waste gas contains relatively high hydrogen (~65% to 75%) and is available at high pressure from the synthesis loop. This allows the designer the flexibility to select between keeping the fuel piping and controls separate and combining the fuels and simplifying the controls.

Similar reasoning can be applied in ammonia plants, wherein depending on the configuration of the purification system, the waste gas can contain a relatively high level of inert (~60% to 70% mostly nitrogen and argon), but is available at not too low a pressure so that mixing of fuels can be considered.

TABLE 6.2
Favored Flame Shape for Various Reformer Types

Reformer Type	Burner Capacity (Typical), MM Btu/h	Flame Characteristics
Top fired	8–10 (2.35–2.93 MW)	Round, compact flame
Side fired	1.2–2.0 (0.35–0.6 MW)	Flat flame, sweeping along the radiant wall
Bottom fired	6–8 (1.75–2.35 MW)	Round, narrow "pencil" flame

In general, the burners in steam reformer service will need to meet the following demands of the process:

- *Flame shape to mitigate tube impingement*. Due to the close spacing between adjacent burners and between burners and process tubes, compact flames are favored, the exact shape dependent on the steam reformer (see Table 6.2).[5]
- *Fuel flexibility*. In most instances, the burners will need to be of the dual-fuel type.
- *Draft*. Steam reformer applications can vary from natural draft (or induced draft) to forced draft. In this instance, the available draft will have an influence on the flame length. For natural draft applications, it may be necessary to limit the heat release to achieve the desired flame length.
- *Combustion air*. Typical design for burners is 15% excess air for natural (or induced) draft applications and 10% excess air for forced draft. In some instances, the design may call for turbine exhaust gas (TEG) to be used as the oxygen source in lieu of combustion air. Depending on the fuel fired in the gas turbine, the hot exhaust stream can contain around 13%–14% oxygen (wet basis), sufficient to supply all or part of that needed for combustion in the steam reformer.
- *Environmental considerations*. In order to meet the new environmental regulatory requirements, the burners in a steam reformer service are typically specified now as the low emission type. While the exact NOx level required will be site specific, it is not uncommon to achieve an expected value of 0.03 lb/MMBtu (HHV) (12.9 kg/MJ) heat released for burners in hydrogen plant service using PSA off-gas.

6.3.5 Steam Cracking Furnaces

6.3.5.1 Cracking Furnace Design

The typical firebox layout is shown in Figure 6.13. Radiant coils are placed in the center of the firebox with burners placed on both sides in order to provide uniform heating. Several variations of this design are shown in Figure 6.14.

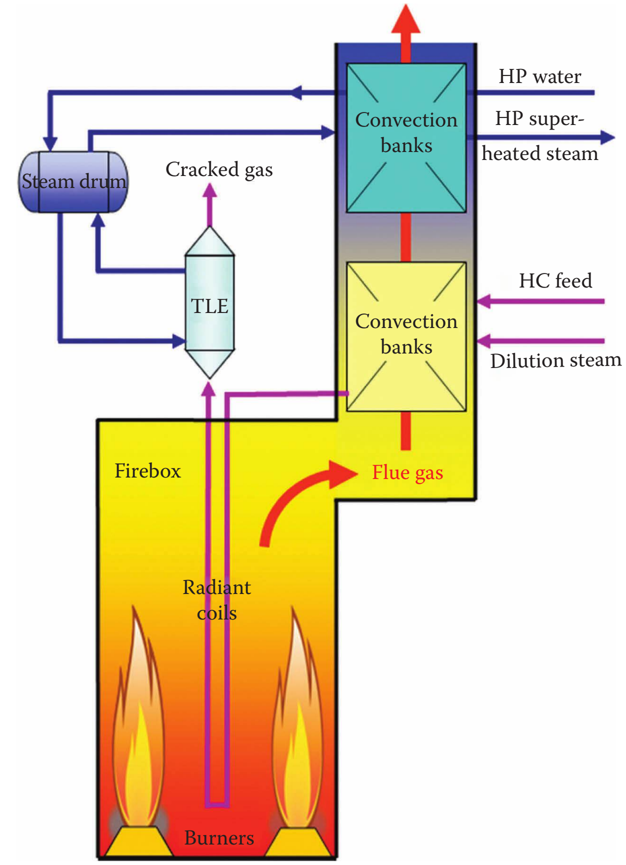


FIGURE 6.13
Typical cracking furnace firebox layout.

6.3.5.2 Burners for Cracking Furnaces

In order to understand the critical role of burners in a steam cracking furnace, it is important to keep in mind that, unlike most process heaters, the steam cracking furnace is the main reactor in the plant. A deviation from the ideal process temperature profile has a direct impact on plant throughput and profitability since it increases coking rates and lowers the coil selectivity. The process temperature profile inside the cracking coils is, to a certain degree, determined by the layout of the cracking coils, but the heat distribution in the box, determined by the number, type, location, and heat release of the burners, remains the key parameter.

Cracking furnaces typically come in four different burner layouts:

1. 100% radiant wall fired
2. 100% floor fired
3. Combination floor + wall fired
4. Floor + terrace fired

The layout depends on licensor, age of the furnace, type of radiant coil, size of the firebox, and NOx emission limitations. In early firebox designs, the fired duty was mostly provided by radiant wall burners (see Figure 6.15).

Since the heat release of a single burner was limited to about 1 MMBtu/h (0.3 MW), many burners were needed to provide the total firing duty. The number of burners in a naphtha cracking furnace would typically range from 160 in the 1970s up to 240–300 in the late 1980s. The low firing duty meant that a single burner could be considered as a point source of heat. The heat distribution could be adjusted by varying the fuel pressure between fuel headers (zone firing was, historically, a popular feature), by adding more or less secondary air to individual burners, or simply by turning burners off.

The ongoing tendency to reduce investment cost and maintenance cost has significantly changed the character of these furnaces since the late 1980s. The firing capacity of a single firebox increased tremendously, which made the use of 100% sidewall burners very costly and maintenance-intensive. Although all the main licensors have moved away from the use of 100% wall firing, there is no clear consensus on the ideal system. The use of large heat

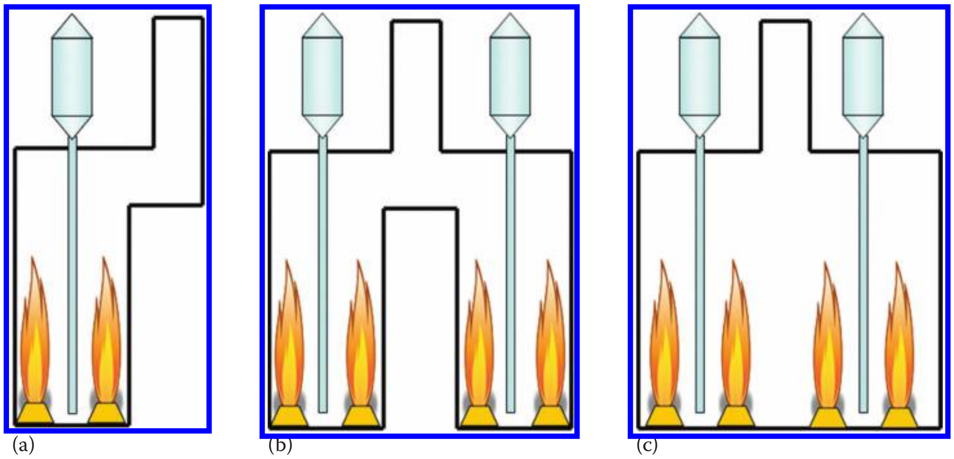

(a) (b) (c)

FIGURE 6.14
Variations of typical cracking furnace firebox layout. (a) Single cell. (b) Twin cell. (c) Twin cell without inner walls.

FIGURE 6.15
Radiant wall burners.

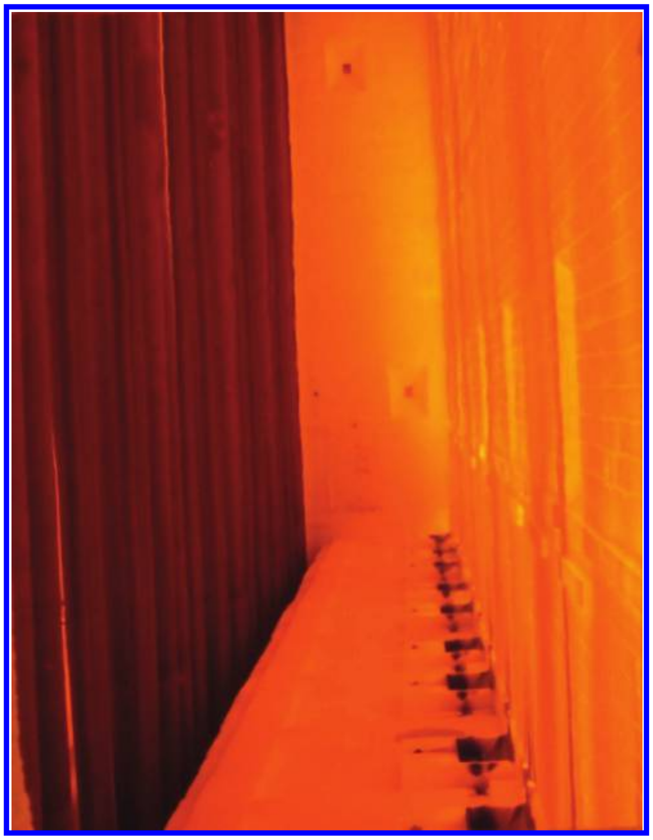


FIGURE 6.16
Large heat release floor burners.

release floor burners, typically in the range 5–15 MMBtu (1.5–4 MW), is common (see Figure 6.16). Sometimes a few rows of wall burners are specified as well; sometimes the furnace is all-floor fired; sometimes a second row of burners ("terrace") is added, situated about halfway up the wall, consisting of burners of similar design to the floor burners. A key parameter that is commonly specified by the furnace designer, and demonstrated by the burner supplier, is the vertical normalized incident heat flux profile.[6]

While the firebox duty has increased, the firebox size has not kept equal pace due to efforts to reduce investment costs. As a result, the firing density (expressed in fired duty per firebox volume or floor surface area) has increased faster than the firebox duty, and burners have been placed so close together that they have started influencing each other in a negative sense. These burner–burner interactions have been the reason for a number of designs for which at start-up the combustion system has exhibited poor flame quality—severe rollover of the flames into the cracking coils (see Figure 6.17). The continuous flame impingement resulted in excessive local coking of the process tubes, short run-lengths, carburization of the material, and coil cracks. Even the NOx emissions have been higher than anticipated based on results measured during the burner test (more correctly termed "witnessed performance demonstration").

Like all other process furnaces, cracking furnaces have become the subject of tighter NOx and CO emissions limits. Cracking furnaces, however, are especially being scrutinized because of their high firing rates, which make them very large producers of NOx in an absolute sense. Moreover, the firebox temperatures are the highest in the petrochemical industry, with bridgewall temperatures up to 2300°F (1260°C) compared to the typical firebox temperatures in refinery furnaces of 1400°F–1600°F (760°C–870°C). This means that even on a relative basis, the thermal NOx production of a cracking furnace is often two to three times higher than that of a refinery furnace.

There are other factors that make the design of burners for cracking furnaces especially challenging, such as fuel flexibility. The normal (design) fuel is usually a by-product of the process, referred to as residue gas.

FIGURE 6.17
Example of flame rollover in a pilot-scale cracking furnace.

The main components are methane and hydrogen. Depending on the design of the plant, the hydrogen content can be as low as 5 vol% or as high as 80 vol%. The burners also must have the flexibility to burn other fuels in other scenarios such as "start-up," "backup," and "emergency backup." Sometimes the alternate supply is natural gas (mostly methane). More often, the other fuel supply is LPG. The plant has an integrated fuel system, the consequence of which is that the design fuel can be contaminated with higher molecular weight species (C4s, C5s), which might be olefinic, diolefinic, or worse. The result is often tip plugging during these incidents.

For reasons of energy efficiency, the burner is occasionally integrated with either a combustion air preheat system or with TEG.

In order to properly account for all of the factors mentioned earlier, single-burner testing has almost completely been superseded by multiple-burner testing. Because problems at start-up are so damaging and costly,[7,8] the test is often complemented by computational fluid dynamics (CFD) simulations (see Volume 1, Chapter 13), the accuracy and sophistication of which is developing rapidly.[9]

6.3.5.3 *Integration with Turbine Exhaust Gas*

Gas turbines typically run at a high excess air. Turbine material temperature limits require a very lean combustion, and the turbine exhaust gas (TEG) also contains cooling air for the turbine blades. As a result, the TEG contains high levels of oxygen, typically between 13 and 17 vol%.

In power plants, the waste heat in the TEG is used to generate steam in a heat recovery steam generator, reheating it using burners that consume this excess oxygen. As an alternative for such a combined cycle power generation system, the TEG can be used as an oxidant for industrial burners. Although the TEG oxygen content is less than that of air, it is still sufficient for stable combustion, especially since its temperature is typically 950°F–1050°F (510°C–565°C).

Steam cracking furnaces and steam methane reformers are very good candidates for integration with gas turbines, since the amount of flue gas from a single gas turbine is compatible with the firing requirements of a world-scale plant. While firing with TEG requires some investments such as forced draft fans for makeup/backup air and the ducting between furnaces and gas turbine, reduction in firing rates due to its high temperature is in the range of 15%–20%. In addition, firing with TEG increases the amount of flue gas relative to firing with air, due to its low oxygen content. A higher amount of flue gas through the convection section of a cracking furnace results in a higher production of superheated high-pressure steam (1800 psi, 980°F) (124 bar, 527°C), which further improves the energy balance of the plant. Finally, the high overall thermal efficiency that is achieved since the flue gas in a cracking furnace is typically cooled down to ~250°F (120°C) makes this a very attractive concept, especially when energy costs are high.

Combustion tests with ultralow NOx burners firing 1000°F (540°C) TEG have shown a 20% reduction in NOx (on a concentration basis) compared to the ambient air case.[10] However, on an absolute basis (expressed in lb/h), the combined NOx emissions of the gas turbine and cracking furnace are only ~10% higher than the cracking furnace firing ambient air.

6.4 Process Heater Design

6.4.1 Radiative Heat Transfer

This section summarizes some of the background given in Volume 1, Chapter 7 on heat transfer, which is relevant for the rating of fired heaters. For more detailed information, please refer to that chapter.

The radiation heat flux $E_{\lambda,b}$ from a blackbody that has an absolute temperature T is given by Planck's equation[11]:

$$E_{\lambda,b}(\lambda,T) = \frac{2\pi c^2 h}{\lambda^5}\frac{1}{\exp(hc/\lambda kT)-1} \tag{6.1}$$

with h, c, and k as constants and λ as the wavelength in which the radiation is emitted. A blackbody is defined as a perfect emitter and absorber of radiation; it absorbs all incident radiation, and no surface can emit more for a given temperature and wavelength. Planck's equation integrated over all wavelengths yields the simple and well-known Stefan–Boltzmann equation for radiation from a blackbody:

$$E_b = \int_0^\infty E_{b,\lambda} d\lambda = \sigma T^4 \tag{6.2}$$

with $\sigma = 5.670 \times 10^{-8}$ W/m^{2}K^4 as the Stefan–Boltzmann constant.

6.4.1.1 *Emissivity*

Radiation from a gray (nonblack) surface is less than from a black one. This difference is expressed by the emissivity ε for a real surface, which is defined as the ratio of its emissive power to that of a black surface at the same temperature:

$$E = \varepsilon\sigma T^4 \tag{6.3}$$

Absorptivity α of a surface is the fraction of incident radiation that is absorbed. For radiation at a particular wavelength, the spectral emissivity ε_λ is equal to α_λ. For systems in thermal equilibrium (i.e., wall temperature is equal to the radiating gas temperature) and for gray surfaces, ε is equal to α.

6.4.1.2 Effective Radiant Coil Plane

The radiative heat transfer between two black surfaces depends on the difference in the emissive powers of the two surfaces and the relative geometrical orientation of the surfaces. The geometry, which determines how the surfaces view each other, is introduced through a quantity called the view factor, designated by F. For two black surfaces, the radiant transfer is given by

$$Q_{1-2} = A_1F_{12}\sigma T_1^{\,4} - A_2F_{21}\sigma T_2^{\,4} \tag{6.4}$$

with A_i as the respective surface areas.

For gray surfaces $A_1F_{12} = A_2F_{21}$, so for radiant heat transfer from surface A_1 to surface A_2, the Stefan–Boltzmann law can be written as

$$Q_{1-2} = \sigma A_1F_{12}\varepsilon_1\left(T_1^{\,4} - T_2^{\,4}\right) \tag{6.5}$$

with F_{12} as the shape factor, A_1 as the surface area, and ε_1 as the emissivity. For the special case of radiation between two infinite parallel plates[7]:

$$Q = \frac{\sigma A\left(T_1^{\,4} - T_2^{\,4}\right)}{(1/\varepsilon_1) + (1/\varepsilon_2) - 1} \tag{6.6}$$

6.4.1.3 Radiation from Nonluminous Flames

The predominant fuel source for fired heaters is fuel gas due to its ease of handling and maintenance. Combustion of fuel gas gives nonluminous flames. When calculating the radiation from nonluminous flames, it is important to consider that a lot of gases are "nonparticipating"; they are completely transparent to thermal radiation. Examples are symmetric molecules such as N_2 and O_2. Asymmetric molecules such as H_2O, CO_2, CO, and hydrocarbons are participating, but radiation absorption and emission occur in narrow wave bands (see Figure 6.18). The energy in these wave bands corresponds to different oscillation states of the molecule and the discrete steps to go from one to the other.

As a result, the radiative properties of the flue gas depend on

1. Concentration of the participating gases
2. Temperature
3. Pressure
4. Effective thickness (also called optical thickness or mean beam length)

The emissivities of radiating gases have been determined experimentally and are typically shown as curves. See Sections 7.5.6 and 7.5.7 of Volume 1 for calculations and more information.

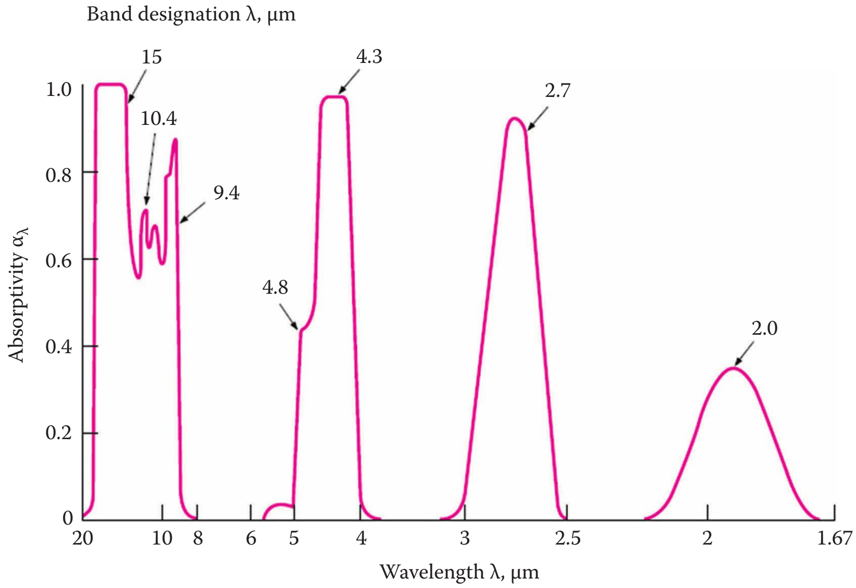


FIGURE 6.18
Spectral absorptivity of CO_2 at 830 K and 10 atm for a path length of 38.8 cm.

6.4.2 Lobo–Evans Method

There are several ways to calculate the radiant transfer in a fired heater, ranging from a relatively simple overall model to very complex and detailed calculations. The more complex models require that the radiant chamber is divided into (a few or very many) zones for which the heat balance is solved. The zone model[12] is one of these approaches, and of course the CFD approach (see Volume 1, Chapter 13) is the version that takes zoning to the extreme with millions of calculation cells.

This chapter will demonstrate a simple well-stirred model that can achieve relatively good predictions for a wide range of furnace models. It only calculates the overall performance of a radiant chamber, so it will not give information about the flux and temperature profiles inside a firebox. It was first proposed by Lobo and Evans[13] and later further refined by Hottel.[14,15]

It is based on the following assumptions:

- The firebox is well stirred, which means that the gas in the radiant chamber has a uniform temperature, which is equal to the temperature of the gas at the radiant exit.
- The flue gas is gray.
- The heat sink area is gray and at a single temperature.
- Heat loss through the refractory walls is negligible.
- Convective heat transfer is negligible.
- The sink and refractory surfaces are intimately mixed.

The theory and examples in this section will focus on the case of radiant tubes backed by a refractory wall, which is the most common type of arrangement. For other cases such as multiple tube rows against a wall or tubes in the center of the firebox, the analysis is more complex.

6.4.2.1 Gas–Surface Exchange Area

Similar to Equation 6.6, one can define a gas–surface exchange area $(GS)_R$ that accounts for tube surface emissivity, gas emissivity, firebox shape factor, and mean beam length:

$$(GS)_R = \frac{A_T}{(1/C_s\varepsilon_s)+(1/\varepsilon_g)-1} \tag{6.7}$$

where

ε_g is the furnace gas emissivity
ε_s is the sink emissivity
A_T is the total enclosing surface area (ft^2)
C_s is the heat sink fraction of the total enclosing area (ft^2)

C_s is not just based on the projected area of the tubes; it accounts for the tube surface *plus* the refractory behind it. By using the product of $C_s\varepsilon_s$, the problem of radiation to a complex system of tubes backed by a wall is reduced to a simpler version of radiation to an equivalent plane. It is basically saying that the radiative power of this continuous plane is the same as the tubes and the refractory behind it while having the same temperature as the tubes and an effective emissivity ε_s. The effective emissivity itself accounts for the direct radiation to the tubes and the reradiation from the back wall. It is therefore dependent on the tube spacing X_t. This is shown in Figure 6.19.

Some fraction of the radiation is directly incident onto the tubes, while the rest is absorbed and reemitted by the refractory. A fraction of the reemitted radiation will also be

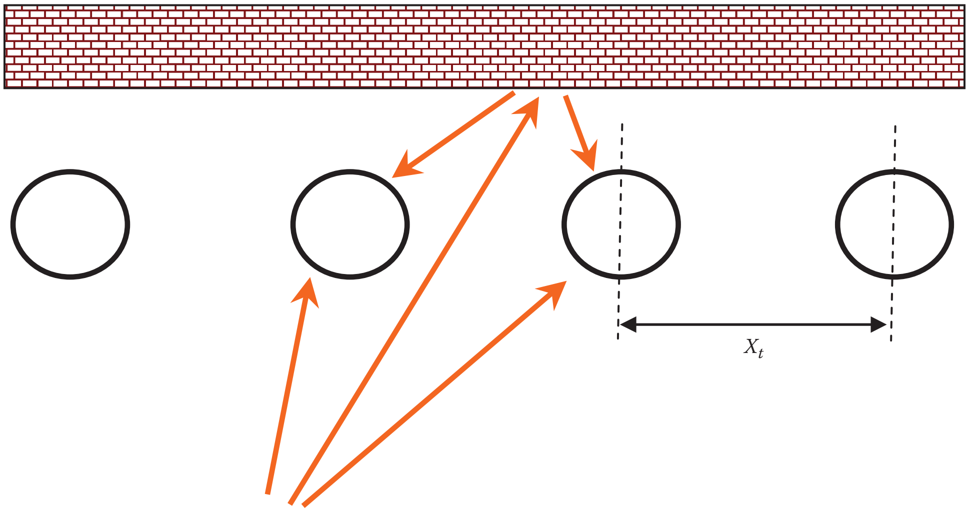


FIGURE 6.19
Radiation to a single row of tubes backed by a refractory wall.

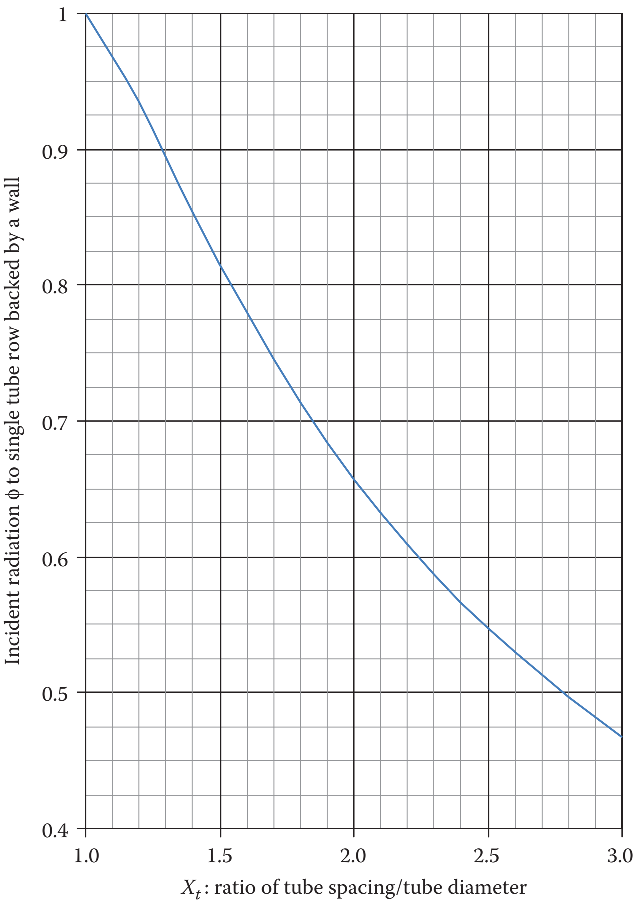


FIGURE 6.20
Incident radiation to a single-tube row, which is backed by a refractory wall.

absorbed and reflected by the tubes. The incident radiation ϕ to the tubes can be expressed by the following equation:

$$\phi = 1 - \frac{1}{X_t}\left[\sqrt{\left(X_t^2 - 1\right)} - \cos^{-1}\left(\frac{1}{X_t}\right)\right] \qquad (6.8)$$

This equation is also plotted in Figure 6.20. The effective emissivity of this arrangement is now given by Equation 6.9 and plotted in Figure 6.21 for various tube emissivities:

$$\varepsilon_s = \frac{1}{1/\phi(2-\phi) + (X_t/\pi)\left((1/\varepsilon_t) - 1\right)} \qquad (6.9)$$

6.4.2.2 Reduced Firebox Efficiency and Firing Density

Using the exchange factor $(GS)_R$, the radiative heat exchange between the furnace gas and the heat sink is

$$\dot{Q}_R = \sigma(GS)_R\left(T_g^4 - T_s^4\right) \qquad (6.10)$$

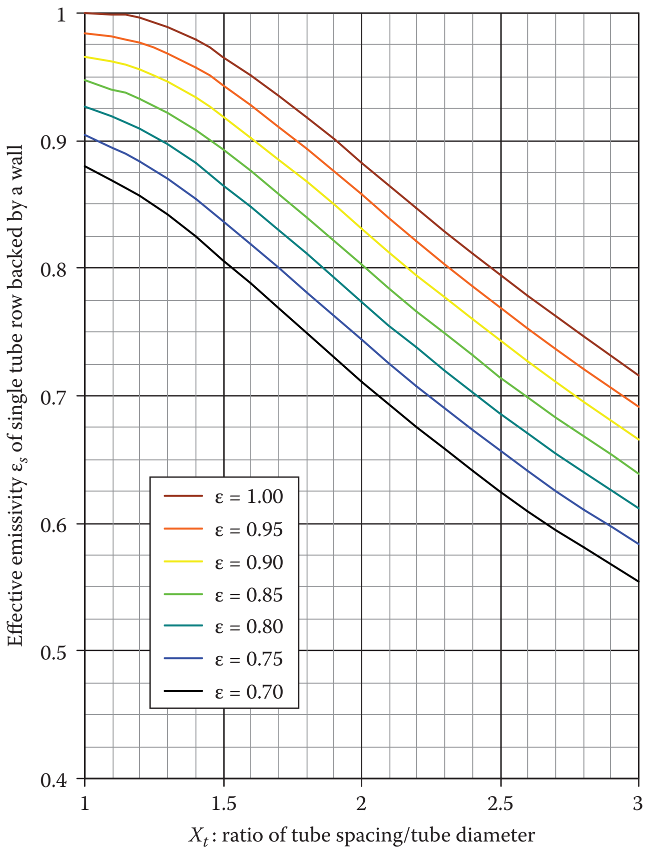


FIGURE 6.21
Effective emissivity of a single-tube row backed by a refractory wall, plotted for various tube emissivities.

A heat balance over the radiant section is given by

$$\dot{H}_F - \dot{Q}_R = \dot{m}C_p(T_g - T_0) \qquad (6.11)$$

where

T_0 is a base temperature, for example, 60°F (520°R absolute)
$\dot{H}_F$ is the enthalpy of the flue gas at the radiant temperature
$\dot{m}$ is the mass flow rate of the flue gas through the radiant chamber
C_p is the average heat capacity of the flue gas over the range T_0–T_g

Using the same average, C_p,T_{AF}, which is a pseudoadiabatic flame temperature, can be calculated as follows:

$$T_{AF} = T_0 + \frac{H_F}{\dot{m}C_p} \qquad (6.12)$$

Combining Equations 6.10 and 6.11 in order to eliminate T_g yields the following normalized equation:

$$Q'D' + \tau^4 = (1 - Q')^4 \qquad (6.13)$$

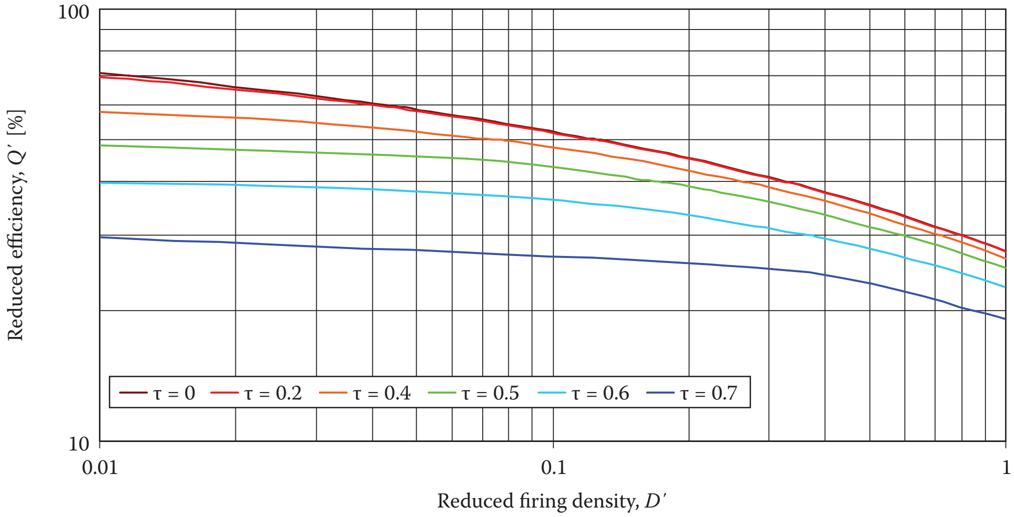


FIGURE 6.22
Relationship of reduced firing density and reduced efficiency.

with Q' the reduced furnace efficiency:

$$Q' = \frac{\dot{Q}}{\dot{H}_F}\frac{T_{AF}-T_0}{T_{AF}} \tag{6.14}$$

D' the reduced firing density:

$$D' = \frac{\dot{H}_F}{(GS)_R\sigma T_{AF}^3(T_{AF}-T_0)} \tag{6.15}$$

and τ the reduced sink temperature:

$$\tau = \frac{T_s}{T_{AF}} \tag{6.16}$$

According to Equation 6.13, the furnace efficiency is only a function of firing density and heat sink temperature. The firing density accounts for fuel composition, excess air, and all other parameters that affect adiabatic flame temperature and flue gas emissivity. The relationship between Q' and D' is plotted in Figure 6.22.

The furnace gas temperature is now given by

$$T_g = T_{AF}(1-Q') \tag{6.17}$$

The furnace efficiency is the ratio of the flue gas enthalpy and the radiant duty, so using Equation 6.14,

$$\eta = \frac{\dot{Q}}{\dot{H}_F} = \frac{T_{AF}}{T_{AF}-T_0}Q' \tag{6.18}$$

6.4.2.3 Example of a Well-Stirred Firebox Calculation

Calculate the furnace gas temperature and radiant efficiency of an oil-fired process heater with a cylindrical combustion chamber 10 ft (3 m) in diameter and 20 ft (6 m) high. There are 30 tubes of 5 in. (13 cm) nominal diameter and 18 ft (5.5 m) length, spaced at 11 in. (28 cm) center–center intervals in a single row in front of the cylindrical walls. The tube emissivity is 0.90 and the surface temperature is 600°F (320°C).

The burner liberation rate with fuel oil is 15 MMBtu/h (4.4 MW) with 25% excess air and 0.3 lb of steam per lb of fuel oil. The fuel oil has a lower heating value of 17,400 Btu/lb (40,500 kJ/kg) and the ambient combustion air is preheated from 60°F to 400°F (16°C to 200°C). Fuel oil composition is 87 wt% C, 13 wt% H.

6.4.2.3.1 Combustion Calculations

The required fuel oil flow rate is $15 \times 10^6/17{,}400 = 862$ lb/h. The stoichiometric air requirement is 14.46 lb air per lb of fuel oil. Including the excess air, the total airflow rate becomes 18.08 lb/lb fuel oil, for a total of 15,588 lb/h of air. Including the atomization steam, the oil produces 16,708 lb/h flue gas. The flue gas composition (based on dry combustion air) is as follows:

Flue Gas (vol%)	Wet (%)
N_2	73.3
O_2	3.9
CO_2	10.7
H_2O	12.05

6.4.2.3.2 Radiant Chamber Calculation

The surface area of the enclosure is $A_T = \pi \times 10 \times 20 + 2\pi \times 5^2 = 785.4$ ft².

Using the effective heat sink area gives $C_s = 10 \cdot \pi \cdot 18/785 = 0.72$.

A 5 in. nominal diameter pipe has an outside diameter of 5.563 in. The ratio of tube spacing and outside diameter X_t = 11/5.563 = 2. Using X_t in Figure 6.21 or Equations 6.7 and 6.8 for the effective emissivity of the heat sink gives $\varepsilon_s = 0.83$.

From the analysis of the fuel, steam quantity, and excess air, the partial pressures of CO_2 and H_2O in the furnace gas are 0.106 and 0.107, respectively.

The mean beam length of a totally enclosed gas volume can be calculated as 3.5 times the hydraulic radius or $L_m = 3.5V/A_T = 3.5 \cdot \pi \cdot 5^2 \cdot 20/785.4 = 7.0$ ft, with V as the volume of the enclosure.

Using a guessed gas temperature of 1800°F to find $\varepsilon_g = 0.\ 25$.

$$\text{Total Exchange Area } (GS)_R = \frac{785}{(1/0.72\cdot 0.83)+(1/0.25)-1}$$
$$= 168\,\text{ft}^2$$

Using a base temperature T_0 = 520R and a mean air-specific heat of 0.24 Btu $\text{lb}^{-1}{}^\circ\text{R}^{-1}$, the input enthalpy is

$$H_f = 862\times 17{,}400 + 15{,}588\times 0.24\times(400-60)$$
$$= 16.27\ \text{MMBtu/h.}$$

Assuming a mean furnace gas-specific heat C_p = 0.31 Btu $\text{lb}^{-1}{}^\circ\text{R}^{-1}$, the adiabatic flame temperature is

$$T_{AF} = 520 + \frac{16.27\cdot 10^6}{16{,}708\cdot 0.31} = 3{,}661^\circ\text{R}$$

The reduced firing density is now

$$D' = \frac{16.27\cdot 10^6}{168\cdot 0.173\cdot 10^{-8} 3661^3 (3661-520)} = 0.363$$

The reduced sink temperature is

$$\tau = \frac{1060}{3661} = 0.29$$

Now use Figure 6.22 or solve Equation 6.13 to find that Q' = 0.383.

The firebox gas temperature is then

$$T_g = 3661\ (1-0.383) = 2259^\circ\text{R} = 1800^\circ\text{F}$$

and the radiant efficiency is

$$\eta = \frac{3661}{3661-520}\cdot 0.383 = 0.446$$

If the calculated firebox temperature is different from the assumed firebox temperature, it should be used to reestimate the firebox gas emissivity and repeat the subsequent steps.

6.4.3 Refinements of the Lobo–Evans Method

Over the years, the well-stirred firebox method has been refined with more realistic assumptions. The following section will highlight some of these refinements.

6.4.3.1 *Effect of Nongray Gas Properties*

The assumption of a gray gas oversimplifies the radiative properties of the furnace gas. For example, in a gray gas, the emissivity ε is equal to gas absorptivity α. This actually only applies to systems in thermal equilibrium, which means that the tube wall temperature is equal to the radiating gas temperature. The simplest approach to a nongray problem is to model the gas by one clear (nonparticipating) and one gray gas. The gray gas occupies the energy fraction, a, of the blackbody spectrum and the clear gas occupies $(1-a)$. It can be calculated by evaluating the gas emissivity at the mean beam length L_m and at twice the mean beam length:

$$a = \frac{\varepsilon_g(L_m)}{2-\varepsilon_g(2L_m)/\varepsilon_g(L_m)} \tag{6.19}$$

Instead of using Equation 6.7, the total exchange area is now calculated by

$$(GS)_R = \frac{aA_T}{(1/C_s\varepsilon_s)+(a/\varepsilon_g)-1} \tag{6.20}$$

6.4.3.2 *Convective Heat Transfer*

The dominant mode of heat transfer in the firebox is by radiation, but the role of convection is not negligible. Equation 6.10 can be modified to include a convective heat transfer coefficient:

$$\dot{Q}_{RC} = \sigma(GS)_R\left(T_g{}^4 - T_s{}^4\right) + h_s A_s (T_g - T_s) \tag{6.21}$$

Since convection is a small part of the total heat transfer rate, the two terms can be combined as follows:

$$\dot{Q}_{RC} = \sigma\left[(GS)_R + \frac{h_s A_s}{4\sigma T_{gs}^3}\right]\left(T_g{}^4 - T_s{}^4\right)$$
$$= \sigma(GS)_{RC}\left(T_g{}^4 - T_s{}^4\right) \tag{6.22}$$

where

T_{gs} is the arithmetic mean of T_g and T_s

A_s is the total surface area of the radiant tubes (not the effective heat sink area)

The convective heat transfer coefficient h_s is typically set to a value of 7 Btu/(h · ft² · °F).

6.4.3.3 *Imperfectly Stirred Firebox*

One of the main assumptions of the well-stirred firebox is that the exit temperature is equal to the mean firebox temperature. In almost all practical cases, this is not the case, as the exit temperature is typically well below the mean radiating temperature by an amount Δ. In order to account for this temperature correction, Equation 6.11 is modified as follows:

$$\dot{H}_F - \dot{Q}_R = \dot{m}C_p(T_g - \Delta - T_0) \tag{6.23}$$

The dimensionless relationship between Q' and D' can then be rewritten as

$$\dot{Q}'D' + \tau^4 = (1 + \Delta' - \dot{Q}')^4 \tag{6.24}$$

with $\Delta' = \Delta/T_{AF}$.

As a first estimate, $\Delta' = \dot{Q}'/4$ has been shown to give satisfactory results for practical applications.

6.4.3.4 Wall Losses

In many high-temperature furnaces, wall losses cannot be ignored. The wall loss through the refractory is approximated by

$$\dot{Q}_{\text{loss},R} = UA_R(T_g - T_0) \tag{6.25}$$

with U being the overall coefficient for the heat transfer of the furnace gas to the outside:

$$U = \frac{1}{(1/h_{c,\,\text{inside}}) + \sum W_i/k_i + (1/h_{c,\,\text{outside}})} \tag{6.26}$$

where

W_i is the thickness of the refractory layer

k_i is the thermal conductivity of that refractory layer

h_c is the respective heat transfer coefficients on the inside and outside

6.4.3.5 Generalized Firebox Model

Combining Equation 6.24 with these losses and accounting for convective heat transfer and temperature gradients lead to the following generalized furnace model:

$$\dot{Q}'D' + \tau^4 = (1 + \Delta' - \dot{Q}')^4 + L'(1 + \Delta' - \dot{Q}' - T_0') \tag{6.27}$$

with L' called the wall loss group and T_0' the normalized base temperature:

$$L' = \frac{UA_R}{T_{AF}^3\sigma(GS)_{RC}} \tag{6.28}$$

$$T_0' = \frac{T_0}{T_{AF}} \tag{6.29}$$

References

1. Newnham, R., *Direct Fired Heaters: A Practical Guide to their Design and Operation*, Kingsley Knowledge Publishing, Alberta, Canada, 2012.
2. ANSI/API Standard 560, *Fired Heaters for General Refinery Service*, 4th edn., August 2007 [also ISO 13705:2006 (Identical)], *Petroleum, Petrochemical and Natural Gas Industries—Fired Heaters for General Refinery Service*, Section 4.3, pp. 12–13.
3. McAdams, W.H., *Heat Transmission*, 3rd edn, McGraw-Hill, New York, p. 69, 1954.
4. ANSI/API Standard 560, *Fired Heaters for General Refinery Service*, 4th edn., August 2007 [also ISO 13705:2006 (Identical)], *Petroleum, Petrochemical and Natural Gas Industries—Fired Heaters for General Refinery Service*, Section 14, p. 42.
5. McAdams, J.D. and Waibel, R., Methanol furnace combustion troubleshooting, *Proceedings of International Methanol Technology and Operators Forum (IMTOF)*, Edinburgh, Scotland, June 2007.
6. Sundaram, K.M. and Albano, J.V., Firebox modeling of SRT cracking heaters, Article 22d, *Proceedings of 6th Annual Ethylene Producers' Conference*, Atlanta, GA, April 17–24, 1994.
7. Schmidt, G., Combustion system modeling of a cracking furnace to reduce NOx emissions, Article 136b, *22nd Annual Ethylene Producers' Conference*, San Antonio, TX, April 21–25, 2010.
8. Just, R., Flame rollover and other flame shape problems, Article 63b, *16th Annual Ethylene Producers' Conference*, New Orleans, LA, April 25–29, 2004.
9. Tang, Q., Denison, M., Adams, B., and Brown, D., Towards comprehensive computational fluid dynamics modeling of pyrolysis furnaces with next generation low-NOx burners using finite-rate chemistry, *Proceedings of the Combustion Institute*, 32, 2649–2657, 2009.
10. Platvoet, E., Integrating gas turbines with cracking heaters—Impact on emissions and energy efficiency, Article 153a, *24th Annual Ethylene Producers' Conference*, Houston, TX, April 1–5, 2012.
11. Perry, R.H. and Chilton, C.H., *Heat Transmission by Radiation, Chemical Engineers Handbook*, 6th edn, McGraw-Hill, New York, Section 10, pp. 66–68, 1984.
12. Khalil, E.E. and Truelove, J.S., *Calculation of Heat Transfer in a Large Gas Fired Furnace*, AERE–R8747 (HTFS-RS225), 1977.
13. Lobo, W.E. and Evans, J.E., Heat transfer in the radiant section of petroleum heaters, *Trans. AIChE.*, 35, 743–778, 1939.
14. Hottel, H.C. and Sarofim, A.F., *Radiative Transfer*, McGraw-Hill, New York, 1967.
15. Hottel, H.C., First estimates of industrial furnace performance—The one-gas-zone model re-examined, in *Heat Transfer in Flames*, Afgan, N.H. and Beer, J.M. (eds.), Scripta Book Company, 1974.

7

Air Heaters

Carl A. Connally, Lothar Schulz, and Timothy Webster

CONTENTS

7.1 Introduction

Typical air heater applications in the refining and chemical process industries are start-up heaters for fluid catalytic cracking (FCC) units and sulfuric acid and nitric acid plants or to supply heated air for dryers, kilns, fluidized bed reactors, or fluidized bed dryers.

The air heaters can be direct fired or indirect fired designs. A direct fired unit mixes the flue gases from the burner directly into the stream to be heated. An indirect fired system heats air or, more typically, a recycled flue gas, to provide the hot-side flue gas for a gas-to-gas heat exchanger, in which the cold side is the gas (air, nitrogen, etc.) to be heated.

As implied earlier, the term "air heater" can refer to systems other than those specifically for air. Flue gas reheaters would typically use the same design as an air heater, and since the flue gas would normally be mostly inert, the burner would use a separate air source. Some of the air heater systems for dryer applications would be heating the dryer flue gas, either directly or indirectly to provide heat for the process.

While many air heaters operate at atmospheric or near-atmospheric pressure, they can also be designed for high-pressure systems. The start-up air heaters for FCC systems normally operate in the range of 1.5–3.5 barg (21.8–50.8 psig), and other processes, such as gasifiers, require operating pressures in excess of 7.0 barg (101.5 psig).

7.2 Direct Fired Systems

A typical fuel gas-fired air heater design is illustrated in Figure 7.1. The incoming air is split internally by a baffle system with a portion of the air (~50% excess air) routed through the burner and the bulk of the air passing around the burner. Combustion is initiated in the burner chamber, but the flame fills a large portion of the quench chamber. The combustion products and the quench air mix together in the latter part of the quench chamber to achieve the required outlet temperature. The air split is fixed and generally cannot be adjusted. The outlet temperature is controlled by adjusting the burner heat release.

Figure 7.2 illustrates a horizontal design with a side or top outlet. This is typical of designs for dryer applications as well as start-up air heaters for FCC systems. Since catalyst fines can collect in the air heater, a cleaning nozzle is provided on the vessel. Pressurized combustion poses no special problems, but the vessels are designed and usually stamped according to Section VIII of the ASME pressure vessel code. Figure 7.3 illustrates a design with separate combustion air and quench air inlets. This provides more flexibility in turndown, since the combustion air can be separately controlled to match the fuel gas flow. This design is also necessary when combustion air control is required to meet specific low NOx (see Volume 1, Chapter 15) or CO (see Volume 1, Chapter 14) requirements, as well as when the quench medium is not air, but a low oxygen flue gas or other "inert" stream.

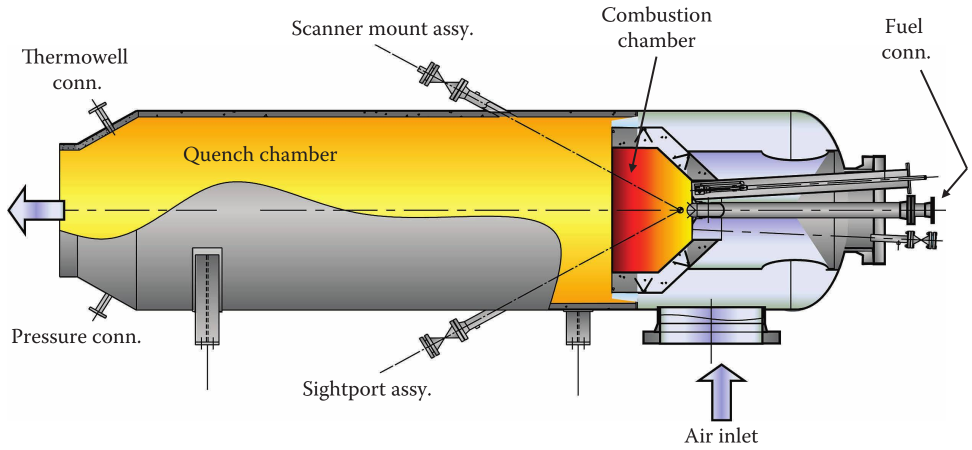


FIGURE 7.1
Direct-fired air heater.

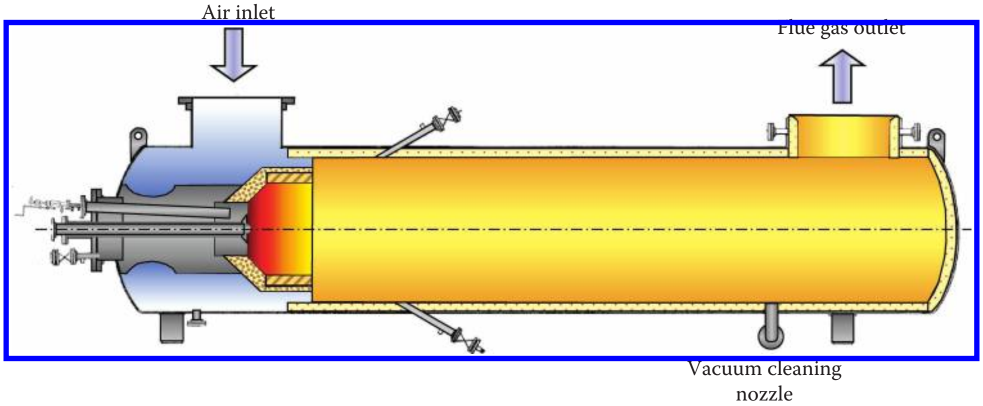


FIGURE 7.2
Horizontal direct-fired air heater with a side outlet.

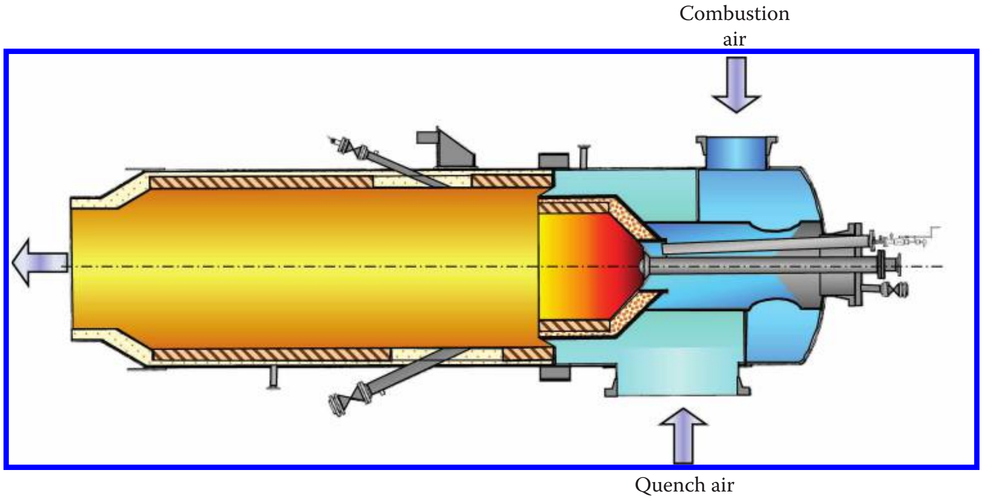


FIGURE 7.3
Direct-fired air heater with a separate combustion air inlet.

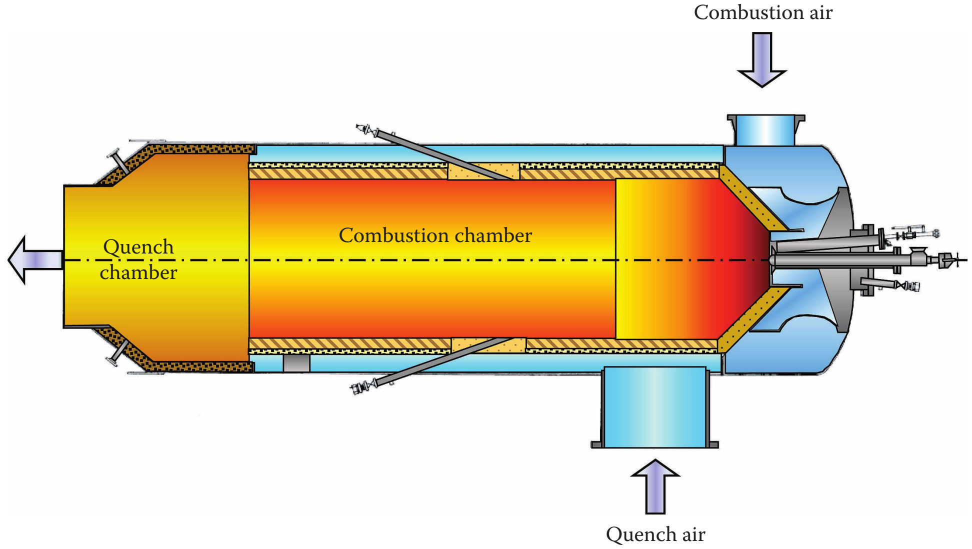


FIGURE 7.4
Oil-fired air heater.

A fuel oil-fired air heater is illustrated in Figure 7.4. A separate combustion air inlet is required, since oil firing requires a more closely controlled fuel-to-air ratio to maintain combustion stability and to minimize unburned hydrocarbon emissions and smoke formation. Secondly, a longer combustion chamber is required to prevent quenching of the flame before the combustion is complete.

Figure 7.5 shows the design for an all-metal air heater. These air heaters are used, where refractory spalling will cause a problem in downstream equipment, such as an expander or turbine or contaminate a product, such as in a drying application. The air heater is designed with an extended combustion chamber in order to shield the outer shell from the flame, until part of the quench gas can mix with the flame and reduce the temperature. The combustion chamber is usually constructed of 310 stainless steel, alloy 800, or some similar high-temperature metal alloy. The combustion chamber is also ported to allow the gradual mixing of the quench stream with the combustion products. The maximum design outlet temperatures with this design would be limited by available metal alloys to around 650°C–760°C (1200°F–1400°F).

The most common application for pressurized air heaters is for start-up of fluidized catalytic cracking systems in refineries. These air heaters usually operate in a pressure range of 1.5–3.5 barg (21.8–50.8 psig) and operate for only a few days, long enough to slowly heat up the system to the operating temperature, where the cracking reactions can begin. Once the system is online, the air heater burner is shut down, but the system

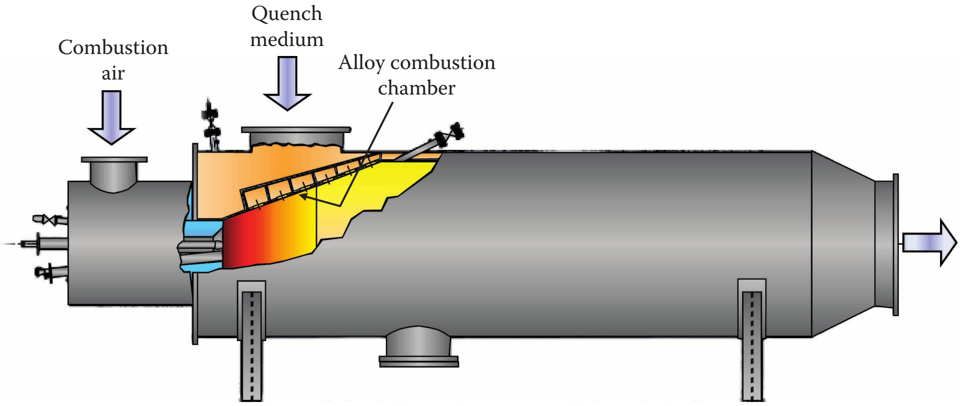


FIGURE 7.5
All-metal air heater.

air is still circulated through the air heater. The system will operate this way for months or ideally for years, without the need to restart the burner. The vessel and burner are generally horizontal units or vertically up fired, but a few have been down fired. For accessibility and ease of maintenance, the horizontal orientation is preferred. For trouble-free operation, the down-fired position is best. Vertical up firing exposes the burner and pilot to accumulation of catalyst debris and condensation, which can make relighting the burner problematic after an emergency shutdown. Even in the horizontal layout, catalyst debris and buildup can complicate relighting of the burner.

For some air heaters operating in continuous service, NOx emissions (see Volume 1, Chapter 15) may need to be minimized. Since air heaters operate with high excess air levels or quench medium flows to meet their design outlet temperatures, the NOx emissions from the burner can be driven to extremely low levels (less than 10 ppm firing natural gas or similar fuels) with suitable low NOx or ultralow NOx burners and proper design and control of quench air. To achieve lower NOx levels from the burner requires adequate mass flow through the burner to lower the adiabatic flame temperature. Firing natural gas reaching single-digit NOx levels typically equates to between 50% and 60% excess air. Fuels with higher adiabatic flame temperatures (see Volume 1, Chapters 3 and 4), such as propane, will require higher excess air levels.

For systems that use recycled flue gases or other "hot" gas streams for quench air, additional mass flow will be required to achieve the same target flame temperature. In these cases, the burner may operate with 15%–20% excess air and then have a portion of the quench flow mixed into the combustion air to achieve the required mass flow through the burner. This scheme requires more complex controls since both the air and the quench medium must be controlled to ensure that the flow through the burner has adequate oxygen levels for combustion and also remains in the mass flow range required. If the mass flow through the burner is too low, higher NOx emissions will result. If the mass flow through the burner is too high, the burner flame will be subject to excessive quenching and may generate high levels of CO or be extinguished.

Since the excess air levels, or quench airflows, required to achieve the desired outlet temperature are typically much higher than those required through the burner, provisions must be taken to ensure that they are not injected in such a manner that they will quench the burner flame. This can be achieved in different ways depending on the size and layout of the air heater. A separate quench air inlet can be located downstream of the burner primary combustion zone. This location is based on the estimated flame length and usually allows for the first 80% of the flame length to be protected from quenching.

For applications where all of the quench airflow is to be introduced at the same end of the air heater as the burner, it is common to install a "flame tube" inside the heater. The burner flame is contained in the flame tube, with quench airflow in the annular space between the flame tube exterior and the air heater interior to provide cooling of the tube. Sizing of the flame tube must allow an adequate diameter to contain the burner flame and avoid impingement on the tube walls and adequate length to prevent flame quenching. If the interior of the flame tube is not refractory lined, then it will typically be designed to allow a small amount of quench air to flow into the base of the tube and stream down the walls to provide internal cooling.

7.2.1 Air Heater Features

Air heater equipment can be arranged to operate in any position. While a horizontal position is most common and in most cases provides easier access to the burner and attached equipment, air heaters can be arranged to fire vertically up or vertically down. The firing arrangement does not drive the basic design or size.

The quench section of the air heater is usually lined with an insulating castable refractory (see Volume 2, Chapter 5) of moderate strength. Typically 100 mm (3.9 in.) will provide enough insulating effect to reduce the shell temperature to less than 150°C (300°F). This represents a heat loss of less than 3% of the total release. A personnel shield can be used around accessible areas. If the air heater is in continuous use, then a thicker layer of refractory may be warranted. If desired, insulating fire brick can be used in place of the insulating castable material.

The combustion chamber is typically lined with 150 mm (5.9 in.) of a heavy duty castable refractory, suitable for 1650+°C (3000+°F). It will be cooled by the quench air flowing around it and the casing, which is usually carbon steel, and will normally operate at less than 400°C (750°F). In systems that operate continuously or for the oil-fired designs, involving longer combustion chambers, it may be desirable to use a backup insulating layer to reduce the chamber shell temperature for longer life. As with the quench chamber, fire brick and insulating brick can be used in place of castable material.

Air heaters are provided with sight port assemblies, suitable for the operating pressure, and provided with an isolating valve, so that the glass (either Pyrex or quartz) can be isolated from the vessel pressure and cleaned or replaced if it becomes damaged. Sight ports can be used to view and monitor the burner flame to detect any early signs of deterioration to the flame quality, to burner parts, or to the refractory. Similarly, designed flame scanner nozzles with isolating valves are also provided so that faulty flame scanners can be replaced while the system is in operation.

Air heaters for pressurized operation are typically designed to some pressure above the normal operating pressure that would provide some safety factor, based on worst case upset conditions or maximum capability of the system. In some cases, the vessel may be designed to contain a detonation. In which case, a vessel design pressure is usually set at eight times the maximum operating pressure, but the shell thickness calculations are based on 90% or 100% of the material yield strength, rather than the ASME allowable stress values. For purposes of an ASME stamp or hydrostatic test pressure, the equivalent ASME design pressure can be derived, using the ASME allowable stress values for the shell material and the shell thickness determined in the detonation pressure calculation.

In general, the size of the air heater is determined by the vessel operating pressure and the air pressure drop available to the air heater. The flame is compressible and subject to the same $PV = nRT$ relationship as any compressible fluid. The correlations for flame dimensions, based on burner air pressure drop, are modified by the operating pressure of the air heater to determine the necessary internal length and width of the air heater. The typical air pressure drop for pressurized operation is from 0.035 to 0.07 bar (0.5–1 psi). The necessary fuel gas pressure drop for the burner is in the range of 0.5–2.0 bar (7.3–29 psi). Most pressurized applications are in the range from 1.0 to 7.0 barg (14.5–101.5 psig) operating pressure. While an air heater can be designed and constructed of any size, those designed for catalytic cracking systems are bracketed by a heat release range of 3.5 MW (12×10^6 Btu/h) with an approximate size of 1.0 m (3 ft) OD × 4.0 m (13 ft) long up to 120 MW (410×10^6 Btu/h) with an approximate size of 3.5 m (11 ft) OD × 13.0 m (43 ft) long. The smaller air heaters (<30 MW $= 100 \times 10^6$ Btu/h) would normally require a separate manway for access to the interior of the vessel, while access can usually be gained through the burner throat on the larger designs.

A problem common for many air heater installations is pilot light off. In many cases, the pilot is only used a few times a year and may not be properly maintained.

Figure 7.6 illustrates an igniter design that is common to many low-pressure air heaters, in which the pilot is ignited by a retractable electric igniter. If the igniter is not retracted, the ignition element can become overheated, sag, and ground out against the side of its guide tube. In extreme cases, the ignition element can burn off. Another common problem stems from the buildup of condensate in the guide tube or on the ignition rod itself; this will cause grounding or premature arcing. If condensate is a problem, the guide tube must be air purged or the igniter should be completely removed from the vessel when not in use.

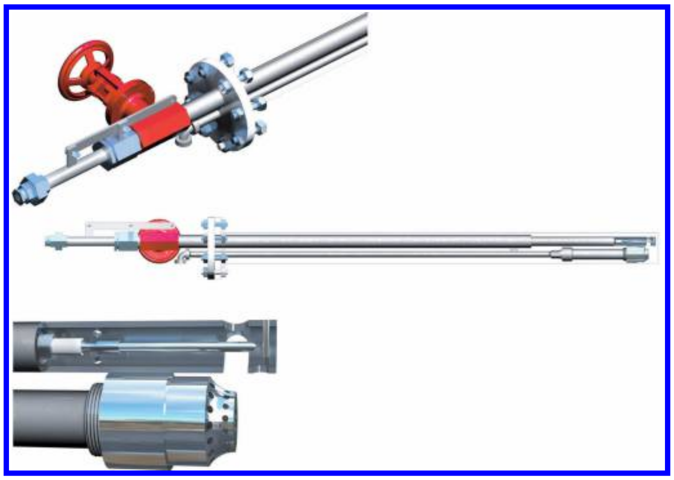


FIGURE 7.6
AH 1 1/2 pilot with retractable igniter.

Another style of igniter is shown in Figure 7.7 and is termed the "high-energy igniter." This igniter operates on the same principle as the electronic ignition in an automobile; it stores a charge in a capacitor and then releases the energy all at once across a hermetically sealed spark gap. After passing through the spark gap, the charge travels to the igniter tip where it is discharged. Catalyst fines or carbon deposits fouling the tip will not disrupt the "spark" and neither will condensate. In fact, the tip can be immersed in water and still emit a spark. This igniter is not affected by condensate, but it does have its own problems. It is susceptible to heat and must be retracted after igniting the pilot to prevent overheating and breakdown of the insulation around the tip. Due to the high charge being carried, any minor breakdown in the insulator or terminal connections will cause loss of spark. Thirdly, the tip position in relation to the pilot is critical, and being too close or too far away from the pilot tip will prevent ignition.

Over the years, the forced draft style pilot using a high-energy igniter has proved to be the most reliable design for both ambient pressure systems and pressurized systems. Typically, instrument air is used for the

FIGURE 7.7
Retractable high-energy igniter.

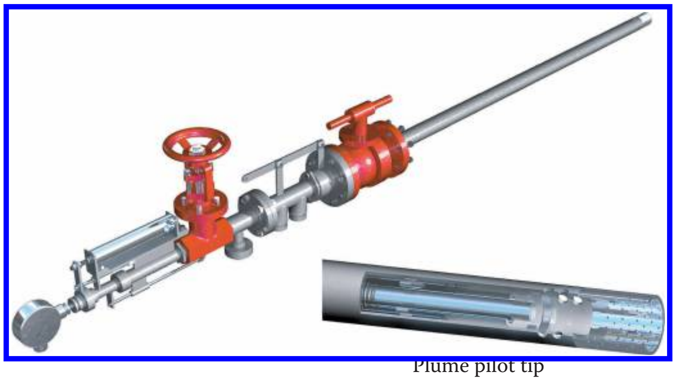


FIGURE 7.8
Plume pilot.

pilot air supply. Pressure regulators for the pilot fuel and air are used to ensure that these utilities are delivered to the pilot from 0.7 to 2.0 bar (10–29 psi) above the vessel operating pressure. If the vessel operating pressure is variable, then differential pressure regulators should be used to maintain the proper air and gas pressures under all conditions. The igniter can be retractable or fixed and the pilot can be retractable. Figure 7.8 illustrates a typical design of this type of pilot.

A "splitter damper" is sometimes used on air heaters that have a single inlet. It is comprised of a shaft through the inlet nozzle with a half blade attached to it. The blade is positioned over the internal baffle that splits the air between the burner and the quench zone of the air heater. The purpose of the damper is to provide some adjustment on the airflow to the burner during pilot ignition and light off of the main burner. It can be difficult to light the pilot or the burner when the design air rate is flowing. Closing the damper will decrease the air to the burner and force more air to the quench side. Once the burner has stabilized, the damper should be returned to the open position to allow the designed air split to be utilized. Figure 7.9 illustrates the configuration of the splitter damper. The damper blade is biased in the nozzle toward the burner end. Roughly 1/3 of the total air is used as combustion air, so the damper is designed to block off 1/3 of the inlet nozzle area. The damper blade is not designed for tight shutoff and there will always be considerable leakage, even when the blade is fully closed. A tab is welded in the inlet nozzle to prevent the blade from being rotated past the "closed" position. The damper should only be rotated 90° between the fully open

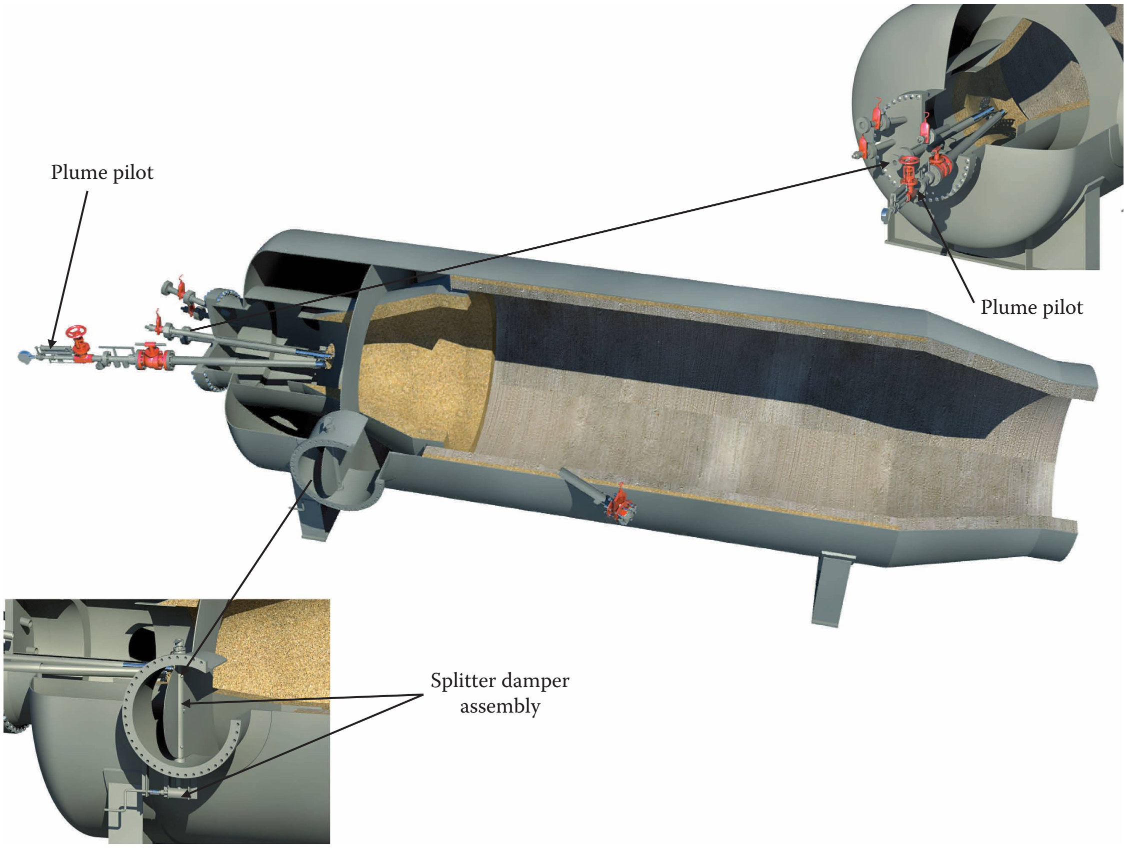


FIGURE 7.9
Splitter damper.

position and the closed position and always returned to the fully open position, when the burner has stabilized or when the burner is shut off.

7.3 Indirect Fired Systems

Indirect fired process gas heaters are mainly used in heating processes, which require drying temperatures above 200°C (392°F). Below this temperature, steam or hot oil heaters are usually more economical.

For many clients, it is mandatory that their products are in contact with clean air, process gas like N_2, or steam only. Flue gases with soot, CO, or CO_2 can damage the structure or color of the end product.

Typical applications are drying processes for

- Milk, coffee, cacao
- Chemicals, salt
- Tobacco
- Drying ovens for paint booths and automotive products

In the range between 50 kW and 6.0 MW (170,000–20 × 10^6 Btu/h), John Zink KEU supplies compact heater units for a maximum process air outlet temperature of 500°C (900°F).

Natural gas, LPG, light fuel oil (LFO), or heavy fuel oil (HFO) can be used as fuel in combustion equipment similar to boiler burners.

All heater units consist of a stainless steel combustion chamber and an integrated convection tube bundle.

Limited use of refractory material leads to short start-up times and high turndown ratios.

In the combustion chamber, the fuel will be burned with between 10% and 30% excess air, which results in flame temperatures of approximately 1400°C (2550°F).

The radiation from the flame transfers heat to the combustion chamber wall, which will be cooled by the incoming process air.

By the end of the combustion chamber, the flue gases are cooled down to 700°C–800°C (1300°F–1500°F). Below this temperature, radiation is not sufficient for further heat transfer. The flue gases then flow through the downstream convection tube bundle to complete the balance of the heating.

The efficiency depends on the fuel used and can be above 90%.

For higher heat capacities up to 20 MW (68 × 10^6 Btu/h) or temperatures up to 750°C (1400°F), John Zink KEU uses a combination of a hot gas generator with proprietary recuperators.

7.4 Air Heater Types

7.4.1 GSX Air Heaters

GSX air heaters may be delivered as an insert unit for installation in a customer's drying channel or supplied with an outer casing as stand-alone solution.

The airflow is around the combustion chamber and in cross flow over the tube bundle (see Figures 7.10 and 7.11). Cleaning of the flue gas tubes can be done after dismantling of part of the front plate.

The maximum heat capacity is approximately 1 MW (3 × 10^6 Btu/h) with the outlet temperature of the process air a maximum of 250°C (480°F). Only gas and LFO can be used.

7.4.2 VTK Heater

The process air enters on top of the convection tube bundle in countercurrent flow to the flue gases in the tubes and also in countercurrent flow in the gap formed between the combustion chamber and outer casing (see Figures 7.12 and 7.13).

A cleaning opening in front of the flue gas tubes is provided.

The maximum heat capacity is approximately 2.5 MW (8.5 × 10^6 Btu/h) with the outlet temperature of the process air a maximum of 550°C (1020°F). Only gas and LFO can be used.

7.4.3 VTN Heater

The process air enters on the burner side in tangential flow to get efficient cooling of the combustion chamber. The airflow is cocurrent in the gap formed between the combustion chamber and the outer casing. A large cleaning door is provided for easy and quick cleaning of the combustion chamber and tubes especially for HFO combustion (see Figures 7.14 and 7.15).

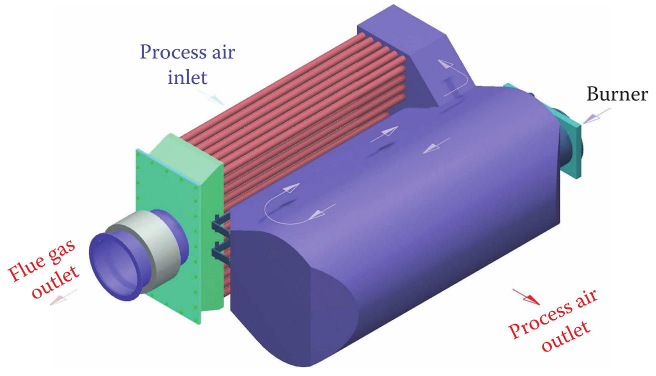


FIGURE 7.10
Combustion chamber with tube bundle.

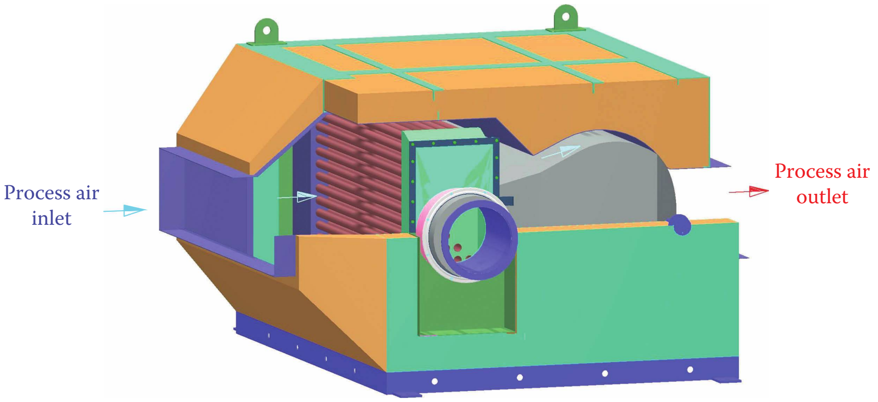


FIGURE 7.11
GSX heater with housing.

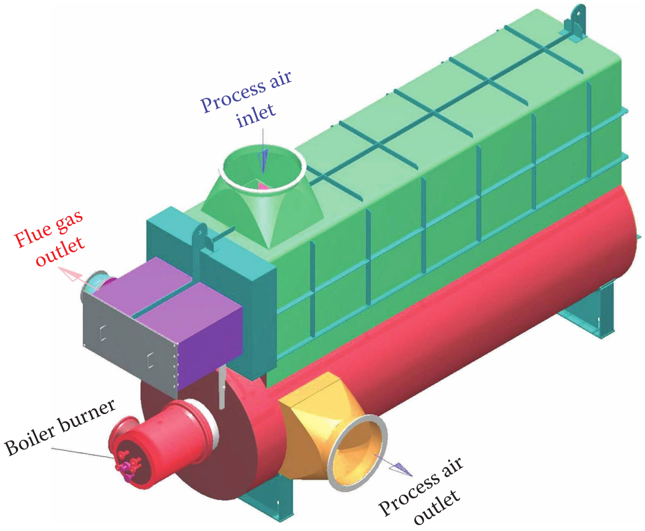


FIGURE 7.12
VTK heater with air inlet and outlet sockets.

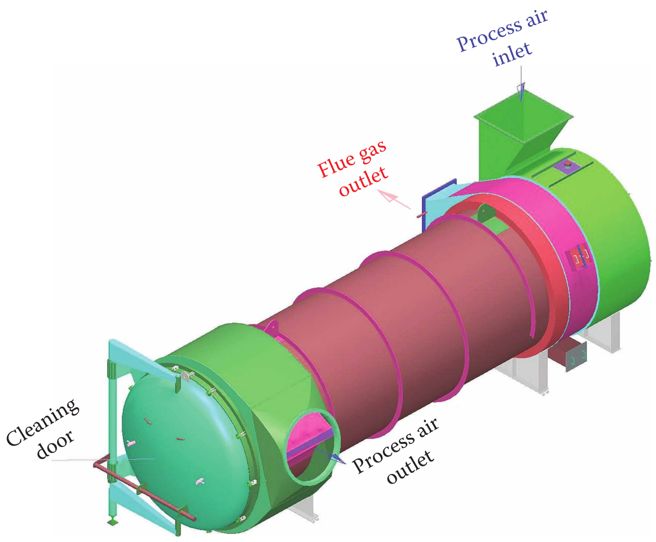


FIGURE 7.14
VTN heater, view to the cleaning door.

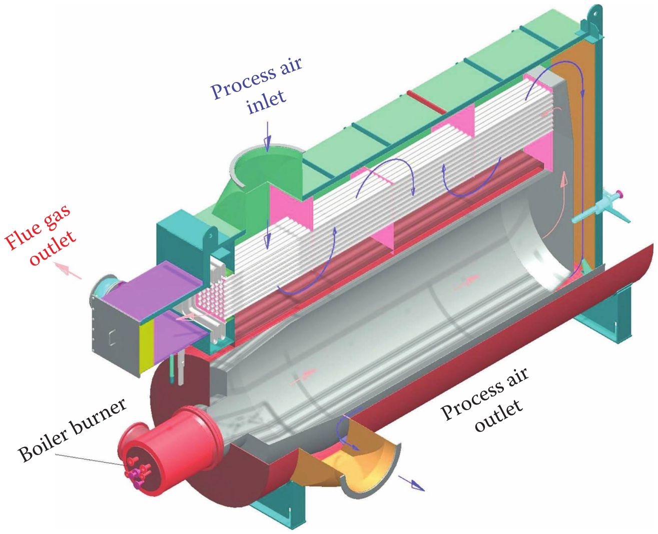


FIGURE 7.13
VTK heater in sectional view.

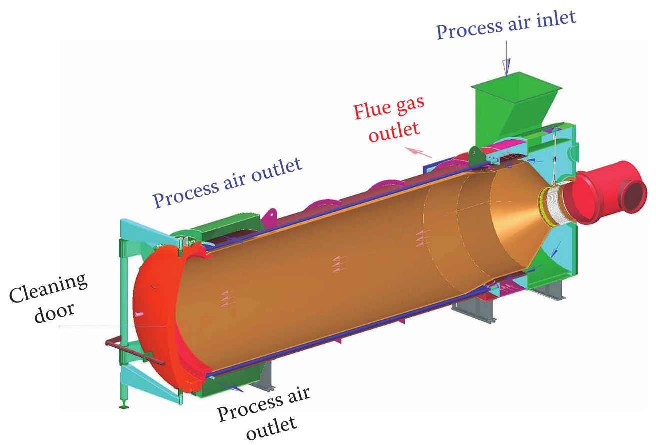


FIGURE 7.15
VTN heater in sectional view.

The maximum heat capacity is approximately 6.0 MW (20 × 10^6 Btu/h) with the outlet temperature of the process air a maximum of 500°C (900°F). Gas, LFO, and HFO can be used.

7.4.4 HG-WT Heaters

For higher heat capacities, the heaters consist of two parts, the combustion chamber and a separate downstream heat exchanger. The maximum heat capacity is approximately 25 MW (85 × 10^6 Btu/h) with the outlet temperature of the process air a maximum of 600°C (1100°F).

The flue gas will be produced in a combustion chamber with a mixture of flue gases from the burner and recirculated flue gases from behind the heat exchanger. The flue gas temperature in front of the heat exchanger is approximately 800°C (1500°F) and will be cooled down to approximately 150°C (300°F). The main part will be recycled back into the combustion chamber with the help of a recirculation fan. The rest will be sent through the stack to the atmosphere.

The incoming process air flows in the tubes in countercurrent flow to the flue gases (see Figures 7.16 and 7.17).

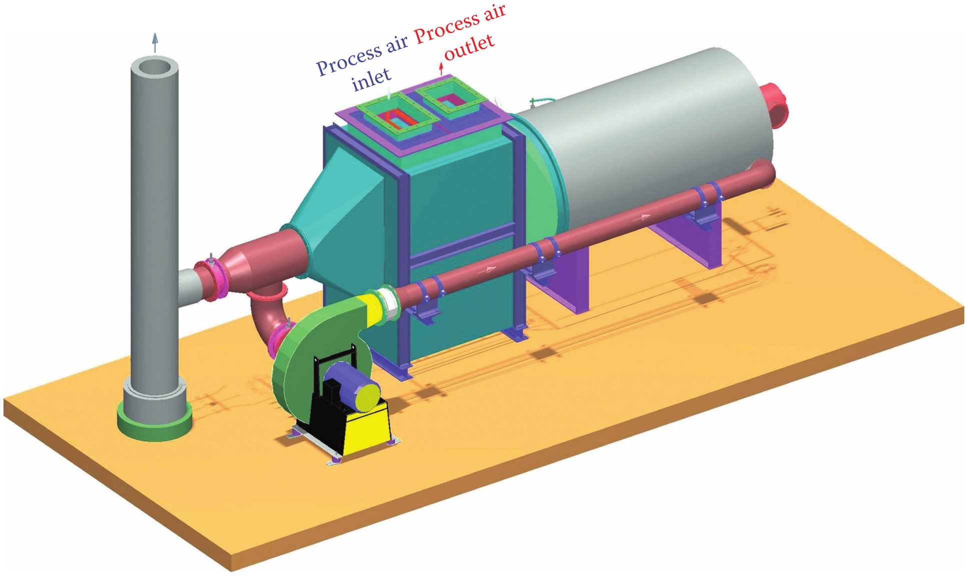


FIGURE 7.16
HG-WT heater.

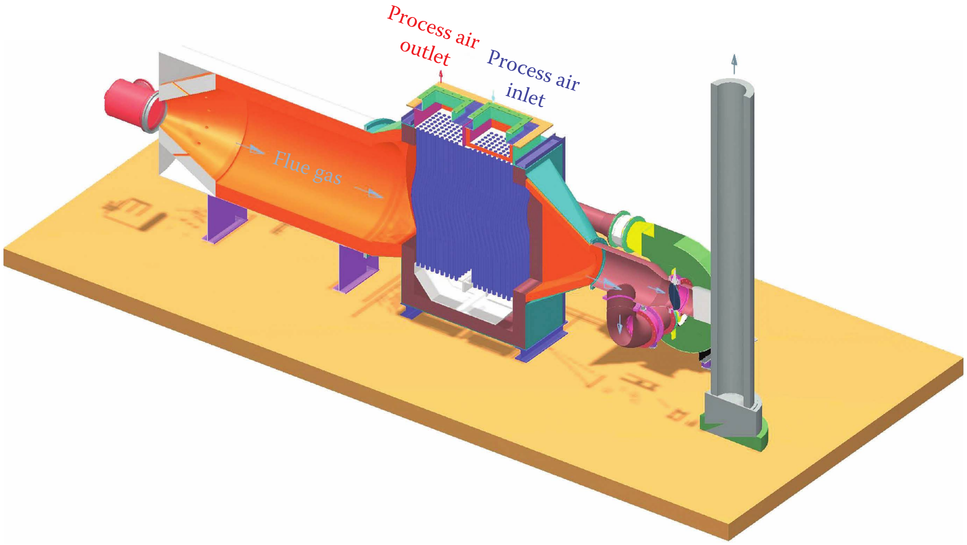


FIGURE 7.17
HG-WT heater in sectional view.

8

Thermal Oxidizer Basics

Jay Karan, Bernd Reese, Klaus-Dieter Zschorsch, and Wolfgang Klaus

CONTENTS

8.1 Introduction

Improvements are continually being made to the efficiency of the many chemical and mechanical manufacturing methods used to produce an ever-increasing number of compounds and products. However, unwanted by-products, or wastes, are produced by virtually all methods. Such by-products can exist in the vapor, liquid, or solid phase. Many are hydrocarbons although some non-hydrocarbon materials are also produced. Regardless, the many different by-products must be safely contained or destroyed to prevent potential environmental damage.

Almost all industrial sectors produce by-products, or wastes. Petroleum and natural gas, petrochemicals, pulp and paper, agricultural chemicals, pharmaceuticals, distillation, automotive, plastics molding, carbon fiber, and fiber optics production are just a few of the diverse areas that produce by-products. The by-products can be the remains of the less than 100% efficient chemical processes used to create hydrocarbon-based products. The by-products can also be impurities and catalysts present in feedstocks that are not fully consumed during the manufacturing process. In addition, manufacturing processes in widely varying fields often require significant ventilation resulting in streams of contaminated air that must be treated. By-products may vary from a few parts per million in air or water to nearly 100% concentration of a hydrocarbon.

Many methods are available to treat by-products. If the quantities are small, non-soluble solids and liquids can be put into sealed drums and isolated in secure landfills. The larger amounts of liquids, primarily contaminated water, have been injected into deep wells. For all practical purposes, vapors cannot be stored and must be treated as they are generated. Activated carbon, for instance, can be used to adsorb organic materials from gases (see Chapter 15). Stripping and absorption can also remove contaminants from liquid and gaseous streams. Filtration methods are used to remove solid materials from gases and liquids. None of these methods, however, actually destroys the waste material. Chemical and biological treatments are used to destroy organic waste, but may not be the most cost-effective choice for rapid and efficient treatment of large amounts of material. The most effective method for rapidly eliminating a high percentage of hydrocarbon contaminants is to oxidize the organic materials at an elevated temperature (at or above 1500°F/800°C). Such high-temperature oxidation is known as thermal oxidation, which is essentially combustion of the contaminants. For some contaminated airstreams, effective oxidation can also be achieved at lower temperatures using a catalyst to increase the oxidation reaction rate.

High-temperature thermal oxidation quickly and efficiently destroys hydrocarbon-based waste materials, converting the carbon and hydrogen to carbon dioxide and water vapor. However, during this process, some elements in the wastes, such as sulfur or chlorine, form compounds such as sulfur dioxide and hydrogen chloride that, if present in sufficient quantities, have to be recovered or removed from the combustion products by post-oxidation treatment, in order to meet various federal, state, and local air quality guidelines. Other elements or compounds in the wastes, such as sodium, sodium chloride, catalysts, or other inert solids, will produce particulates. If significant quantities of particulates are produced, they will have to be removed by post-oxidation treatment also. If nitrogen is molecularly bound to any organic waste compound (commonly called "bound nitrogen"), special staged oxidation methods may have to be used to prevent formation of large amounts of NOx (see Volume 1, Chapter 15). The same staged oxidation method can be used to break down existing NOx that is part of an incoming waste.

A number of factors determine the design of thermal oxidation systems. Process variables such as the waste composition and flow rate affect the size, the materials of construction, and the performance of the system. Economic considerations influence decisions regarding capital expenditure versus operating costs. A typical example of one economic consideration is determining the benefit and feasibility of installing a heat-recovery system. Regulations set the required destruction efficiencies, emission rates, and acceptable ground level concentrations of emissions. Very few applications have identical requirements or specifications. Thus, most thermal oxidation systems are custom designed to satisfy the process, economic, and regulatory requirements of a particular application.

Regardless of the specific design, most thermal oxidation systems (see Figure 8.1) consist of some or all of the following components:

- A device and method to supply oxygen to the process to initiate and sustain oxidation using burner or catalyst
- A vessel (combustion chamber/thermal oxidizer [TO]) to provide the residence time required for oxidation of the hydrocarbons
- A heat-recovery (heat exchanger or boiler) and/or flue-gas conditioning system
- Emission control equipment (filters, scrubbers, etc.) to treat the flue gas prior to discharge to the atmosphere
- An elevated exhaust point (stack) through which the flue gas can be dispersed into the atmosphere
- Control hardware and logic to automatically maintain and monitor the various process parameters to ensure safe operation

The purpose of this chapter is to provide a better understanding of the use of thermal oxidation to destroy fume and liquid wastes. To accomplish this, the following is needed:

- An explanation of the basic practical thermal oxidation and post-oxidation processes and components that can be combined into complete systems to destroy hydrocarbon wastes and treat the combustion products to achieve required emission limits
- Examples of practical complete system configurations that can be applied to treat different waste compositions and combinations

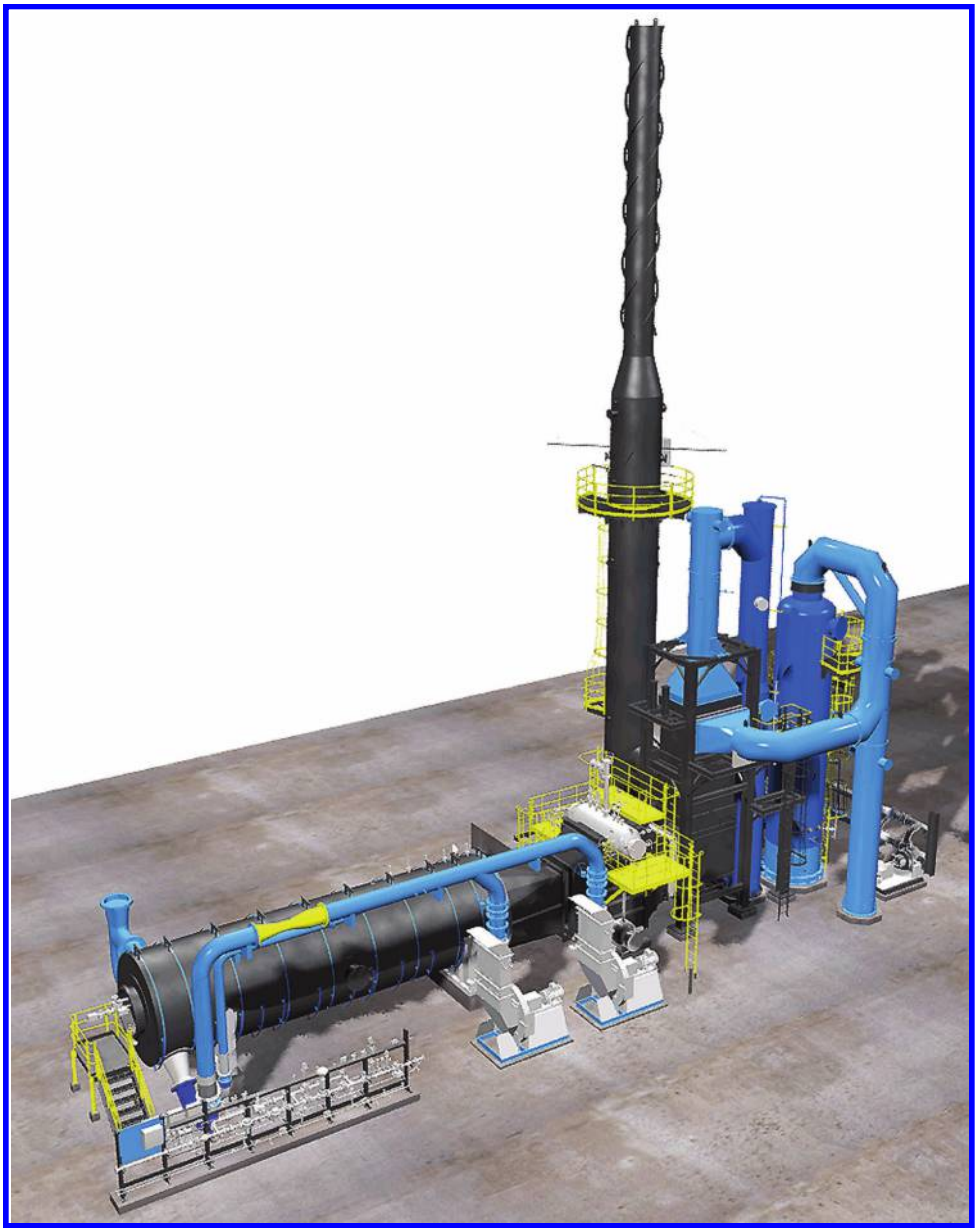


FIGURE 8.1
Example of a comprehensive thermal oxidizer system.

8.2 Combustion Basics

The prerequisites for, the actions during, and the results of combustion must be known in order to design a thermal oxidation system that will achieve the destruction and removal efficiencies (DRE) needed to protect the environment.

8.2.1 Material and Energy Balance

A certain amount of process information must be developed in order to correctly choose and design the components of a thermal oxidation system. For the burner, the designer must know the amount and properties of the fuel/waste and the amount of air required to provide stable, effective combustion. For the combustion chamber, the operating temperature, volume, and properties of the combustion products must be determined. For post-combustion processing of the combustion products, such as heat recovery or flue-gas conditioning, the mass flow and composition of the combustion products must be known. For emission control applications such as acid-gas removal or particulate removal, the amount of the pollutant must be known to predict the level of removal required. By completing material and energy balances at different equilibrium points throughout the overall process, all of this information can be generated. This information is then used to configure systems and to design and dimension the individual components.

8.2.2 Oxidizing/Reducing Combustion Processes

Oxidation reactions occur during both excess-oxygen (excess air) and sub-stoichiometric (starved air) processes. These processes are used separately and in combination to destroy hydrocarbon wastes. The high-temperature, excess O_2 process is by far the most commonly used (see Volume 1, Chapter 4). The excess O_2 process converts virtually all of the carbon and hydrogen in hydrocarbon wastes to harmless CO_2 and H_2O. To be certain that the conversion is maximized, the amount of combustion air supplied to this process is greater than the theoretical amount needed to oxidize the compounds. If sulfur compounds are present, the excess air also provides oxygen to convert the sulfur compounds to SO_2/SO_3 (SO_X) (see Volume 1, Chapter 14). Additionally, some nitrogen is converted to NO/NO_2, often referred to as NOx (see Volume 1, Chapter 15). SO_X and NOx have a detrimental effect on the environment, and stack emission levels of these compounds are regulated.

The sub-stoichiometric (reducing) process operates with less than 100% of the theoretical amount of O_2 needed to oxidize the hydrocarbon compounds. Some of the oxygen is combined with the carbon and hydrogen to form CO_2 and H_2O. However, partial reactions also occur, forming CO and H_2 as well as other products of incomplete combustion (PICs). SO_X production and NOx production are significantly reduced. Eventually, the partially oxidized products of combustion must be completely oxidized. This final oxidation is completed at temperatures significantly lower than the flame temperature, thereby minimizing NOx formation, although the remainder of the unconverted sulfur from the reducing process will be fully converted to SO_X.

8.2.3 NOx Formation

In Volume 1, Chapter 15, NOx formation is discussed at length. Basically, three mechanisms for NOx formation are present.

8.2.3.1 Thermal NOx

The first mechanism, thermal NOx, refers to how NOx is formed during high-temperature reaction of nitrogen with oxygen. Thermal NOx formation increases exponentially with increasing operating temperature. At greater than 2000°F (1100°C), it is usually the primary source of NOx if the waste does not contain "bound nitrogen" (nitrogen that is molecularly bound to the hydrocarbons).

8.2.3.2 Prompt NOx

Prompt NOx refers to NOx that is formed by hundreds of rapid reactions between nitrogen, oxygen, and hydrocarbon radicals early in the flame. Hydrocarbon radicals are produced by the decomposition of hydrocarbons and are present in large quantities at the base of the flame, but exist for a very short time before they are oxidized. For example, methane (CH_4) decomposes to the following radicals: CH_3^-, CH_2^-, and CH^-. Although these radicals exist for a very short time, they are extremely reactive and easily decompose the otherwise stable nitrogen molecule and provide chemical reaction pathways for the nitrogen to be rapidly converted to NOx. So as the name implies, prompt NOx is formed very early in the flame and is not affected by the techniques used to control thermal NOx. Prompt NOx can be a large contributor in lower-temperature combustion processes.

8.2.3.3 Fuel NOx

Fuel-bound NOx refers to the NOx formed by the excess-oxygen combustion of hydrocarbon compounds that contain molecularly bound nitrogen (e.g., NH_3). The fuel-bound NOx mechanism is very similar to the prompt NOx mechanism, in that the nitrogen present with the hydrocarbons is readily exposed to the hydrocarbon radicals because, being bound to the hydrocarbons, it is in the immediate vicinity of the radicals when the hydrocarbon begins to decompose. However, the rate of conversion of fuel-bound nitrogen to NOx is much greater than the conversion of the nitrogen (N_2) in air to NOx via the prompt NOx mechanism. For this reason, the high-temperature thermal oxidation of organic nitrogen containing compounds will contribute far more to the overall NOx formation than either of the other two mechanisms. However, as noted in the preceding section, NOx formation from fuel-bound nitrogen can be significantly reduced by utilizing an initial stage of sub-stoichiometric combustion followed by an oxidation stage that operates at a lower temperature.

8.2.4 Carbon Monoxide

Small amounts of carbon monoxide (CO) are produced in all combustion systems (see Volume 1, Chapter 14). The amount of CO produced is minimized if proper combustion design is implemented. As the products of combustion travel through the high-temperature TO chamber, the CO produced from the flame will further partially be oxidized by the excess O_2 present in the products of combustion. For typical TO designs, by the time the combustion products exit the chamber, the resulting CO is generally at levels well below 100 ppmv. If large amounts of CO are produced because of poor combustion at the burner, only part of it will be oxidized as the gases pass through the oxidizer chamber, and the outlet concentration will be much greater than 100 ppmv. Although a high level of CO in the combustion products exiting a TO is most often a result of insufficient combustion of the hydrocarbons in the waste, it can also be a good indicator of poor combustion of the burner fuel due to poor burner operation or due to quenching of the burner flame by the waste.

Some wastes contain CO as a component in the incoming waste. The recognized autoignition temperature of CO is greater than 1200°F (650°C), so systems designed to handle wastes containing CO must operate at sufficiently high temperature (higher than about 1400°F or 800°C) and have sufficient residence times (which is dependent on the vessel volume) to achieve good performance. The attempts to reduce operating cost by reducing operating temperature or attempts to reduce capital cost by using smaller vessels will result in poor CO destruction performance.

8.2.5 Acid Gases

Many waste gases and liquids contain sulfur or chlorine compounds. When these compounds are oxidized, the sulfur is converted to SO_2/SO_3 and, with sufficient moisture, the chlorine converts mostly to HCl and a small amount of free Cl_2. To meet emission limits, these compounds must be removed from the flue gas. HCl can be recovered or removed while SO_2 is normally just removed. Although sulfur and chlorine are the most common acid-producing components, compounds containing phosphorus, fluorine, and bromine are occasionally encountered. Phosphorus can cause corrosion problems in heat-recovery equipment because of high dew points. HF is very reactive and will attack virtually every part of a system until neutralized. Bromine in organic compounds is very difficult to convert to HBr that is much more easily removed than pure bromine (Br_2). Much of it converts to Br_2 in normal oxidizing conditions. A special process must be used to achieve high conversion to HBr in order to facilitate high removal efficiency.

8.2.6 Particulates

Often, wastes will contain noncombustible solids that remain after the waste material is oxidized or will contain compounds that convert to solid particles during combustion. Common examples of noncombustible solids would be salts dissolved or suspended in wastewater or caustic liquors, catalyst material carried over in the off-gas stream from a catalytic cracking unit, silica carried over from a biomass drying operation, etc. Common inorganics found in wastes are compounds of elements like sodium or silicon or pure elements like cobalt, manganese, and nickel. The examples of a few combustible compounds that produce particulates (see Volume 1, Chapter 14) during combustion are silane (SiH_4), germane (GeH_4), or phosphine (PH_3), which when oxidized produce SiO_2 (silica), GeO_2, and P_4O_{10}, respectively.

Depending on the point of introduction into the system, the particulates formed will vary in size from several microns to submicron in diameter.

8.3 Basic System Building Blocks

A simple thermal oxidation system may consist of only combustion components, that is, a burner mounted on a vertical combustion chamber with an integral stack. A complex system may include the combustion components and all possible heat-recovery and flue-gas treatment components such as boilers, hot oil heaters, waste preheaters, flue-gas conditioning equipment, acid-gas-removal and particulate-removal equipment, and stack. A catalytic system with a preheater, burner, catalytic oxidation chamber, and stack lies somewhere between a simple and a complex system. Each of the components in the system is a stand-alone process block. When necessary, the blocks are combined to build a complete thermal oxidation system. A description of each of the components in a thermal oxidation system, including comments on benefits and limitations, is provided as a guide to proper utilization of these components.

8.3.1 Burners

The burner is the component required to mix and ignite the fuel (and waste, if the waste has a high enough heating content to support a stable flame) and air and to provide a stable flame with appropriate shape and combustion characteristics throughout the design operating range of the system (see Volume 2, Chapter 6). The basic parts of a burner include a pilot to provide the initial source of ignition; the assemblies to introduce the fuel, waste, and air; and the means to ensure flame stability once lit. Burners are used over a wide range of heat releases and can burn gas and/or liquid fuels and combustible waste streams. The mixing of fuel/waste with combustion air is accomplished by the combination of air velocity through the burner (usually referred to as "pressure drop") and the velocity and distribution of the combustible material as it is introduced into the burner.

8.3.1.1 Pilots

A pilot is essentially a very small burner that provides the ignition source for the main burner fuel. Pilots normally utilize natural gas or propane gas that is mixed with air in the pilot assembly. Two methods of mixing the air and fuel are used, namely, inspirated air and compressed air. An inspirated-air pilot utilizes the momentum of the fuel in a venturi (see Volume 1, Chapter 9) to educt air into the venturi throat where it mixes with the pilot gas. For oxidation chambers that operate at a pressure that is higher than the delivery pressure of the venturi, the venturi is installed in the burner air plenum so that the supply air to the venturi is at a pressure higher than the chamber pressure. For others, the venturi can be installed outside the burner where it educts atmospheric air.

A pressurized-air pilot requires a supply of compressed air, at a pressure greater than the operating pressure of the burner. Pressurized air is metered by use of an orifice or valve into a small mixing chamber where it mixes with similarly metered pilot gas. The mixture of air and fuel then travels through a tube to a special high-temperature pilot tip where it is ignited.

There are two common methods for pilot ignition. The simplest ignition method utilizes a high-voltage (>6000 V) electric spark to ignite the air/fuel mixture at the pilot tip. This arrangement is inexpensive, but heat exposure over time can damage the spark delivery hardware and lead to ignition difficulties. A common alternative is the flame-front ignition system, in which a spark, located outside of the burner, ignites an air/fuel mixture flowing to the pilot tip through a steel pipe of about 1 in. (2.5 cm) diameter. The spark initiates a flame front that travels through the flowing mixture in the pipe finally emerging near the area of the pilot tip, igniting the pilot flame. Only the open pipe from which the flame front emerges to ignite the pilot air/fuel mixture is exposed to heat. The end of that flame-front pipe is made of stainless steel and is far less likely to be damaged and result in ignition difficulties.

8.3.1.2 Fuel Introduction

To be considered a fuel, the material must have sufficient heating value to sustain stable combustion once ignited. The material can be a gas or a liquid. In order to burn quickly and efficiently, a fuel gas need only be mixed with the appropriate amount of air, but a liquid must first be atomized into fine droplets and then mixed with air. Unlike the pilot, main burners used in TOs almost never use premixed air and fuel. Therefore, a method must be employed to quickly mix the fuel and air. The most effective method is to separate the total fuel flow, whether gas or atomized liquid, into smaller "jets" of flow by using a tip or multiple tips with orifices (ports) drilled at the proper size and orientation. This serves two purposes. First, having more individual jets provides more fuel surface area exposed to the combustion air, regardless of the jet velocity. Second, as the jets of fuel exit the tip at significant velocity, the combustion air is promptly drawn into the rapidly dispersing fuel jet. Thus, air is mixed quickly with the fuel. Once the fuel is ignited, a mechanism is required to provide flame stability, that is, continuous and relatively stationary ignition of the fuel/air mixture near the point of fuel introduction. This is commonly accomplished by establishing an airflow disturbance adjacent to some portion of the fuel discharge. The flow disturbance creates localized reduction of velocity such that the flame speed exceeds flow velocity at that point and creates recirculation of a portion of the reacting flame constituents, thus continuously igniting the incoming fuel as it mixes with air.

8.3.1.2.1 Gas Tips

For fuel gas or higher-pressure combustible waste-gas introduction, specially designed tips made of heat-resistant alloys are utilized. They are mounted on the ends of pipes, commonly referred to as risers, pokers, or lances, which are often removable through the front of the burner, so the tips can be easily accessed for maintenance and replacement. Based on the type and amount of gas to be introduced and the amount of gas pressure available at the tips, a calculated number of firing ports are drilled into each tip in an engineered pattern. Smaller ports known as ignition ports are also drilled into each tip to aid in stabilizing the flame. In addition to fuel/air mixing, a very important function of the firing and ignition ports is to direct and shape the gas discharge from the tips, thereby directing and shaping the flame. For lower heat-release burners, a single gas tip, located at the center of the burner, is often used. For higher heat-release burners, multiple gas tips, arranged symmetrically around the circumference of the burner, are used, just as multiple ports on a single tip are used, to increase the rate of mixing of the fuel and air, thereby increasing the oxidation reaction rate. The rapid oxidation of fuel is important because it must be burned before nonflammable wastes, such as contaminated water, or inert gases can be introduced into the system. The gas pressure drop through the tips is usually in the range of 10–25 psig (0.7–1.7 barg).

8.3.1.2.2 Liquid Tips

Liquid fuel and liquid waste tips serve the same purposes as gas tips, but are more complex. Mechanical atomization, requiring a 200–300 psig (14–20 barg) liquid pressure drop at the tip, can be used. However, the turndown for mechanical atomization is only about 3:1. At that point, the atomized liquid droplets become larger than preferred for optimum burning. To maintain the small droplet size over a larger operating range and reduce the amount of liquid pressure needed, a dual-fluid atomizer using an assist medium such as steam or air is required. Droplet sizes similar to that achieved in a mechanical tip at 200–300 psig (14–20 barg) can be achieved in a dual-fluid atomizer at only 60–100 psig (4–7 barg) pressure drop for both fluids. The dual-fluid atomizer presents an important advantage because of the lower liquid pressure. The lower liquid pressure allows the use of larger liquid passages through the tip, making the tip less susceptible to plugging.

8.3.1.3 Waste Introduction

Waste gases and waste liquids that are not capable of burning as a stable fuel are usually introduced downstream of the burner. However, even though a waste does not have sufficient heating value to sustain stable combustion, it can still be classified as either endothermic or exothermic, for a specific operating temperature. Waste is considered to be endothermic if the hydrocarbon content (heating value) is small and much more than a minimum amount of auxiliary fuel must be burned to maintain the required operating temperature in the combustion chamber. If the heating value is high enough that a cooling media must be added to control the maximum operating temperature, the waste is considered exothermic. In other words, if the waste contains more heat content than the energy required to raise its mass, as well as the mass of air required for oxidation of the waste hydrocarbons, to the operating temperature of the oxidizer, then it is considered exothermic. A waste can be exothermic at a lower operating temperature, but endothermic for a higher operating temperature. Waste liquids are normally available at higher pressure and can be sprayed into the system using hardware (atomizers, etc.) similar to those used for liquid fuels. Waste gases, on the other hand, are normally available only at lower pressures, so injection hardware must be designed for the lower pressure drop. Waste-gas injection "tips" often are simply open pipes. As with fuel gas, more pipes will better distribute the waste gas and mix it more rapidly with available oxygen. Endothermic wastes may not support stable combustion, but a significant amount of organic material (heating value) may still be present. If so, regardless of whether the waste is liquid or gas, the waste injection hardware may have annular spaces around them for local introduction of air to react with the organic material. The air entering through the annulus also serves to cool the injection hardware as well as providing some or all of the air required for oxidation. The positioning of the waste injection hardware is critical.

The reason for introducing wastes containing low organic content, whether liquid or gas, downstream of the burner flame zone, is to avoid impairing oxidation of the burner fuel. Impaired fuel oxidation can result in the formation of carbon monoxide, unburned hydrocarbons, or worse, soot. If the waste is air that is only slightly contaminated with organics, it may be used as the combustion air source and introduced directly through the burner's air passages. The direct injection of the waste through the burner's air passage is typically used in catalytic systems because the waste is generally slightly contaminated air.

8.3.1.4 Low-Pressure-Drop Burners

Low-pressure-drop burners are designed to operate with a very low pressure drop/air velocity through the combustion zone. Generally, the motive force that "pulls" the air through the burner is created by the buoyancy of hot flue gas in the stack relative to the cooler ambient air surrounding the base of the stack. This "pull," or "draft," created depends on stack height, flue-gas temperature, flue-gas flow rate, and stack diameter. Because the draft is a natural result of system configuration, it is designated that "natural" draft and burners that utilize this motive force are natural draft burners. Natural draft systems typically generate 0.15–1.5 in. (3.8–38 mm) water column (W.C.) negative pressure at the stack base. The majority (75%–80%) of that pressure drop is consumed as the air passes through the throat of the burner into the combustion zone. At such a low pressure drop, the velocity of the air at the burner throat can be no more than 15–50 ft/s (4.6–15 m/s). As a result, the ratio of the burner's throat area to the airflow volume is relatively large, making airflow control and distribution more difficult. Short flames are usually desired, but with little energy available in the form of combustion air velocity, usually multiple gas tips must be used for the fuel gas in a natural draft burner. In liquid-fueled natural draft burners, to achieve short flames, an increased number of ports and wider-angle port orientations are required in the atomizer tip.

Natural draft systems are impractical if any equipment, for example, boiler or scrubber, with substantial pressure drop, is required downstream of the furnace. Hence, most incinerator systems require at least one blower to move the gas through the system. Figure 8.2 illustrates a typical natural draft burner configuration.

8.3.1.5 Medium-Pressure-Drop to High-Pressure-Drop (Forced Draft) Burners

Medium-pressure-drop to high-pressure-drop burners operate at much higher pressure drops than natural draft burners because the combustion air supplied to the burner is pushed, or "forced," into the system by a blower or compressor (see Volume 2, Chapter 3). Thus, such burners are known as forced draft (FD) burners. Generally, the medium-pressure-drop burner would be designed for 1.0–8.0 in. W.C. (0.25–2.0 kPa) pressure drop. A high-pressure-drop burner may be designed for pressure drops of 16–30 in. W.C. (4–7.5 kPa) or more. With much more energy

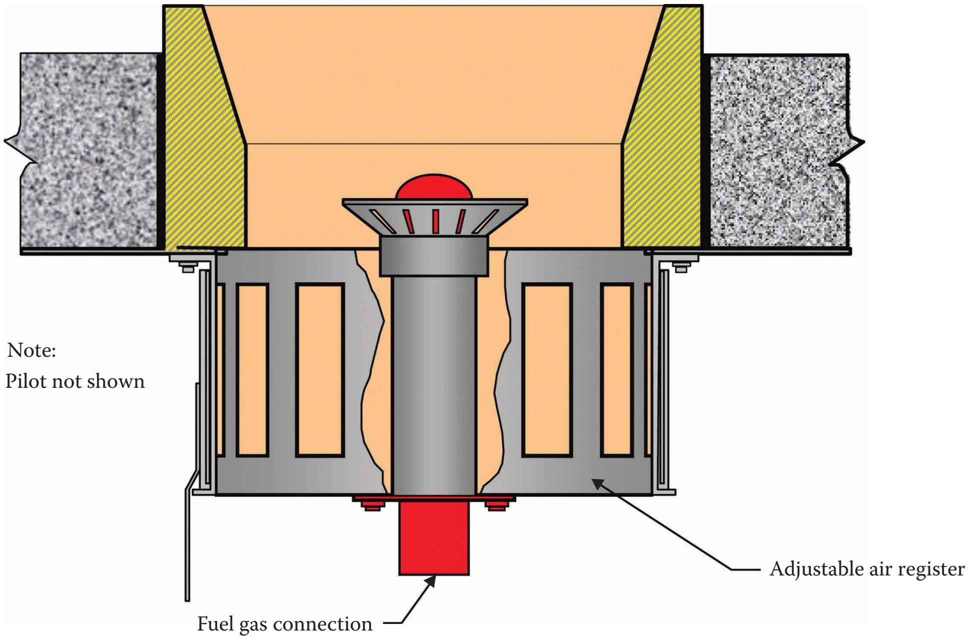


FIGURE 8.2
Typical natural draft burner.

available to provide fuel and air mixing, the heat-release-to-flame-length ratio can be much greater for both gas and liquid fuels. Low-pressure combustible waste gases can be more easily introduced because the greater energy of the air provides a large portion of the fuel-to-air mixing. Combustible waste liquids that are more difficult to burn are more easily burned in a higher-pressure-drop burner. With more pressure available, the flow area to air volume ratio is smaller, so the burner is smaller. Also, proper combustion airflow distribution is more easily achieved. Some of the available pressure may be used to impart an angular velocity component to the combustion air to further aid in mixing. Figures 8.3 and 8.4 are typical configurations for medium- and high-pressure-drop burners.

In addition to the burners shown in Figures 8.3 and 8.4, another high-pressure-drop burner is the John Zink KEU "Combustor." The Combustor is an assembly integrating a burner and a short combustion chamber, specifically for use in a TO. The Combustor is designed with three concentric stainless steel cylinders. At the Combustor inlet, oxygen-rich waste air or ambient air is supplied through a tangential nozzle, thereby generating a rotating flow of air (vortex). This creates a short rotating flame that fills the burner chamber (see Figures 8.5 and 8.6).

The intensive mixing of the reactants leads to a homogenous temperature profile across the reaction chamber and ensures an excellent burnout result even at low reaction temperatures. The prevention of hot flame zones diminishes the formation of NOx.

The standard Combustor has a pressure drop of 12 in. W.C. (3 kPa) on the combustion air side with a turndown of 3:1. If a higher turndown rate up to 5:1 is required, a Fluid Dynamic Combustor in combination with a pressure control loop that controls the position of the inlet spin valve can expand the turndown rate by maintaining the pressure drop over the turndown range.

8.3.1.6 Combination Gas and Liquid Fuel Burners

Combination fuel burners are virtually the same as dedicated gas or liquid burners. The difference is the special care required when locating the liquid and gas tips. A typical configuration would be for the liquid injection point to be at the center of the burner, through a stabilizer cone, with gas tips located at the circumference of the circular throat of the burner. This configuration is shown in Figure 8.3. By separating the fuels, better air mixing and faster burning occurs. If the gas tips are located near the liquid tip, the gas will consume the air more readily, delaying mixing of air with the atomized liquid. This would delay the oxidation reaction of the liquid resulting in a longer flame zone. In general, the faster the gas and liquid fuels are burned, the sooner noncombustible wastes can be introduced to the system.

8.3.2 Furnace/Thermal Oxidizer/Incinerator/Combustion Chamber

Once all the fuel, wastes, and air are combined, several intimately connected and simultaneous conditions must exist to achieve the DRE required. The mixture must be (1) exposed to a sufficiently high temperature

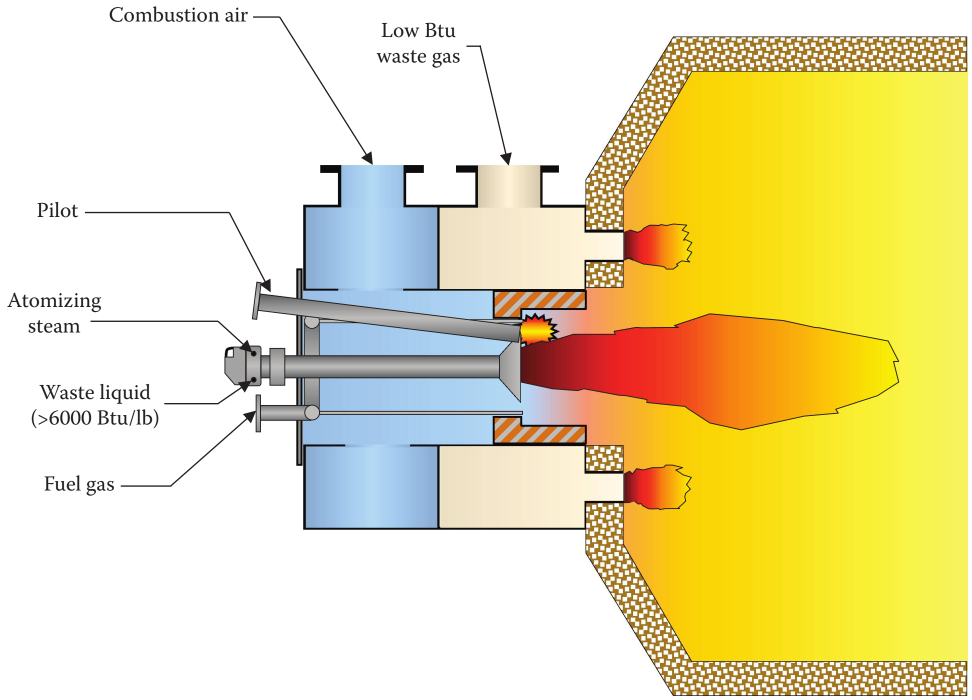


FIGURE 8.3
Typical medium-pressure-drop burner.

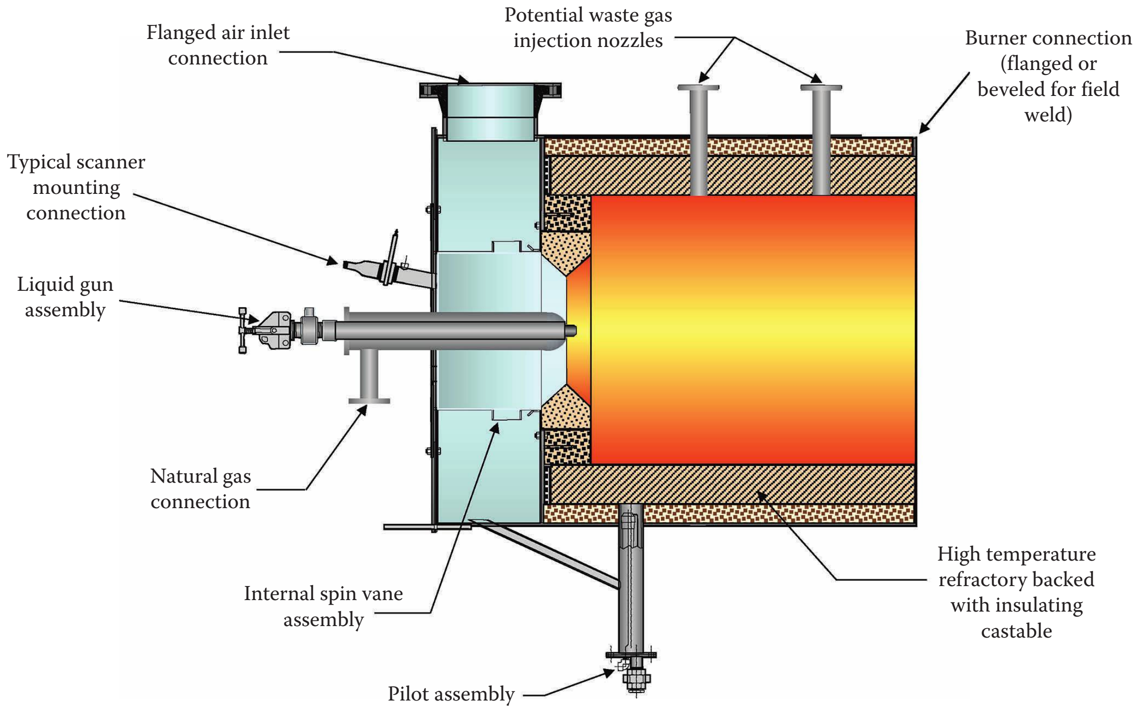


FIGURE 8.4
Typical high-pressure-drop burner.

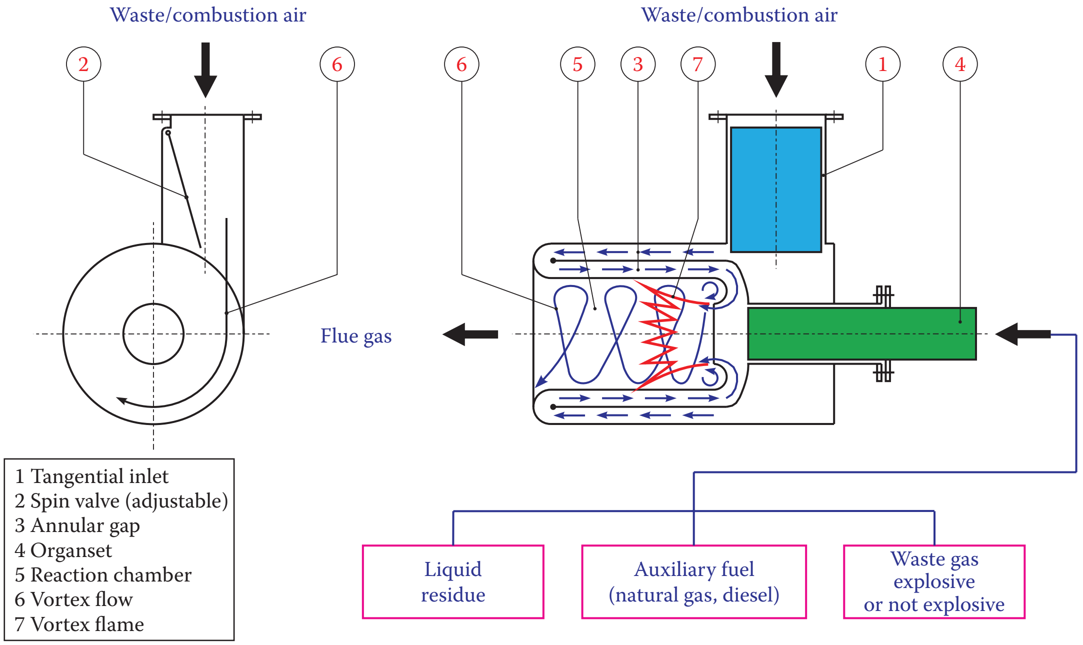


FIGURE 8.5
Schematic of a KEU combustor.

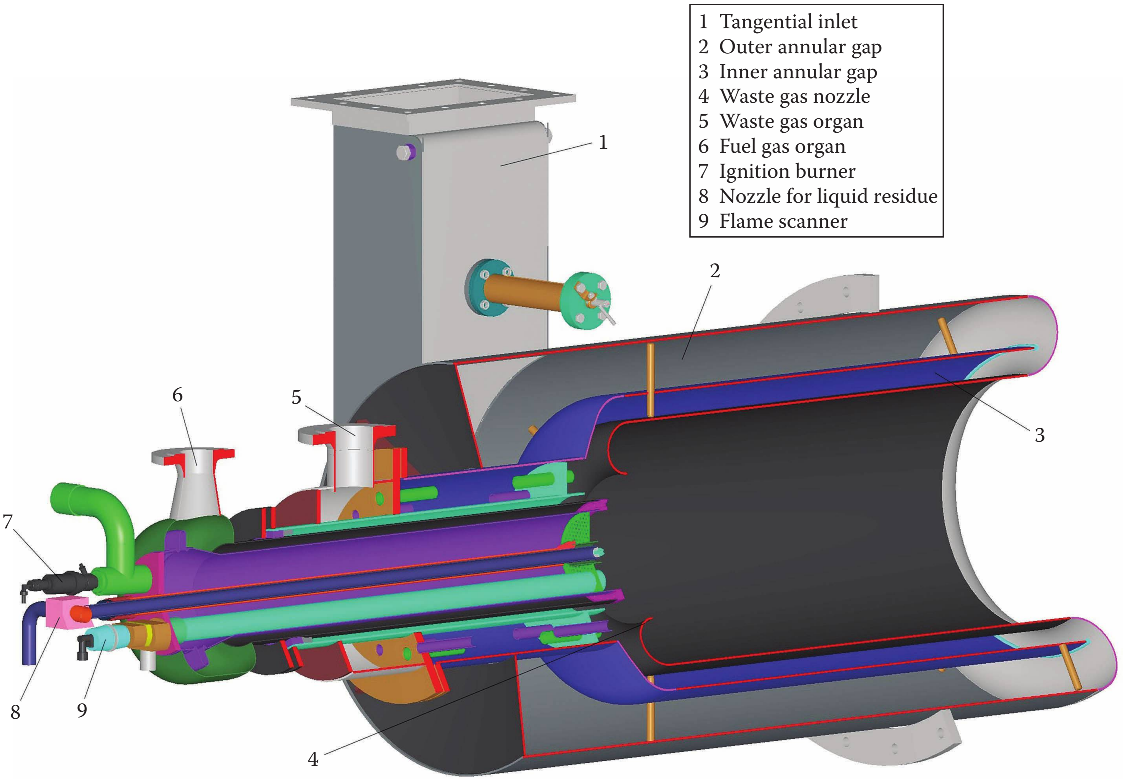


FIGURE 8.6
KEU combustor.

(2) for an adequate period of time (3) in a relatively turbulent environment to enable the oxidation reactions to reach the degree of completion needed to achieve the waste destruction efficiency required. These conditions are known as the “three Ts” in the combustion industry. The vessel that provides the environment for these conditions is known by various designations depending on the application. Furnace, TO, combustion chamber, and incinerator are all common names for the oxidation environment. For consistency, in this chapter, the vessel will be referred to as a TO. To provide the environment needed for the three conditions to be optimized, the TO must have the correct volume and geometry.

8.3.2.1 Size

The optimal residence time, or working volume, required to meet the target DRE in the TO is a function of many factors, the most important of which are waste composition, waste characteristics, degree of turbulence, DRE required, and TO operating temperature. The working volume is generally considered to start at the point at which the final amount of waste is introduced and extends to the point nearest the incinerator outlet where the operating temperature is measured. Size can also be influenced by the capital and/or operating costs, which are not directly related to the destruction efficiency of hydrocarbons. Modifying design factors to reduce size may increase undesirable emissions, such as CO or NOx.

Complex hydrocarbons present in some wastes require greater residence times and/or higher temperatures than an easily oxidized compound such as H_2S. A longer residence time means a larger vessel, which increases capital cost. Higher temperature means more fuel usage, unless the waste is exothermic, which increases operating cost.

For a given operating temperature, a waste stream containing primarily water contaminated with a little hydrocarbon will require more time than would a vapor waste stream containing the same hydrocarbon contaminant, because the liquid stream has to vaporize and reach the oxidation temperature before it begins reacting. Again, more residence time means a larger, more expensive TO.

If the flow in the TO is not relatively turbulent or contains areas of stagnant flow, more residence time will be required for the available oxygen to come into contact with the organic material so it can be oxidized and destroyed. This poor mixing of waste with air delays the onset of oxidation. A larger, more costly vessel would be needed to achieve the desired DRE. TO designers are careful to avoid such areas of stagnant flow to minimize wasted volume whenever possible.

TABLE 8.1

Typical TO Operating Conditions

Waste Type	Operating Temp. (°F)	Retention Time (s)	Typical DRE
Lean gases containing hydrocarbons or sulfur compounds	1200–1400	0.3–0.6	99
Lean gases containing common chlorinated solvents	1600–1800	1	99.9
Liquid streams	1600–2000	1.0–2.0[a]	99.99
Halogenated hydrocarbon liquids	1800–2000	1.5–2.0[a]	99.99
Dioxins and polychlorinated biphenyls (PCBs)	2200	2	99.9999

[a] Extra time for liquids to allow for droplet evaporation.

A high DRE requirement may result in the need for greater residence time and/or higher operating temperature, assuming the mixing is adequate. The greater the TO temperature, the greater the oxidation reaction rate. Again, higher temperatures may require additional fuel, adding to the operating cost. A higher temperature may also result in the need for more expensive refractory material. On the other hand, the increased reaction rate could reduce the residence time needed and result in a smaller volume TO, decreasing capital cost. For exothermic wastes, the highest operating temperature possible should be selected to minimize use of quench air, which leads to reduced TO sizes and, therefore, cost.

Increasing the temperature has the aforementioned pros and cons, but also an overall drawback to higher temperatures is that it may lead to higher nitrogen oxide (NOx) emissions.

It is obvious that for TO design, it ultimately becomes a trade-off between the capital cost of increased residence time and the costs and problems associated with an increase in operating temperature.

Table 8.1 shows typical TO operating conditions for a variety of cases. Although the ranges of operating conditions have been developed by testing, experience with actual operating systems allows specific conditions to be set on a case-by-case basis, depending on past experience with similar waste streams.

8.3.2.2 Flow Configuration

The orientation and flow configuration of the TO must accommodate the user’s space restrictions, the characteristics of the waste being burned, and the downstream flue-gas treatment requirements and again provide the most cost-effective solution. The TO can be arranged for vertical upflow, horizontal flow, or vertical downflow of the combustion products. Vertical upflow is preferred for the simplest situations when the combustion products are vented directly to the atmosphere with no downstream treatment needed. This is also the arrangement of choice if the equipment located immediately downstream of the TO requires an elevated entry. Horizontal flow is the most utilized configuration when a heat-recovery boiler or other equipment with side entry near grade is located immediately downstream of the TO. Vertical downflow is required in many systems when the waste contains ash-forming materials or salts. The vertical downflow aids in discharging the solids by gravity and prevents the accumulation of these materials in the TO. Online solids collection and removal equipment must be installed at the base of the TO in these cases.

Most TO vessels are cylindrical in design with length-to-inside diameter of refractory (L/D) ratios ranging from 2:1 to 4:1. For a cylinder of a specific volume, the surface area is the least when the L/D ratio is 1:1. As the L/D increases for the same volume, the surface area of the cylinder increases. From an L/D of 1:1 to 2:1, the increase is almost 26%; from 1:1 to 3:1, the increase is more than 44%; and from 1:1 to 4:1, the increase is almost 59%. The result is that the cost of the cylinder increases as the L/D increases. Also, the greater the L/D ratio, the longer the TO, thus requiring a larger plot area for a horizontal configuration. A positive result of greater L/D is a smaller cross-sectional flow area that increases flue-gas velocity. As discussed in the previous section, the greater velocity helps minimize dead zones in the TO and improves mixing and consequently DRE, which in turn enables the use of a smaller working volume. A smaller volume reduces the capital cost of the equipment. Another important consideration that affects the L/D is that flame or liquid impingement on the walls of the TO will cause incomplete combustion and refractory deterioration. So, in some cases, a greater diameter or length may be required to maximize DRE or equipment life regardless of the oxidation reaction rate.

8.3.3 Refractory

Virtually all TO vessels are internally lined with heat-resistant refractory material (see Volume 2, Chapter 5). Installation of a refractory lining provides three important benefits. The steel shell is protected from the high-temperature environment inside the TO, the external temperature of the vessel is reduced to levels that are safe for personnel and equipment in the vicinity, and the oxidation process is insulated against heat loss so that the vessel is a reasonably adiabatic chamber.

Refractories used in TO vessels are primarily ceramic materials made from combinations of high melting oxides such as aluminum oxide or alumina (Al_2O_3), silicon dioxide or silica (SiO_2), or magnesium oxide or magnesia (MgO). Refractories containing primarily alumina and silica are "acid" refractories and are by far the most common type used for TO linings. Refractories containing large amounts of MgO are "basic" refractories and are used for their good resistance to specific reactive ash components, particularly alkali metal compounds, which result from burning some inorganic salt-laden wastes. Because MgO refractories are significantly more expensive than the alumina–silica materials, the TO configuration is often optimized to allow the use of and maximize the life of the less-alkali-resistant alumina–silica refractories.

Refractories may be further categorized into "hard" and "soft" refractories. Hard or soft refers to the state of the refractory when ready for service. Hard refractories can be further subcategorized as bricks, plastics, or castables. Brick refractory is available in a wide variety of compositions ranging from high-alumina-content aluminosilicates to magnesites. The binding material in brick can be calcium cement based or phosphoric acid based. A brick lining is held in place by gravity and/or the compressive forces resulting from proper placement (as in the construction of an arch). The linings must be installed in a vessel by skilled craftsmen and require more time to install, especially if special shapes have to be assembled by cutting bricks. Brick has high density and low porosity (i.e., good penetration resistance to molten or refractory-attacking materials), which offers the best abrasion and corrosion resistance of any form of refractory. However, the high density results in the brick usually being heavy (120 lb/ft^3 = 1920 kg/m^3 or more) and the insulating value being lower, resulting in greater lining thickness being required to achieve the same thermal resistance. An additional consequence of a thicker lining is that a larger, more expensive TO shell is required for a given vessel volume requirement.

Plastic refractories have similar alumina and silica content as bricks and are so called because the binder in plastic refractories, usually a water-wetted clay or phosphoric acid–based binder, is not set and the material is very malleable or "plastic" in the ready-to-install condition. Once in place, however, the binder is set by exposure to air or heat. Plastic refractories are shipped ready-to-install in sealed containers to provide shelf life. Once opened, the material must be used immediately. Because the refractory is in a "plastic" state, it can be forced or "rammed" into place and can be formed into almost any shape needed. It is held in position by an anchor system that consists of a metal piece welded to the shell and a prefired anchor (a special ceramic refractory shape) held by the metal portion attached to the shell. The anchor extends through the lining to the surface of the refractory. In addition, because plastic refractory is dense (140 lb/ft^3 = 2240 kg/m^3 or more), has low porosity, and is relatively easy to install, it is often used in difficult-to-brick locations of vessels that will encounter a high-temperature, corrosive environment. The total cost of plastic refractory, based on material and installation, insulating capabilities, and erosion and corrosion resistance is usually less than that of the brick with a comparable composition, especially if the final shape needed is unusual.

Castable refractories consist of fireclay or high-alumina-content aggregates that are held together in a matrix of hydraulic calcium aluminate cement. Castable refractories are the least expensive to install among the hard refractories. Castable refractory is shipped in bags like dry cement, mixed with water prior to installation (a variable that can affect its final properties), and either poured or gunned (slightly dampened and blown through a nozzle) into place. The castable refractory is held in place by alloy steel anchors that are welded to the furnace shell. The castables used for incinerator applications usually weigh between 50 and 120 lb/ft^3 (800 and 1920 kg/m^3). Compared to the other hard refractories, castables generally have the best insulating properties and the poorest corrosion and erosion resistance. To minimize lining thickness, a layer of insulating castable refractory is often installed as a backup to a brick layer, the brick layer being the layer that forms the internal surface of a TO. The brick provides high resistance to temperature, corrosion, and abrasion, while the castable provides the higher insulation properties needed to reduce overall refractory thickness.

Soft refractories are composed of ceramic fibers formed into a blanket, a soft block module, or a stiff board. They remain soft when in service. The blanket and board are usually held in place with stainless or other high-temperature alloy anchors (or pins) welded to the inside TO wall. The soft refractories are easily installed by pressing onto the steel shell with the pins projecting through. Self-locking washers are then placed on the pins to keep the material from coming loose. The block modules have an internal frame that is attached to an anchor welded to the shell. Soft refractories are much lighter (usually less than 12 lb/ft^3 = 190 kg/m^3), are much better insulators, and can be heated rapidly without fear of damage due to thermal shock. Thermal shock is the rapid thermal expansion of the hot surface of hard refractory. That layer then separates and falls off, reducing the refractory thickness. Soft refractories are limited to 2300°F (1260°C), are susceptible to erosion, and do have poorer resistance to alkali liquids and vapors. Ceramic fiber refractory is very cost-effective in certain applications.

Common problems related to all refractories include the following:

- Improper operation or design can lead to thermal shock or erosion damage.
- Normal acid gases (SO_2 or HCl) in the combustion products can raise the gas dew-point temperature to be as high as 400°F (200°C). Excessive refractory thickness, rain, or extremely cold weather can result in a furnace shell temperature below the dew point, which can result in acid condensation and consequent corrosion of the steel oxidizer vessel. To minimize weather-related effects, a ventilated sheet metal rain shield often is used to prevent rain contact and limit external convective heat transfer from the vessel.
- Flame impingement on refractory surfaces can result in higher-than-expected temperatures, frequent temperature fluctuations, and locally reducing conditions, all of which can shorten refractory life.
- Liquid impingement on hot refractory will cause spalling and erosion, which decreases refractory life. Spalling is similar to thermal shock separation and is caused by steep thermal gradients in the refractory.
- At higher temperatures, salts (those containing Na, Ca, K, etc.) and alkaline earth oxides (e.g., K_2O, Na_2O, CaO, and MgO) will react with most acid refractories. The result of these reactions can be a loss of mechanical strength, crumbling, or even a “fluxing,” that is, liquefaction, of the exposed surface. In any case, refractory life is shortened.

8.3.4 Catalytic Systems

In a typical thermal incinerator, waste destruction occurs in the flame or the TO because of high-temperature, gas-phase oxidation reactions. In a catalytic unit, waste destruction occurs in a catalyst bed at much lower temperatures via oxidation reactions that are promoted by the catalyst. The lower operating temperature in a catalytic unit is its main advantage. The lower temperature reduces the need for auxiliary fuel to maintain furnace temperature, thus reducing operating cost. Another advantage to lower operating temperature catalytic oxidation is that NOx formation during oxidation is reduced. The lower temperature also eliminates the need for internal refractory lining, reducing the shell diameter. However, the use of a stainless steel shell and external insulation is necessary and generally cancels any benefit gained from eliminating the refractory.

A major limitation of catalytic oxidation is that the catalyst is susceptible to damage from certain compounds in the waste or from overheating. Most waste streams for which catalytic oxidation is considered are contaminated airstreams that have more than enough O_2 to complete oxidation of the hydrocarbons in the waste. It is possible to treat inert gas streams, such as nitrogen contaminated by a small amount of hydrocarbon, but enough air must be blended with the gas stream prior to entering the catalyst bed to maintain approximately 2% O_2 in the flue gas after the oxidation has occurred. The addition of air to waste streams that are mostly inert greatly increases the volume and overall capital and operating costs. Another limitation is that a DRE of >99% requires a significant amount of catalyst, which increases the capital cost. Recuperative heat exchangers are often used downstream of the catalytic unit to preheat the incoming waste gas to further reduce the fuel requirement.

Catalysts used in catalytic oxidation systems actually consist of a ceramic or stainless steel base material (carrier or substrate) covered with a thin coating of a catalyst material. In a given volume of enclosure, the greater the catalyst surface area available for contact with the waste gas, the greater the amount of oxidation reaction that can be achieved. The catalyst material is generally one of two types, either noble metals or transition metal oxides. The noble metals are generally preferred. Catalyst type is also based on the ability of the catalyst to resist chemical deactivation (poisoning) from compounds present in waste streams. Typical compounds responsible for poisoning are HCl, HBr, HF, and SO_2 (reversible poisons) and elements such as lead, bismuth, mercury, arsenic, antimony, and phosphorous (irreversible poisons). While catalysts have been formulated that will retain their activity in the presence of many of these poisons, there is no single catalyst that is best for all applications. An additional reversible situation is fouling by fine particulate, which could be fine rust particles, refractory dust, or particulate in the waste stream. To avoid rust and refractory dust entering the catalyst, bed, the system can be designed without refractory upstream of the catalyst bed and with stainless steel as the vessel material upstream of the catalyst. Fine particulate quite simply covers the surface of the catalyst reducing the amount of surface available for reaction. When the DRE has degraded too much, the particulate can often be removed by removing and washing the catalyst blocks or by washing in place.

There are generally two types of catalyst carrier media, ceramic beads and honeycomb monoliths. Virtually all new applications utilize catalysts that are applied onto honeycomb monoliths because they require less pressure drop and allow more flexibility in furnace design and orientation. If bead catalyst is used, the flow must usually be in a vertical (up or down) direction. The honeycomb monoliths can be installed in any orientation and are usually found in horizontal flow catalytic oxidizers, which are convenient to maintain due to easier access.

Below some minimum threshold temperature, all oxidation catalysts become ineffective. At higher temperatures, the oxidation rate increases rapidly until the rate becomes limited only by the catalyst surface available for interaction with the waste gas. The temperature at which this rapid increase occurs varies depending on the hydrocarbon, but is typically between 400°F and 700°F (200°C and 400°C) and is usually referred to as the "ignition" temperature. Catalytic units are designed such that the inlet temperature to the catalyst bed is maintained above the ignition temperature. As the waste moves through the catalyst bed, oxidation occurs and the gas and catalyst temperatures rise. The temperature rise in the catalyst bed depends on the heating value of the waste stream. If subjected to temperatures between 1200°F and 1350°F (650°C and 730°C) for very long, many catalysts will begin to suffer significant damage as a result of sintering. Sintering is the melting and coalescing of the active catalyst material, which results in a loss of available catalyst surface area and, consequently, a loss of catalytic activity. The rate of sintering increases rapidly with increasing temperature. A catalyst that shows the first signs of damage at 1200°F (650°C) likely will be severely damaged in a matter of hours at 1500°F (820°C). Therefore, for long-term operation and best DRE, the catalyst bed needs to be maintained above the temperature at which high-rate reactions occur, but below the temperature at which significant sintering occurs. Typical catalyst outlet temperatures are in the range of 600°F–1000°F (320°C and 540°C).

Destruction efficiency in a catalytic incinerator depends on the waste-gas composition, the catalyst type and configuration, the waste-gas temperature at the entrance to the catalyst bed, and the amount of time the contaminant is exposed to the catalyst (catalyst surface area). Changes in destruction efficiency are achieved by changing the amount of catalyst or by changing the waste flow rate for a given amount of catalyst, either of which changes the effective exposure or residence time. Residence time in a catalyst bed is often expressed as its inverse and is called space velocity. Space velocity is defined as volumetric flow rate of waste (SCFH) divided by catalyst bed volume (ft^3 or m^3). Typical catalytic units are designed with space velocities of less than 30,000, inlet temperatures less than 700°F (370°C), and outlet temperatures less than 1200°F (650°C). Practical catalytic systems typically achieve destruction efficiencies of 90%–99%.

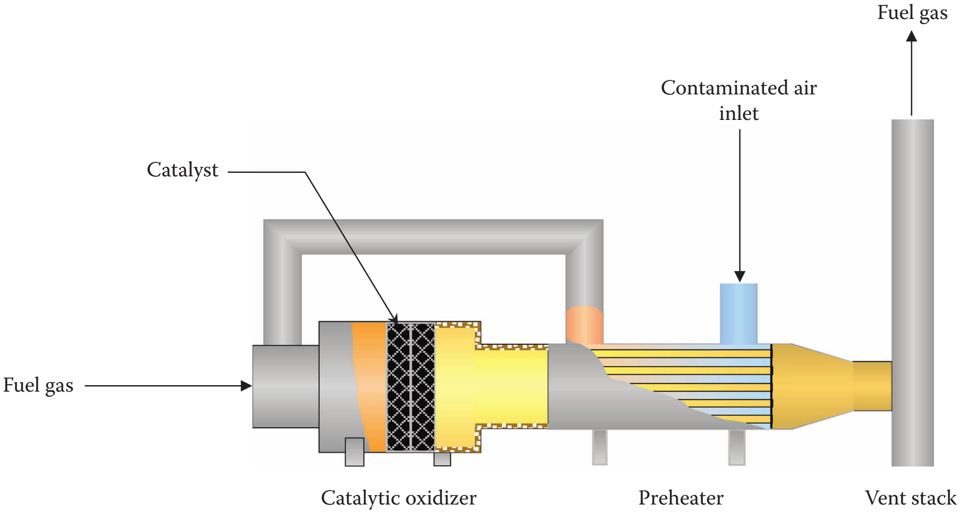


FIGURE 8.7
Typical horizontal system with a preheat exchanger.

Catalytic oxidizers must be configured to provide well-mixed, uniform waste-gas flow at the catalyst bed entrance and to avoid flame impingement on the catalyst bed by the heat-up burner. As noted previously, units constructed from catalyst-coated monoliths can be oriented in any flow direction, but usually are mounted in horizontal flowing units to facilitate catalyst loading and maintenance. Systems using either a fixed or a fluidized bed of beads are mounted vertically. Figure 8.7 shows a typical horizontal system with a preheat exchanger.

8.3.5 Flue-Gas Processing Methods

In some incineration systems, the flue gas does not require treatment to reduce emissions of acid gases or particulates, and use of heat recovery is not cost-effective. In such cases, the hot flue gas leaving the furnace is vented directly to the atmosphere. For many systems, however, some form of flue-gas cooling is utilized, either by heat recovery and/or by conditioning of the flue gas before it enters downstream equipment. Flue-gas cooling is accomplished indirectly or directly. Indirect cooling is achieved by heat transfer from a higher-temperature mass to a lower-temperature mass through the use of heat-recovery devices such as boilers, recuperative preheat exchangers, heat-transfer medium exchangers, or regenerative preheat exchangers. Heat-recovery devices remove heat from the flue gas to lower the temperature, but do not change the mass flow rate. Direct cooling is accomplished by adding a cooler material directly to the flue gas to achieve the necessary temperature reduction. Adding cooling material to the flue gas increases the total mass flow rate while reducing the temperature of the final mixture. The added material can be water, air, or recycle flue gas, depending on the downstream equipment.

Flue-gas processing for emission control includes wet and dry particulate removal, wet and dry acid-gas removal, and NOx reduction.

8.3.5.1 *Cooling by Heat Recovery*

The different types of indirect heat-recovery devices are boilers, recuperative heat exchangers, process oil heaters, and regenerative preheat systems.

8.3.5.1.1 Boilers

There are two basic types of boilers: fire tube and water tube.[1] In a fire-tube boiler, the combustion products pass through the inside of the boiler tubes, while water is evaporated on the outside. Conversely, in a water-tube boiler, the hot combustion products flow over the outside of the tubes, while the steam is generated on the inside of the tubes. Cooling flue gas from 1800°F to 500°F (1000°C to 260°C) with a boiler can result in substantial steam production. Adding an economizer downstream of the boiler will recover even more heat by reducing the flue-gas temperature to about 350°F (180°C). (An economizer is a lower-temperature heat exchanger used to heat the boiler feedwater from its normal supply temperature of about 220°F (100°C), before it is injected into the boiler.)

8.3.5.1.1.1 Water-Tube Boiler There are several important differences between the water-tube and the fire-tube boilers. A water-tube boiler (Figure 8.8) is generally less expensive to build for applications that require high steam pressure (i.e., greater than 700 psig or 48 barg) and/or large steam flows (i.e., greater than 50,000 lb/h or 23,000 kg/h), because the pressure-retaining parts are relatively small-diameter tubes and tube rows lend themselves to modular expansion. On the other hand, the large-diameter shell of a fire-tube boiler is a pressure-retaining part, making it more expensive to build large-diameter shells for higher pressures and higher capacities. Extended surfaces (finned tubes) and superheaters are more easily incorporated into water-tube boilers often resulting in smaller space requirements. Most importantly, the heat-transfer surfaces of a water-tube boiler are accessible to soot blowers (high-pressure steam or air lances) that are used for periodic cleaning of the flue-gas side of the tubes to prevent loss of efficiency due to fouling by non-molten particulate resulting from waste combustion. Thus, for an application that requires high-pressure steam production from a large flow of combustion products containing significant amounts of particulate, the boiler design of choice is the water-tube. The typical flue-gas pressure drop through a water-tube boiler is 2–6 in. W.C. (0.5–1.5 kPa). Thus, the water-tube boiler is also used when pressure drop must be minimized.

8.3.5.1.1.2 Fire-Tube Boiler Fire-tube boilers have the important advantage that virtually all the surfaces in the boiler are maintained at the steam temperature. Thus, there are no cold spots on which acid may condense. Consequently, the fire-tube boilers are well suited to those applications where low-pressure to medium-pressure steam is to be produced from an acidic flue gas. The typical flue-gas pressure drop through a fire-tube boiler is 8–12 in. W.C. (2.0–3.0 kPa). Figure 8.9 shows a typical fire-tube boiler.

There are times when the flue gas contains molten particulate, but it is cost-effective to cool the flue gas to "freeze" molten particulate, so the remaining heat can be recovered in a boiler. Even if the flue gas is cooled to 1200°F (650°C), for instance, significant heat recovery is still possible. It is important to understand that the choice to use a direct cooling (quenching) medium upstream of the boiler will affect both the size of the boiler and the heat recovery. If clean, recycled flue gas from the outlet of the system can be used for quenching, but the mass of recycle flue gas required will be large because of its elevated temperature (350°F–500°F or 180°C–260°C).

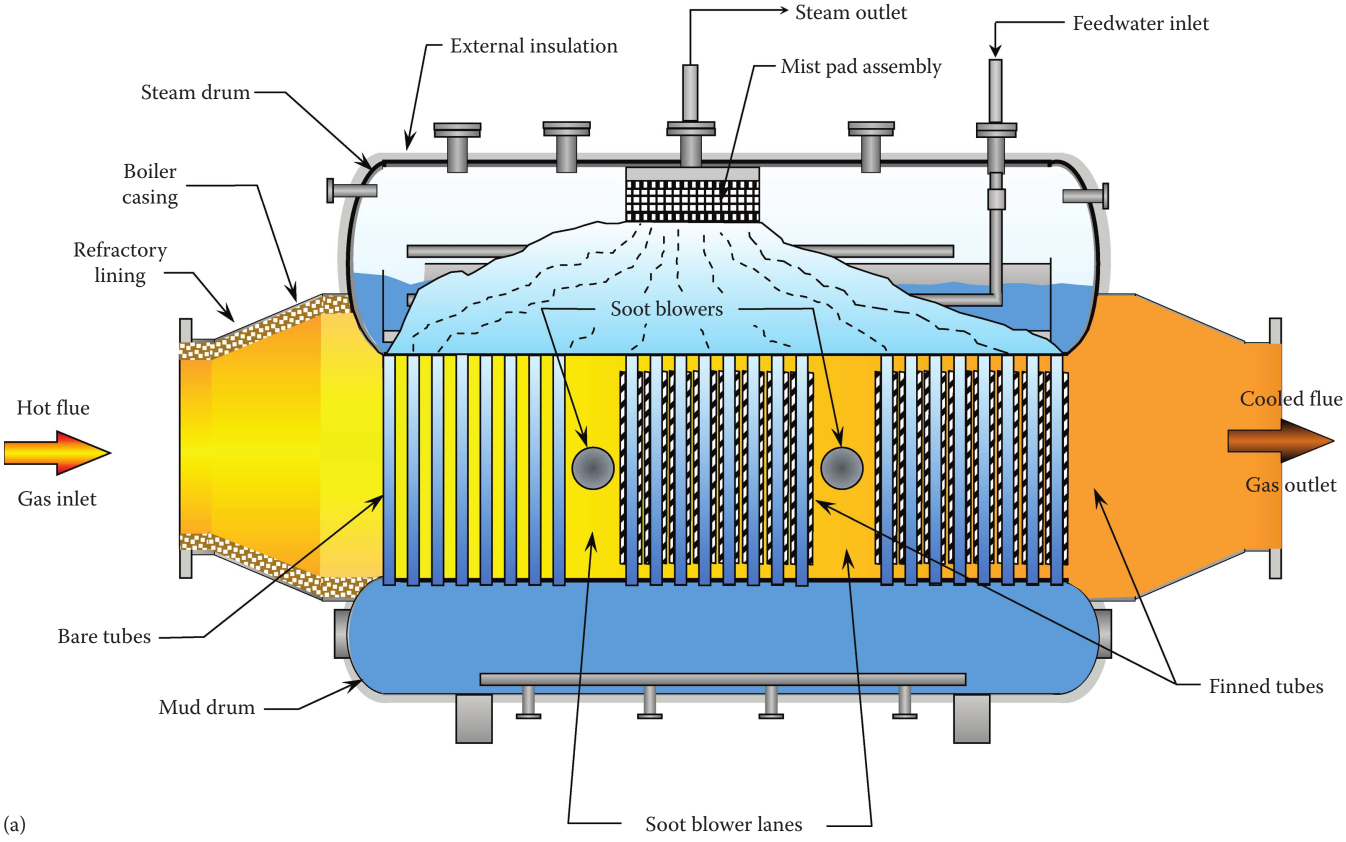


FIGURE 8.8
Water-tube boiler: (a) side view.

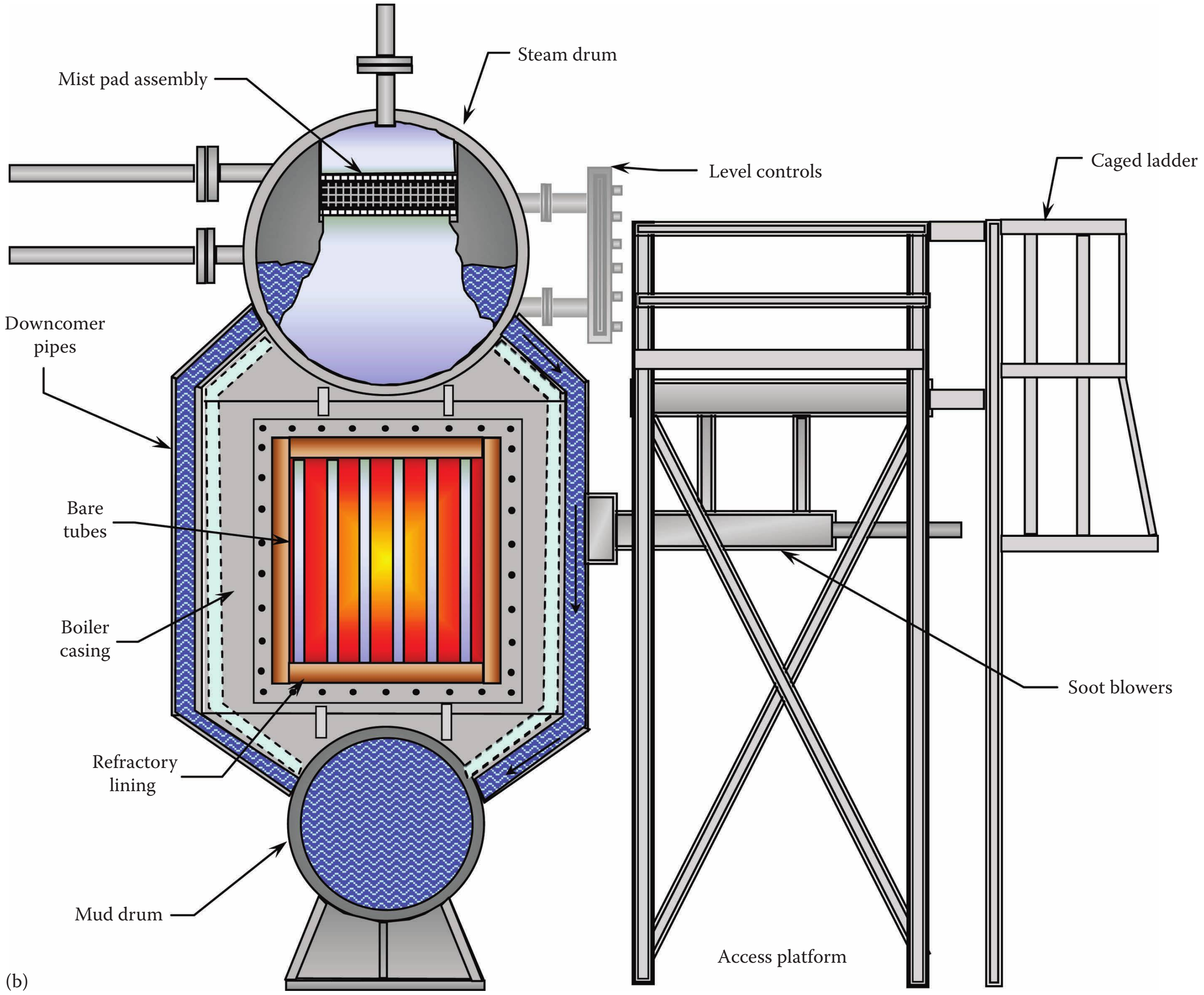


FIGURE 8.8 (continued)
Water-tube boiler: (b) end view.

The flow through the rest of the system could be twice the mass of the flue gas from the TO. If water is used to quench, a much smaller mass is required resulting in a much smaller mass flow through the rest of the system. However, using recycled flue gas for the quenching medium results in much greater heat-recovery efficiency than if water quench is used because the added water leaves the system with its latent heat of vaporization as well as the sensible heat at the exit temperature.

8.3.5.1.2 Recuperative Preheat Exchanger

If no steam is needed, waste heat boilers are not a viable heat-recovery option. However, if a low heating value waste gas is being treated and a large amount of auxiliary fuel is needed to maintain operating temperature, a preheat exchanger may be used to minimize the auxiliary fuel requirement by transferring heat from the flue gas to the incoming waste gas or combustion air. Figure 8.10 shows a typical all-welded shell-and-tube heat exchanger. The furnace exhaust flows through the tubes in the exchanger while the waste gas, or combustion air, flows around the tubes inside the shell. Up to 70% of the energy released in the furnace can be recovered economically by this method. Normal recovery efficiencies are in the 55%–60% range. Structural limitations (thermal expansion) typically constrain the hot flue-gas temperature to no more than 1600°F (870°C). However, more expensive U-tube-type heat exchangers exist that can be designed for higher temperatures.

Recuperative exchangers may also be plate-and-frame-type construction. This type can withstand higher-temperature expansion differences because of its

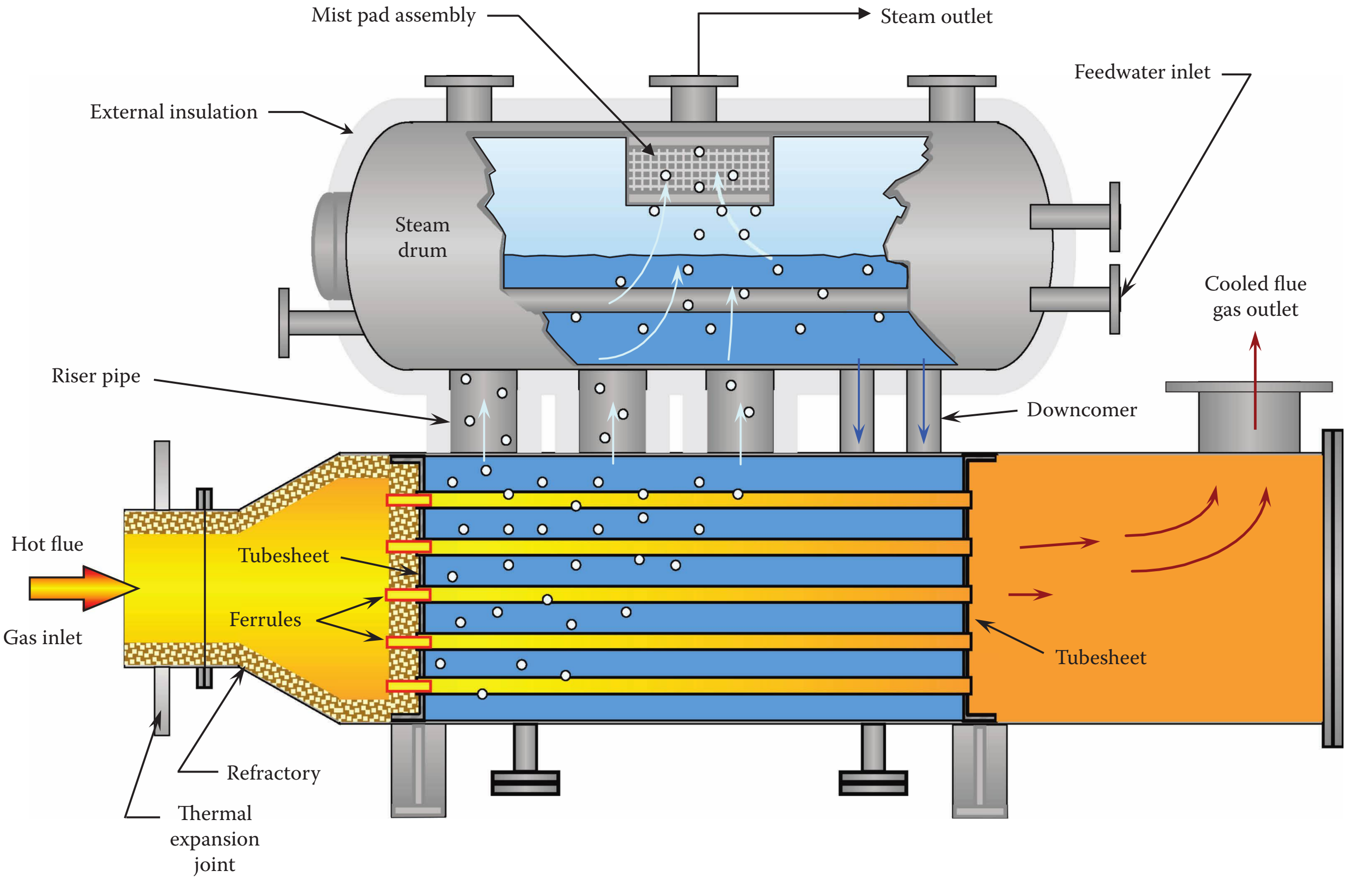


FIGURE 8.9
Fire-tube boiler.

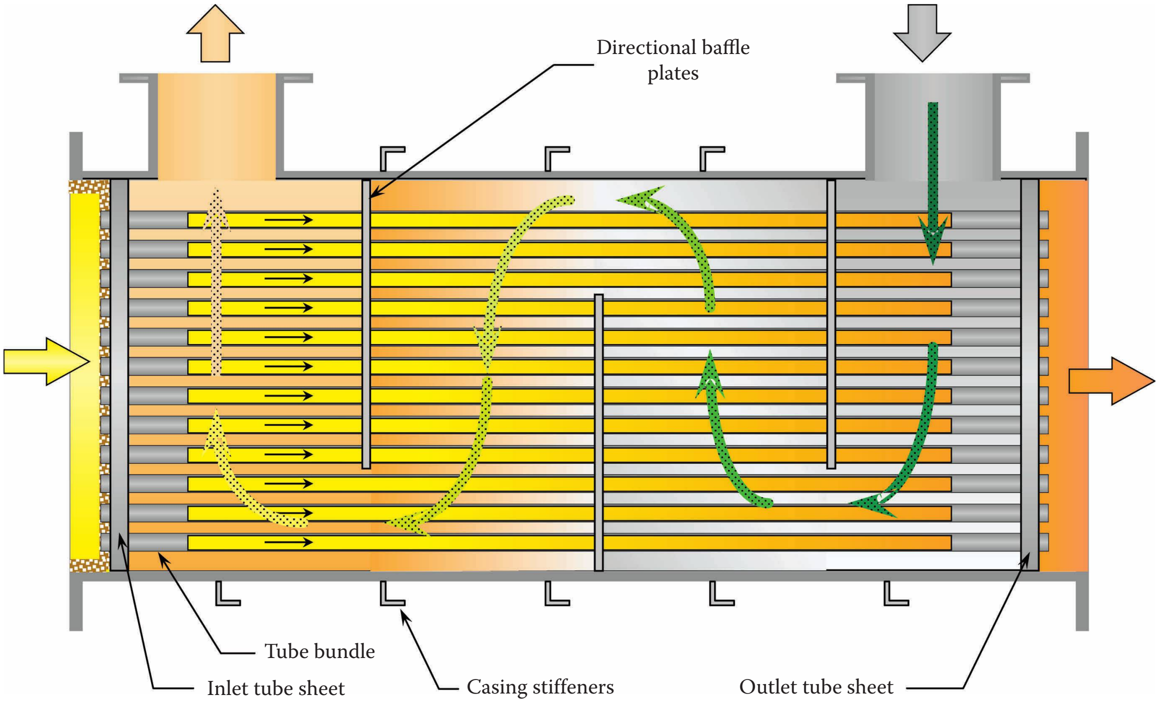


FIGURE 8.10
Typical all-welded shell-and-tube heat exchanger.

non-welded construction. However, some leakage of waste gas into the clean combustion products will occur, increasing the unburned waste hydrocarbon emission to the point where if the DRE requirement is high, it may not be met. Also, although the initial leakage may be low enough when the exchanger is first put into service, it will likely increase with time, particularly in systems that operate intermittently (shutdown daily or weekly). To avoid leakage problems, the more expensive shell-and-tube-type heat exchangers with all-welded construction should be used when high DRE is required.

8.3.5.1.3 Regenerative Preheat Exchanger

A TO system with regenerative preheat exchange consists of a refractory-lined TO connected to three or more vessels containing a ceramic packing (often ceramic scrubber bed packing) that alternately functions to preheat the waste and to cool the flue gas exiting the TO section. Figure 8.11 shows the configuration of such a system. This system is primarily for contaminated airstreams. The temperature of the gases is measured at the inlet and outlet of each ceramic-packed vessel. Several valves must be used to control the direction of flow at all times during operation. The system is somewhat larger and more expensive than a recuperative system, but it is much more efficient. Up to 95% heat recovery is possible if the incoming hydrocarbon content is very low. However, the normal rates are in the 85%–90% range. The potential fuel savings make these systems attractive for large waste flows that have little heating value.

In operation, the waste gas flows into the system through a hot bed of packing (chamber 1) before it enters the TO. The incoming waste-gas temperature is monitored at the hot end (nearest the TO) of chamber 1. The flue gas exiting the TO flows through an identical, but cooler bed of packing (chamber 2) before it is vented to the atmosphere. When the packing in chamber 2 has absorbed heat to the point that the exit gas temperature rises above a preset maximum, typically 300°F–350°F (150°C–180°C), the hot gas is redirected to the third bed of packing (chamber 3) that was out of service and is cool. At the same time, the incoming waste gas is switched from chamber 1 to chamber 2, which is now the hot bed, to pick up stored heat before flowing into the TO. Chamber 1, which is now temporarily out of service, was in preheat service when the flows were switched. Consequently, chamber 1 is filled with untreated waste gas. The waste gas in chamber 1 is purged into the TO with "cool" recycle flue gas while it is out of heat exchange service. If the most recently used incoming bed was not purged or if only two beds were used, a bed full of waste gas would be vented at each flow reversal. The result would be similar to the plate exchanger problem mentioned previously, in that the high DRE cannot be achieved. Thus, for high DREs, at least three beds are required in order to allow the cool inlet bed to be purged before it becomes an outlet bed. In practice, some large systems are constructed of five or more beds to overcome shipping restrictions and to allow prefabrication. However, the DRE requirement for many common applications is low enough that two-bed systems or rotating single-bed systems are frequently used.

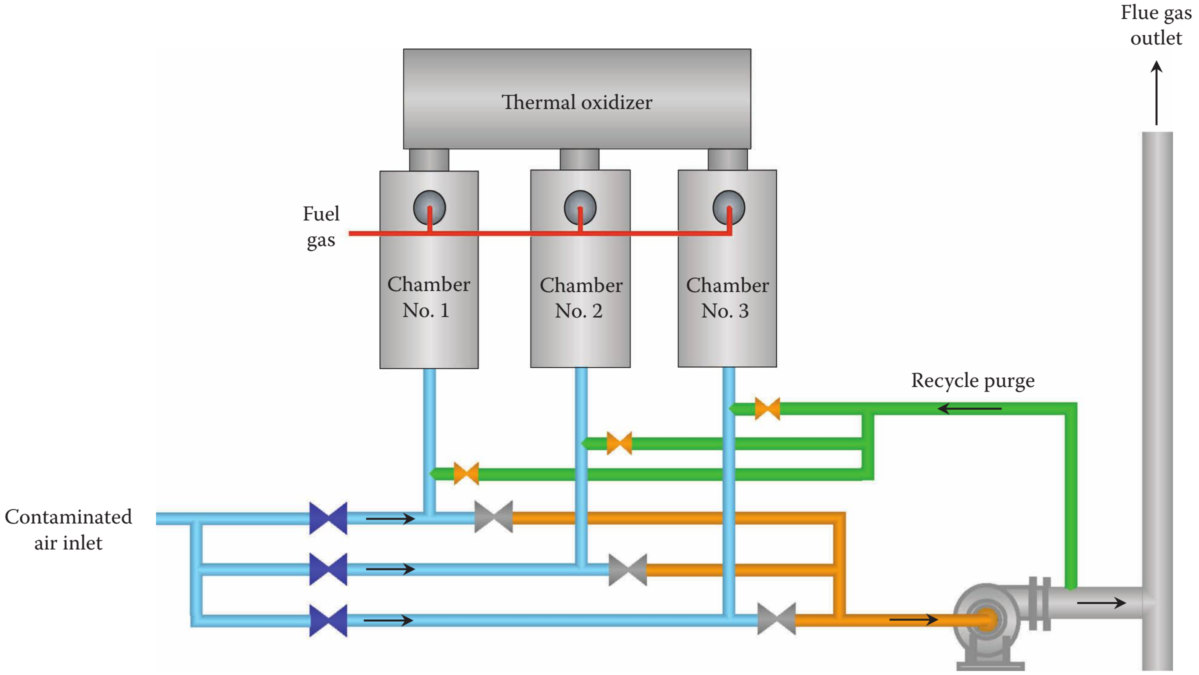


FIGURE 8.11
Regenerative preheat exchanger.

8.3.5.1.4 *Organic Heat-Transfer Fluid Heat Exchangers*

Many plants use organic heat-transfer fluids (hot oil) to provide a controlled heat source for plant processes. If the TO system can be operated with or without waste so that a continuous heat source is available, the flue gas can be used to heat this fluid in a shell-and-tube design exchanger. However, there may be periods when the system is generating more flue gas than is needed because the heat demand of the hot oil is reduced. Also, during normal operation, the temperature of the flue gas entering the exchanger may be limited to some maximum because of the properties of the fluid. For such cases, the system may have to be designed to bypass some or all of the hot flue gas directly to the atmosphere and to cool the flue gas by mixing in some ambient air to avoid coking the organic heat-transfer fluid. Figure 8.12 shows the necessary configuration.

In the system in Figure 8.11, the waste is fired into a horizontal TO that is connected directly into the base of a hot vent stack. Another connection located in the base of the stack allows flue gas to be drawn from the base of the stack into a duct connected to the hot oil heat exchanger. An induced draft (ID) blower is located downstream of the hot oil heater to "pull" the flue gas from the base of the stack and through the exchanger. The cooler flue gas exiting the exchanger enters the inlet of the ID fan and is then pushed by the fan back into the stack at a location that is at least two stack diameters above the flue-gas extraction nozzle in the base of the stack. A hot oil temperature controller monitors the fluid temperature and modulates the flue-gas flow through the heat exchanger, by means of an inlet damper to the ID fan or a variable-frequency drive (VFD) speed control on the ID fan's motor.

For additional temperature control flexibility, the incoming hot flue gas can be cooled by inducing ambient air into the stack-to-oil heater duct. A flue-gas temperature controller monitors the hot oil heater inlet temperature and modulates the valve that controls the amount of ambient air drawn into the stack-to-oil heater duct.

This bypass configuration can be used for any heat-transfer device when more flue gas is available than the heat exchanger can process.

8.3.5.2 *Cooling without Heat Recovery*

Flue-gas cooling by means other than heat removal is often necessary. When the flue gas has to be processed to remove emissions, it must be cooled to a temperature that will not harm the downstream equipment or not reduce the efficiency of the downstream equipment. If particulate is to be removed by a dry process such as an electrostatic precipitator (ESP), the flue gas must usually be cooled (conditioned) to below 650°F (340°C). If it is to pass through a baghouse, it will have to be conditioned to 400°F (200°C) or less. For wet particulate removal or wet acid-gas removal, the flue gas will likely have to be cooled (quenched) to its saturation temperature for treatment.

8.3.5.2.1 *Conditioning Section*

In most applications, the purpose of the conditioning section is simply to cool the flue gas exiting the TO to some required downstream temperature. In other cases, the flue gas may contain molten droplets of material that must be cooled below their melting point (frozen), so they will not adhere to downstream boiler tubes or other cooler surfaces. In other words, the "condition" of the flue

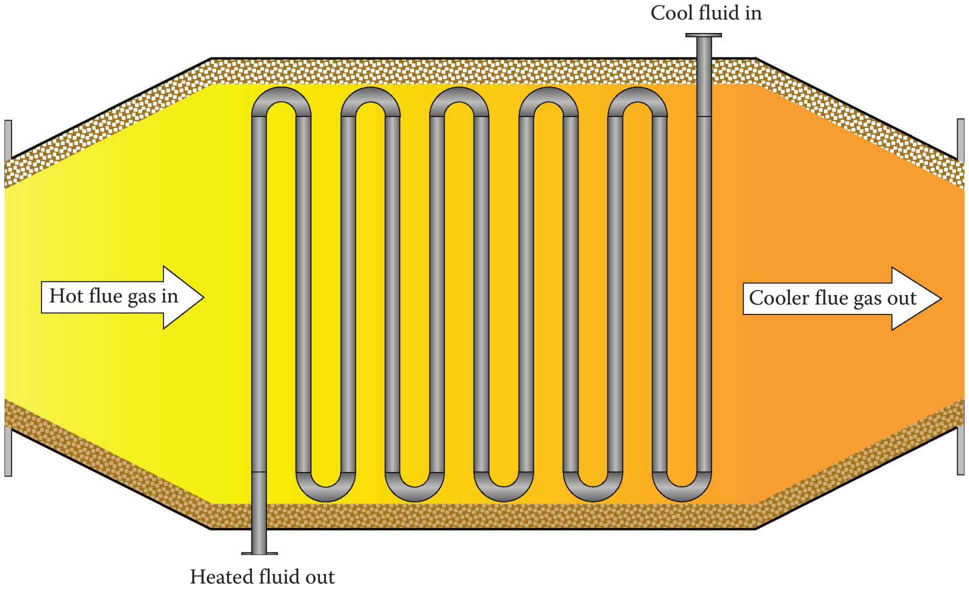


FIGURE 8.12
Organic fluid transfer system configuration.

gas must sometimes be altered before it can be further treated. Removing heat with heat-recovery devices (indirect conditioning) has already been discussed. As noted previously, removing heat reduces the temperature, but does not change the mass flow rate or composition. This section will review direct conditioning heat-transfer methods that reduce the temperature and also change the mass flow rate as a result of adding a cooling material to the flue gas to which heat is transferred.

The cooling medium that adds the least mass to the flue gas is water. Each pound of water sprayed into the flue gas absorbs almost 1000 Btu (1055 kJ) as it vaporizes (latent heat of vaporization) as well as in the form of sensible heat. For example, assuming that no other heat losses occur, cooling 10,000 lb/h (4,500 kg/h) of flue gas from 1800°F to 600°F (980°C to 320°C) requires about 2655 lb/h (1200 kg/h) of water at 70°F (21°C). Cooling the same amount of flue gas with 70°F (21°C) air requires about 26,190 lb/h (11,900 kg/h) of air, almost 10 times the cooling mass! If the flue gas was only cooled to 1200°F (650°C), the water required is about 1110 lb/h (500 kg/h) and the air needed is about 6150 lb/h (2800 kg/h), only about 5.5 times more mass. Minimizing the flue-gas flow to downstream equipment is normally desired, so the smallest size equipment with the lowest capital cost can be used.

When water is being used to quench the flue gases, all the water is expected to vaporize, so the conditioning section can be oriented in any direction. However, the vertical up or downflow configuration is usually utilized. This configuration also allows a hopper to be placed at the base of the unit, regardless of flow direction, to collect and remove water online if any of the removable spray tips fail to properly atomize the water. The hopper will also provide storage volume for particulate that disengages from the flue gas and does not exit the conditioning section.

If the purpose of the conditioning section is to cool flue gas to only 1200°F (650°C) to freeze molten particulate before the flue gas goes into a waste heat-recovery device, more heat can be recovered if air or recycle flue gas is used to provide the cooling instead of water. As noted in the previous section, although the flue-gas flow will be greater when using air or cooled recycled flue gas, than with water, and the downstream equipment size will be larger, increasing equipment cost, the value of the additional heat recovered will likely exceed the additional cost of larger equipment in just a few years. It should be noted that because of the higher flue-gas temperature, the entire conditioning section would have to be refractory lined.

If the flue gas is being quenched to a low temperature (400°F–600°F or 200°C–320°C), the vessel is usually made of carbon steel with internal refractory lining for part of its length at the hot inlet end and external insulation, but no refractory for the rest of its length. External insulation is used at the cooler end to prevent condensation in the cool unlined portion.

The hot flue gas is usually passed through a reduced diameter refractory-lined section to increase the velocity just before the atomized water is injected. This is done to improve mixing and heat transfer, which increases the evaporation rate of the atomized water droplets traveling through the conditioning section. The conditioning section's outlet temperature is continuously monitored and the water flow adjusted to maintain the desired temperature. Although the temperature of the conditioned flue gas may be well above the saturation temperature, if the flue gas has to be cooled to less than 400°F (200°C), the sheer amount of cooling water being injected may cause liquid water to come into contact with the particulate, make it damp, and cause it to stick to the outlet duct between the conditioning section and the dry particulate-removal device. In addition, the chance of condensation on the walls of the outlet duct and the dry particulate-removal device increases as the water content of the flue gas increases. To reduce the possibility of such problems, the flue gas is sometimes cooled the last 100°F–150°F (38°C–66°C) by the addition of ambient air.

Figure 8.13 is a general representation of a vertical, downflow conditioning section that can use water, air, etc., as the cooling medium.

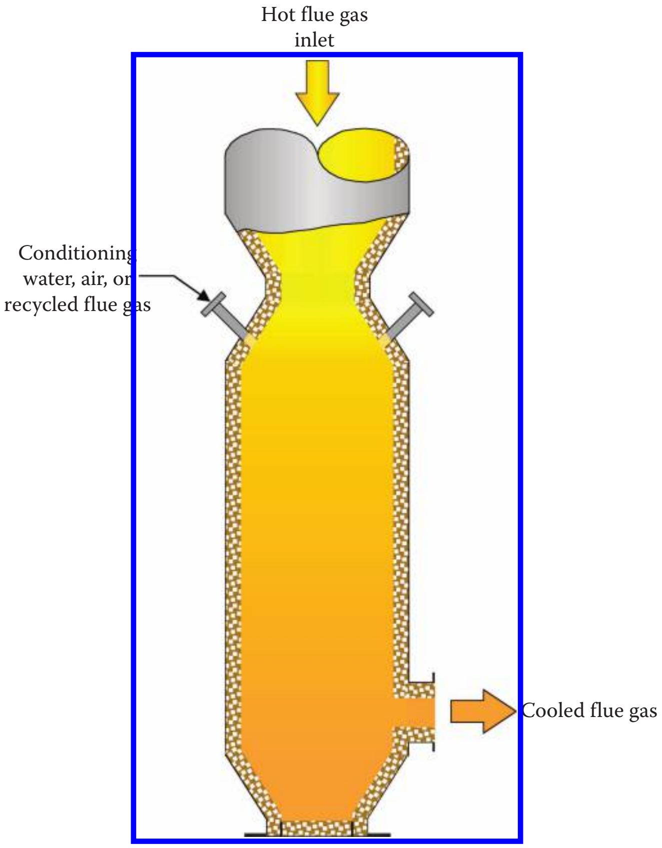


FIGURE 8.13
Vertical, downflow conditioning section.

8.3.5.2.2 Saturation Quench Section

If hot flue gas is to be treated by a wet particulate or acid-gas removal device, it is usually best to complete the heat-transfer task by quenching to full saturation before the subsequent mass-transfer process is initiated. Although this is not always done, most mass-transfer equipment is designed for fully saturated flue gas. The saturation temperature could typically be as high as 210°F (100°C) and usually is no less than about 135°F (57°C), depending on the composition and temperature of the flue gas when it enters the quench section.

Three basic saturation quench configuration options exist. They are (1) a direct spray contact quench section, (2) a submerged quench section, and (3) a combination of adjustable-plug-type quench and venturi scrubber section with an integral droplet separator. A description of the three options follows, including advantages and disadvantages of each. In each configuration, a steep temperature gradient will exist between the furnace exit gas and the quenched gas. As waste rates to the incinerator change, the location of this gradient will shift slightly, presenting a challenge to the hardware designer since radical temperature variations may eventually result in refractory damage. For this reason, a hot-to-cool water-cooled metal interface is often used in this area.

8.3.5.2.2.1 Direct Spray Contact Quench As shown in Figure 8.14, the hot flue gas flows downward through an annular overflow assembly (weir), then through a brick-lined water spray contact duct (contactor tube) fitted with guns spraying recirculated water, then through a downcomer tube (downcomer), and then into the water collection and droplet separator vessel (quench tank).

The weir is the hot-to-cool junction in the system. It is used to feed fresh (make up) water into the quench system and to form a wetted wall in the contact tube

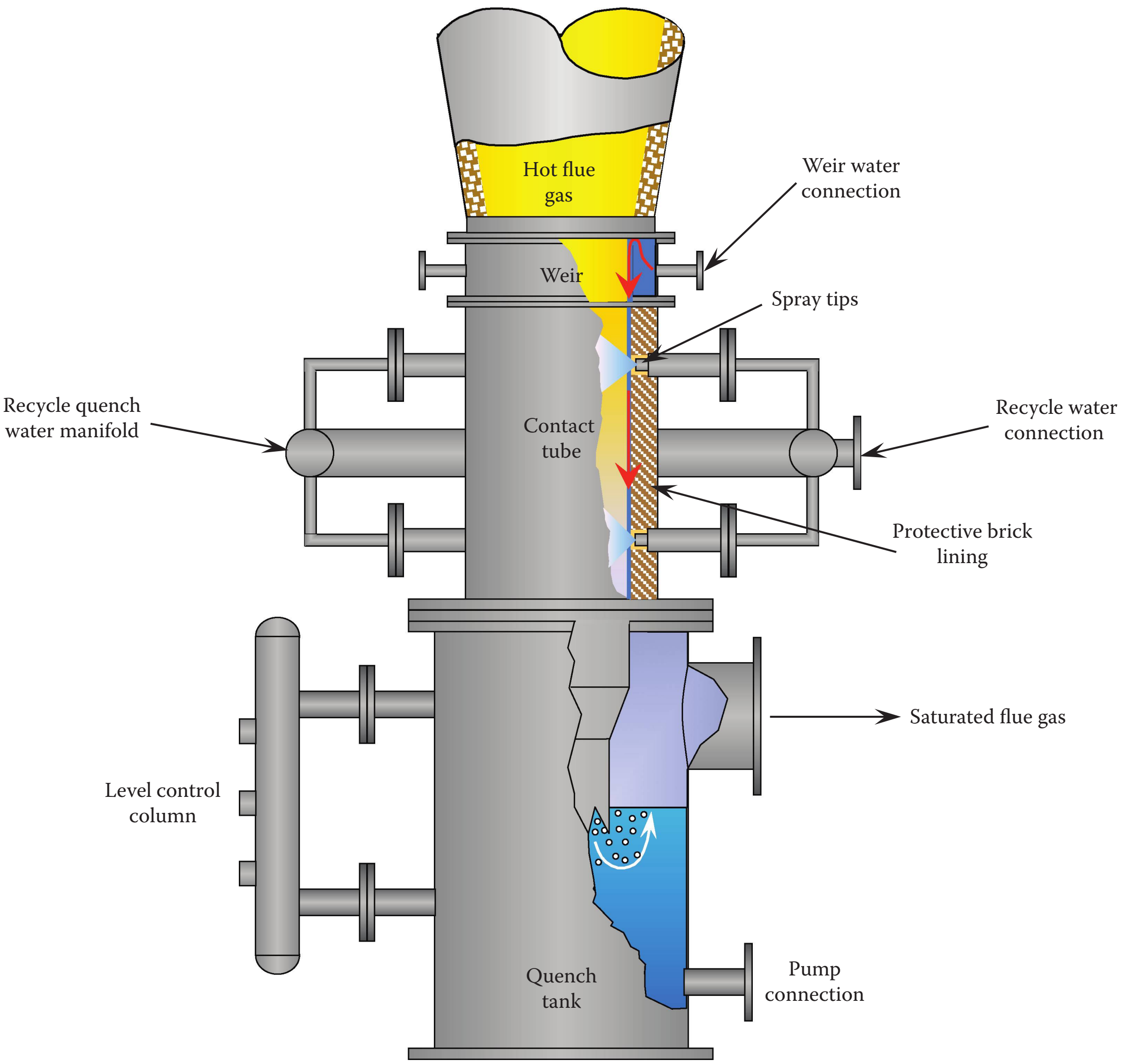


FIGURE 8.14
Direct spray contact quench.

and downcomer. The wetted wall provides a cooling film on the wall in the contact tube and downcomer. The weir is essentially an annular lip over which water flows, thereby ensuring uniform distribution of a curtain of water around the weir and the contact tube and downcomer. The material of construction for the weir is usually stainless steel. For more severe service, such as wastes containing chlorine compounds, a more corrosion-resistant metal such as Hastelloy, Zirconium, or Inconel may be required (see Volume 2, Chapter 4).

The contactor tube, located directly under the weir, is a short, brick-lined, spray tower into which a large quantity of recycled water (approximately four times the calculated amount needed for saturation) is injected. The spray guns must provide efficient atomization and penetration across the full cross section of the contactor tube if quenching is to be effective and virtually complete saturation of flue gases is to be achieved. The high-temperature flue gas is cooled, primarily through vaporization of atomized recycle water, to near the adiabatic saturation temperature. If the flue gas contains particulate, some will drop out in the quench tank. The recycle water may absorb some acid gas as long as the equilibrium concentration of acid in the water is less than in the flue gas. The contactor-tube shell can be constructed of carbon steel with a corrosion barrier, special corrosion-resistant alloys, or fiberglass reinforced plastic (FRP).

After the nearly saturated flue gas and extra water exit the contactor tube, they pass through the downcomer, which extends down to the water level in the quench tank. The downcomer is smaller in diameter than the contactor tube, thus increasing the velocity and mixing energy of the flue gas and atomized recycle water to complete the cooling and saturation of the flue gas. The flue gas makes a 180° direction change at the downcomer outlet as it turns to travel upward in the annular space between the downcomer and the quench tank wall to the quench tank outlet. The higher velocity through the downcomer and the subsequent 180° direction change along with the low velocity in the annulus allow water droplets to disengage from the flue gas and drop to the bottom of the quench tank. The downcomer should be constructed of a corrosion-resistant metal such as stainless steel or other more exotic alloys as required for the service. The quench tank can be made of carbon steel with a corrosion barrier, special corrosion-resistant alloys, or FRP.

Although the primary purpose of the quench tank is as the collection sump for recycling of water, it also captures some of the particulate coming from the TO. Thus, if necessary, the configuration of the quench tank must allow for continuous removal of solids and continuous withdrawal of recycle water. To accomplish this, the base of the quench tank should be conical, with a solids blowdown nozzle at the base and the suction point for the recycle pump(s) in the conical section. Furthermore, the recycle line should be equipped with a dual-basket strainer upstream of the pump(s) and a cyclone separator (hydroclone) downstream of the strainer. The strainer openings are sized to remove only large pieces from the recycle water that might damage the pumps, while the hydroclone removes the smaller pieces that might plug the atomizing spray tips. A continuous blowdown from the hydroclone carries out the collected solids, but the strainer must be manually cleaned periodically. The blowdown from the base of the conical section can be automatic or manual.

The advantages of the direct spray contact quench are (1) faster, more efficient cooling of the flue gas as a result of the large heat-transfer surface area of the atomized water; (2) very low pressure drop across the entire section, usually less than 2 in. W.C. (0.5 kPa); (3) efficient water droplet separation from the flue gas; and (4) large open-flow area unlikely to be affected by any obstruction.

The disadvantages include loss of cooling flow as a result of pump failure and potential plugging of the spray tips. However, if a hydroclone separator is used, plugging is minimized. Also, utilizing two recycle pumps with auto-start on the spare when flow drops below a minimum greatly reduces the potential loss of flow.

8.3.5.2.2.2 Submerged Quench The submerged quench, shown in Figure 8.15, is similar mechanically to the direct spray design, but instead of atomizing water to provide a large contact surface area between the water and the hot flue gas, this method divides the hot-gas flow and "bubbles" it through the water. To accomplish this, the hot flue gas enters the quench section traveling downward through an annular overflow section (weir), directly into a downcomer tube (downcomer) that extends several feet below the liquid surface in the separator vessel (quench tank). The lower, cylindrical portion of the downcomer contains a number of smaller holes through which the hot gas exits the downcomer. As the flue gas bubbles upward through the water, cooling occurs. Often, an additional droplet separator vessel is used. The total pressure drop across this section is usually between 24 and 30 in. W.C. (6 and 7.5 kPa).

The weir is again the hot-to-cool interface in the system. It is used to feed fresh (make up) water into the quench system and, unlike the spray quench, is also used where the large volume of recirculated water from the quench tank is introduced. All the water flows down the wall of the downcomer, flooding, and therefore, cooling the downcomer wall. The material of construction for the weir is usually stainless steel for nonacid-gas service and Hastelloy, Zirconium, or Inconel for acid-gas service, such as chlorinated hydrocarbon service.

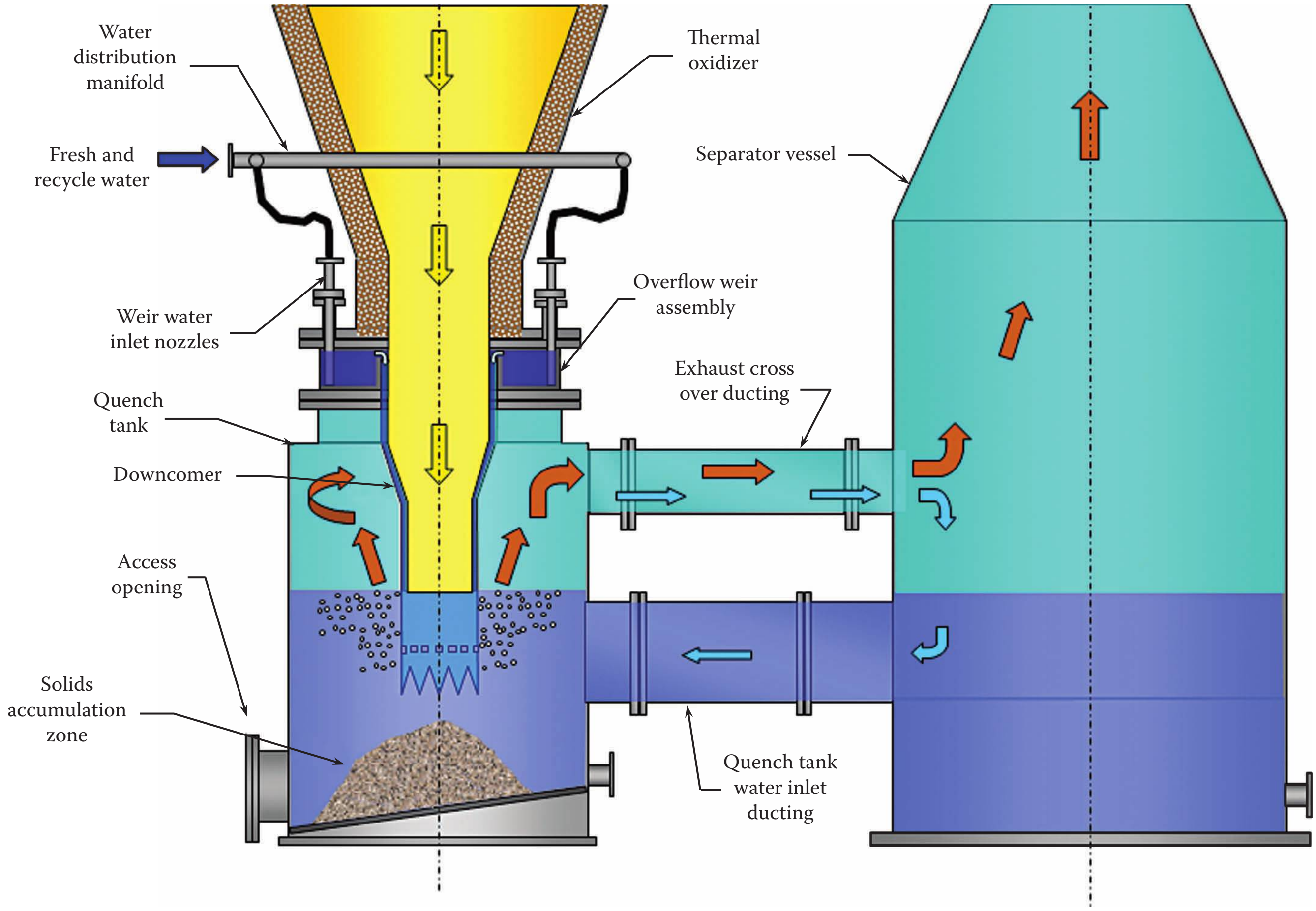


FIGURE 8.15
Submerged quench.

The downcomer is usually an extension of the weir for this process. It is open ended and extends several feet below the water level in the quench tank. Although the downcomer is open at the bottom, the flue gas actually passes through holes cut in the circumference of the lower portion of the downcomer, well below the water surface, but above the open end. As the flue gas passes through the openings, it is distributed evenly and separated into smaller volumes that further divide into smaller bubbles providing the contact surface area needed to quickly evaporate water and quench the flue gas to saturation temperature. The openings must be properly sized and located at the proper depth below the surface for the downcomer to function properly. The downcomer is fabricated from the same material as the weir.

The quench tank serves as the source of the water for quenching. Although a significant amount of water is entering the vessel through the downcomer, the internal design of the quench tank insures proper internal recirculation of the water and prevention of excessive vaporization of the water in any location. The bubbling flue gas creates significant turbulence and splashing at the surface making droplet separation in the quench tank more difficult, and often a second vessel is used as a separator to remove droplets before the cooled flue gas travels to the next flue-gas treatment section. The separated water usually flows back into the quench tank by gravity from the separator. As with the direct spray quench, if particulate enters the quench section, some will remain in the quench tank and some of the acid gas will be absorbed by the recycle water. The configuration of the base of the quench tank and recycle pumping system components (strainer and hydroclone) should be similar to that of the direct spray quench to allow for continuous removal of solids from the bottom of the vessel as well as continuous withdrawal of particulate from the recycle water to prevent filling the overflow weir with solids. The quench tank (and separator, if used) can be made of carbon steel with corrosion barrier, special corrosion-resistant alloys, or FRP, depending on the service.

The primary advantages of the submerged quench are that (1) no spray tips are required, (2) large flow area is unlikely to be affected by obstructions, and (3) even with pump failure and loss of recirculation water, little or no downstream equipment damage is likely to occur because the high liquid level in the quench tank will prevent hot gas from traveling downstream.

The disadvantages include (1) high pressure drop across the section and (2) often, another vessel that must be used for effective droplet separation.

8.3.5.2.2.3 Adjustable-Plug Venturi Quench This downward flow–oriented device, shown in Figure 8.16, combines both heat transfer and mass transfer. The upper portion of this one-piece section utilizes the familiar hot-to-cool interface, an annular overflow weir, followed by a short converging inlet tube, then the adjustable-plug/throat section and an outlet tube. After passing through the adjustable-plug venturi, the quenched flue gas and the water then travel through a separator/quench tank and onto the next section.

Once again, a weir is the hot-to-cool interface in the system. It is used to feed fresh (make up) water into the quench system and recirculate some of the needed large volume of recycle water.

A converging duct, wetted by the weir overflow, provides an entrance for the flue gas to the throat section. A tapered, vertically adjustable plug varies the throat area, in order to maintain the gas velocity and pressure drop across the section relatively constant in spite of changes in the flow rate of the flue gas. A large volume of recycle water is added to the center of the throat through a pipe that directs the water onto the center of the adjustable plug. The water and the flue gas pass through the annular space between the plug and the throat wall. The high velocity (up to 500 ft/s = 150 m/s) at the throat provides the energy needed to atomize and mix the water with the flue gas, cooling the flue gas to saturation. The downstream of the throat, a 90° elbow, directs the quenched flue gas and water to a separator/quench tank. A shaft extends from the plug, through the bottom of the elbow, and is attached to an actuator, which is used to automatically adjust the position of the plug to maintain a set pressure drop regardless of flow variation. The whole venturi assembly can be fabricated using stainless steel, acid-resistant metals, or in some cases, a combination of FRP and metals.

The separator/quench tank usually has a tangential inlet to help separate the droplets and water more effectively from the saturated flue gas. Recycled water and solids removed from the flue gas collect in the separator/quench tank, so vessel configuration and recycle pumping configuration should also be very similar to that of the two previously described quench processes.

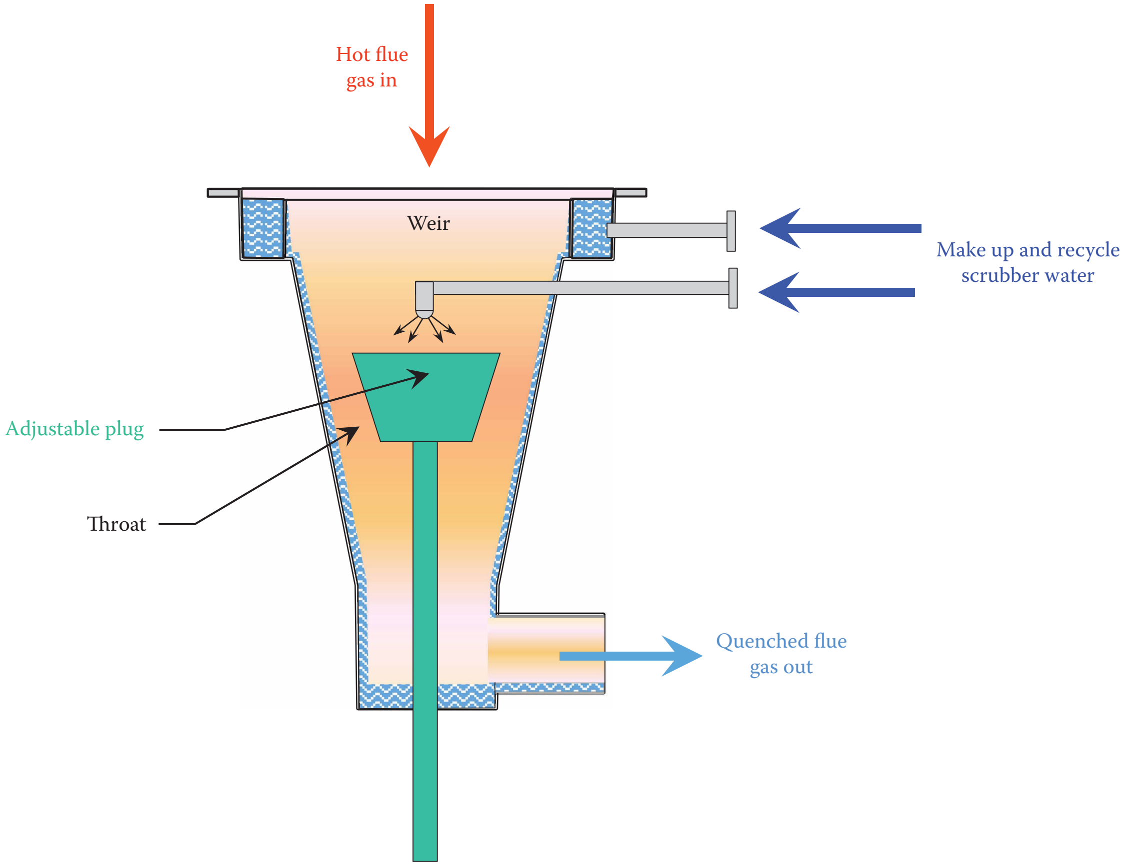


FIGURE 8.16
Adjustable-plug venturi quench.

The separator/quench tank can be built of the same materials noted earlier.

The venturi quench is more effective at wetting the particulate in the flue gas than the two previously described methods, making it the preferred approach when particle removal is required or desired, especially if the particulate is relatively large and easy to remove. Greater acid-gas absorption is also more likely with this method of quench because of the extra mixing energy at the throat.

The total pressure drop across this section could vary between 20 and 70 in. W.C. (0.5 and 1.8 kPa), depending on whether the device is being used primarily as a quench or if it is being used for particulate removal also.

The advantages of the adjustable-plug venturi quench are that (1) quenching and scrubbing in one section reduces plot space and (2) no spray tips are required.

The disadvantages include (1) higher pressure drop than other quenching methods; (2) loss of cooling water flow as a result of pump failure; (3) more complexity of the design including the moving plug and shaft, with a shaft seal that must be maintained; and (4) the solids removal efficiency of the combination quench/venturi that may not be sufficient to meet the particulate emission limit and a second particulate-removal device that may be needed, adding to the system pressure drop. A potential disadvantage is that the pieces of refractory material falling from the refractory-lined duct or TO above the venturi could block the annular space in the venturi section. However, this form of blocking is not as likely if the adjustable plug is operated in the automatic mode to maintain a preset pressure drop. This is because blocked flow area would cause an increase in pressure drop above the set point, resulting in the plug being moved upward to open the annular space allowing most pieces to flush through the throat. On the other hand, very large pieces will not pass through the plug/throat annulus as they would with the direct spray or submerged quench designs, making this a comparative disadvantage for the venturi quench.

8.3.5.3 Particulate/Acid-Gas Removal

The equipment used to remove pollutants from TO system flue-gas streams are known as air pollution control (APC) devices. The most common pollutants that require removal after burning liquid and gaseous wastes are particulate matter and acid gases such as SO_X and HCl. The most commonly used devices for the removal of particulate matter are dry-removal devices such as baghouses and ESPs, and the common devices for acid-gas removal are wet-removal devices such as venturi scrubbers and packed columns. As noted before, particulate removal can also be accomplished in venturi scrubbers.

8.3.5.3.1 Particulate Removal: Dry

The most common methods of dry particulate removal in the combustion-related area are filtration and electrostatic collection. Reasonable, cost-effective efficiencies are achieved even when submicron particles must be removed. High-pressure-drop cyclonic separation is also available, but rarely used with combustion systems.

Dry particulate recovery is often preferred if the material recovered is to be reutilized. If a wet process is used to recover catalyst particles, the particles have to be separated from the water and dried before any purification or refining process can begin. Dry removal minimizes the volume and weight of the material that must be handled after recovery. Much less water is used in the dry-removal process, making dry removal more attractive in locations where water is scarce or expensive. For a given removal efficiency, dry removal requires much less pressure drop than wet removal.

8.3.5.3.1.1 Filtering Device (Baghouse) A common method of dry particle recovery is by collection on the surface of fabric bags. The principal design parameters for a baghouse (see Figure 8.17), assuming a particle-size distribution is known, are the fabric type, the air-to-cloth ratio, and the cleaning method used. The gas to be cleaned enters the baghouse, flows through the bags from the outside (depositing the particles on the outside surface of the bags), and flows inside the bags up to the clean gas plenum and out to a stack or to another treatment device. Many bag fabrics are available. The fabric weave is tight enough that some of the particles are initially captured in, or on, the surface. Once a base coat of particulate (filter cake) has been collected, the even finer filter media than the original fabric now exist, allowing high efficiency capture of even small particles (up to 99.5% for 0.1 μm particles and up to 99.99% for 1–10 μm particles) with a relatively low pressure drop of about 6 in. (15 cm) W.C. As the particulate accumulates, the pressure drop increases and eventually the bags must be cleaned. The commonly used cleaning methods are pulse jet, shaker, or flow reversal. The dislodged dust falls to the bottom of the baghouse and is removed during operation through special valves.

None of the baghouse design parameters are independent, and all are based on testing and previous experience. Fabrics may be woven or felted and can be made from a variety of materials. Fabrics differ in their particle capture efficiencies, corrosion resistance, erosion resistance, temperature range, pressure drop, strength, durability, and ease with which they may be cleaned. The fabric is chosen based on the exhaust gas conditions, the type and size distribution of the particles to be filtered, the particle loading, and the cleaning method.

A typical bag used in a TO system would be a 6 in. (15 cm) diameter by 10 ft (3 m) long cylinder closed at

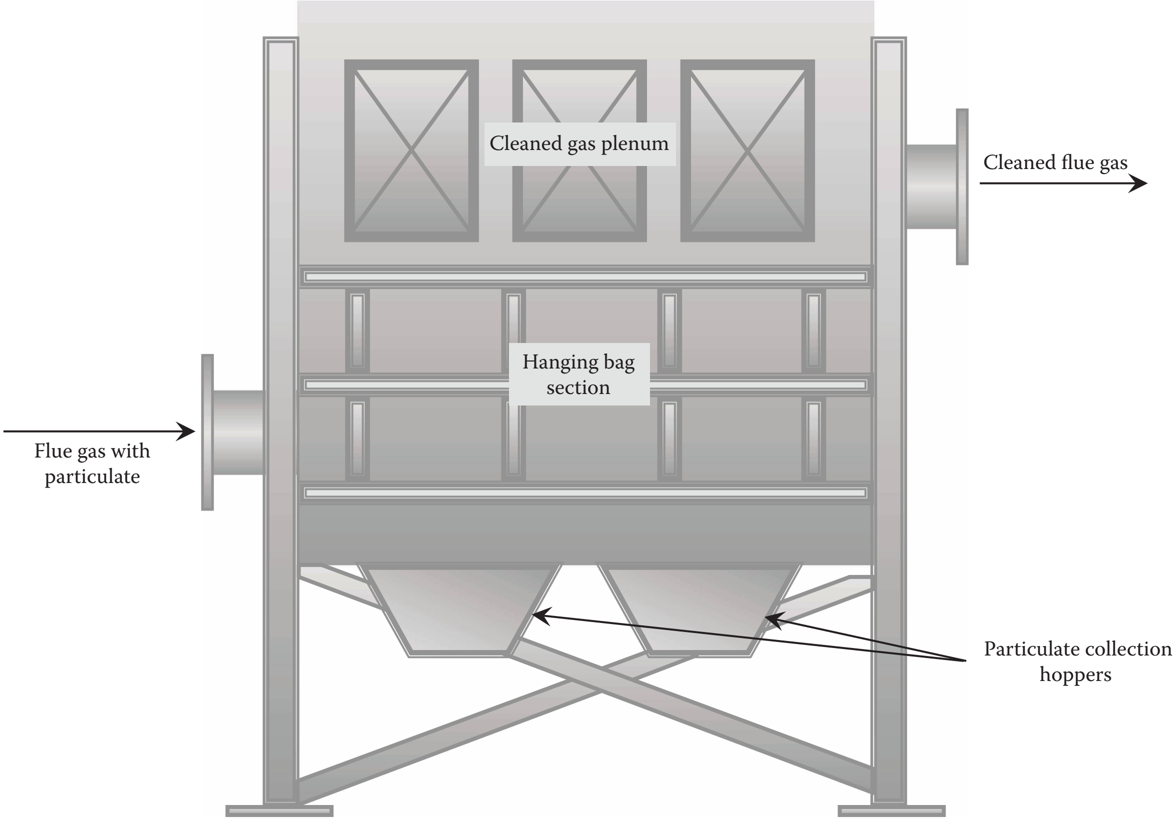


FIGURE 8.17
Baghouse.

the bottom, supported by an internal wire cage and suspended from a tube sheet forming the top of the dirty gas chamber. The number of bags is set by the desired gas/cloth ratio, which usually falls between 2 and 5 ACFM/ft^2 (0.6 and 1.5 ACMM/m^2).

The choice of a cleaning method (pulse jet, shaker, or flow reversal) may be determined by the fabric strength and durability, but is often a compromise between capital and operating expenses. The pulse-jet technology has relatively high energy requirements, but can usually operate at higher filtration rates, thus requiring a smaller filter area. Conversely, the shaker and the flow reversal technologies typically have lower energy requirements and lower filtration rates leading to larger filter area. Pulse jet is the cleaning method normally used for TO systems. To minimize air usage, all bags are not cleaned simultaneously. Also, care must be taken to avoid excessive cleaning that removes the filter cake, reducing collection effectiveness until it builds up again.

Baghouses are sometimes compartmentalized so that valves can isolate the compartment or the section of the baghouse being cleaned (offline cleaning). While this action requires increasing the number of bags by 50% and adding inlet and outlet manifolding and valves, it does reduce the amount of material being drawn back onto the filter immediately after cleaning compared to when cleaning is done online and increases the period of time needed between each cleaning operation.

In general, baghouses are only used in applications where the gas is dry and the temperature is below 450°F (230°C). A typical TO system application would utilize a pulse-jet baghouse operated at 400°F (200°C) with a Nomex fabric designed for an air-to-cloth ratio of between 2 and 5 ACFM/ft^2 (0.6 and 1.5 ACMM/m^2) and a pressure drop of 5–10 in. W.C. (1.3–2.5 kPa).

Baghouses are usually used in nonacid-gas service and are therefore usually constructed of carbon steel. For non-halogen acid-gas service, stainless steel has also been used. Any material can be used as long as it can withstand the occasional short-term temperature excursion to more than twice the design flue-gas temperature. Bag filter material can be polyester, polyaramid, cellulose, fiberglass, Nomex, Gore-Tex, or any other proven fiber that meets the operational criteria. The most commonly used baghouse filter materials are polyester (275°F = 135°C [continuous operating], 300°F = 150°C [temporary max.]), Nomex (400°F = 200°C [continuous operating], 425°F = 220°C [temporary max.]), and Teflon (475°F = 250°C [continuous operating], 500°F = 260°C [temporary max.]), in ascending order of cost.

The advantages of using a baghouse are that (1) the particle removal efficiency is high, (2) the pressure drop to collect the material is low, (3) the material is collected "dry" and does not have to be separated from water, and (4) although it contains some moisture, basically the recovered material is at its final volume and weight.

The disadvantages are that (1) the baghouse is relatively expensive and occupies a lot of plot space, (2) field construction may be required for large flue-gas flows, (3) the large baghouse surface area requires extensive insulation to minimize acid-gas or moisture condensation, and (4) the maximum treatable flue-gas temperature is about 450°F (230°C).

8.3.5.3.1.2 Electrostatic Precipitator An ESP is a device that removes particles from a gas stream by means of electrostatic attraction. A high voltage potential, usually applied to weighted vertical hanging wires (emitters), causes the particles to be charged. Once charged, the particles are exposed to grounded collecting electrodes (plates) to which the particles are attracted, separating (precipitating) them from the flue gas. The particles must then be separated from the collecting plate, while minimizing re-entrainment, and removed from the ESP collection hopper.

An ESP is a large, often rectangular-shaped, chamber containing numerous flat parallel collecting plates with emitter wires located midway between the plates (see Figure 8.18). The flue gas entering the ESP must be uniformly distributed across the chamber for effective treatment. As the flue gas passes between the plates, the high voltage potential, 40–50 kV, carried by the emitter wires creates a corona discharge, making a large number of both positive and negative gas ions. The positive ions are attracted to the negatively charged emitter wires leaving the space between the plates rich in negative ions. The particles passing through the negative ion-rich space quickly acquire a negative charge. Smaller particles are, however, more difficult to charge.

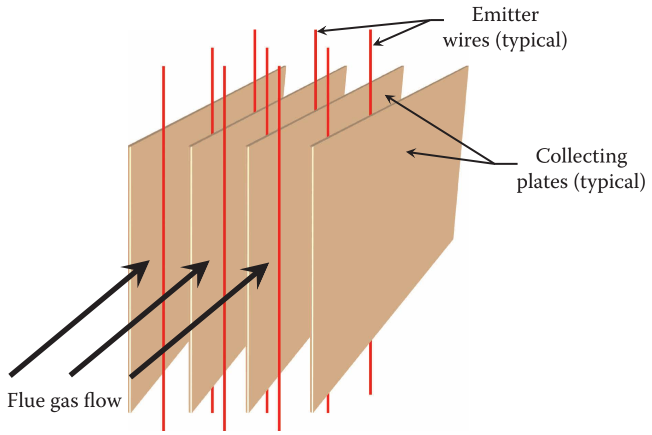


FIGURE 8.18
Dry ESP.

NOTE: Negative discharge electrodes are normally used for industrial ESPs because of the higher potentials available and more predictable performance.

The force that moves the particles to the collection plates results from the charge on the particles and the strength of the electric field between the emitters and the collectors. For smaller particles, the electric field strength must be greater to remove the same percentage of the total. The force can be up to several thousand times the acceleration of gravity, so the particles move rapidly to the collection surfaces. Because most of the particles retain a portion of their negative charge even after contacting the collecting plate, some remain on the plates until a physical action dislodges them.

The weight of the particulate on the plates causes some of the particulate to fall into the collection hoppers, but the rest has to be dislodged by vibration (or rapping) of the plates and emitters. Once collected in the hoppers at the bottom of the chamber, the particulate is removed by the same methods used to remove particulate from baghouses.

Design considerations include the following:

1. The electric characteristics of the particulate, that is, how well the material will accept and hold a charge (the particles must have a resistivity in the range of 10^4–10^{10} Ω cm for efficient removal by electrostatic means)
2. The gas and particle velocity (very important with submicron particulate), including gas velocity in the unit, drift velocity of the particulate induced by the electric field, and particle settling velocity
3. The gas distribution
4. The electric sectioning, that is, the increase in power input in sequential zones or cells through the length of the ESP to achieve the desired removal efficiency
5. Particle re-entrainment

The flue gas must also contain readily ionizable species such as O_2, CO_2, and SO_2. Particle resistivity can be a strong function of the flue-gas temperature, composition, and moisture content. Thus, ESP performance can be quite sensitive to changes in upstream process conditions.

As with the baghouse, most of the design parameters for ESPs have been developed empirically and then fit to equations to equip engineers to develop the physical equipment design. The result of a good design is a chamber containing the correct number and size (length and width) of collector plates spaced appropriately to allow locating emitters between the plates. The capital cost of the ESP is directly proportional to its physical dimensions. Both sides of each collector plate function as collecting surface area (CSA), which is also referred to as specific collector area (SCA) or specific collector surface (SCS). This area is often expressed in terms of surface area per 1000 ACFM (actual cubic feet per minute) of flue gas through the ESP. Depending on the particle-size distribution and other particle-related parameters, the removal efficiencies of more than 99% can be achieved at less than 300 ft^2 (28 m^3) of collector area per 1000 ACFM (28 ACMM) of flue gas. For very high particulate loading, upwards of 900 ft^2 (84 m^3) of collector area per 1000 ACFM (28 ACMM) of flue gas may be necessary.

Since the temperature of the flue gas entering the ESP is normally greater than 650°F (340°C), corrosion is not usually a problem if the exterior is well insulated and sealed from rainwater intrusion. Therefore, the materials of construction would be carbon steel or stainless steel for the emitters and carbon steel for the plates and casing.

The advantages of using an ESP include (1) particulate discharged in dry form, (2) relatively low energy usage, (3) very low pressure drop, and (4) efficient small particle removal.

The disadvantages are (1) relatively high capital cost; (2) relatively large space requirements; (3) field construction that may be required for units with large flows; (4) large surface area that encourages heat loss, leading to potential acid-gas condensation problems with certain wastes; and (5) multiple stages that may have to be used to achieve high removal efficiency.

8.3.5.3.2 *Particulate Removal: Wet*

The most common methods of wet particulate removal utilized in TO systems are venturi-type devices and wet ESPs (WESP). Reasonable removal can be achieved with the simple venturi-type device, if particles are larger than one micron. For smaller particles, multistage venturi-type devices with subcooling can be effective at high pressure drops. To treat a flue gas containing a high percentage of very small particles, the WESP is the better choice. The pressure drop is also much less than with a venturi-type device.

8.3.5.3.2.1 Venturi-Type Scrubber The venturi-type particulate scrubber is very similar to the venturi quench described in the previous section of this chapter, that is, a constricted passage or "throat" that increases flue-gas velocity to achieve a desired pressure drop. Flue-gas pressure drops range from 20 to 100 in. W.C. (5 kPa to 25 kPa) depending on various factors. If the flow rate varies significantly, the cross-sectional area of the throat must be adjusted to maintain the necessary velocity (pressure drop). Another approach to maintaining a relatively constant pressure drop through the venturi is to use a fixed throat, but increase or decrease recycled flue-gas flow to maintain a constant flow rate.

Liquid is injected either in the throat or just upstream of the throat. Typical liquid injection rates are in the range of 10 gal/1000 ft^3 (1.34 L/m^3) of gas. Because of the high scrubbing water flow rate, venturi scrubbers require a liquid recirculation system. The recirculation system will include equipment to control the blowdown rate, in order to maintain the total (suspended and dissolved) solids content of the recycle water at about 5% by weight. Additional equipment may also be utilized to cool the recirculated water and control pH.

When venturi scrubbers are operated at saturation with relatively high pressure drop, typical particle removal efficiencies are >90% for particles with aerodynamic diameters of ≥1 μm and ±50% for particles with aerodynamic diameters of 0.5–1.0 μm. Overall particle collection efficiency obviously depends on the particle-size distribution and will range from 80% to above 99%. For a properly designed, subcooled system with efficient droplet separation, the overall particle collection efficiency should be ≥99%.

The basic principle of operation for the venturi-type particulate-scrubbing device is to provide small water droplets that will wet and surround the particulate matter with water by inertial impaction. Given the time, the small water-encapsulated particle droplets will then agglomerate (i.e., droplets contact other droplets and combine to form larger droplets). The larger droplets can be separated from the flue gas, downstream of the venturi, by a cyclonic separator, a mist eliminator, a settling chamber, or a combination of two or all three separation methods. Assuming the flue gas is fully quenched to saturation, the overall removal efficiency depends on how effectively the particles are wetted, how much droplet agglomeration time is allowed, and how effectively the larger droplets are separated from the flue gas. As with other types of particulate-removal devices, many of the design parameters have been developed from empirical data.

The inertial impaction of the particle into the droplet is the dominant mechanism for removal of larger particles with an aerodynamic diameter greater than 1 μm. Two important factors determine the effectiveness of capture: (1) the relative velocity difference between the particle and the water droplet and (2) the droplet diameter. A greater relative velocity difference improves impaction effectiveness. Similarly, a smaller water droplet also improves impaction effectiveness.

The relative velocity difference between the particles and the liquid droplets increases with higher flue-gas pressure drop (i.e., increased energy consumption) that increases the velocity of the flue gas and the particles suspended in it. Smaller scrubbing water droplets can be produced by mechanically atomizing the water or by using an atomizing media.

The shearing effect of the high velocity flue gas also atomizes the water more finely.

If the flue gas is not fully saturated with water vapor, scrubbing fluid will be vaporized until saturation is achieved and removal performance will be poorer. The vaporization of the scrubbing fluid causes problems such as reducing the amount of scrubbing fluid available for particle removal, forming new particles from the previously captured particulate matter as the droplet evaporates, and reducing the diffusive capture because of a net flow of gas away from the evaporating particle.

The vertical, downflow adjustable-plug venturi described previously utilizes mostly pressure drop to shear/atomize the water and to provide the relative velocity difference. Other types use a combination of pressure drop and water atomization to achieve the velocity difference and smaller droplet size. Figure 8.19 shows a horizontal venturi scrubber with atomized water injection upstream of the throat.

Although particles larger than one micron are easily removed, smaller, submicron size, particles are much more difficult to capture. The capture effectiveness can be enhanced by (1) subcooling saturated flue gas to below the saturation temperature and (2) using colder water for scrubbing. The purpose is to take advantage of thermophoresis and diffusiophoresis effects to produce a directional preference in the Brownian motion toward the target droplet by these submicron particles.

Basically, thermophoresis is the migration of a particle away from a higher temperature zone and toward a lower-temperature zone.

Diffusiophoresis is a more complicated phenomenon that occurs when a mixture of particles of varying weight exists in a gas stream and a concentration gradient within the gas stream occurs for the heavier particles. The diffusion of the heavier particles from the higher concentration zone to the lower concentration zone occurs in accordance with Ficke's law. The net motion of the lighter particles is also altered toward the low concentration zone due to the momentum imparted during collisions with the heavier particles traveling in that direction. By injecting colder water and subcooling the flue gas below saturation temperature, a temperature gradient and a water vapor concentration gradient are created resulting in a net increase in particle motion toward the target water droplets.[2,3]

The advantages of the venturi scrubber include (1) reasonably good particulate-removal performance, (2) relatively simple design, (3) usually lower capital cost, and (4) relatively low plot space requirement.

The disadvantages are (1) high energy consumption, that is, high flue-gas pressure drop; (2) high removal efficiency for small particles that requires additional cost of subcooling; (3) continuous blowdown to maintain low total solids content that is undesirable if water is scarce; and (4) the effluent that contains a large volume of water, which has to be treated.

8.3.5.3.2.2 Wet Electrostatic Precipitator A wet ESP (WESP) functions very similarly to the previously described dry ESP. It removes particles from a flue-gas stream by means of electrostatic attraction. As before, particles

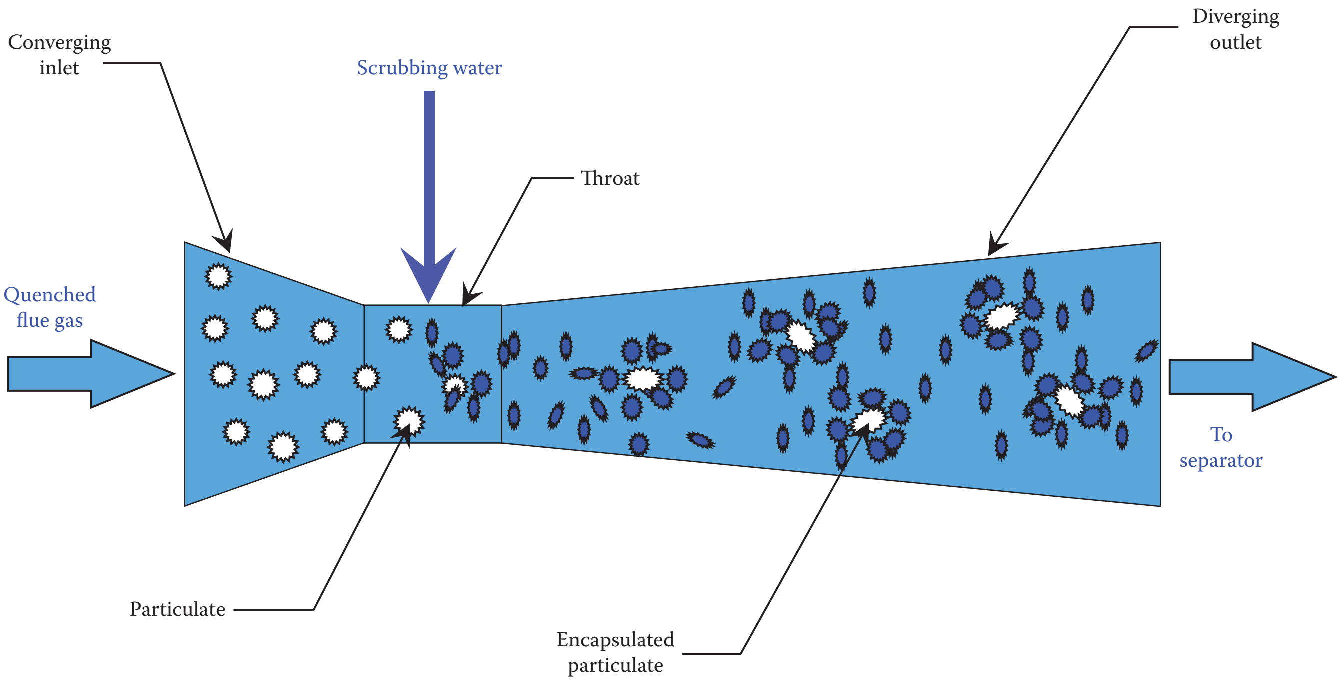


FIGURE 8.19
Horizontal venturi scrubber.

first must be charged, and as in the dry ESP, vertically oriented emitter wires are used to generate the corona. Once charged, the particles are drawn to the oppositely charged, grounded collecting electrodes, which for the WESP are tubes through which the emitter wires hang. The particulate collects on the inside surface of the tubes. The particles are then removed from the collection surface while minimizing re-entrainment. The primary process difference is that the flue gas must be saturated when it enters the emitter/collector section.

Flue gas coming into the WESP first enters a chamber under the vertically oriented collector tube section (see Figure 8.20). The saturated flue gas is uniformly distributed before reaching the tubes. The collector tube section is similar in construction to a shell-and-tube heat exchanger in that it has an inlet and outlet tube sheet and a sealed casing (shell) around the tubes. The emitter wires are positioned at the centerline of each tube. Water at ambient temperature is circulated through the shell side of the collector tube section, ensuring that the collector tube temperature is less than the temperature of the saturated flue gas. This causes water vapor in the flue gas to condense on the inside of the tubes and flow downward into the chamber under the collector tube section. As the wetted particles are charged and move to the tube wall, they are washed down the tube by the condensing water. Occasionally, the power is shut off to a part of the collector tube section, and water is sprayed downward through that section of tubes to further clean them. The distribution section in the inlet chamber can also be designed like a packed column in order to absorb acid gases.

The design criteria for emitting and collecting electrode areas in a wet ESP are similar to those for the dry ESP. However, the potential for sparking/arcing is greater in the WESP, so proper spacing between emitters and tubes, or anything else, is very important.

Because the WESP is wet, the materials of construction would be stainless steel or Hastelloy for emitters and tubes and stainless steel or FRP for the housing and water collection section.

The advantages of the WESP are (1) high removal efficiency for submicron particles; (2) low gas-side pressure drop, usually less than 6 in. W.C. (1.5 kPa) (normally about 4 in. W.C. = 1 kPa) if the distribution section is designed to absorb acid gases as well as distribute flow; (3) acid-gas absorption section that can be built-in; (4) "cool" low operating temperature; and (5) less problems with re-entrainment because the wet particles stick to the tube wall.

The disadvantages include (1) greater capital cost than most other wet-removal devices, (2) larger plot space required, (3) more complicated operation than with other equipment, (4) multiple stages that may have to be used to achieve extremely low particulate emission levels if the particulate loading is high, and (5) more maintenance that may be required due to corrosion and complexity.

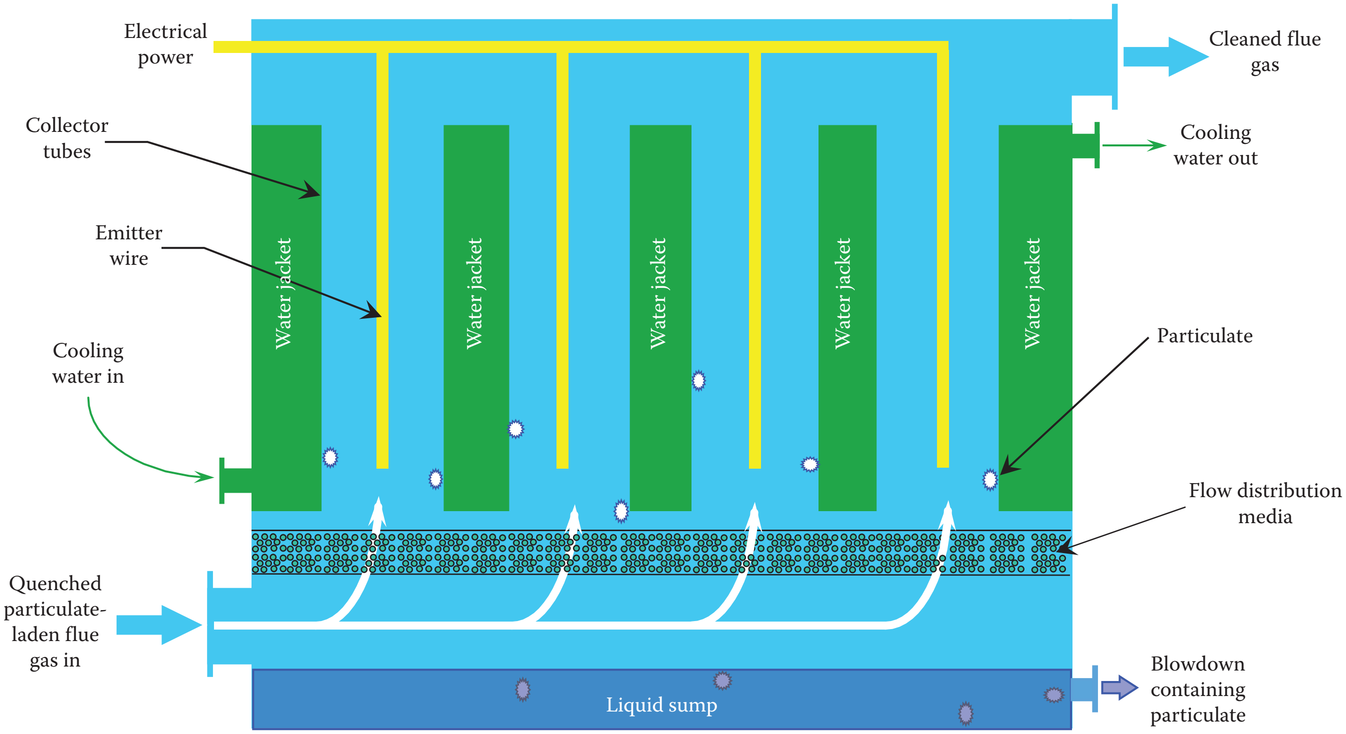


FIGURE 8.20
Wet ESP.

8.3.5.3.3 Acid-Gas Removal: Dry

Acid gases can be removed from flue gas by reaction with or adsorption by dry alkaline materials such as limestone/lime. This method of acid-gas removal is commonly used on large power boiler applications that have a high flue-gas volume. For most TO systems, dry removal is not cost-effective and does not achieve the removal efficiency needed for the amount of combustion products generated. Some of the drawbacks are that (1) the reactant/adsorbent has to be disposed of after use, thereby creating a solid waste; (2) the removal efficiency is not high; and (3) the capital cost of equipment needed for dry injection is quite high. For this reason, no detailed description of dry systems will be covered in this chapter.

8.3.5.3.4 Acid-Gas Removal: Wet

The most common methods of wet acid-gas removal used in TO systems are packed columns and the previously described venturi-type devices. Packed columns contain packing material that distributes the water over a large surface area for contact with the flue gas. Venturi-type devices utilize many small water droplets to provide the large amount of liquid surface area for contact by flue gas. Each has its advantages and disadvantages.

8.3.5.3.4.1 Packed Column A packed column is the device of choice that is most often used to recover or remove acid gases from a flue-gas stream. The device consists of a vertical, usually cylindrical, vessel containing a section filled with packing material supported by internal hardware. Recirculated water is pumped to the top of the packed section and flows downward through the packing and collects in the base of the vessel. Flue gas, quenched to its saturation temperature, enters the vessel under the packed section and flows upward through the packing and out to the atmosphere or to another treatment section. Figure 8.21 shows a simple packed column.

The purpose of the packed column is to transfer compounds from the flue gas to an absorbing liquid. For combustion systems, the compounds are acid gases such as HCl or SO_2. The absorbing liquid is typically water or a weak caustic solution. As the liquid flows downward through the randomly placed packing, it is distributed over the large amount of packing surface.

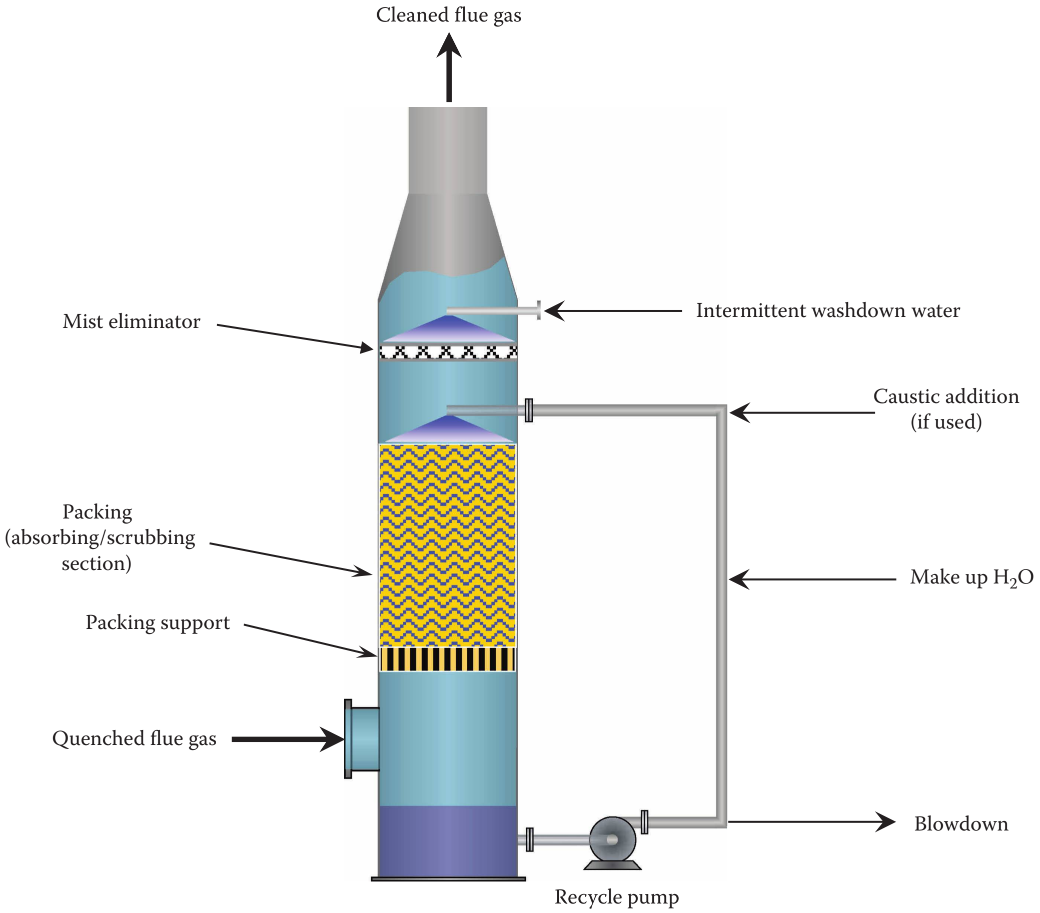


FIGURE 8.21
Simple packed column.

The countercurrent flow of flue gas traveling in the open spaces between packings comes into contact with a large amount of liquid for a relative long period of time. As the flue gas flows through the absorbing liquid, soluble gases are dissolved in the liquid. Just as in the wet particulate-removal cases, the flue gas must be saturated prior to coming into contact with the absorbing liquid to eliminate evaporation during contact.

The rate of acid-gas absorption at any point within the packed section is limited by the mass-transfer rate across the gas-phase boundary layer. Thus, a large mass-transfer coefficient, a high concentration of the pollutant in the flue gas, a low concentration of the dissolved pollutant in the absorbing liquid (i.e., low pollutant vapor pressure over the absorbing liquid), and a large amount of interfacing contact area will all increase the rate of absorption. The countercurrent flow of flue gas to absorbing liquid, which puts the cleanest absorbing liquid in contact with the cleanest flue gas, is the primary reason absorption can be so effective in a packed column. The packing type, the flue-gas velocity, the type of substance (i.e., acid gas) to be dissolved into the absorbing liquid, and the type of absorbing liquid determine the mass-transfer coefficient. The amount of interfacial area is determined by the type of packing in the column and by how well the packing is "wetted." Operational testing has been used to develop virtually all of the packing performance parameters.

The upstream incineration process determines the concentration of acid gas in the flue gas entering the column, the acceptable outlet gas concentrations are set by environmental regulations, and the liquid purge stream concentration is either specified by the customer or determined by design to achieve adequate acid-gas removal. The diameter of a packed column is typically designed to give a superficial gas velocity of 7–10 ft/s (2–3 m/s). This range of velocities is high enough to create enough pressure drop (about 0.3–0.5 in. W.C. = 75–125 Pa per foot of bed depth) to prevent poor flue-gas flow distribution (channeling) yet low enough to avoid flooding. A recirculation rate of 7–10 GPM (26–38 LPM) of liquid per square foot of bed cross-sectional area is usually required to assure adequate "wetting" of the packing. The remaining variables that are determined by the designer are the column height, the packing type, and the type and temperature of absorbing liquid.

The packed columns using water to recover acid are commonly referred to as "absorbers," and those operated with a caustic solution to remove acid are commonly referred to as "scrubbers."

Absorbers yield an acid solution and are typically used in applications in which the acid solution may be reused in a process or when the wastewater treatment plant can make use of the acid solution to neutralize a caustic waste. Packed bed absorbers can be designed to produce acid purge streams of up to 22% HCl by weight (the azeotropic maximum concentration) assuming the HCl vapor concentration in the flue gas leaving the absorber is sufficiently high. The concentration of HCl in the flue gas determines the maximum strength of the acid blowdown stream. A typical 10 ft (3 m) deep absorber section could remove about 99% of the HCl present in the flue gas while producing a 2% HCl blowdown stream.

Scrubbers not only remove acid gas from the flue gas, but also neutralize the dissolved acid. By adding NaOH to maintain a pH of between 6.8 and 8.0 in the absorbing liquid, the vapor pressure of the acid gas in the outlet flue gas is greatly reduced, thus increasing the absorption rate. The advantages of the scrubber over the absorber are a less corrosive blowdown stream and either a greater acid-gas removal for the same blowdown rate or a much lower blowdown rate for the same percentage of acid-gas removal. A typical 10 ft (3 m) deep scrubber could remove 99.9% of the HCl present in the flue gas and produce a blowdown stream containing up to 5% by weight total solids, most of it being NaCl.

The neutralized effluent from a scrubber also contains some sodium hypochlorite, which is a result of NaOH reacting with the free chlorine generated during the combustion process. The sodium hypochlorite is a strong oxidant and may require treatment to meet effluent requirements. A reducing agent like SO_2 or a solution of sodium bisulfate may be added to the recirculating stream in the scrubber to reduce the sodium hypochlorite to sodium chloride ratio.

It is important to remember that CO_2 in the flue gas also reacts with NaOH. The reaction rate, however, is relatively low until the pH of the recirculating water becomes greater than 8.0. Above that concentration, the CO_2/NaOH reaction rate increases significantly, greatly increasing the consumption of NaOH. Proper pH control of the recycle water is an economic necessity.

In some applications, both acid production and a very high level of acid-gas removal are required. This can be accomplished with a two-stage system. Figure 8.22 illustrates this application. The first stage is an absorber that removes 80%–95% of the acid gas and produces an acid blowdown stream. The second stage is a scrubber, which achieves the desired level of acid-gas removal and with a significant portion of the acid gas removed in the absorber section, produces a relatively low-volume blowdown stream. Such a system is advantageous when (1) there is a need to achieve very high levels of acid-gas removal and/or (2) there is a use for the acidic blowdown stream or because of the amount of acid gas to neutralize, it is more economical to neutralize it externally with a less expensive reagent such as lime (CaO) or slaked lime ($Ca(OH)_2$).

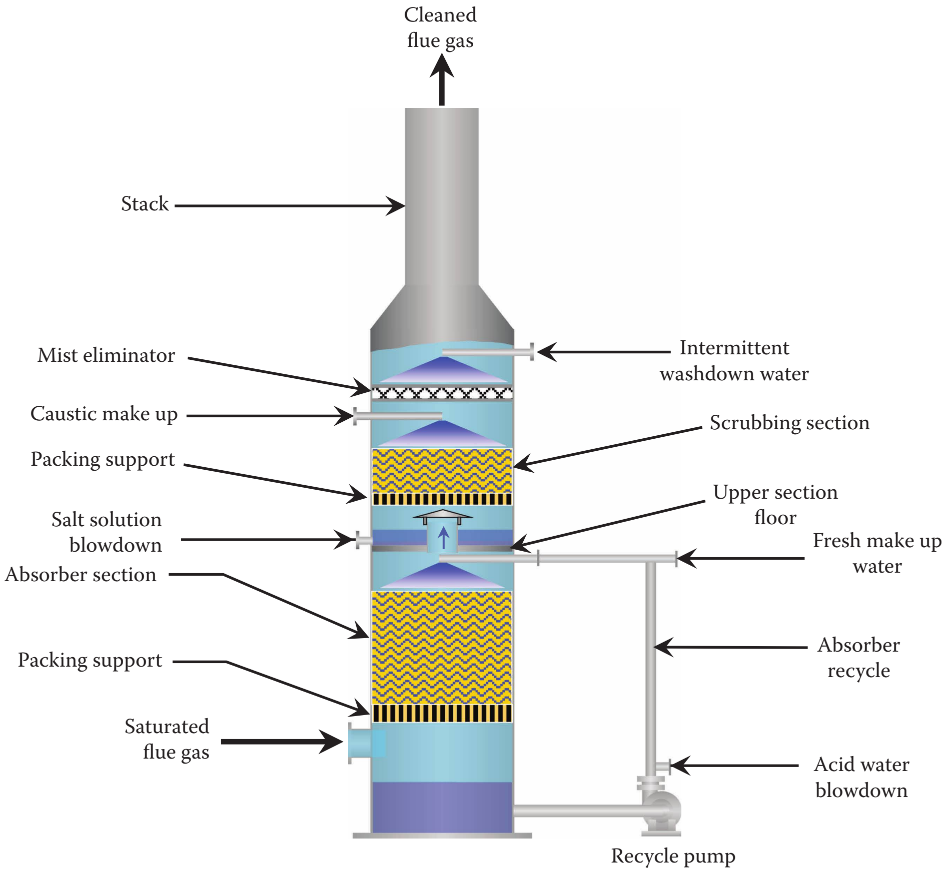


FIGURE 8.22
Two-stage acid-gas removal system.

In some applications, the flue-gas quench process can be incorporated in the packed column. For this condition, the lower section of the vessel and the packing material must be able to withstand both high temperatures and acidic conditions. A high-temperature and acid-gas-resistant lining must be used in the lower portion of the vessel, and the packing material and internal support hardware must be made of ceramic or graphite. Figure 8.23 represents this application.

While the packed column is primarily designed for acid-gas removal, it also will remove some particulate matter. However, the basic mechanism of entrained particle removal is inertial separation from the gas stream followed by entrapment in the absorbing liquid. Because the gas velocities through the packed section are far too low for effective inertial separation and entrainment of particles with an aerodynamic diameter of less than 10 µm, a packed column is not an effective particulate-removal device.

The common materials of construction for a packed column assembly are FRP for the vessel and ceramic, FRP, plastics (such as polypropylene, PVC, CPVC, Teflon, Kynar), or some combination of these for the packing, internal support hardware, and water circulation equipment. If the recirculating liquid is subcooled to enhance recovery/removal, the heat exchanger would have to be built of corrosion-resistant materials.

The advantages of using a packed column include (1) high removal efficiency, (2) low pressure drop/energy cost (less than 10 in. W.C. = 2.5 kPa drop for a 10 ft = 3 m deep packed section), and (3) no moving parts in the column itself.

The disadvantages are (1) poor small particulate-removal capability; (2) dependent on recirculation pumps, caustic feed pumps, etc., to operate; and (3) momentary loss of quench that can cause "meltdown."

8.3.5.3.4.2 Venturi Scrubber The venturi scrubber described earlier as a quench or a particulate-removal device can also function as an acid-gas removal device. One of the important factors for effective mass transfer is the intimate contact between the flue gas and the liquid used to absorb the acid gases, and the venturi scrubber can provide relatively good gas-to-liquid

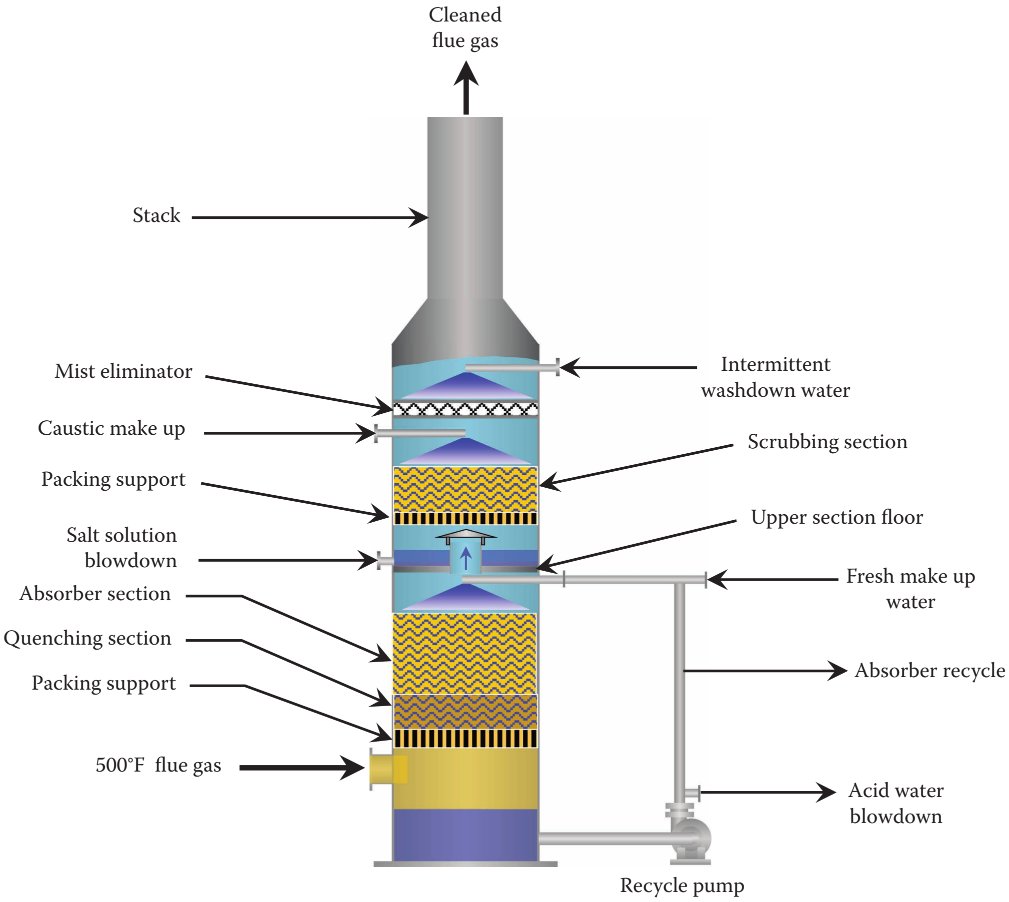


FIGURE 8.23
Combination quench/two-stage acid removal system.

contact. However, because of other factors such as acid-gas solubility and solubility rates (which are affected by gas-film and liquid-film resistance) and the difference in concentration of the pollutant in the flue gas compared to the concentration of the dissolved pollutant in the absorbing liquid (i.e., low pollutant vapor pressure over the absorbing liquid), the venturi scrubber, although good, is not the best overall choice for acid-gas removal.

HCl is a good example of a highly soluble acid gas. It is easily absorbed in water, to create hydrochloric acid. If necessary, the HCl can then be reacted with sodium hydroxide (NaOH) added to the recycle water to neutralize the HCl and form NaCl.

SO_2 is an example of an acid gas that is only moderately soluble in water. Therefore, it must be in contact with the recycle water for a longer period of time for high removal efficiency. Adding NaOH to the recycle water will improve the solubility by decreasing liquid-film resistance, but a long period of contact is still needed to achieve high removal efficiency. The concentration of NaOH must be as low as possible to avoid excessive reaction of NaOH with CO_2.

The physical configuration of the venturi allows only a relatively short gas/liquid contact time. The short period of time reduces the amount of absorption, especially for moderately soluble acid gases. Also, because the flue gas and the absorbing fluid have to travel in the same direction (cocurrent flow), at the venturi outlet, the cleanest gas is exposed to recycle water that has the highest concentration of acid gas, which further reduces the effectiveness of the scrubber. Despite these drawbacks, a properly designed venturi system can effectively remove much more than 90% of the acid gas from a flue-gas stream.

The advantages of the venturi as an acid-gas removal device include (1) high effectiveness for highly soluble acid gases, (2) low capital cost, and (3) low plot space requirement.

The disadvantages as an acid-gas removal device consist of (1) only moderate effectiveness for lower-solubility acid gases, (2) high operating cost (pressure drop),

(3) short gas-to-liquid contact period, and (4) cocurrent flow that reduces ratio of concentration of the pollutant in the flue gas relative to the concentration of the dissolved pollutant in the absorbing liquid.

8.3.5.4 NOx Control Methods

NOx can be controlled during or after the combustion process. The most effective method will be determined by the NOx emission allowed and by the capital and operational costs of various methods.

The three NOx formation mechanisms in combustion systems, namely, thermal NOx, prompt NOx, and fuel-bound NOx, were described in Section 8.2.3 of this chapter. In addition, more detail information about NOx formation is available in Volume 1, Chapter 15.

Minimizing the temperature and size of high-temperature zones is the primary defense against thermal NOx formation. It is worth noting that many of the burner design methods used to mitigate NOx formation in other applications such as process furnaces or boilers can also be used to minimize thermal NOx formed in the burners of TO systems.

The NOx formation from fuel-bound nitrogen is more difficult to prevent since combustion of organics with bound nitrogen in the presence of excess oxygen at any temperature will result in significant NOx production. In addition, the NOx produced from nitrogenous organic wastes will usually be much greater than the burner fuel–generated thermal NOx. Also, some waste streams contain incoming NOx that will not be destroyed in an excess-oxygen environment. For these conditions, addressing burner fuel–generated NOx is virtually inconsequential. Either the combustion process has to be changed to reduce the NOx exiting the combustion section or the flue gas must be treated after the NOx is formed (post-combustion treatment).

If the waste streams do not contain large quantities of noncombustible materials such as air, water, or other inerts, a modified combustion process, using a two-stage oxidation, that is, a sub-stoichiometric (reducing) first stage followed by a lower-temperature oxidizing second stage, can be used very effectively eliminating the need for post-combustion NOx removal.

If post-combustion NOx removal is required, although many different post-combustion treatment methods are available, including some wet techniques that are very efficient, the methods most commonly used for typical thermal oxidation systems are selective non-catalytic reduction (SNCR) and selective catalytic reduction (SCR).

8.3.5.4.1 Combustion-Process Modification

For wastes and/or fuels with a high concentration of bound nitrogen, or for wastes that contain NOx, the single most effective practical process modification is the aforementioned two-stage combustion that is a form of staged-air combustion. The most common implementation of staged-air combustion is accomplished in a three-stage combustion process,[4] as shown in Figure 8.24. As noted previously, this method may not be the most cost-effective if the waste stream(s) contains significant quantities of oxygen, water, or inerts, because of the large amount of auxiliary fuel it would take to drive the first-stage sub-stoichiometric (reducing). If the steam is needed in the

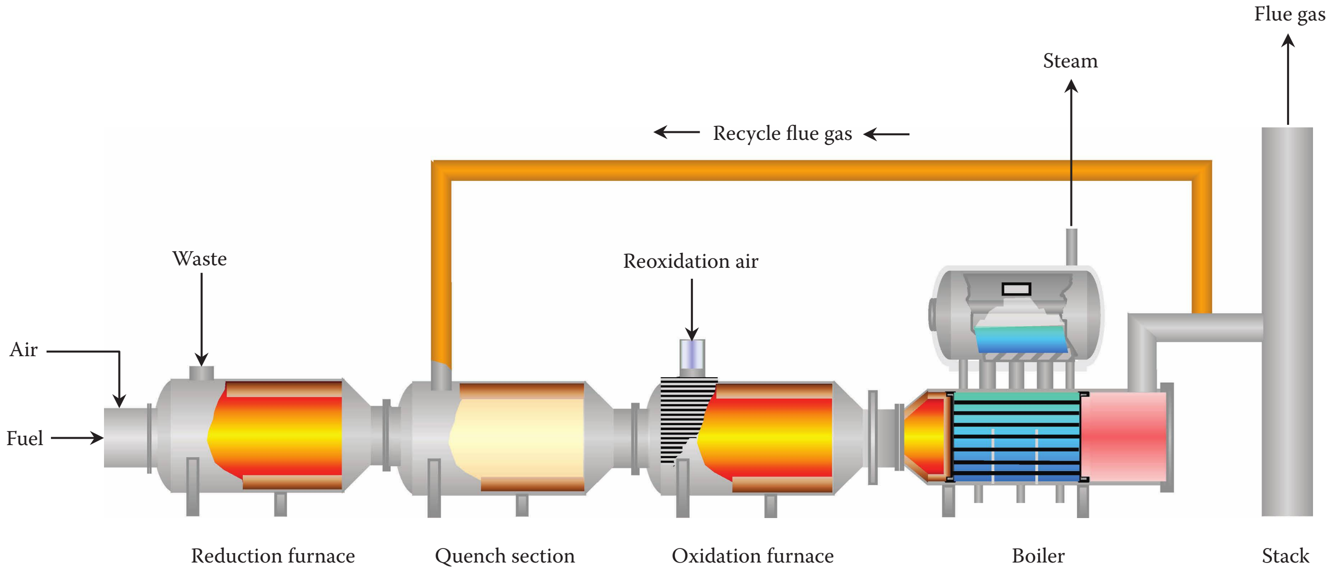


FIGURE 8.24
Three-stage NOx reduction process.

plant, it is possible that the high fuel use could be justified based on the amount of steam generated.

NOx reduction by this method is achieved in the high-temperature sub-stoichiometric first stage by dissociation of the organic compound and effective conversion of the released nitrogen atoms to N_2. The flue gas is cooled in the second (quench) stage resulting in a lower peak temperature in the third (oxidizing) stage. The lowered peak temperature minimizes reformation/formation of thermal NOx in the oxidizing stage.

Fuel, waste(s), and less-than-stoichiometric combustion air are introduced into the first stage (reduction furnace) to produce a high-temperature (2000°F–2800°F = 1100°C–1500°C) reducing atmosphere. The excess combustible material in the high-temperature reducing zone provides the driving force for the reduction of the oxides of nitrogen and conversion of bound nitrogen to N_2 instead of NOx. Some inert (low or no oxygen content) gasses such as recycle flue gas, steam, or water may also be introduced into the reducing zone to allow more consistent control of the operating temperature so as not to exceed the limits of the refractory. The optimum level of oxygen deficiency in this stage depends on the waste composition. Although the oxygen is usually supplied by combustion air, the oxygen in NOx and in waste-gas streams will also be utilized. The residence time in this stage is usually in the range of 0.5–1.0 s. The primary components of the gas leaving this stage are CO, H_2, CO_2, N_2, and H_2O. Most of the fuel-bound nitrogen is converted to N_2 with the remainder exists as low levels of HCN, NH_3, and NO.

The hot flue gas then enters the second process stage (the quench chamber) by passing through a venturi mixing section. An inert cooling medium such as recycle flue gas, water, or steam is injected through multiple openings in the venturi throat to quickly mix with the flue gas and reduce the temperature to 1300°F–1750°F (700°C–950°C). The temperature must be high enough so that rapid ignition of the combustibles occurs by simply adding air, but low enough to limit the temperature achieved in the final oxidation stage to less than 2000°F (1100°C). Although cooling is rapid, sufficient time must be allowed in this section to ensure that the bulk gas temperature is uniform.

As the cooled flue gas exits the quench section, air in excess of the stoichiometric requirement is introduced at the entrance into the third stage (the oxidation zone), again by using a venturi mixing section. In this final stage, the carbon monoxide, hydrogen, and any remaining hydrocarbons produced in the first stage are oxidized to CO_2 and H_2O. The flue-gas cooling step performed prior to introduction of the oxidation air controls the peak oxidizing operating temperature, thereby limiting formation of thermal NOx. The residence time in the oxidation stage is usually in the range of 0.5–1.0 s. The flue gas may then be treated by any of the previously discussed methods or exhausted directly to the atmosphere for dispersion.

These three process steps, reducing, quench, and oxidation, often result in levels of NOx less than 150 ppmv (parts per million by volume) at excess-oxygen conditions of less than 1%–2% (dry).

A wide variety of bound-nitrogen-containing gas and liquid wastes can be incinerated using this treatment method, especially if the waste contains significant heating value. Nitrogen in the chemical waste can be in the organic or inorganic form. Examples of the organic form are HCN, amines, nitriles, and nitroaromatics; examples of the inorganic form are ammonia and NOx. A small process stream containing some quantity of NOx can be treated to produce a cleaner stream that contains comparably less NOx than the original waste. This method is not suitable for NOx reduction in flue gas from large combustion processes, such as utilities. It also is not appropriate for waste streams containing large quantities of air, water, or inerts. Either case would require an excessive amount of auxiliary fuel to create the sub-stoichiometric environment in the first stage.

The three-stage systems using the process described earlier have been operating in a variety of industrial applications for more than 30 years. The destruction efficiencies of incoming compounds of more than 99.99% are achieved. Carbon monoxide is usually in the 50–100 ppmv (dry, corrected to 3% O_2) range. NOx in the flue gases can vary from 50 to 200 ppmv, dry, corrected to 3% O_2, depending on the composition of the waste stream being treated.

A modified version of this process, a two-stage process, can also be used if the NOx level required does not have to be as low. In the two-stage process, the first-stage reduction furnace remains the same. The difference starts at the reduction furnace outlet. Instead of adding an inert cooling media then adding just enough combustion air to oxidize combustibles and maintain 1.5% excess O_2, a large amount of air is introduced to cause the oxidation reactions to occur while simultaneously cooling the oxidation section outlet temperature to 1800°F (1000°C) or less. This process is shown in Figure 8.25.

The high excess O_2 (up to 10%) causes more equilibrium NOx to form in the oxidation of the two-stage system than would form in the lower O_2 oxidation section of a three-stage system. In addition, the method of control is slightly different, in that the two-stage system requires the use of a sometimes maintenance-intensive combustibles analyzer to measure and control the combustibles level in the reduction furnace. For the three-stage system, the combustibles level is controlled by measuring the differential temperature between the quench section

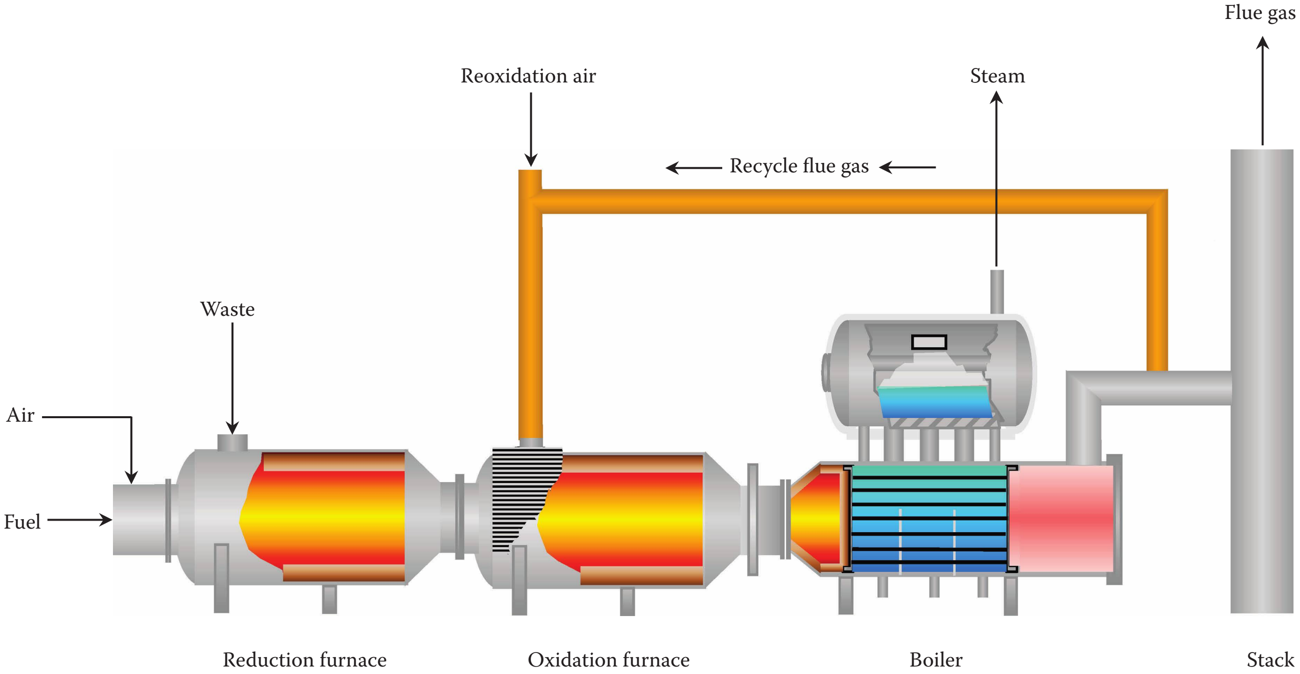


FIGURE 8.25
Two-stage NOx reduction process.

and the outlet of the oxidizing section, thus avoiding the instrument maintenance issue. Another minor drawback is that recycle flue gas cannot be used to control temperature in the reduction furnace, which results in lower heat-recovery efficiency in the event that heat recovery is used. In spite of these detractions, the two-stage system is still far better for treating wastes containing bound nitrogen than a single-stage, oxidizing-only combustion process.

8.3.5.4.2 *Selective Noncatalytic Reduction*

The selective noncatalytic reduction (SNCR) system directly reduces nitrogen oxides ($NOx = NO + NO_2$) downstream of the combustion process. Typical reduction agents used are ammonia (NH_3) or urea ($(NH_2)_2CO$).

The reduction agents react selectively with NOx without the help of a catalyst. The total reactions are complex with many conversions of molecules with free radicals but can be simplified to the following global reactions:

For ammonia[5]:

$$4NO + 4NH_3 + O_2 \rightarrow 4N_2 + 6H_2O \tag{8.1}$$

$$6NO_2 + 8NH_3 \rightarrow 7N_2 + 12H_2O \tag{8.2}$$

For urea:

$$4NO + 2(NH_2)_2CO + O_2 \rightarrow 4N_2 + 2CO_2 + 4H_2O \tag{8.3}$$

$$2NO_2 + 4(NH_2)_2CO + 4O_2 \rightarrow 5N_2 + 4CO_2 + 8H_2O \tag{8.4}$$

The reducing agents must be mixed uniformly into the hot flue gas downstream of the combustion stage in the presence of excess oxygen. The normal operating temperature range for the DENOX reaction is between 1652°F and 1922°F (900°C–1050°C). If the temperature is above the optimum value, the agent will oxidize and might create NOx. For example, the reaction from ammonia to NOx can be summarized with the following equation:

$$4NH_3 + 5O_2 \rightarrow 4NO + 6H_2O \tag{8.5}$$

$$4NH_3 + 3O_2 \rightarrow 2N_2 + 6H_2O \tag{8.6}$$

At lower DENOX reaction temperatures, the reaction is too slow. As a result of this, the ammonia would not react and will slip toward the stack. German environmental laws[6] allow, for instance, a maximum ammonia emission of 30 mg/N-m^3. Furthermore, if the flue gas contains acid components like sulfur trioxide (SO_3) or hydrogen chloride (HCl), the ammonia slip will result in the formation of ammonia salts, which might sublimate and block downstream heat recovery systems.

8.3.5.4.2.1 Comparison of DENOX Reduction Agents Typically reduction agents are dissolved in water. The advantage of using urea-based systems compared to ammonia is that urea is nontoxic and the liquid solution is less volatile and can be stored and handled more safely.[7]

Another benefit of urea is based on its physical properties. During evaporation, the urea has a longer residence time in the spray than ammonia due to its lower vapor pressure. This results in higher penetration depth and better distribution of the agent.

On the other hand, studies on urea-based DENOX systems sometimes show the production of nitrous oxide (N_2O), also known as a greenhouse gas. Nitrous oxide concentrations are found up to 36 ppm (70 mg/N-m^3) at very low DENOX temperatures and/or very high molar ratios (mol NH_3/mol NOx).[8]

8.3.5.4.2.2 Design The challenge is to reach high reduction efficiency without NH_3 slip. This can be influenced by a homogenous temperature distribution in the reaction chamber (DENOX chamber), a minimum of heat loss (quasi-adiabatic reaction), a sufficient residence time, a controlled and steady DENOX temperature, a controlled agent dosing, and a good mixing momentum and equal distribution of the agent. All these requirements are realized in a specially designed John Zink KEU DENOX chamber. To prevent radiation from equipment up- or downstream of the DENOX chamber, the chamber is arranged in an angle. The cylindrical reaction chamber is designed with a multilayered refractory lining to achieve a minimum of heat loss. Mixing devices like vortex flow, orifices, and spin and antispin injection are used. The atomizing is done with replaceable stainless steel guns. The design of the guns in combination with the use of compressed air as atomizing medium allows a high turndown ratio. The overall volume of the agent is relatively low compared to the total flue-gas volume. For equal distribution of this low-flow, a solution of 20%–25% urea or ammonia in water is used. The dosing of the reduction agent is based on the calculated NO value in the raw gas and is controlled by the actual NOx value in the clean gas.

8.3.5.4.2.3 Results In the field, proven best results are at temperatures between 1740°F (950°C) and 1900°F (1000°C). Under this condition, the John Zink KEU design reaches a very high efficiency with molar ratios of $[NH_3]/[NO]$ between 3 and 5. John Zink KEU SNCR units achieved NOx reduction rates of approx 80%–95% without noticeable NH_3 slip[9] (see Figure 8.26).

Depending on the absolute amount of the NOx input concentration and the flue-gas flow rate, the John Zink KEU DENOX systems, as described earlier, show emission values of less than 49 ppm (100 mg/N-m^3).[9,10] The advantage of the investment costs of an SNCR is significant, compared to the costs of an SCR unit. The SCR would also be sensitive to particulates and have the potential issue of poisoning the catalyst.

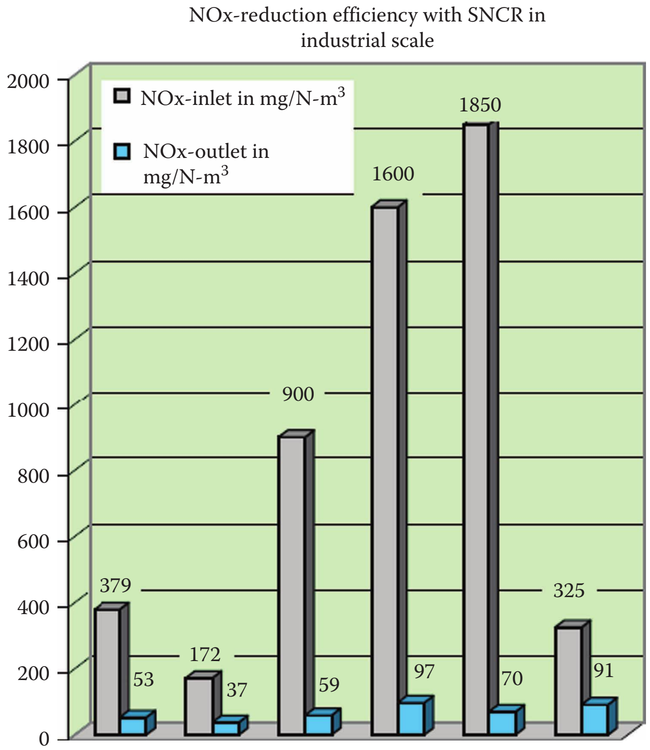

NOx-reduction efficiency with SNCR in industrial scale

FIGURE 8.26
Results of three different DENOX units. The DENOX temperatures are between 1800°F and 1830°F (980°C–1000°C), the ammonia slip is <2 ppm, and the flue-gas flow rates are between 3730 and 9960 scf/min (6000–16000 N-m^3/h).

8.3.5.4.3 Selective Catalytic Reduction

In the SCR process for reduction of nitrogen oxides, a corresponding quantity of ammonia source is injected into the flue gas at a temperature of typically 170°C–550°C. The mixture of ammonia and flue gas passes through a catalyst where the nitrogen oxides are converted to nitrogen and water vapor. More than 99% of ammonia then reacted in contact with the catalyst. The residual unreacted ammonia is called ammonia slip. The SCR unit consists of the ammonia handling system, flow mixing and stratification system, reactor, solid catalyst, and a control system.

The nitrogen oxides, which primarily consist of NO and to a minor extent NO_2, are converted according to the following stoichiometry:

$$4NO + 4NH_3 + O_2 \rightleftarrows 4N_2 + 6H_2O \qquad (8.7)$$

$$2NO_2 + 4NH_3 + O_2 \rightleftarrows 3N_2 + 6H_2O \qquad (8.8)$$

These reactions are exothermic; a small temperature rise of the flue gas occurs as it passes through the SCR catalyst. The conversion of nitrogen oxides does not create any secondary pollution as the products formed are only

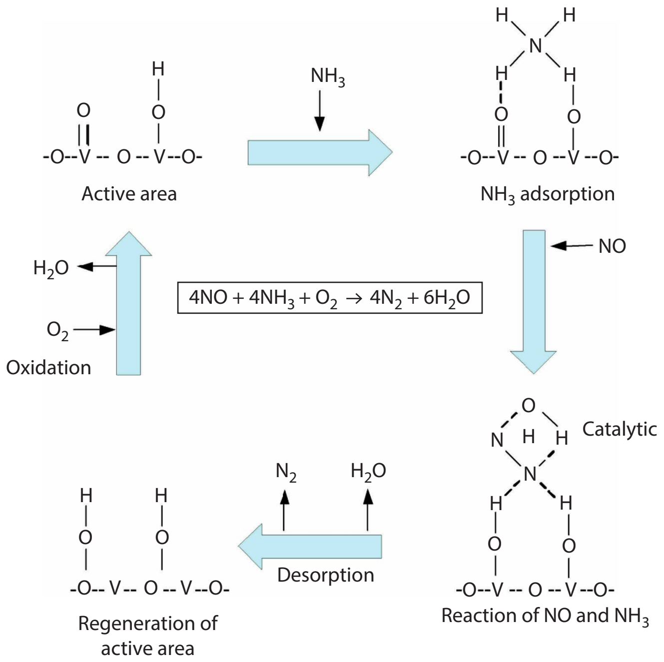


FIGURE 8.27
Scheme regarding catalytic reaction of NOx and NH_3 on the catalyst.

nitrogen and water vapor, which are already present in the atmosphere in large quantities. The conversion efficiency is typically 90% but may be as high as 99%.

The reaction scheme of NOx and NH_3 on the catalyst is shown in simple form in Figure 8.27.

NH_3 adds to the V_2O_5 molecules of the catalyst. NOx reacts with the chemisorbed NH_3, forming nitrogen and water. After the desorption of the reaction products, the reduced catalyst is reoxidized to the active initial state.

The active component vanadium pentoxide of SCR catalysts is able to accelerate other reactions than wanted reactions regarding SCR process. That means in addition to wanted chemical reaction, other secondary reactions also take place.

These unwanted chemical reactions can be the following:

$$SO_2 + \tfrac{1}{2}O_2 \rightarrow SO_3 \tag{8.9}$$

$$NH_3 + SO_3 + H_2O \rightarrow NH_4HSO_4 \tag{8.10}$$

$$2NH_3 + SO_3 + H_2O \rightarrow (NH_4)_2SO_4 \tag{8.11}$$

The reaction equation (8.9) describes the catalytic oxidation of possible flue gas component SO_2 to SO_3, also called SO_2/SO_3 conversion. The reaction equation (8.10) indicates the ammonia bisulfate (ABS) forming in case of SO_3 excess, and Equation 8.11 describes the ammonia sulfate forming in case of ammonia excess.

Figure 8.28 shows a schematic diagram of SCR process. The main components of the SCR process consist basically a reactor with the catalyst and an ammonia storage and injection system. The reducing can be either liquated, water-free ammonia under pressure or it can be 25% aqueous ammonia solution at atmospheric pressure. A solution of urea can be used as well. The anhydrous ammonia is normally stored in a storage tank at ambient temperature and at the corresponding vapor pressure (approx. 10.5 bar at 25°C ambient temperature). From the storage tank, the liquid ammonia flows by its vapor pressure to the evaporator, which can be heated by hot water, steam, or electricity. A controlled flow of evaporated ammonia is then passed to the NH_3 dilution air mixer, where the ammonia is diluted with air supplied by the dilution air blower before injection into the hot flue gas stream. The ammonia vapor is diluted with air to about 6 vol. % in order to eliminate the risk of ammonia ignition when injected into the hot flue gas. The upper and lower explosion limits for diluted ammonia vapor are 15 vol. % and 28 vol. %, respectively. Secondly, the dilution with air improves the mixing of the ammonia vapor and the flue gas.

A static mixing element is located in the flue gas duct before the inlet of the reactor to ensure a homogeneous mixing of the flue gas and the diluted ammonia vapor. It is important to obtain homogeneous mixing in order to attain a high efficiency of the SCR process and minimize the NH_3 slip (unused NH_3) from the SCR reactor. That means the SCR process requires precise control of the ammonia injection rate. Insufficient injection results in low conversion of NOx and an injection rate that is too high results in an undesirable release of unconverted ammonia to the atmosphere referred to as ammonia slip.

The injection of the ammonia–air mixture normally takes place through a grid of nozzles in order to achieve a uniform mixing of the ammonia with the flue gas. A homogeneous distribution of the ammonia in the flue gas is of crucial importance to achieve efficient NOx conversion. Uneven ammonia distribution can cause poor performance of the SCR unit because some portions of the flue gas are overtreated with ammonia, resulting in higher ammonia slip, whereas other portions of the flue gas is undertreated, resulting in a lower than designed DeNOx efficiency. The requirements to mixing become increasingly important as demands to NOx removal efficiency and low ammonia slip become more stringent. Today, 90%–95% NOx reduction with just 2 ppm ammonia slip is not unusual.

To ensure efficient optimum mixing, the distance from the ammonia injection grid (AIG) to the catalyst should be as long as possible. Long mixing distance is costly, and therefore a system of one or more static mixers may be placed in the flue gas duct to enhance the mixing. However, static mixers introduce an additional pressure loss, which is not desirable. Developed gas distributor can be also round plate, or plate assembly that

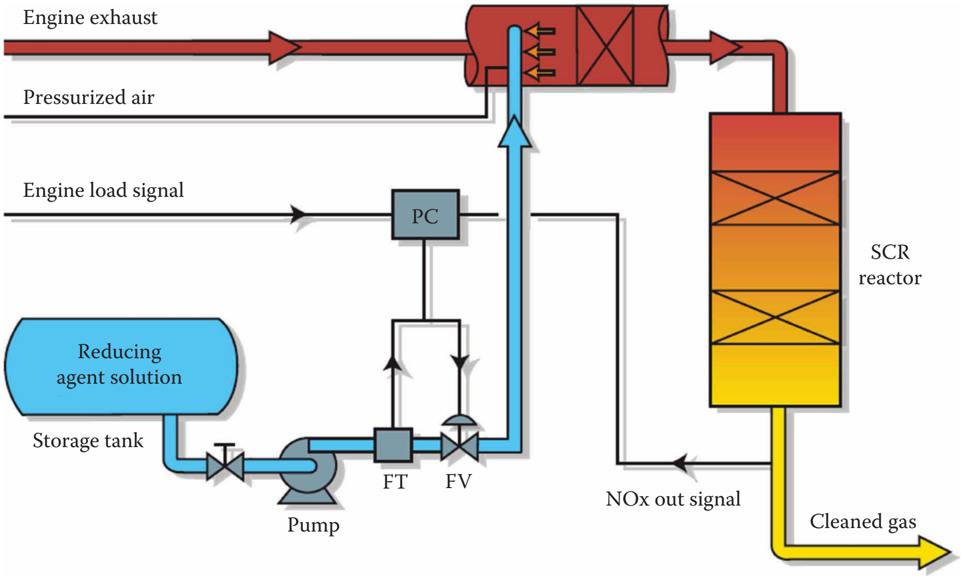


FIGURE 8.28
Schematic process diagram for SCR of NOx.

is installed at the inlet of the SCR reactor is designed to provide a uniform distribution of the gas mixture over the entire cross section of the reactor.

A prerequisite for the catalyst to perform at its best is the proper design of the SCR reactor, gas channels, and the ammonia injection and mixing system. In a high-dust SCR unit, care must be taken to ensure an even gas flow rate and fly ash distribution across the reactor in order to avert damage to the catalyst in the form of plugging or erosion.

In practice, the degree of conversion of the nitrogen oxides depends on the amount of ammonia added, and it increases with increasing NH_3/NOx ratio. One should not, however, add NH_3 above the stoichiometrically required amount in order to avoid the slip of unreacted ammonia.

As the homogeneous mixture of flue gas and ammonia vapor passes through the channels of the catalyst, the nitrogen oxides are converted through the catalytic reactions described earlier.

8.3.5.4.3.1 Design The SCR unit is designed in principle in two steps. Initially, the catalyst volume is calculated, which is necessary under ideal conditions. Since inhomogeneous distributions and fluctuations, this can be for example in the gas velocity, the NOx/NH_3 ratio and the temperature arises in the operation, margins must be added to the calculated volume. The required service life and the expected deactivation rate of the catalyst necessitate further margins and hence give the final volume.

The SCR catalyst volumes are designed for a specified ammonia NH_3 slip during steady operation and accurate ammonia dosing; this can be normally maintained. However, at sudden changes of load, minor overshooting may occur.

When determining the layout of an SCR system, it is important to review the ducting to verify that the flow is distributed properly and in the case of high-dust applications. That dust does not cause erosion of the duct walls or settles in the ducts. It is often necessary to install guide vanes, and for high-dust installations, a rectifier grid can be installed above the first catalyst layer.

The reactor is normally designed with several layers of catalyst, for instance with initially two layers and one spare for future use. This principle is applied to minimize catalyst consumption, in that the third layer is installed when the first two layers reach their design activity. Subsequently, the first layer is changed then the second and so on. In cases where dust is present in the flue gas, soot blowers or sonic horns should be installed in order to keep the catalyst clean from dust buildup and thereby maintain a low pressure drop.

Flow modeling: The design may be verified by the use of flow modeling. The use of gas-flow modeling by computational fluid dynamics (CFD; see Volume 1, Chapter 13) or in physical scale models (see Volume 1, Chapter 11) (typically 1:10–25, depending on the size of the installation) has proven an efficient and often necessary tool to accomplish these goals. The general objectives of the model work are to ensure a high degree of velocity uniformity upstream the ammonia injection and at the entrance to the catalyst layers and to verify proper mixing of ammonia into the flue gas. The model work further assists in optimizing the design of guide vanes, the layout of ducts, the reactor, and the necessary flow control devices to minimize overall pressure loss and to ensure that formation of dust deposits is not be promoted.

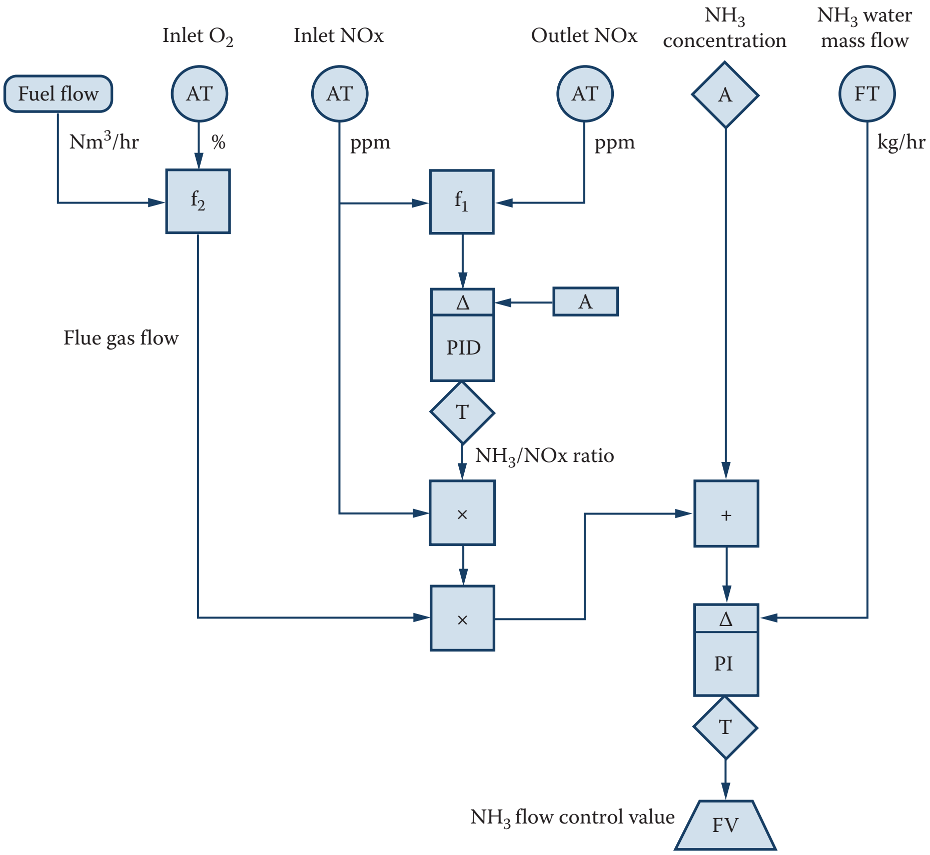


FIGURE 8.29
Control parameter/control of reducing agent (NH_3).

Process control: The control of the system relates to the addition of the reducing agent. The control (see Volume 2, Chapter 2) is a combined feed forward/feed backward, where the set point is generated based on a load signal indicating the flue gas flow rate and an NO measurement. This is fine-tuned by the use of a feedback signal giving the NOx level in the outlet of the reactor. The main challenge in the process control is to establish locations in the duct, providing reliable and representative measurements.

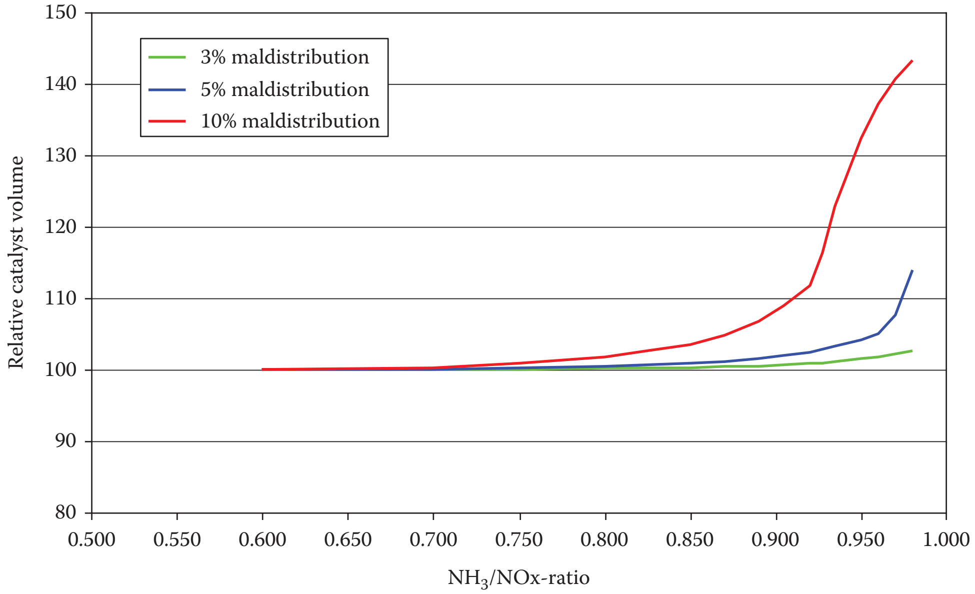
Ammonia flow control: The flow of ammonia is controlled by a flow controller including allow transmitter and a control valve. The set point of the flow controller, which controls the position of the control valve in relation to the actual measured flow, is given by a process computer, which is also fed with information about the actual flue gas flow (feed forward signal) and the NOx concentration at the reactor outlet ("feedback" signal). If the inlet concentration of NOx varies, continuous measurement of this concentration is also included. These data are compared by the PLC with preprogrammed parameters for different flue gas flows, related NOx emission levels, and required NH_3/NOx ratio as measured and calculated during commissioning of the unit. On this basis, the PLC continuously calculates the required flow of ammonia and adjusts the set point of the flow controller accordingly (for more details, see Figure 8.29).

Injection of the reducing agent: It is mandatory that a good mixing of ammonia and NOx is obtained, especially for high removal efficiencies. For flue gas ducts where the stratification is minor and for low removal efficiencies, an AIG may be sufficient to secure a satisfactory mixing (see Table 8.2).

For large ducts, such as those in large coal-fired power stations, it often proves necessary to install a static mixer that reduces any stratification in NOx, temperature, and flow velocities and further ensures a good mixing of ammonia and NOx. The drawback of mixer installation is an increased pressure drop. The requirement

TABLE 8.2
Types of Mixing Elements and Related NH_3/NOx Maldistribution

Element Type	NH_3/NOx Maldistribution
Static mixer	3%
NH_3 injection grid—long mixing distance	5%
NH_3 injection grid—short mixing distance	10%



The maldistribution is defined as the standard deviation in a normal (Gauss) distribution:

$$\sigma = \sqrt{\frac{SUM\,(X_i - X_{average})^2}{(N-1)\,X_{average}^2}}$$

FIGURE 8.30
Ammonia maldistribution.

to mixing is carefully reviewed in each individual case before a solution is implemented. There are a variety of mixer options available, ensuring that the most optimal solution, given the requirements, is selected. Figure 8.30 illustrated the how the NH_3/NOx maldistribution will have effects on the relative catalyst volume.

8.3.5.4.3.2 Reducing Agent In short, the SCR DeNOx process utilizes ammonia as a reducing agent. The ammonia source for the reduction of NOx may be anhydrous ammonia, aqueous ammonia, or an aqueous solution of urea.

Anhydrous ammonia is the preferred reducing agent for medium- and large-sized SCR units. But ammonia is a medium, however, that poses a high risk when handled inappropriately or in poorly designed facilities, since it has a highly irritating effect on eyes and the breathing system even at very low concentrations. Additionally, spillages can contaminate ground and flowing waters. Safe storage and handling, as well as an optimal conditioning for the SCR process, are therefore the key issues for the design of ammonia storage and handling facilities.

Anhydrous ammonia: Before the anhydrous ammonia is injected into the gas stream, it is evaporated and mixed with air to ensure that the two do not form an explosive mixture. For the evaporation, there are a number of options that need to be evaluated in each case. Anhydrous ammonia is considered the cleanest and most cost-effective reducing agent, but storage of anhydrous ammonia under pressure may imply a potential hazard.

Aqueous ammonia: Aqueous ammonia stored at atmospheric pressure is used as reducing agent in small- to medium-sized SCR units. The system is always designed in stainless steel. Truck unloading is usually performed through hoses and unloading pumps. Storage vessels are either double wall or single wall, placed in a containment basin. When applying aqueous ammonia (NH_4OH) typically 25% wt solutions, it can be either evaporated or injected by using atomizing air directly into the gas stream via nozzles. The disadvantage of the direct injection is an often poor turndown ratio of the nozzles.

Urea: A urea solution can be injected directly into the duct via nozzles, in those cases where there is sufficient distance for an in-duct decomposition of the urea into CO_2 and ammonia and when the gas stream is relatively clean. As an alternative, there are a number of ammonia on demand (AOD) systems available that produce ammonia from urea. On the other hand, urea solution is considered safe to handle and may therefore be the preferred reducing agent for SCR units in residential areas.

8.3.5.4.3.3 Deactivation Deactivation of the catalyst describes all phenomena reducing the efficiency of the catalytic reaction by chemical reaction of the active compounds inhibiting for the desired reaction, interference with film or pore diffusion of the reactants or products, and sintering and recrystallization of the catalyst. For most of the standard applications, these phenomena are well known and result in a comprehensive set of deactivation rates. The activity of the catalyst may decrease during operation due in detail to one or more of the following factors.

Poison by alkali metals: Caustic compounds such as the alkaline metals sodium (Na) and potassium (K) can reduce the catalyst activity if they get into direct contact with the catalyst surface. The mechanism for this reaction is that the alkaline metals react with active sites on the catalyst.

For most applications, avoiding condensation of water vapor can eliminate this risk of alkaline metal poisoning. For coal, the risk is considered minor, as most of the alkaline metals in the coal ash are not soluble. For oil, the risk of poisoning is larger due to a higher content of water-soluble alkali metals; for biofuels like straw and wood, a very severe deactivation is observed due to the high content of soluble potassium in these fuels.

Poisoning by arsenic: Arsenic (As) can reduce the catalyst activity. The poisoning by arsenic is due to capillary condensation in the small pores. Arsenic (as As_2O_3) can then react with the active sites on the catalyst surface. On dry bottom boilers, no or only small severe arsenic poisoning has been observed during test on SCR catalyst. The effect of arsenic poisoning may be a significant problem for wet bottom boilers recycling fly ash after the electrostatic filters. For other boiler types, the presence of arsenic is considered just another factor contributing to the deactivation.

Sintering of the carrier: Longtime exposure to temperatures above 450°C (840°F) may cause sintering of the catalyst's active sites (surface area) whereby the activity is reduced. The sintering is minimized by stabilizing the catalyst with tungsten.

Plugging of the pores: Small particles may deposit in the pore system of the catalyst and thereby restrict the transportation of nitrogen oxides (NOx), ammonia (NH_3), and oxygen (O_2) to the catalyst's active sites. This type of deactivation is considered unavoidable, and it is therefore a part of the normally expected deactivation rate, which is taken into consideration when designing the catalyst loading.

For high-dust applications, installation of soot blowers is necessary in order to assure that the channels in the catalysts are not plugged.

Condensation of water, sulfuric acid, and ammonium sulfates: Ash may contain poisonous compounds, which could be transferred to the catalyst surface in case of water condensation. In addition, water may react with the ash resulting in the formation of a hard layer blocking the pores. Furthermore, vanadia (the active component) is water soluble and may be washed out. Condensation of water should therefore be avoided.

Condensation of sulfuric acid may take place in case of operation below the acid dew point (see Figure 8.31). This is not the case during normal operation and may only take place during start-up and shutdown. The importance of possible acid condensation is therefore considered minor. Operation below the temperature of the ammonium sulfate dew point leads to a temporary reactivation of the catalyst. Only in case of longtime operation below this temperature, the performance may be affected permanently. The process is reversible, and operation above the temperature of condensation causes the ammonium sulfate to evaporate.

Sulfur is omnipresent component of both coal and oil and results in a significant environmental impact when

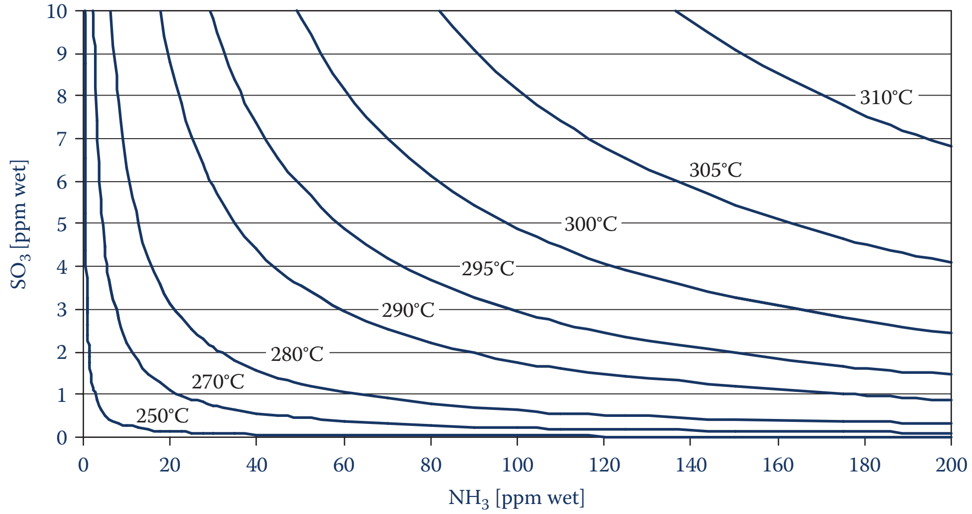


FIGURE 8.31
Example of ammonium bisulfate formation and reversible plugging of the pores at low operation temperature.

combusted, and the management of sulfur fossil-fuel-fired power plants or units where sulfur-consisted waste gas is combusted is therefore of great importance.

During the combustion of coal in the boiler or incinerators, virtually all sulfur is converted into SO_2, but a small fraction is further converted or oxidized into SO_3. The SO_3 reacts with residual ammonia to form ABS, which can foul downstream equipment. Also, SO_3 reacts with water vapor as the flue gas cools down and forms sulfuric acid mist that cannot be removed in common flue gas desulfurization (FGD) and therefore ends up in the stack as visible opaque plume.

It is a general fact that the active component in SCR catalysts, vanadium, is also catalyzing the formation of SO_3 from SO_2. This reaction is an unwanted reaction. The choice of SCR catalyst and the selection of the operating conditions are therefore crucial to the power plant owner or incinerator units. There are two possible ways of controlling the SO_2 oxidation: by varying the percentage of vanadium in the catalyst and by limiting the temperature. Most often, the temperature is not a variable in power plants or in thermal incinerator units, and the designer of a SCR unit therefore has to carefully adjust the catalyst properties to reach his or her goal.

The designer's dilemma is that reducing the SO_2 oxidation by lowering the percentage of vanadium in the catalyst also lowers the specific activity and the total catalyst volume therefore increases and the plant becomes more costly.

8.3.5.4.3.4 Catalyst The catalyst itself and the reactor that accommodates the catalyst are the most important components of the SCR method. The form and types of catalysts are described in this chapter. The catalysts used predominantly for industrial applications are honeycomb, plate type, and corrugated type.

Honeycomb catalysts: Homogeneous extruded honeycomb catalysts (full-ceramic honeycomb catalyst) consist entirely of catalytically active materials like titanium dioxide, vanadium oxides, and tungsten oxides. The honeycomb elements with its porous structure character consist approx 80% TiO_2. The catalysts have often the same cross-sectional dimension of 150 × 150 mm.

The typical application for honeycomb catalysts is the abatement of NOx under low-dust conditions. Examples are boilers firing gas and oil as well as gas turbines and internal combustion engines firing heavy fuel oil, low fuel oil, or gas. Exhaust gas from municipal waste incineration plants is another field of application. Tail end and low-dust configurations often employ honeycomb catalysts.

The extremely high specific surface area of honeycomb catalysts ensures a low catalyst volume and therefore a compact design of your SCR installation. The catalyst type and the required catalyst volume are individually designed and optimized based on the specified exhaust gas conditions. The elements are characterized by the length and by the pitch. The honeycomb pitch can most easily be represented as the number of honeycomb openings. The catalyst length mainly varies depending on operating conditions. The range of variation extends from 20 × 20 to 40 × 40 openings, which corresponds to a pitch of 7.5 and 3.7. Due to different applications, honeycombs are available with a pitch between 2.7 and 5 mm (0.11 and 0.20 in.) for low-dust applications and between 6.4 and 8.2 mm (0.25 and 0.32 in.) for high-dust applications.

The elements are arranged in element frames referred to as modules. The number of elements can variously depend on the supplier and application. Typically, the size of a module is 2000 × 1000 × 1000 mm (79 × 39 × 39 in.) (L × W × H).

Plate catalyst: Plate catalysts consist of catalytically active material composed of titanium dioxide, vanadium oxides, and tungsten oxides or molybdenum oxides rolled onto stainless steel mesh. The catalyst plates are integrated into element frames, which are installed in a steel module. The size of cassettes is 465 × 465 mm (18.3 × 18.3 in.). The height variously depends on plate length. Usually a length of 500 or 600 mm (20 or 24 in.) is used. Typically, the modules consist of two layers of elements and each layer arranged with 8 cassettes. The dimension of such kind of module is 2000 × 1000 × 1000 mm (79 × 39 × 39 in.) (L × W × H).

The typical application for plate catalysts is the abatement of NOx under high-dust conditions. Examples are plant heavy fuel oil, firing coal, and residual oil from refineries as well as industrial high-dust processes and wood-fired boilers.

Plate catalysts have some advantages compared to conventional honeycomb catalysts. Plate catalysts show minimal dust depositions even under extreme high-dust conditions and also exhibit a high erosion resistance. Additional benefits include low SO_2 conversion rates and low pressure drop. These result in a long service life with high NOx removal rates and minimal operating costs. The most appropriate catalyst type and the required catalyst volume are individually designed and optimized depending on the exhaust gas conditions. By varying the number of plates per element frame, the distance between two plates and thickness can be optimized over a wide range. The plate height varies depending on operating conditions. The pitch referred from the distance between two plates and thickness of plates characterized the catalyst. According to the height of the plate and distance between the plates, there are various specific active surfaces of the catalyst (indicated as surface m^2/catalyst volume m^3). The plate structure is housed in steel casings forming elements (cassettes). For SCR applications, modules consisting different numbers of cassettes are used.

Corrugated-type catalyst: The corrugated-type catalyst based on fiber-reinforced titanium dioxide (TiO_2, 5–10 wt%) carrier is impregnated with the active components in the SCR reaction such a tungsten trioxide (WO_3, 80–90 wt%) and vanadium pentoxide (V_2O_5, 0–5 wt%). These compounds are finely dispersed over the catalyst surface. The physical characteristic of corrugated-type catalyst is the low-pressure loss. The fiber-reinforced titanium dioxide carrier is flexible and thermoshock and erosion resistant, resulting in low deactivation rates and high mechanical durability.

The catalyst is shaped to a monolithic structure with a large number of parallel channels. The unique catalyst design provides a highly porous structure with a large surface area and an ensuing large number of active sites. A substantially lower weight than for conventional plate or extruded catalysts allows a fast response to changes in operation.

The active corrugated structure (see Figure 8.32) is housed in steel casings forming elements. The single cassettes are normally used in small installations where they are placed on tailor-made support frame. For bigger unit, modules that consist of different numbers of cassettes are used.

The corrugated-type catalyst is offered in a number of different models with varying channel size (often referred to as pitch), wall thickness, and chemical composition adapted to specific operating conditions. The choice of pitch and wall thickness for a given SCR installation is determined mainly by the dust content of the flue gas. For low-dust applications, channel sizes of 2–4 mm (0.08–0.16 in.), for example, with 0.4 mm (0.016 in.) wall thickness, are selected. Larger channel catalysts (7–10 mm or 0.28–0.39 in. pitch) with minimum 0.8 mm (0.03 in.) wall thickness should be selected for operation in dust-laden gases in units on coal-fired boilers.

8.4 Blowers

Blowers (also referred to as fans; see Volume 2, Chapter 3) are used to overcome the pressure drop required to move air/flue gas through a TO system consisting of multiple flue gas treating processes. Although blowers are also used to move waste gases, this section will discuss only combustion air and flue-gas blowers. The location of the blower is a major factor in blower selection. If a blower is located at the front end of a system, "pushing" flow through, the process is described as FD, the blower is called an FD fan, and the blower will be handling clean air. The FD blower will draw in air at ambient pressure and raise it to the pressure needed to push combustion products through the system.

If a blower is located at the back end of a system, "pulling" flow through, the process is described as ID, and the blower is called an ID fan. At that point, the gas handled will be combustion products, cooled sufficiently by heat recovery or quenching to prevent damage to the fan. Thus, an ID fan draws in conditioned flue gas at a pressure below ambient (vacuum) and raises it to just greater than ambient to exhaust it to the atmosphere.

Occasionally, a blower will be located at the front end, "pushing" through part of a system, while a blower at the outlet "pulls" through the rest of the components. In that case, process is called balanced draft. Obviously, the same criteria for blower selection apply as noted earlier.

When the blower location is determined, the final selection is based on inlet/outlet composition and volumetric flow rate, inlet/outlet pressure requirements, and inlet/outlet temperature ranges during operation, flow, and pressure.

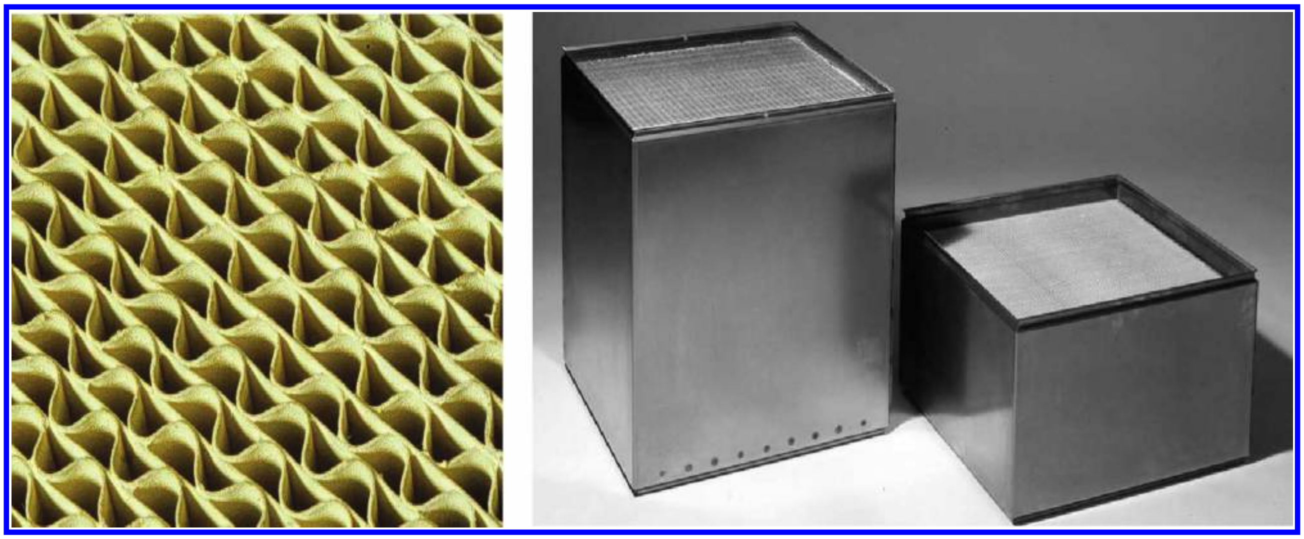


FIGURE 8.32
Corrugated catalyst and elements.

Two general classes of blowers exist for moving gas volumes. They are axial and centrifugal. Axial blowers use propellers to move the gas parallel to the axis of rotation of the blower. Although axial blowers are inexpensive to buy and install and occupy little space, they are very limited in pressure capability (less than 20 in. W.C. = 5 kPa) and very noisy at higher pressures. For these reasons, axial-type blowers are seldom used for multicomponent TO systems and will not be examined further in this chapter.

Centrifugal blowers use an impeller that draws gas in at the center of the impeller, parallel to the axis of rotation, and turns the gas 90° through the vanes in the impeller. Using vanes that can be as simple as paddle-like blades or more complex like aerodynamic wing-shaped blades, the centrifugal blower delivers radial-outward velocity to the gas. The impeller is mounted eccentrically within the casing of the fan so that the distance between the impeller and the casing forms a diverging nozzle with the maximum nozzle area being at the outlet of the fan. The gas that is accelerated by the impeller emerges from the periphery of the impeller and flows through the diverging nozzle (volute) of the casing effectively converting kinetic energy (velocity) into pressure energy. The centrifuge effect of throwing the air to the periphery of the impeller causes a low pressure at the inlet of the impeller, thereby drawing more gas into the blade space.

Centrifugal blowers can generate well over 100 in. W.C. (25 kPa) pressure, more than sufficient for almost any multicomponent system. As the gas is centrifuged to the periphery of the housing, more gas is drawn into the blade space.

There are three distinct types of centrifugal blowers. The difference is basically the blade configuration used. The three basic blade types are straight (or radial), forward curved, and backward curved. Figure 8.33a illustrates various types of fan wheel designs available. These are all variants of the three basic blade types. Each has its advantages and disadvantages. Table 8.3 summarizes those relative characteristics.

The proper selection of a centrifugal blower is not accomplished by any single straightforward formula. Experience, usage, and careful evaluation of each application are necessary to ensure proper fan selection. It is always best to work with blower manufacturers to get the most cost-effective recommendation.

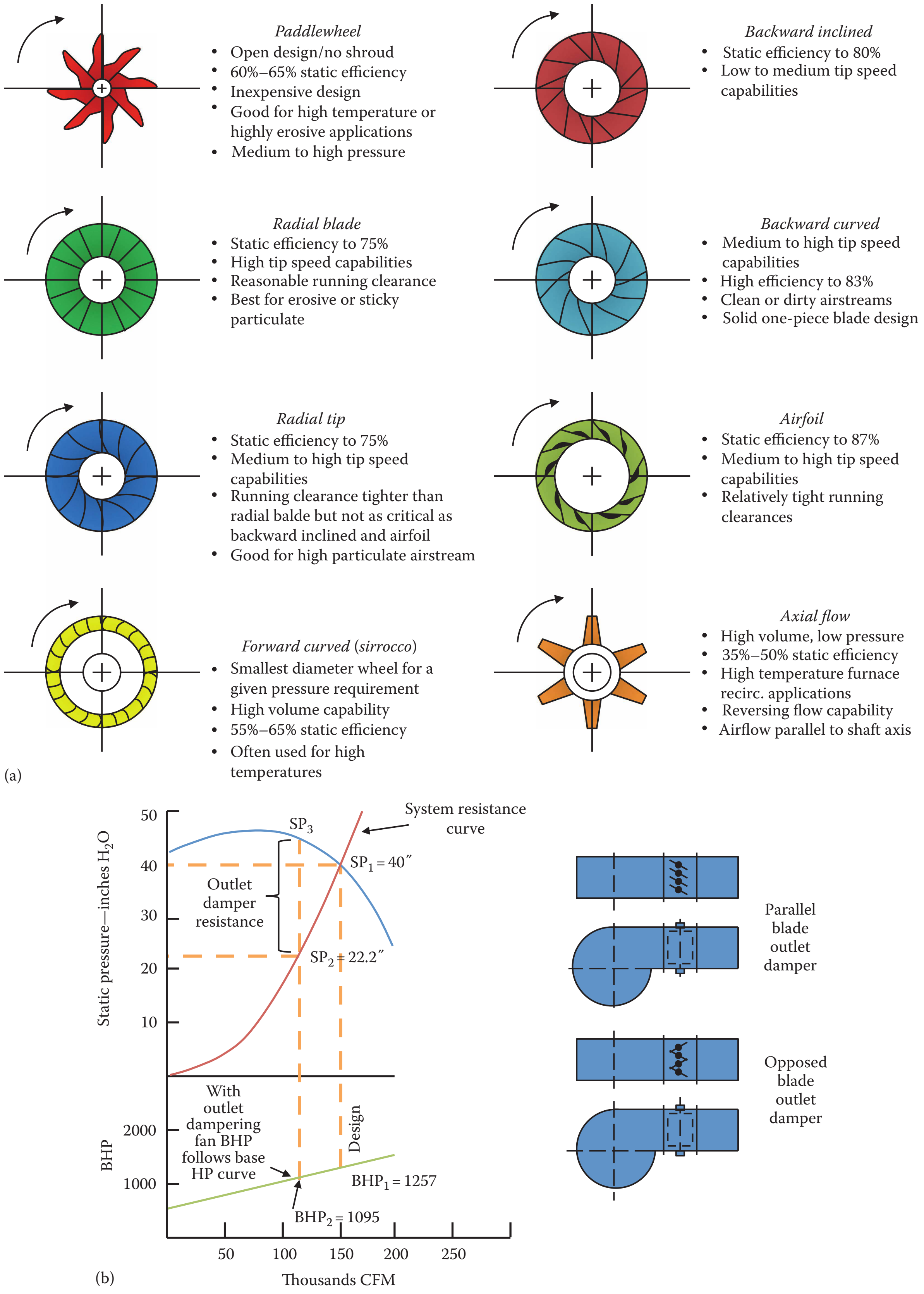
Another aspect of proper blower selection is utilizing the best-suited method of flow control for the blower to match the combustion system. Three primary methods are used. They are discharge-damper control, inlet damper/vane control, and fan speed control. The pressure flow–horsepower curves for each are shown in Figure 8.33b–d.

A discharge damper consists of one or more sliding or pivoting blades (such as a butterfly damper) that reduce flow area in a duct. Closing the damper increases resistance to flow and reduces flow. However, when flow is reduced, the operating condition of the blower (and the point on the fan curve) is shifted to lower flow and a corresponding higher pressure. This pressure is greater than required, and so some energy is wasted in the pressure drop taken across the damper. The blower performance curve is not changed by using a discharge damper. The horsepower usage ratio is reduced, but by less than the flow ratio change (see Figure 8.33b).

Inlet vane control is accomplished with a special damper that consists of multiple, adjustable vanes arranged either in a parallel array in a box (called simply, inlet vane damper) or radially from the centerline of the damper and mounted at the inlet to the blower's impeller (called radial vane damper). The radial vane damper is better than the parallel inlet vane damper because it can impart pre-spin to the air in the direction suitable to the impeller's action. In either case, as the vanes are rotated (adjusted) closed, the amount of air allowed into the blower inlet is reduced. In the case of the radial vane damper, the pre-spin modifies the basic characteristics of pressure output and power input, resulting in new and reduced pressure and horsepower characteristics. As the vanes of the radial vane damper are closed further, the flow of air is further reduced, but the spin is increased. This further reduces the pressure and horsepower characteristics. Effectively, the radial inlet vanes change the blower performance curve so that the horsepower reduction ratio is actually greater than the flow ratio change. A specially designed inlet box with a parallel blade damper directing the flow to one side of the box, effectively providing pre-spin at the blower inlet, provides very similar results (see Figure 8.33c).

Fan speed control for most combustion system blowers is accomplished by utilizing variable speed drivers (motors or turbines). Controlling the blower by varying its speed is ideal because very little energy is wasted. The theory behind variable speed control is that the volume of air flowing is proportional to the blower speed, the pressure developed is proportional to the square of the speed, and the horsepower required is proportional to the cube of the speed. Thus, unlike inlet vane control, for which the blower curves start at the same low flow point for each vane setting and change the end point for the high flow, variable speed control results in a completely separate performance curve for each blower speed. The net effect is that the horsepower reduction ratio is even greater than with inlet vane control (see Figure 8.33d).

As with selection of blowers, the selection of the method of flow control is not necessarily based simply on cost or efficiency. Often, a combination of two methods is needed, such as discharge dampers controlling flow to different parts of a combustion system, while



Paddlewheel
- Open design/no shroud
- 60%–65% static efficiency
- Inexpensive design
- Good for high temperature or highly erosive applications
- Medium to high pressure

Backward inclined
- Static efficiency to 80%
- Low to medium tip speed capabilities

Radial blade
- Static efficiency to 75%
- High tip speed capabilities
- Reasonable running clearance
- Best for erosive or sticky particulate

Backward curved
- Medium to high tip speed capabilities
- High efficiency to 83%
- Clean or dirty airstreams
- Solid one-piece blade design

Radial tip
- Static efficiency to 75%
- Medium to high tip speed capabilities
- Running clearance tighter than radial balde but not as critical as backward inclined and airfoil
- Good for high particulate airstream

Airfoil
- Static efficiency to 87%
- Medium to high tip speed capabilities
- Relatively tight running clearances

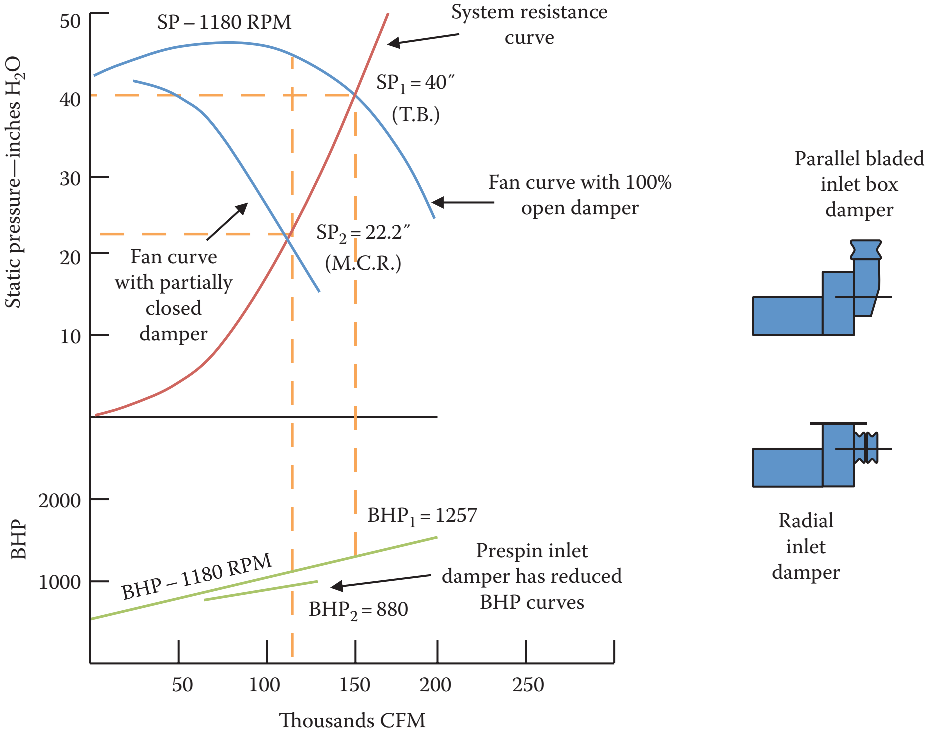
Forward curved (*sirrocco*)
- Smallest diameter wheel for a given pressure requirement
- High volume capability
- 55%–65% static efficiency
- Often used for high temperatures

Axial flow
- High volume, low pressure
- 35%–50% static efficiency
- High temperature furnace recirc. applications
- Reversing flow capability
- Airflow parallel to shaft axis

(a)

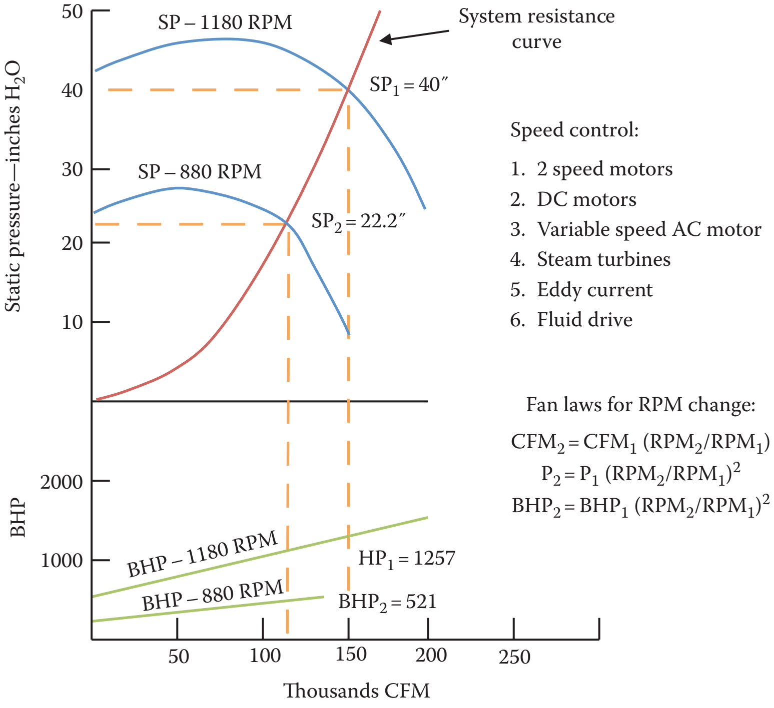
(b)

FIGURE 8.33
(a) Fan wheel designs, (b) outlet damper flow control.



Annual operating costs: (880 HP)(0.746 kW/HP)(365 days/year)
(24 h/day)(0.05\$/kWh) = \$275,722/year

(c)



Annual operating costs: (880 HP)(0.746kW/HP)(365 days/year)
(24 h/day)(0.05\$/kWh) = \$275,722/year

(d)

FIGURE 8.33 (continued)
(c) Radial inlet damper flow/inlet box damper flow control, and (d) blower speed control.

TABLE 8.3

Advantages and Disadvantages of Various Types of Fan Wheel Designs

	Radial	**Forward-Curved Radial Tip**	**Backward Curved**
Efficiency	Medium	Medium	High
Tip–speed	High	Medium	Medium
Size[a]	Small	Medium	Large
Initial cost[b]	Small	Medium	Large
HP curve	Medium rise	Medium rise	Power limiting
Accept corrosion coating	Excellent	Fair to poor	Good (thin coat)
Abrasion resistance	Good	Medium	Medium
Sticky material handling	Good	Poor	Medium
High-temperature capability	Excellent	Good	Good
Running clearance	Liberal	Medium	Minimum req'd
Operation without diffuser	Not efficient	Must use	Good efficiency
Noise level	High	Medium	Low
Stability/non-surge range[c]	Medium	Poor	Medium
	20%–100%	40%–100%	20%–100%

[a] The size is based on fans at the same speed, volume, and pressure.

[b] The cost is based on fans at the same speed, volume, and pressure.

[c] More a function of operating point along a curve than fan type.

the pressure upstream of the dampers is maintained constant by the use of an inlet vane damper.

Inlet vane dampers, be they parallel or radial, provide larger turndown than outlet dampers. Generally 3:1 turndown is possible with inlet vane dampers. The addition of an outlet vane damper to an inlet vane damper provides a further step of turndown resulting in 4:1. The speed control of electric motors using VFDs can easily achieve 5:1. In principle, up to 10:1 is possible with a VFD, but turndown ratios higher than 5:1 should be handled with care as overheating of the motor is likely.

As mentioned previously, in TO systems, it is common to have more than one air user fed by a single fan. Consequently, a discharge damper is required for each user, and a pressure control loop using either an inlet vane damper or a VFD is required to maintain constant pressure upstream of the discharge dampers, in order to ensure that enough air reaches each user regardless of the demands of the other users. In such a configuration, it is important to note that at extreme turndown conditions such as during start-up, the pressure control loop will drive the fan to maintain high pressure at low flow causing the fan to run at a point extremely to the left on its static pressure curve, thus driving it into surging.

Surging as the name implies is drastic fluctuation of pressure due to loading of the impeller to its maximum and resulting depressurization due to internal recirculation (churning) within the impeller and casing of the fan. Surging will generally be audible near the fan and will be evident in the flowmeter readings. Importantly, surging results in much reduced flow output from the fan and should be avoided because it is generally accompanied by high vibration in the fan. Generally, the possibility of surging occurring during high turndown conditions is avoided by supplying a lower-pressure set point in the pressure control loop when low flow is required.

If the system design calls for high turndown beyond the capability of the usual controls, a vent valve may be added that allows the fan to run safely in its design region, but vents some of the output of the fan to reduce the flow to the system. This, of course, wastes energy in the vented fraction of the fans output. However, in large systems where ranges of flow can be extreme sometimes, venting is the most operation-friendly design solution.

8.5 Closing

The purpose for presenting this chapter is to attempt to provide a better understanding of the use of thermal oxidation processes to destroy vapor and liquid wastes. Although specific design details were not included, the important aspects of the "big picture" related to design of good TO systems have been addressed.

Nomenclature

APC	Air pollution control
CSA	Collecting surface area
DRE	Destruction and removal efficiency
ESP	Electrostatic precipitator
FRP	Fiberglass reinforced plastic
LEL	Lower explosive (flammability) limit
MMBtu/h	Million Btu/h
PCB	Polychlorinated biphenyl
SCR	Selective catalytic reduction
SCS	Specific collector surface
SNCR	Selective non-catalytic reduction
TO	Thermal oxidizer
WESP	Wet electrostatic precipitator

References

1. S. Londerville and C. Baukal (Eds.), *The Coen-Hamworthy Combustion Handbook*, CRC Press, Boca Raton, FL, 2013.
2. C. D. Cooper and F. L. Alley, *Air Pollution Control: A Design Approach*, 2nd edn., Waveland Press, Prospect Heights, IL, 1994.
3. W. Licht, *Air Pollution Control Engineering: Basic Calculations for Particulate Collection*, Marcel Dekker Inc., New York, 1980.
4. R. D. Reed, *Furnace Operations*, 3rd edn, Gulf Publishing, Houston, TX, 1981.
5. M. Schultes, *Abgasreinigung Verfahrensprinzipien, Berechnungsgrundlagen, Verfahrensvergleich*, Springer-Verlag, Berlin, Germany, 1996.
6. GMBI Nr. 25-29 S. 511-Gemeinsames Ministerialblatt, Erste Allgemeine Verwaltungsvorschrift zum Bundes-Immissionsschutzgesetz (Technische Anleitung zur Reinhaltung der Luft-TA Luft), Carl Heymanns Verlag KGKöln, Germany, 2002.
7. EPA-452/F-03-031, Air Pollution Control Technology Fact Sheet SNCR. http://www.epa.gov/ttn/catc/dir1/fsncr.pdf (accessed on March 19, 2013).
8. R. Bühler, Erfolgskontrolle SNCR-Verfahren zur Entstickung von Holzfeuerungen, Bundesamt für Energiewirtschaft ENET, Bern, Switzerland, 1996.
9. B. Reese, W. Klaus, and D. Müller, Schadstoffgemische: Stickoxidarme Entsorgung, Verfahrenstechnik 30 Nr. 3, Vereinigte Fachverlage GmbH, Mainz, Germany, 1996.
10. European Commission, Integrated Pollution Prevention and Control. Reference document on best available techniques for the manufacture of organic fine chemicals, Chapter 4, 2006. http://www.ineris.fr/ippc/sites/default/files/files/ofc_bref_0806.pdf (accessed on March 19, 2013).
11. H. Bosch, and F. Janssen, *Catal. Today* 2, 369, 1988.

9

Thermal Oxidizer Control and Configurations

Bernd Reese, Wolfgang Klaus, Jay Karan, and Juergen Foelting

CONTENTS

9.1 Introduction

Chapter 8 discusses the basics of thermal oxidizers (TOs) including some of the important system building blocks. This chapter discusses some of the specific equipment used in TOs. Regardless of the specific design, most thermal oxidation systems consist of some or all of the following components:

- A device and method to supply oxygen to the process to initiate and sustain oxidation using burner or catalyst
- A vessel (combustion chamber/TO) to provide the residence time required for oxidation of the hydrocarbons
- A heat recovery (heat exchanger or boiler) and/or flue gas conditioning system
- Emission control equipment (filters, scrubbers, etc.) to treat the flue gas prior to discharge to the atmosphere
- An elevated exhaust point (stack) through which the flue gas can be dispersed into the atmosphere
- Control hardware and logic to automatically maintain and monitor the various process parameters to ensure safe operation

9.2 Control Systems and Instrumentation

All TO systems require some sort of control, if only a flame failure switch to ensure waste and fuel shutoff for safety reasons. Controls can be classified into flame safeguard and process control functions (see Volume 2, Chapter 2, for a discussion of combustion controls).

Flame safeguard requirements evolved from insurance and general safety regulations for fuel-fired burners in general. This part of a control system is usually designed to satisfy detailed rules published by the National Fire Protection Association (NFPA), Industrial Risk Insurers (IRI), or Factory Mutual (FM). Flame safeguards are designed to ensure that the fuel and waste flows to an incinerator are stopped if the flame is lost and that the furnace is fully purged of combustibles prior to ignition so that the potential for explosions is eliminated. Generally, the major components of these systems include

- Flame sensor (flame scanner, flame rod, or other methods) to ensure that either a stable flame is maintained (normal operation) or no flame is present (during purge)
- Fuel supply pressure switches to verify the fuel supply is within the design range
- Air supply flow or pressure switches to verify an adequate supply of combustion air is available
- Automatic shutoff block valves to turn off the fuel flow and waste flow

Process controls are provided to keep the system operating within boundaries, to meet legal emission requirements, and to protect the equipment from operational damage. Some form of automatic temperature control is normally used to adjust fuel and/or quench (air, water, or steam) flow to the unit so that the waste is burned properly without exceeding the temperature limits of the refractory lining. In many cases, the outlet O_2 concentration is monitored and used to control the combustion airflow. Downstream flue gas treating equipment will be built for a much lower temperature than the TO, so the flue gas cooling process parameters (boiler water level, water spray flow rate, etc.) and the resulting flue gas temperature are always monitored to avoid expensive thermal damage. Automatic steps (system shutdown, hot gas diversion, etc.) can also be provided to deal with various failures. For scrubbers, the flow rate and pH of the circulating liquids are controlled automatically to ensure proper removal of acid gases. Less obvious are controls applied to deal with specific variations in incinerator waste feed streams. In some applications, the waste flow and composition are expected to change abruptly. When waste flow and composition are expected to change abruptly, the control method required to maintain effective system performance (or even flame stability) could become very complex. The speed of analyzer or thermocouple response often plays a major part in control system design. In these cases, control system (and burner) design experience is absolutely critical.

9.2.1 Multipurpose Automation Philosophy for TO Plants

To be state of the art, a modern TO for the pharmaceutical and chemical industries needs to be able to automatically handle fluctuating loads with a minimum of operator intervention. In this industry, the TO is usually run as a disposal plant. Customers attempt to minimize investment and maintenance and focus on the operation of the production unit. However, with emission limits becoming ever more stringent, the disposal plant and its reliability become more and more critical.

For this purpose, the disposal plant needs to be designed so that it will handle the load swings coming from the production plant. All necessary process equipment as well as the automation concept needs to be developed individually for each plant in order to meet that requirement and to ensure safe and reliable operation during the plant's lifetime. Installation and commissioning matters are equally to be taken into account to meet the customer's demand. Simulating not only the individual design load cases but also the dynamic behavior in between them, John Zink helps to determine the individual requirements of the TO.

At the beginning of such plant engineering, the complete unit from the feeding lines through the combustion stage and the eventually installed heat recovery unit as well as a flue gas scrubber system needs to be analyzed by process and control technologists. During that effort, all possible dynamics coming from the plant itself or from the production side should be considered and the means to react on them worked out.

Design values are determined by theoretical calculation of the combustion for the specified load cases from the cold start-up of the plant to the maximum load case. This design step is performed by a complete project team consisting of at least a process engineer, instrument and control engineers, and safety engineers, to ensure that all the possible cases are investigated. With that, the necessary information is available to define the remaining load cases and dynamic transition processes, caused either by starting and stopping individual waste streams or by production plant issues. The latter are more difficult to evaluate. After this information is available, the means are selected to realize the maximum automation level for the plant. Sometimes it makes sense to have an additional discussion with the owner of the production plant to adapt the combustion system to the plant operation philosophy.

Some typical means to establish such an automation concept are

1. Permanent stoichiometric air balance
2. Permanent energy balance

Calculation of stoichiometric air demands allows for a short reaction time in the event of fluctuating waste streams entering the chamber. The temperature inside the combustion chamber would not see the energy changes as fast as the flow measurements of the individual streams. With this concept, one can ensure an appropriate reaction against load swings, provided that the compositions of the incoming waste streams are

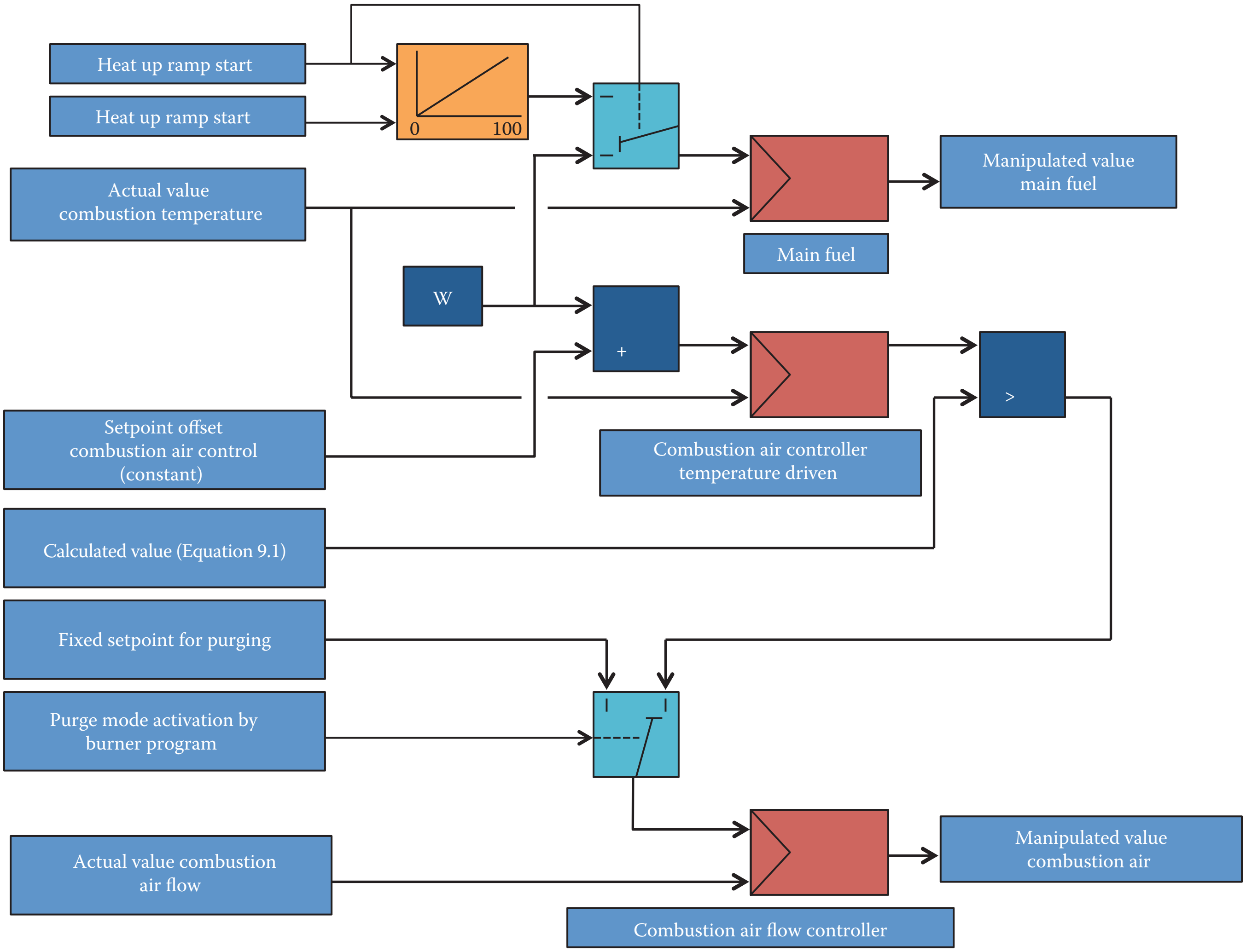


FIGURE 9.1
Typical control schematic of a combined fuel/air controller suitable for set points coming from an external calculation.

known within the specified range. Figure 9.1 shows a typical control schematic of a combined fuel/air controller suitable for setpoints coming from an external calculation (e.g., calculation shown as follows).

Fuel/air calculation used to foresee the actual air demand:

$$\dot{V}_{CAc} = ((\dot{V}_{FG} * k_{FG} + \dot{V}_{WG1} * k_{WG1} + \dot{V}_{WG2} * k_{WG2} + \dot{m}_{WL1} * k_{WL1} + \dot{m}_{WL2} * k_{WL2}) * k_{st}) * k_{ex} \qquad (9.1)$$

where

$\dot{V}_{CAc}$ as necessary flow of combustion air for safe and proper combustion (Nm^3/h)

$\dot{V}_{FG}$ as actual main fuel flow (fuel gas in this example) (Nm^3/h)

k_{FG} as stoichiometric demand of combustion for the main fuel ($\mathrm{Nm}^3/\mathrm{Nm}^{3)}$

$\dot{V}_{WG1}$ as actual flow of waste gases (Nm^3/h)

k_{WG1} as stoichiometric demand of combustion air for the waste gases ($\mathrm{Nm}^3/\mathrm{Nm}^3$)

$\dot{m}_{WL1}$ as actual flow of waste liquids (kg/h)

k_{WL1} as stoichiometric demand of combustion air for the waste gases (Nm^3/kg)

9.2.1.1 *Permanent Energy Balance*

Here the plant's air/fuel ratio follows a calculation based on an energy balance. A typical application for an energy-balance-supported controller design is a two-stage combustion unit, where the upper stage temperature can barely be measured, but is important to know and where the main fuel does not have a constant calorific value as shown in Figure 9.2.

The individual energy amounts are defined as:

$$Q_{FG1} = Q_{FLGstage1} - Q_{WL} - Q_{ATair} - Q_{CA1} \qquad (9.2)$$

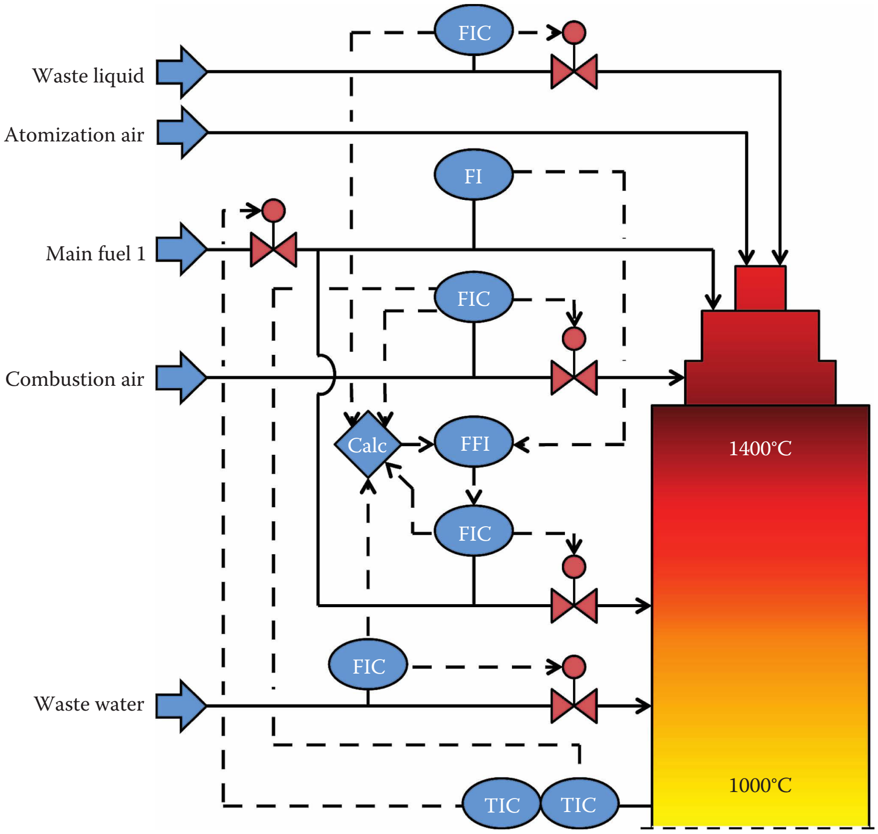


FIGURE 9.2
Example of a two-stage combustion unit (PFD).

and

$$Q_{FG2} = Q_{FLGstage2} - Q_{FLstage1} - Q_{WW} \tag{9.3}$$

The individual energy amounts can be described as

$$Q_{FG1} = \dot{m}_{FG1} * LCV_{FG} \tag{9.4}$$

$$Q_{FG2} = \dot{m}_{FG2} * LCV_{FG} \tag{9.5}$$

$$Q_{WL} = \dot{m}_{WL} * LCV_{WL} \tag{9.6}$$

$$Q_{WW} = \dot{m}_{WW} * LCV_{WW} \tag{9.7}$$

$$Q_{ATair} = \dot{V}_{ATair} * cp_{ATair} * t_{ATair} \tag{9.8}$$

$$Q_{CA1} = \dot{V}_{CA1} * cp_{CA1} * t_{CA1} \tag{9.9}$$

$$Q_{FLstage1} = \dot{V}_{FLstage1} * cp_{FLstage1} * t_{FLstage1} \tag{9.10}$$

$$Q_{FLstage2} = \dot{V}_{FLstage2} * cp_{FLstage2} * t_{FLstage2} \tag{9.11}$$

Taking this into account, and looking for the ratio of heat to stage 2 to overall released heat, one can say that

$$\frac{\dot{m}_{FG1} * LCV_{FG}}{\dot{m}_{FGges} * LCV_{FG}} = \frac{Q_{FLGstage1} - Q_{WL} - Q_{ATair} - Q_{CA1}}{Q_{FLGstage2} - Q_{FLstage1} - Q_{WW}} \tag{9.12}$$

Use the following values:

$Q_{FLGstage1}$ as heat of flue gas in combustion stage one
$Q_{FLGstage2}$ as heat of flue gas in combustion stage two
Q_{FG1} as heat of fuel gas stage 1
Q_{FG2} as heat of fuel gas stage 2
Q_{CA1} as heat of incoming combustion air in stage one
Q_{WW} as heat of incoming waste water
Q_{WL} as heat of incoming waste liquid
Q_{ATair} as heat of incoming atomizing air
m_{FG1} as actual fuel gas mass flow to stage 1 used for the ratio control (kg/h)
m_{FGges} as actual overall fuel gas mass flow used for the ratio control (kg/h)
LCV_{FG} as calorific value for fuel gas that will be cancelled
m_{WL} as actual waste liquid mass flow (kg/h)
LCV_{WL} as calorific value for waste liquid (kJ/kg)

m_{WW} as actual waste water mass flow (kg/h)
LCV_{WW} as calorific value for waste water (kJ/kg)
V_{ATair} as volume flow for atomization air (used as constant in this example) (Nm^3/h)
cp_{ATair} as atomization air cp (kJ/[Nm^3 * K])
t_{ATair} as actual value of atomization air temperature (used as constant in this example) (°C)
V_{CA1} as volume flow for atomization air (Nm^3/h)
cp_{CA1} as atomization air cp (kJ/[Nm^3 * K])
tA_{CA1} as actual value of atomization air temperature (used as constant in this example) (°C)

Knowing this ratio and having the total amount of fuel gas controlled by the temperature, one may split between fuel gas to stage one and fuel gas to stage two. This allows one to run the plant in a wide range without the danger of local overheating.

Right after the plant's behavior is determined, the next step is to translate this behavior into the control philosophy. From experience, the key to a successful "multipurpose" plant is to view the whole waste disposal system as a part of the production plant and to adapt the TO to all possible operating cases the production may require.

With the aforementioned methodology, plants have been built for many years that operate in a fully automatic mode throughout the year, disposing of all individual waste gas streams together with the incurred waste liquids. Figure 9.3 shows a diagram for a typical multipurpose TO application.

Another noteworthy advantage of this technology is the potential to run a disposal plant as energy efficiently as possible within the specified ranges and with increased safety due to minimal operator intervention. The challenge is to find the optimum between energy efficiency and stable operation by establishing a large turndown ratio and safety requirements.

For instance, it is necessary for stable operation at all times to adapt the actual flue gas volume to the load changes. Too much flue gas volume may cause inefficient combustion that wastes energy. On the other hand, the chosen safety devices need to ensure proper and complete combustion for each individual plant load case (see Figure 9.4).

In this example, it is assumed a plant that always has enough combustion air for a stable combustion, even for full load, and in the second case, a plant that controls the combustion air depending on the actual amount of waste liquid. The fuel gas saving is notable especially when the plant is running at low capacity. This concept will also work with several individual waste streams.

It is absolutely essential to develop the safety interlocks of the system together with the control and operational part. This is different than most combustion plants, where a so-called burner management system (BMS) takes care of the interlocks, and separate from that, a distributed control system (DCS) is used to operate and control the plant. This conventional design necessitates a huge number of operator inputs (see Figure 9.5).

The control system of multipurpose plants however should comprise both the safety part and the control part; both are completely integrated. As a result, this control philosophy allows running the plant nearly without operator input (see Figure 9.6).

The big advantages for this philosophy are

- Less battery limits inside the plant
- Less interconnection points to be clarified during project development
- Less commissioning time, due to the fact that the unit is able to run independently from the customer's DCS
- No DCS software engineer necessary for unit combustion
- Less operator inputs during normal operation will cause less possibilities for tripping the plant by improper operation

To complete the system, other parts of the plant should be integrated right from the beginning like waste heat boilers, flue gas scrubbers, and wet electrostatic precipitators (WESPs) or dust filter units.

9.3 System Configurations

The previous sections of this chapter covered the basics of the combustion process, the individual components that make up the combustion section, and the post-combustion flue gas treatment sections. The choice of these components for any total system depends primarily on the nature of the waste stream to be destroyed and the emission requirements for the flue gas ultimately exhausted to the atmosphere. This section is a discussion of suggested system configurations designed for the most common types of waste streams. The types are

- Non-acid gas endothermic waste gas/waste liquid
- Non-acid gas exothermic waste gas/waste liquid
- Sulfur-bearing acid gas (includes pulp and paper)
- Halogenated (chlorine-bearing) acid gas
- Explosive gases and complex liquids with non-continuous/varying compositions
- Salts/solids, i.e., solids that melt (down fired)
- NOx minimization or reduction

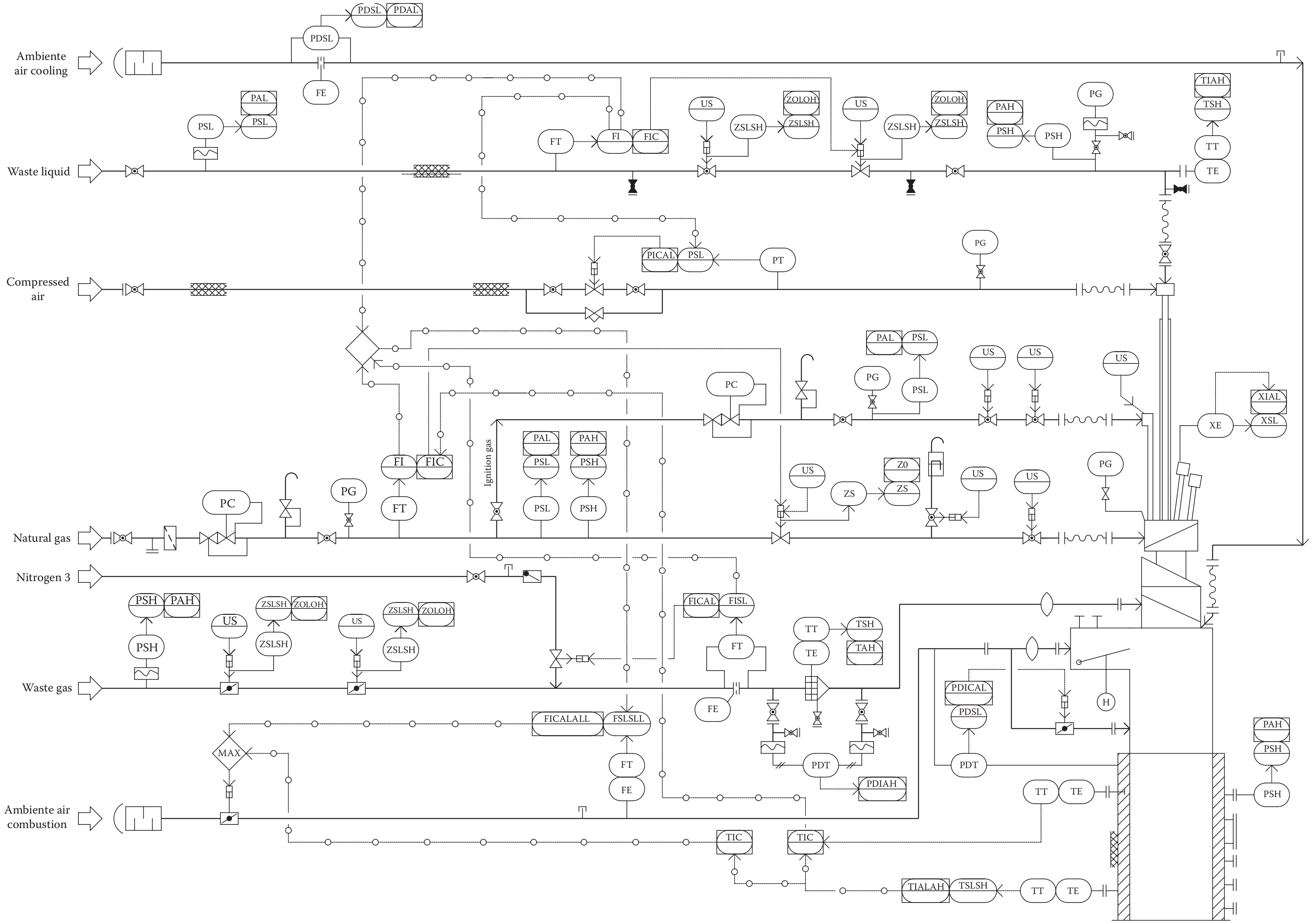


FIGURE 9.3
Example process and instrumentation diagram for a typical multipurpose TO application.

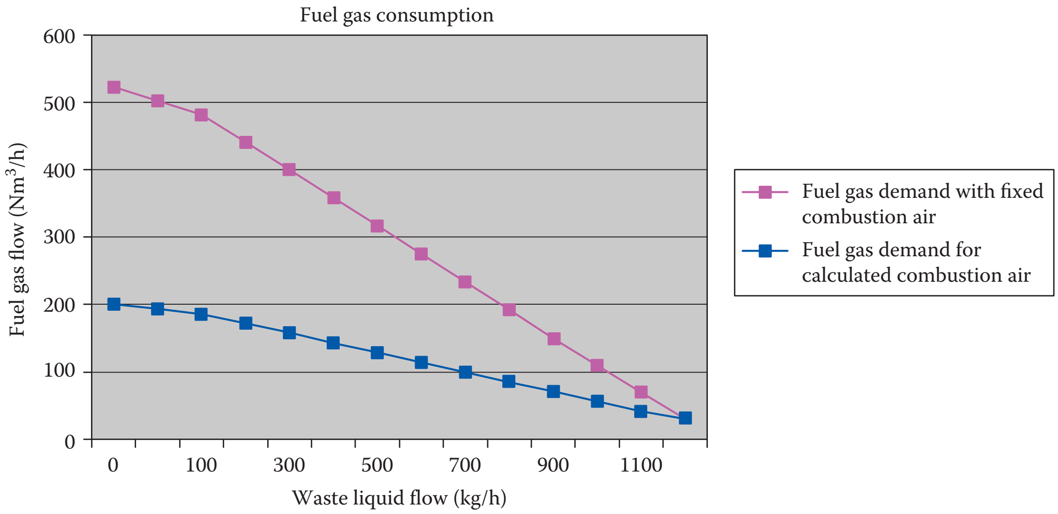


FIGURE 9.4
Example for the difference between conventional combustion air supply and energy-optimized operation in a TO unit with one combustible waste liquid.

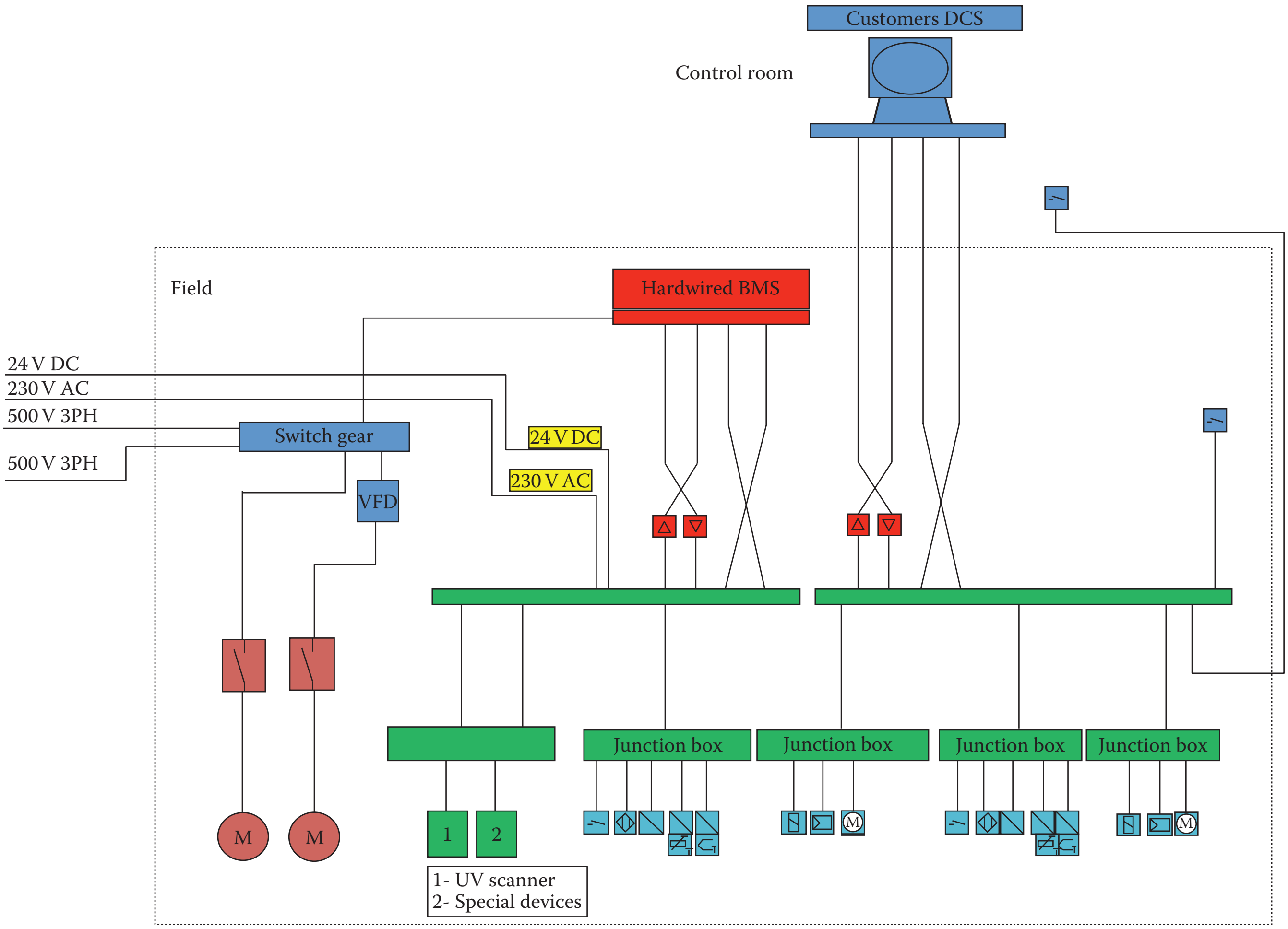


FIGURE 9.5
Typical automation concept with separated BMS and DCS.

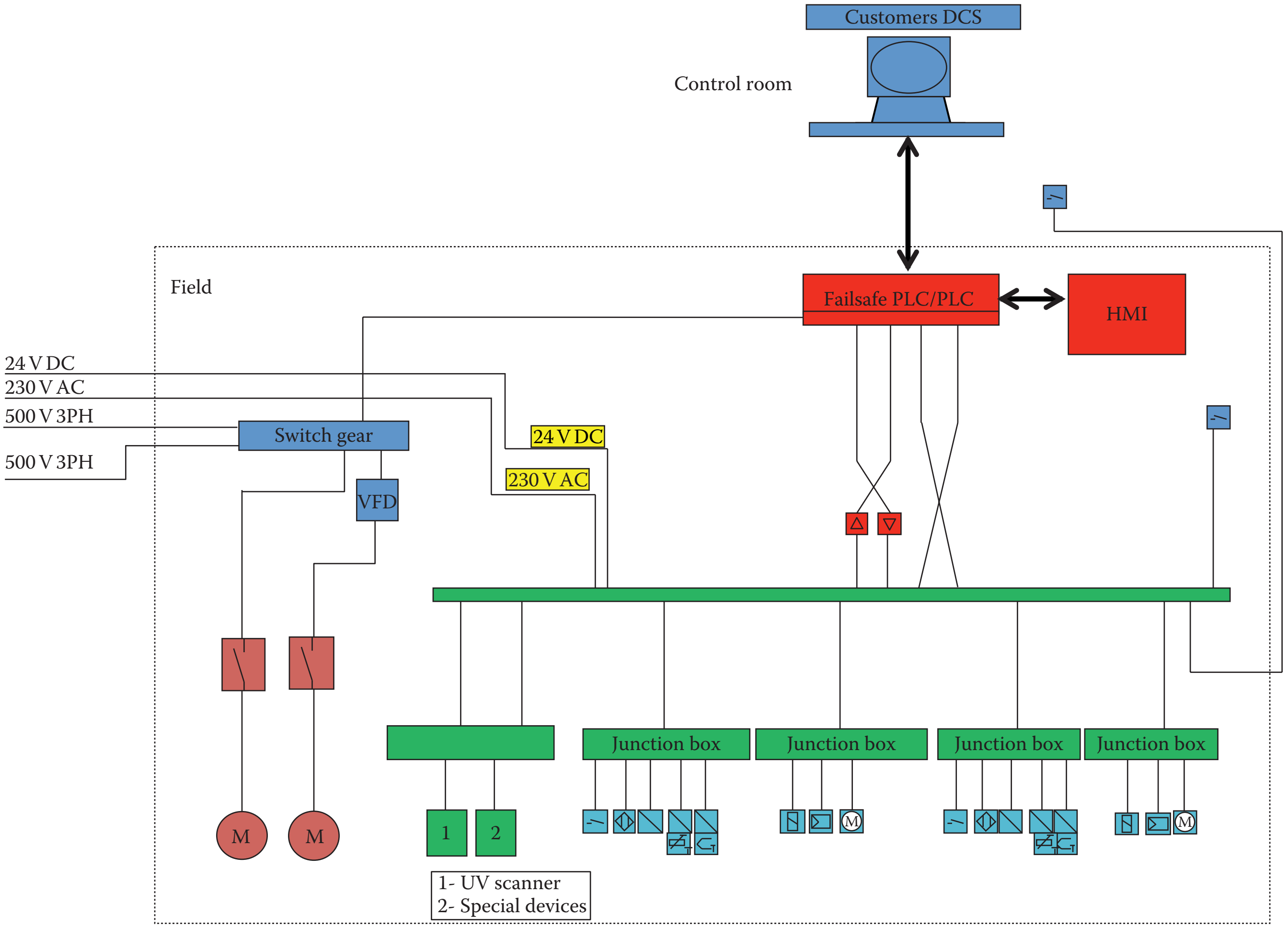


FIGURE 9.6
Example for a combined automation concept.

9.3.1 Non-Acid Gas Endothermic Waste Gas/Waste Liquid System

Endothermic waste gas and waste liquid are often handled similarly. Examples are waste gases containing primarily inert materials such as N_2 (including air) or CO_2 with some hydrocarbon contamination or waste liquid that is water with some small amount of organic. For these cases, the system would be required to introduce significant amounts of fuel and air to bring the waste components to the temperature required to oxidize them. The simplest system option is a vertical TO with either a natural or forced-draft burner. The flue gas exhausts directly to the atmosphere.

Often, it is economically desirable to recover heat from the flue gas of such systems by making steam or by preheating the waste gas stream to reduce fuel usage. Preheating a waste liquid is usually not cost-effective.

If the waste gas is mostly air contaminated with a small amount of hydrocarbon, but with a heat content that is below the lower flammability (or explosive) limit (LEL), standard thermal oxidation is applicable. However, other methods such as catalytic oxidation and regenerative oxidation can be more economical.

In all cases, fuel-firing capability is required to maintain the correct outlet temperature except in the case of the catalytic system, where the fuel firing is used only to heat up the unit prior to introducing the contaminated air, but can be turned off during normal operation.

Figure 9.7 shows a simple TO. Fuel and combustion air enters the burner while the inert waste gas and/or inert waste liquid are introduced downstream of the fuel/air combustion zone. Because no heat recovery or flue gas treatment is needed, the most cost-effective design is a refractory-lined, vertical upflow unit. The refractory-lined vessel functions as both the combustion chamber, for residence time, and the stack, to disperse the flue gas to the atmosphere. The operating temperature is maintained at the minimum possible to achieve destruction and removal efficiency (DRE), so the least amount of fuel will be used. For discussion purposes, a waste

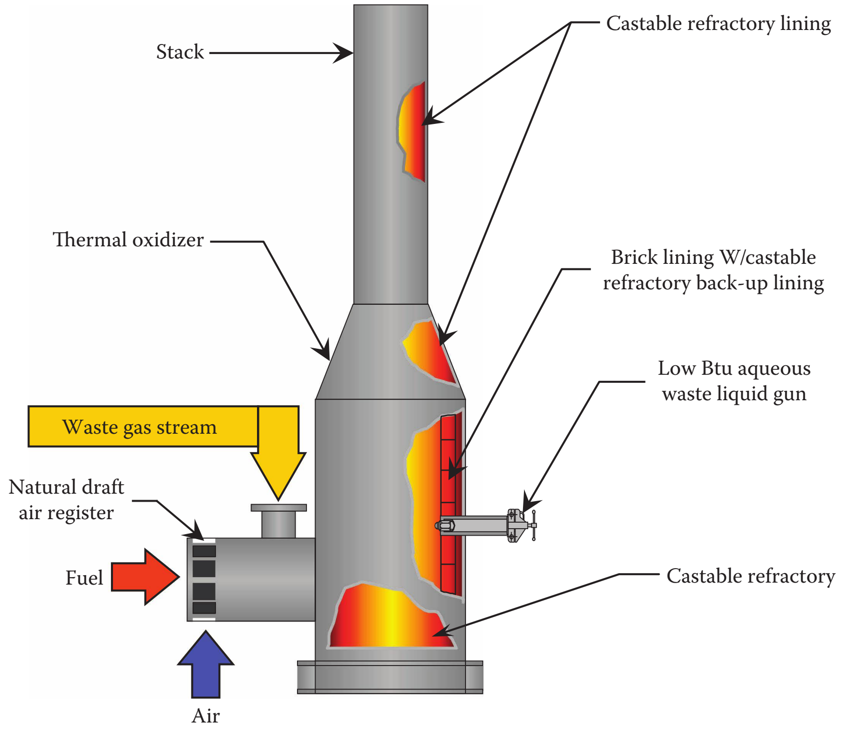


FIGURE 9.7
Simple vertical TO.

gas stream of 100,000 lb/h (45,000 kg/h) of N_2 at 80°F (27°C) with no hydrocarbon content will be considered. The operating temperature chosen is 1500°F (820°C). To heat the waste to the chosen operating temperature, the calculated fuel requirement, ignoring heat losses, is slightly greater than 65 MMBtu/h (19 MW).

In a system that uses a large amount of fuel, recovering some of the fuel heating value can quickly pay for the additional heat recovery equipment. If a TO is to be located in a plant that needs steam for process use, a system configuration represented by Figure 9.8 should be selected over a vertical TO system. The refractory-lined TO itself is horizontal, so flue gas can exit directly into the boiler, which can be either a fire-tube or a water-tube configuration. As noted earlier in Chapter 8, fire-tube boilers are normally recommended for smaller units with lower steam pressure

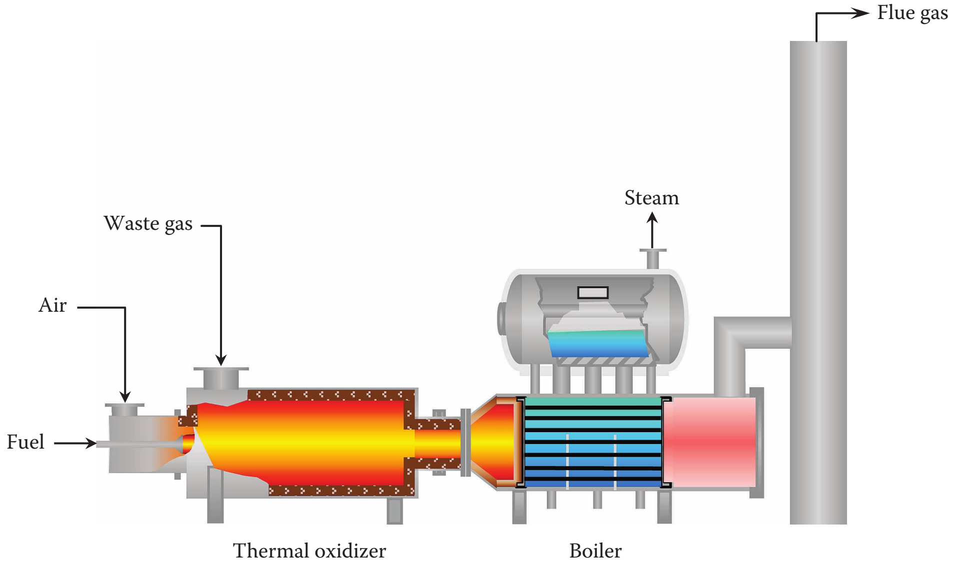


FIGURE 9.8
TO system generating steam.

requirements, while water-tube boilers are preferred for larger systems with higher steaming rates and/or higher steam pressures. For the same N_2 waste case noted earlier, a boiler could recover as much as 47 MMBtu/h (14 MW) and produce as much as 40,000 lb/h (18,000 kg/h) of 150 psig (10 barg) saturated steam by cooling the flue gas from 1500°F to 500°F (820°C–260°C) (a recovery efficiency of about 70%). By utilizing an economizer to further cool the flue gas to 350°F (180°C), about 45,000 lb/h (20,500 kg/h) of steam could be produced (a recovery efficiency of more than 80%). Because the flue gas exit temperature is 500°F (260°C) or less, an economical, unlined carbon steel vent stack can be used.

If there is no use for steam in the plant, the next best choice would be to recover heat into a heat transfer medium (hot oil) used in the plant. The heat recovery capability of the hot oil system can be as large as the boiler, and the cost of the heat recovery equipment may be lower than a boiler.

If there is no use for steam or heating of a heat transfer medium, the heat can be "recouped" by transferring heat from the hot flue gas exiting the TO to the incoming waste gas. Figure 9.9 illustrates such a system. The process is usually referred to as a recuperative process because energy is being removed from the flue gas and put back into the TO itself as opposed to into some other plant system.

Generally speaking, heat recovery into a liquid-based system such as a boiler or heat transfer medium poses less operational problems than a recuperative heat exchanger. Even between a boiler and a heat transfer medium heat exchanger, the boiler provides significant advantages. The boiler can handle wide swings in inlet temperature without sustaining thermal damage. Higher heat inputs just produce more steam. Moreover, since the heat transfer coefficient on the waterside of the boiler is much higher than the gas side, the tube metal temperature always runs very close to the temperature of the water/steam mixture, i.e., the saturated steam temperature, regardless of the inlet temperature of the flue gas. That is not true of a superheater, of course, because it is a gas-to-gas heat exchanger and the heat transfer coefficient on both sides of the tube wall is very similar. The superheater would have the same limitations as other gas-to-gas heat exchangers discussed later. So, a boiler containing a superheater section will have some of the limitations of gas preheaters.

A heat transfer medium heat exchanger benefits from a much higher heat transfer coefficient on the liquid side, thereby reducing risk of tube metal damage due to high flue gas temperatures. However, the heat transfer medium that is often a petroleum-based or synthetic oil has a relatively low cracking temperature that can be reached if the flue gas temperature or flue gas flow rate gets high enough. If the cracking temperature is reached on the inside of the tube metal surface, the oil in contact with it will carbonize (coke) and form solid carbon deposits on the inside of the tube. The carbon deposits are a poor conductor of heat and cause the tube metal temperature to rise drastically causing further coking, eventually leading to accelerated tube failure. So if wide swings in flue gas temperature or flow rate are expected into the heat transfer medium heat exchanger, it is preferable to install it in a bypass arrangement with an ID

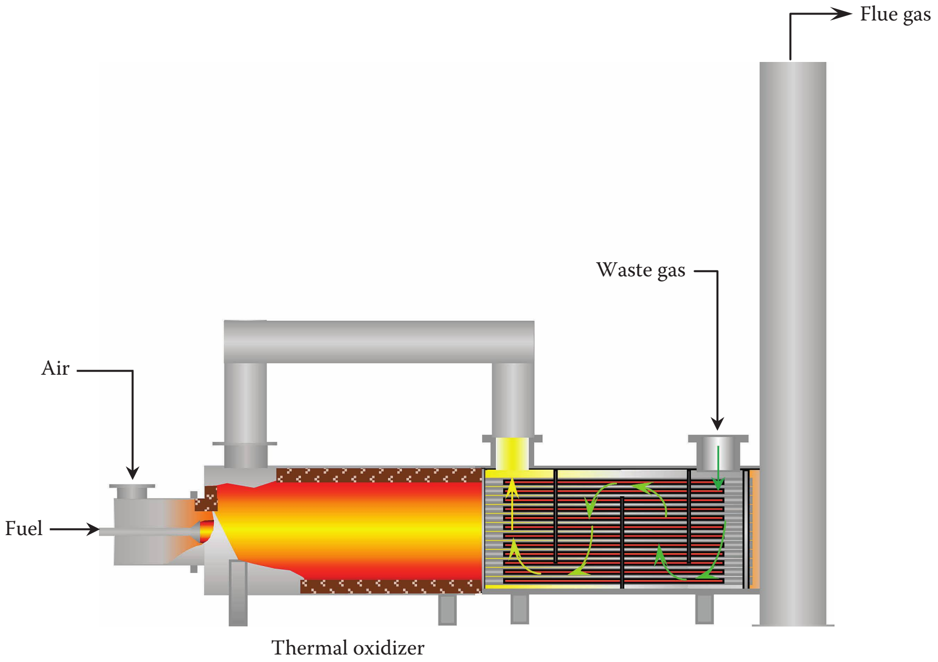


FIGURE 9.9
Heat recovery thermal oxidation system.

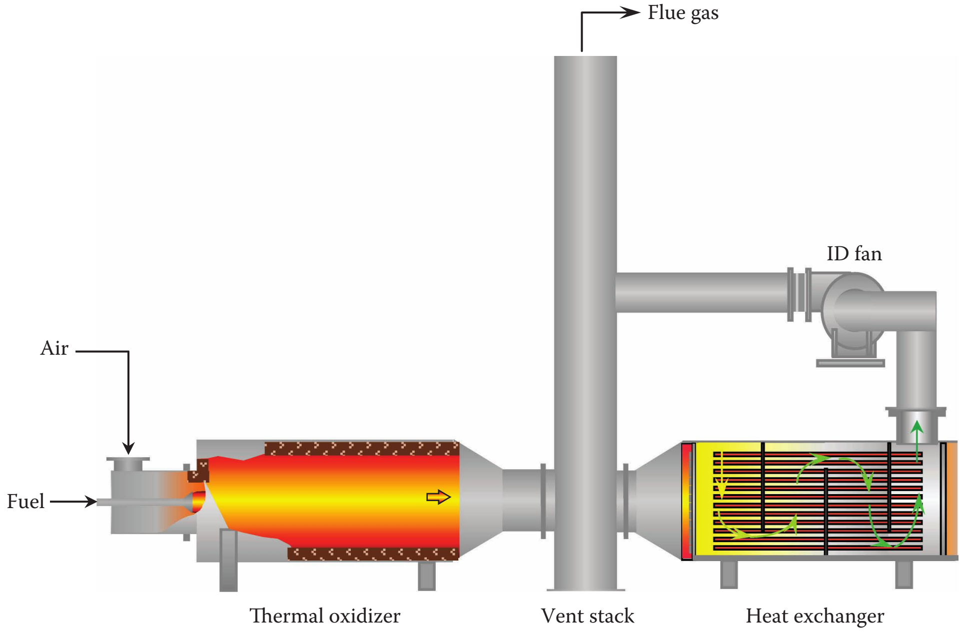


FIGURE 9.10
Bypass recuperative system.

fan (as shown in Figure 9.10) or using a bypass damper, to control the flue gas flow through the heat exchanger. It should be noted that flue gas bypass dampers are commonly available only up to flue gas temperatures of to 1200°F (650°C). Beyond that temperature, exotic valves that are costly and limited in size will be required. The operating temperature of TO systems is usually higher than 1200°F (650°C), so the ID fan approach is usually a better solution, although it is more expensive.

Recuperative heat recovery provides the best economics because it directly reduces the high cost of fuel encountered in endothermic systems, without the need for ancillary equipment such as water purification systems, boiler feedwater pumps, and condensers in the case of boilers or hot oil recirculation pumps and expansion tanks in the case of heat transfer medium systems. However, recuperative heat exchangers pose the greatest operational challenge. This is because the tube metal temperature of recuperative heat exchangers, which are gas-to-gas heat exchangers, runs midway between the temperatures of the hot flue gas on the outside and the gas being heated on the inside. The peak tube metal temperature occurs where the gas being heated exits the heat exchanger because that is where the difference in temperature between the flue gas outside the tube and the gas being heated inside the tube is smallest. Using a parallel flow heat exchanger instead of a counterflow heat exchanger helps to reduce the peak metal temperature at the outlet, but requires larger surface area, resulting in a more expensive heat exchanger.

If a recuperative heat exchanger is designed to maximize heat recovery, small changes in flue gas temperature, i.e., 50°F–100°F (30°C–60°C), or small changes in flue gas flow rate can drive the tube metal temperature to the alarm limit causing a shutdown. Premature failure of recuperative heat exchangers is encountered quite frequently, due to sustained operation at or above the design temperature. Most of the time, the high-temperature operation occurs due to temperature excursions resulting from variations in the heat content of the waste. The ideal configuration for the use of a recuperative heat exchanger is, again, a bypass configuration like the one shown in Figure 9.10. However, for very large endothermic waste flows, the size and cost of the bypass ducting and induced draft (ID) fan may be prohibitive. Other methods to mitigate damage to the recuperative heat exchanger are as follows:

1. As mentioned before, parallel flow can be used instead of counterflow.
2. Installation of a combustion air preheater ahead of the waste gas preheater. The combustion air preheater allows a continuous cooling flow to extract heat from the flue gases right from the beginning of start-up. In addition, up to the full design amount of heat extraction can be done in the combustion air preheater regardless of operating conditions, such as turndown, if a vent valve is installed on the outlet of the combustion air preheater to dump any air that is not required by the process. The vent valve gives the operator/control system considerable control over the temperature of flue gas entering

the downstream waste gas preheater, allowing the system to tolerate wider variations in waste composition. In addition, the combustion air preheater eases start-up operability, widens waste flow turndown range, and reduces the hot soak temperature (max. temperature that the heat exchanger can tolerate without any cooling flow inside the tubes) requirement of the waste gas preheater.

3. Installation of a screen wall (checker wall) to reduce direct line-of-sight radiation from the flame to the first few rows of tubes in the heat exchanger that is located immediately downstream of the TO chamber, be it a combustion air preheater or a waste gas preheater. The screen wall will also help to improve temperature uniformity in the flue gases entering the heat exchanger.
4. Use a hanging tube heat exchanger configuration rather than a heat exchanger with two fixed (attached with expansion joints) tubesheets (see Figure 9.10).

Admittedly, all these protective designs are going to cost more than simply inserting the waste gas heat exchanger downstream of the TO. However, it should be kept in mind that the cost of saving one heat exchanger replacement, with its associated removal and installation costs, and more importantly the lost production during the shutdown will go a long way toward paying for the cost of the more conservative design approaches outlined earlier.

In heat medium heat recovery systems, or recuperative heat recovery systems, usually the refractory-lined TO is horizontal for the same reasons as the boiler system. Some smaller systems may be built utilizing a vertical upflow TO with the preheat exchanger mounted on the top end of the TO with the stack on top of the exchanger. This is the simplest and lowest cost configuration and is suitable to heat medium heat exchangers and superheaters. Although in principle this arrangement can be used for boilers and preheaters, it is usually not practical because of the weight of the boiler and the large size of the preheater and its associated refractory-lined ductwork.

For the same waste case described earlier (100,000 lb/h or 45,000 kg/h of N_2), preheating the waste gas in a counterflow heat exchanger, from 80°F to 900°F (27°C–480°C), would reduce the fuel needed to maintain 1500°F (820°C) in the TO, from more than 65 MMBtu/h (19 MW) to less than 29 MMBtu/h (8.5 MW). That is a reduction of fuel of about 55%. At only $3.50 per MMBtu/h of fuel fired, the savings in fuel cost would be over $1,000,000 per year compared to the simple vertical TO. The amount of heat transferred from the flue gas would drop the exit temperature of the heat exchanger from 1500°F to about 900°F (820°C to about 540°C), which corresponds to a heat recovery of about 40%. When less fuel is required, less combustion air is required, reducing the capital and operating cost of the fan. In addition, the amount of flue gas is reduced, reducing the size of the TO and consequently reducing cost of the TO vessel. The vent stack would have to be refractory lined for this example because the exhaust flue gas temperature is 900°F (480°C), unless it was fabricated using a heat-resistant alloy that can handle 900°F (480°C) flue gas temperature. However, heat-resistant alloy stacks can be more expensive than refractory-lined carbon steel stack depending on the height and design flue gas temperature.

Overall, the value of the steam generated and/or the reduction in fuel used during the first year will typically pay for the additional equipment needed to produce steam or preheat the waste gas.

When the waste gas is contaminated air, the configuration of the simple system, as well as the boiler and preheat systems, is nearly the same as when the waste gas is a non-oxygen-bearing inert. The only real difference in the simple system is that it will have to be forced draft because the oxygen in the waste can be used for combustion of either fuel, waste, or both. Both the boiler and recuperative systems are already forced (or induced) draft to overcome the pressure drop of the heat transfer component. For oxygen-bearing wastes, no additional combustion air is needed for any of the systems, if the waste contains enough oxygen to oxidize all the hydrocarbons. So, the fuel requirement for these cases is reduced.

For example, 100,000 lb/h (45,000 kg/h) of 80°F (27°C) contaminated air with no hydrocarbon content can be heated to 1500°F (820°C) with only 39 MMBtu/h (11 MW) of fuel. Cooling the flue gas to 350°F (180°C) can provide 32 MMBtu/h (9.4 MW) of heat recovery, generating about 27,000 lb/h (12,250 kg/h) of 150 psig (10 barg) steam. For the recuperative case, by preheating the contaminated air from 80°F to 900°F (27°C–480°C), the fuel required would be a little more than 17 MMBtu/h (5 MW). The amount of heat transferred from the flue gas would drop its temperature from 1500°F (820°C) to about 735°F (390°C), which corresponds to a heat recovery of about 54%.

However, if the hydrocarbon concentration in the air is too great, care must be taken to preheat the waste air to a safe margin below the lowest autoignition point. If some of the hydrocarbon oxidizes in the preheat exchanger, severe high-temperature damage may occur if the exchanger material cannot handle the higher temperature. For a simple example, at a concentration of 1% methane (CH_4) in air, the calculated temperature rise is greater than 450°F (230°C). If preheat to 900°F (480°C) is intended, an additional temperature rise of 450°F (230°C) yields 1350°F (730°C). If the TO outlet temperature is 1500°F (815°C), the average tube temperature at the exchanger inlet is about 1425°F (774°C), 225°F (107°C) greater than design. In general, the maximum heating value of contaminated air for a TO system using recuperative heat recovery should be below 20 Btu/ft^3 (0.75 MJ/m^3).

When the hydrocarbon content of the contaminated air is low enough, two other systems become more economically attractive when comparing the sum of the capital and operating costs. They are catalytic oxidation with recuperative heat recovery and regenerative thermal oxidation (RTO). The catalytic process is usually suitable for lower volumes of waste than the regenerative process, and the waste has to be particulate-free. For the catalytic process, the maximum heating value of the contaminated air must be limited, not just to remain below LEL, but also to prevent overheating the catalyst, which would be damaged rapidly at temperatures greater than 1300°F (700°C). The unlined, but externally insulated, chamber upstream of the catalyst receives and evenly distributes the 600°F (320°C) or so, preheated, contaminated air to the catalyst section. Downstream of the catalyst, the chamber is sometimes lined because the temperature after reaction may be 1000°F–1300°F (540°C–700°C) before passing into the heat exchanger.

For the regenerative process, a short-term temperature excursion does not pose a major problem because the refractory in the TO portion and the ceramic media in the heat recovery chambers can withstand operating temperatures greater than 2000°F (1100°C). However, rapid cycling of the recovery chambers will result in excessive wear and tear on the valves and ceramic media and expose the carbon steel ductwork downstream of the chambers to temperatures above the maximum allowable for those parts.

In general, the maximum heating value of contaminated air for a catalytic system with recuperative heat recovery should be limited to 13 Btu/ft^3 (0.48 MJ/m^3). For a regenerative system, which has even greater heat recovery efficiency, the maximum heat content should be limited to 7 Btu/ft^3 (0.26 MJ/m^3). For greater organic content, more air can be added to dilute the overall heat content, but that increases the volumetric flow rate, which increases the capital cost of either type of system. Another design approach for higher heat content wastes is to bypass some of the flue gas around the recuperative heat exchanger in a catalytic system so that less preheat is introduced to the oxidation process. Such a bypass system is shown by Figure 9.10.

Also, depending on the DRE required, strong consideration should be given to using a fired boiler that consists of a radiant and a convective section. The boiler's combustion chamber is essentially a "cold-wall" TO inside a boiler. Heat recovery is greater and the NOx emissions will be reduced, but CO emissions may be higher and DRE lower due to the cold walls of the combustion chamber.

At times, the contaminated air may contain enough hydrocarbon to be greater than 25% of LEL. In such cases, the use of explosion safeguards such as flame arrestors, detonation arrestors, or liquid seals is recommended. The purpose of such devices is to stop flame propagation from the combustion chamber back into the waste pipeline. As such, the flame arrestor device should be mounted as close to the combustion chamber as possible.

9.3.2 Non-Acid Gas Exothermic Waste Gas/Waste Liquid System

Incineration systems for the disposal of high-heating-value hydrocarbon wastes (greater than 200 Btu/ft^3 = 7.5 MJ/m^3 for gases and greater than 5,000 Btu/lb = 11,630 kJ/kg for liquids) that do not have a substantial halogen, ash, nitrogen, or sulfur content are relatively simple. The combustion of these wastes provides more than enough heat to maintain the furnace above the desired operating temperature so that no auxiliary fuel is required during normal operation. On the other hand, if the heat content of the waste is large, significant quantities of quench medium will be required to keep the TO operating temperature from exceeding the design limits of the refractory. Air, water, recycled flue gas, or steam may be used as quench mediums. Air is the most accessible, but will increase the volume and, hence, the capital cost of the TO. The same is true of steam or recycled flue gas. Liquid water, however, benefiting from the high latent heat of vaporization, provides significantly lower volume for the same quench duty.

If the flue gas does not require treatment before discharge, exothermic systems generally consist of a simple natural or forced-draft burner mounted on a vertical refractory-lined TO similar to the non-acid endothermic system (Figure 9.7). If heat recovery in the form of steam or hot oil is desirable, a boiler or oil heater may be used downstream of a horizontal refractory-lined TO configured similar to the system shown in Figure 9.8. Recuperative heat recovery is not an option because there is no need to add heat to an exothermic system. The concerns regarding heat transfer medium heat exchangers or superheaters discussed in the endothermic waste section earlier apply to the heat exchangers in exothermic systems as well.

High-heating-value hydrocarbon wastes, whether gaseous or liquid, have characteristics very similar to those of fuels. They are generally as easy to burn and are typically injected directly into the burner. The burner can be of low-, medium-, or high-intensity type depending on the waste being burned, the destruction efficiency desired, and what, if any, post-combustion treatment is utilized. Liquid wastes are atomized with medium-pressure steam or compressed air and are nearly always fired through the throat of the burner. Waste gases with higher heating values may be fired through a single gun or through multiple burner tips. Fuel is used only to heat up the system during start-up and in principle can be turned off once the waste is turned on. However, because the waste is not a pure fuel and may be subject to changes in composition, safety rules often require that some minimum quantity of fuel must be kept burning to ensure safe burner operation.

The high heating value of exothermic wastes produces high flame temperatures (2800°F+ = 1530°C+). In order to achieve the desired destruction efficiency, the combustion chamber for these wastes is generally maintained at relatively high temperatures (1800°F+ = 980°C+) and has residence times of greater than or equal to 1 s. The destruction efficiency of organic compounds is 99.99% or greater when using 20%–25% excess combustion air.

Additionally, the high operating temperatures result in excessive NOx formation. Therefore, being able to limit the maximum local temperature at any point in the TO is also a consideration. In some cases, the cooling medium may be injected into multiple locations both in the burner portion and in the TO to limit thermal NOx formation. Low-NOx burner techniques can also apply with these wastes.

The refractory (see Volume 2, Chapter 5) used in the TO partially depends on the type of waste and the operating temperature. For gases, a castable refractory of sufficient thickness to protect the TO shell is adequate. A liquid-burning incinerator, however, may be lined with a firebrick backed with an insulating castable to withstand potential impingement of flame or liquid, as well as withstand the high radiation from the intense flames produced by liquids. The difficult-to-brick areas are usually still castable lined, but a higher-density castable is used in those areas. However, less concern is shown for minimizing heat loss through the walls of the TO because excess heat is available. Maintaining a low stack temperature for personnel protection may require a thicker refractory or a personnel protection shield.

Because the exothermic TO systems are usually forced draft, stack height is not required to achieve the required draft in the simple system, and the exit gas velocity is often greater than 50 ft/s (15 m/s). In addition, the high temperature of the flue gas exiting the simple unit has sufficient buoyancy to carry the flue gas to high altitude. Consequently, a tall stack is not necessary for flue gas dispersion and stack height becomes a matter of customer choice based on surrounding structures and/or dispersion modeling.

When heat recovery is implemented, the flue gas is typically vented at less than 600°F (320°C) through an unlined steel stack. Although the buoyancy is not as great because of lower flue gas temperature, exhaust dispersion is still rarely an issue. Depending on the flue gas temperature, external insulation of the stack may be desirable to prevent water condensation and possible corrosion on the inside of the stack.

9.3.3 Sulfur-Bearing Acid Gas Systems

Acid gas systems are so called because the wastes treated contain components that, as a result of the oxidation reaction to destroy the waste, produce acid compounds such as SO_2/SO_3 and HCl. Emission of these acid compounds is limited by national and local air-quality permitting agencies and must be removed before the flue gas is exhausted into the atmosphere. The oxidation equipment along with the acid removal equipment collectively makes up the acid gas system.

During the early stages of the oil and gas refining process, sulfur compounds, primarily in the form of hydrogen sulfide (H_2S), are removed. The H_2S is then converted to elemental sulfur by the Claus process. The final "clean" by-product of that process is known as tail gas. Tail gas is mostly N_2, CO_2, and water vapor. However, although the efficiency of the Claus process has improved over the years, some of the sulfur compounds still remain in the tail gas. The sulfur compounds in the tail gas include hydrogen sulfide (H_2S), sulfur dioxide (SO_2), carbonyl sulfide (COS), carbon disulfide (CS_2), and elemental sulfur vapors. A small amount of CO and hydrocarbon is also usually present. Sulfur compounds have a very strong odor, often likened to "rotten eggs," and even a small amount is detectable by the human nose. H_2S is detectable by odor at concentrations as low as 10 ppm. Above 300 ppm H_2S is lethal, but not detectable by odor because the human olfactory sensors get saturated and lose the ability to sense the presence of H_2S. By destroying a high percentage of the sulfur compounds, the concentration in the flue gas is reduced to acceptable levels. A majority of the thermal oxidation systems supplied for sulfur-bearing waste streams have been simple units for the treatment of tail gas. TOs are very effective for odor control of wastes containing mercaptans and other odoriferous sulfur compounds.

Sulfur plant tail gas incinerators are generally designed to operate with natural draft. A stack tall enough to create the necessary amount of draft is used to provide airflow to the burner. The burners are designed for pressure drops of from 0.25 in. W.C. (1 kPa) to more than 1.0 in. W.C. (4 kPa). Waste heat recovery boilers are also utilized occasionally. For those cases, medium-pressure-drop, forced-draft burners are used.

The simple incineration process is nearly the same as the simple endothermic configuration described earlier. Castable-type refractory is usually adequate for the temperature and environment expected. As before, the lower portion of the stack is the residence time section. One difference from the simple endothermic system is that an internal coating may be applied to the vessel shell before refractory is installed to protect the steel shell from weak sulfuric acid attack. Refractory linings are porous and allow products of combustion such as SO_2/SO_3 to migrate to the steel shell. The outside of the steel shell is cooled by ambient air and so its temperature can be low enough to allow condensation of the SO_2/SO_3 leading to corrosion on the inside of the steel shell. Another alternative is to add an external rain shield around the vessel to keep the shell temperature above the dew point of the acid gases.

Because of the low autoignition temperatures (generally less than 700°F = 370°C), destroying sulfur-bearing compounds is very easy. Traditionally, a TO temperature of 1000°F–1200°F (540°C–650°C) and a residence time of 0.6–1.0 s were used for all sulfur plant tail gases. In the 1980s, the need to reduce the H_2S emissions to less than 5 ppmv required an increase in operating temperature to 1400°F (760°C) in some cases. The higher temperature ensures a higher degree of destruction of the sulfur compounds. It also increases the fuel demand.

During the oxidation of sulfur compounds, a small amount (typically 1%–5%) of the sulfur dioxide is further oxidized to SO_3. The extent of conversion depends on a number of conditions such as the temperature profile the flue gas, the amount of SO_2 in the flue gas, the potential catalytic action of alumina refractory material, and others. Once formed, SO_3 reacts at a temperature of between 450°F and 650°F (230°C and 340°C), with water in the gas stream, to form sulfuric acid. Sulfuric acid can cause several problems. First, it raises the dew point (substantially sometimes) to temperatures greater than 400°F (200°C). Sulfuric acid condensation can lead to rapid corrosion of carbon steel surfaces. Second, when quenched rapidly, sulfuric acid can form an extremely fine aerosol that is difficult to remove in a packed bed scrubber. Third, the fine aerosol can create a visible plume that has a bright blue-white hue.

If flue gas treatment is required, the design of the treatment system depends on the sulfur content of the waste and the regulatory requirements. Typically, tail gas incinerators have such a small SO_2/SO_3 emission that no flue gas treatment is used. The maximum allowable ground-level concentration (GLC) of SO_2 can be achieved by utilizing a tall stack. Stack heights of 100–300 ft (30–90 m) are common. However, if the requirement is to meet a maximum stack emission instead of a GLC, something else must be done.

For applications with very stringent emission limits, or ones that produce higher SO_2 concentration, a scrubber is used to remove a good portion of the SO_2 before the flue gas is dispersed into the atmosphere. To accomplish this, the flue gas from the TO must first be cooled by a boiler then quenched to its adiabatic saturation temperature by one of the quenching methods described earlier and then passed through a packed bed scrubber. If steam is not needed, the flue gas would be quenched directly to saturation. SO_2 is less soluble than HCl in water and therefore more difficult to remove. Adding sodium hydroxide (NaOH), often referred to as "caustic," can enhance the removal efficiency and convert the SO_2 to sulfates and sulfites for further treatment. The pH of the recirculated scrubbing solution must be no greater than 8.0 or a significant amount of caustic will react with the CO_2 in the flue gas, wasting caustic. Also, because SO_2 is more difficult to remove, taller packed beds are required when compared to similar applications (same inlet concentration and removal efficiency) scrubbing HCl. The presence of a significant quantity of SO_3 in the flue gases may also necessitate the use of a mist eliminator downstream of the scrubber.

Because of the low temperatures and lower oxygen content, NOx formation is fairly limited and is not normally a consideration. However, when operating at a TO at a temperature of only 1200°F (650°C), any carbon monoxide coming in with the tail gas will be only partially (40% or less) destroyed. At 1400°F (760°C), substantially more is oxidized (more than 80%), but CO destruction may still not be high enough to meet requirements. At lower temperatures, increased residence time (large TO) can provide greater destruction, but at a greater capital cost. However, if the system requires a tall stack, the stack volume can provide the additional residence time.

Pulp and paper plants also generate waste gases that contain sulfur compounds. Those wastes are handled in the same manner as described earlier. One difference in the equipment is in the material of construction. While the refineries find carbon steel an acceptable material, the pulp and paper industry often prefers stainless steel for many of its installations.

9.3.4 Halogenated Acid Gas Systems

While sulfur is probably the most common acid gas waste constituent found in the petroleum refining industry, chlorine (Cl_2) is probably the most common acid gas constituent encountered in petrochemical plant wastes. Chlorinated hydrocarbons are much more common than other halogenated hydrocarbons and so this section will present a discussion focusing on chlorinated hydrocarbon treatment. Of the other halogens, fluorine is also relatively common. It converts even more readily to hydrogen fluoride (HF), while bromine and iodine have much lower reaction rates (conversion to HBr and HI) and must be handled differently.

Chlorine is added to many hydrocarbon feedstocks to formulate numerous useful compounds. Wastes containing chlorinated hydrocarbons can be in the form of gas or liquid. Waste gases can be air-based, inert-based, or organic-based; waste liquids can be organic-based or water-based. In each of these cases, the waste may contain chlorinated hydrocarbons such as vinyl chloride, methyl chloride, chlorobenzene, and polychlorinated biphenyls (PCB). A typical incineration system for a relatively high-heating-value chlorinated waste consists of a horizontal TO

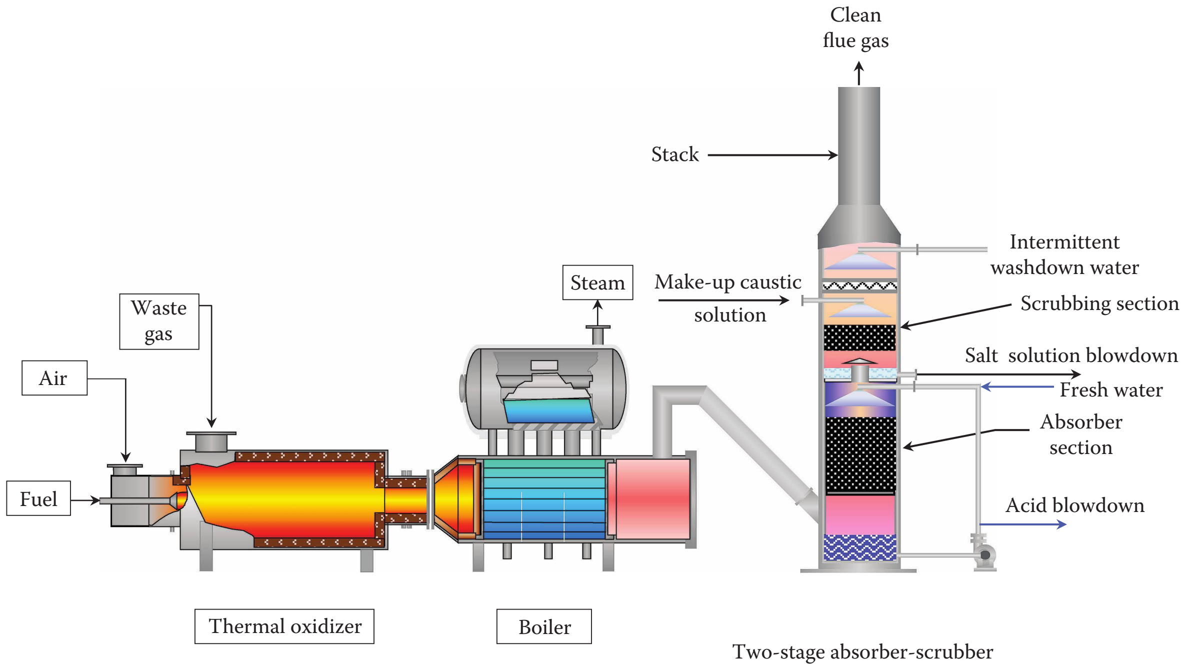


FIGURE 9.11
Horizontal TO with fire-tube boiler and HCl removal system.

to destroy the wastes, a fire-tube boiler to cool the flue gas for further treatment (and for heat recovery), a direct spray contact quench to cool the flue gas to saturation, a packed column to remove HCl, and a stack to vent the cleaned flue gas to the atmosphere. Figure 9.11 depicts such a process. The boiler would not be used if no steam is needed. In that case, the TO could be vertical upflow with a 180° turn into the quench section as shown in Figure 9.12.

Depending on the composition and heating value of the wastes, they may be fired through the burner as a fuel or added peripherally into the TO. The type of chlorinated hydrocarbon waste and the destruction efficiency required dictates the incinerator operating temperature, typically between 1600°F and 2200°F (870°F and 1200°F). Residence time varies from 1.0 to 2.0 s. Generally, air-based waste streams have lower organic content and require lower destruction efficiency, which can be accomplished beginning at 1600°F (870°C). On the other hand, wastes containing PCB require a destruction efficiency of 99.9999%. This level of destruction efficiency usually is accomplished at a temperature of 2200°F (1200°C) with a residence time of up to 2.0 s. The majority of waste streams containing chlorinated hydrocarbons require a destruction efficiency of 99.99%, which is usually obtained at temperatures of 1800°F–2000°F (1000°C–1100°C) and at residence times of 1.0–1.5 s.

The TO refractory can be as simple as ceramic fiber blanket for air-based fume streams with low operating temperature requirements or as elaborate as a high-alumina firebrick with an insulating firebrick backup for waste liquid streams with high operating temperature requirements. Plastic refractory is used in places where brick cannot be easily installed. Castable refractory with calcium oxide binder is generally avoided since the HCl in the flue gases can react with the binder and cause the refractory to disintegrate.

The oxidation process produces HCl and some free chlorine gas, along with the usual combustion products. It is important to limit the quantity of Cl_2 produced since it is very corrosive at higher temperatures and also because it forms sodium hypochlorite, a strong oxidant, when the combustion products are scrubbed with sodium hydroxide. The presence of hypochlorite in the blowdown stream from the caustic scrubber may require special treatment, as noted in the section discussing packed column scrubbers. Cl_2 formation can be minimized by shifting the reaction equilibrium away from Cl_2 formation and toward HCl formation. As can be seen from the reaction equilibrium versus operating temperature curve in Figure 9.13, the equilibrium can be shifted by increasing the TO temperature, by increasing the water vapor concentration, or by decreasing the excess oxygen concentration. In practice, some excess O_2 is required to maintain highly efficient oxidation reactions, so only changes to the temperature and water concentration are used.

A flue gas cooler/waste heat boiler is often used in chlorinated hydrocarbon systems. As noted in Chapter 8, a fire-tube boiler is preferred to a water-tube boiler because all of the heat transfer surfaces in a fire-tube boiler can be maintained at temperatures above the dew point

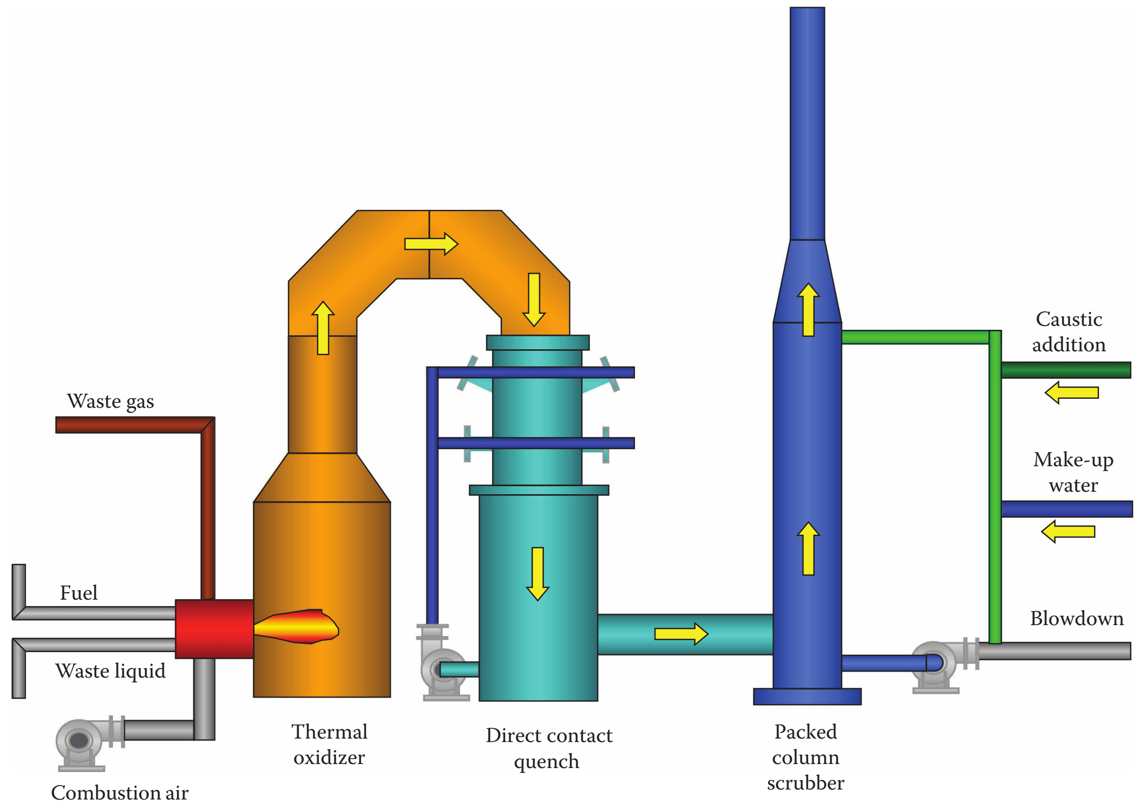
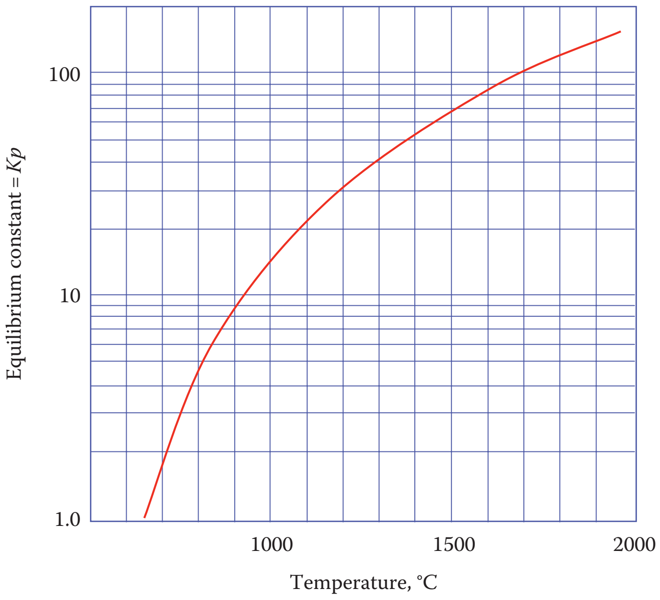


FIGURE 9.12
Vertical TO with 180° turn quench section.

(<250°F = <120°C). Of course, to do this, the steam pressure in the boiler must be high enough to have the saturation temperature higher than the dew point of the flue gas. The material of construction for the boiler tubes, as well as the rest of the boiler, is carbon steel.

The flue gas must be quenched to saturation before the absorption/scrubbing step that removes HCl. When quenching follows a boiler, it can be carried out in the bottom section of the packed column. A two-stage column should be used if HCl is being removed using caustic. With no waste heat boiler, the flue gas is quenched in a direct contact quench. Quench systems are often fabricated using reinforced plastic (FRP) with protection against hot flue gases provided by a dry-laid brick lining. Some users prefer metals such as Hastelloy instead of FRP.

The quantity of HCl in the flue gas dictates whether a single-stage absorber or a two-stage absorber/scrubber is used. For smaller systems where a relatively small quantity of HCl is present in the flue gas, a single-stage water absorber or a caustic scrubber can be used, with sufficient blowdown to maintain the concentration of HCl in the recycle water low enough to allow it to absorb the incoming HCl. If caustic scrubbing is used, the blowdown should maintain the dissolved and suspended solids (mostly NaCl) at no more than 5%, but



$$Kp = \frac{(p_{HCl})^2\,(p_{O_2})^{0.5}}{(p_{H_2O})\,(p_{Cl_2})} \quad \text{or} \quad p_{Cl_2} = \frac{(p_{HCl})^2\,(p_{O_2})^{0.5}}{(p_{H_2O})\,(Kp)}$$

FIGURE 9.13
Cl reaction equilibrium versus operating temperature.

preferably 3%. Again, pH should be kept below 8.0. The single-stage absorber produces a dilute acid stream and the single-stage scrubber produces a stream containing NaCl. For larger systems and those that contain larger quantities of HCl, it is not cost-effective to use large quantities of water or caustic to remove the HCl. In such systems, a two-stage removal system is used. The first stage is an absorber from which the majority of the acid is discharged as a concentrated solution of HCl (up to about 20% by weight). The HCl and Cl_2 remaining in the flue gas after it passes through the absorber are removed in the downstream scrubber, where caustic is used as the scrubbing reagent. HCl removal efficiency of as high as 99.9% can be achieved, although most applications require only about a 99% removal.

While oxidation of chlorinated hydrocarbons produces HCl with a little free Cl_2, fluorinated hydrocarbons produce even less and are easier to remove than HCl. However, burning brominated compounds results in as little as half of the bromine converting to HBr. Iodine compounds are worse. Free bromine and iodine are much more difficult to remove than Cl_2. Brominated and iodine compounds must be treated completely differently. Substoichiometric oxidation is necessary to drive the reactions toward HBr and HI.

9.3.5 Explosive Gases/Complex Liquid Systems

TOs for the chemical and pharmaceutical industry usually handle explosive waste gases. These waste gases contain flammable compositions including hydrogen (H_2), carbon monoxide (CO), and a variety of other hydrocarbon compounds. The hydrocarbons in the waste can also contain other chemically bound ($C_nH_mX_i$) compounds like nitrogen (N), chlorine (Cl), bromine (Br), fluorine (F), sulfur (S), silicon (Si), or titanium (Ti). The major constituents are air or inert gases like nitrogen (N_2), water vapor (H_2O), or carbon dioxide (CO_2). Depending on what type of equipment is producing the emission, such as autoclaves, reactors, vacuum pumps, stripping columns, or tank farms, the flow of waste gases may be continuous or intermittent. Mixing of the waste gases from different sources prior to the TO is often not allowed due to variations in pressure or due to the potential for reactions between chemicals. The TO therefore has to respond flexibly to all variations of composition, concentration, and flow, without any impact on the production process. The heating value of waste gases can fluctuate between zero and highly explosive values. Oxygen levels can also fluctuate over a wide range. As discussed previously, if the waste gases or liquids are exothermic that minimizes the need for additional fuel. Figure 9.14 shows a computer model of a typical Combustor-based TO system for explosive gases. This system shown in Figure 9.14 includes a boiler and scrubber. Figure 9.15 shows a close-up of the Combustor in the actual assembly in the plant.

The TO must fulfill all national regulatory requirements for safety, explosion prevention, and emission limits. The production process and the waste gas treatment system usually form one functional unit. Sustained interruption of the TO operation will result in a shutdown of the production line. Therefore, often plant availability has to be included in the warranty.

9.3.5.1 Design

The "Combustor" (vortex burner), a type of high-pressure-drop burner, described earlier in this chapter and in more detail in Volume 2, Chapter 6, on burner design, is a well-proven burner for this kind of TO system[2–6] that must handle explosive gases and widely varying wastes with intermittent flows.

The cylindrical reaction chamber may be arranged in either a horizontal or a vertical position. The vertical position, usually down fired, is preferred in systems burning large quantities of heavy liquids or liquids that create particulates. The down-fired orientation uses gravity to assist in the transportation of the spray from the atomization guns and the inorganic particles formed during combustion. A rapid change of waste gas composition from high calorific value and/or explosive status to inert status usually has to be considered for the plant design. A high flow of inert gas may adversely impact flame stability because the inert gas delays the rate at which the combustibles come into contact with their reaction partner: oxygen. Unstable flame conditions may result in flame liftoff, pulsations, and noise emissions. Increasing the combustion airflow to stabilize the root of the burner flame is not an economic solution because it requires more fuel as well.

In such cases, where flammable gases are expected to become inert at certain times during the process, waste gas splitting can be used. One portion of the waste gas will then be treated in the Combustor, and the other part of the waste gas will be injected, sometimes along with some supplementary fuel, into the flue gas downstream of the Combustor. Gas-phase reaction (GPR) is a term used to refer to such oxidation of the waste in the downstream flue gas. An additional flame scanner is not required for the downstream waste injection as long as the main burner flame is detected, *and* the reaction temperature after the GPR stage is above 1382°F (750°C).[7]

Additional liquid wastes, waste gases, or partial flows of each might be injected (secondary injection) via suitable nozzles into the hot vortex flow of the flue gas coming out of the Combustor. The nozzle injection systems and mixing devices are designed to maximize mixing efficiency. The oxygen demand for the waste gas that is injected downstream (secondary injection) will be supplied by the Combustor or via an additional air injection

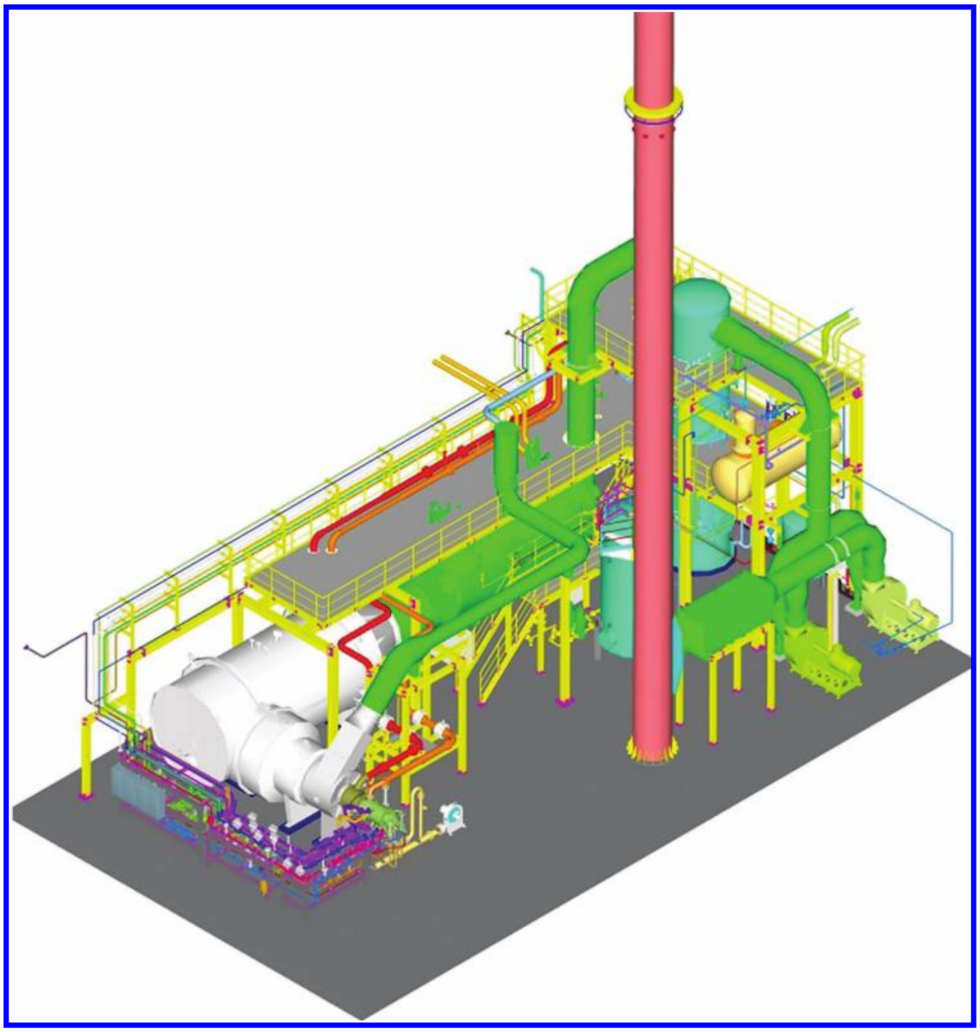


FIGURE 9.14
Computer model of a Combustor-based TO system for explosive waste gases with boiler and scrubber.

system. The rotating flue gas from the Combustor outlet, with its homogenous temperature and oxygen profile, ensures adequate ignition and reaction conditions for the second stage of waste injection. The temperature of the second stage is controlled by a floating temperature setpoint in the Combustor stage. The temperature control in each stage, the calculations of all mixing ratios, and the resulting physical mixing temperatures ensure excellent ignition and reaction conditions. GPR technology is also used if the burner does not have enough space to accommodate all the required injection nozzles. A further advantage of the GPR technology is the reduction of the energy density that reduces the thermal NOx formation. Figure 9.16 shows a computer model of a TO system with GPR technology.

In order to minimize fuel consumption, the lowest temperature that achieves the required destruction efficiency is usually selected as the TO operating temperature. Therefore, the refractory lining should be designed with low heat losses to ensure complete destruction of the hydrocarbons in the waste. A high-temperature gradient from the inner reaction chamber wall to the outside introduces the possibility of waste gases escaping the TO unreacted and should be avoided.

The wall temperature of the carbon steel shell of the TO chamber has to be kept above the dewpoint of the flue gas to prevent corrosion. Furthermore, the refractory materials used have to be resistant to attack by the constituents of the flue gas.

TO applications for the wastes being discussed in this section, which typically produce hazardous or corrosive flue gas compounds like HCl, Cl_2, SO_2, and SO_3, are operated predominantly under negative pressure with an ID fan. Negative pressure conditions in the reaction chamber eliminate leakage of hazardous and corrosive gases. The ID fan's suction can also be used for aspiration of the waste gas into the TO.

In many TO applications, the flue gas will be cooled down by a boiler system. If the flue gas is free of dust, a fire-tube boiler can be used. If particulates are present

FIGURE 9.15
Combustor with a waste gas organ for explosive waste gas.

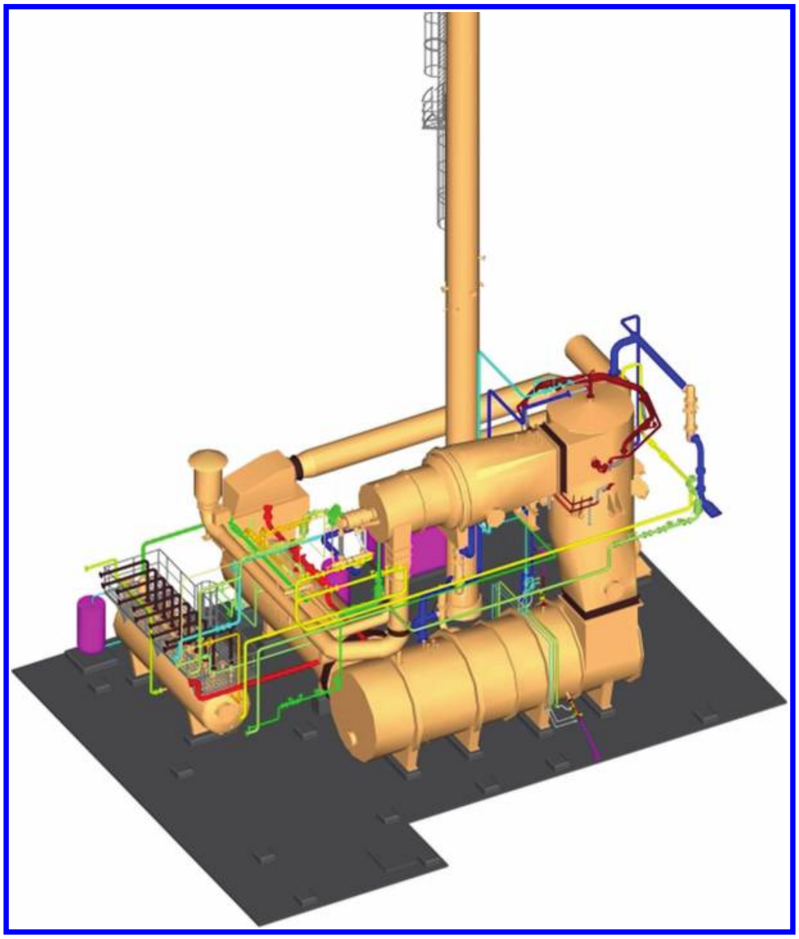


FIGURE 9.16
Computer model of a TO system with GPR technology.

in the flue gas, the preferred solution is a water-tube boiler. The water-tube boiler can be supplied with online cleaning devices like soot blowers. If the flue gas contains inorganic and/or toxic compounds, a downstream flue gas treatment system like a scrubber has to be installed.

9.3.5.2 Explosion Protection Safety Devices

This chapter describes the general manufacturer's safety philosophy applicable to the wastes being discussed in this section, as well as the European guidelines.[8]

Typically, the production plant supplies waste gases to the TO via a header. The flame of the burner in the TO is considered a permanent ignition source[9] and so a system to decouple explosions, using approved safety devices, is necessary to prevent flashback from the TO into the production plant.[10,11] The required number of independent safety devices is specified by area classifications Zone 0, 1, or 2.[9] This classification is defined by the probability of encountering an explosive condition.

In a Zone 0 area, there is a permanent explosive atmosphere. In a Zone 1 area, an explosive atmosphere can be expected to occur regularly. In Zone 2 areas, an explosive atmosphere is seldom expected, and when it occurs, it is expected to persist only for a short time.[12] In addition, the various gases are classified into explosive groups by their specific flammability characteristics. Each group (IIA, IIB, subdivided into IIB1, IIB2, IIB3, and IIC) requires specifically approved safety devices, such as static, dynamic, or liquid sealed flame arrestors.[13] Figure 9.17 shows a controls schematic for explosion protection for Zone 0.

Static flame arrestors are made of wound corrugated metal strips with defined gaps. The large surface area present in the corrugated strips cools the flame rapidly by heat dissipation into the boundary layers. Thus, the temperature of the burning gases in the flame is reduced below the ignition temperature.[14] The maximum allowed gap width (MESG = Maximum Experimental Safe Gap) in the corrugated strips is defined by the explosion group. The longer and the narrower the gap, the more efficient is the quenching effect. The typical gap sizes are between 0.15 and 0.9 mm (0.0059 and 0.035 in.).[14]

Depending on the location of the installation and on the operating conditions, either deflagration or detonation arrestors can be used. Deflagration arrestors may only be installed close to the ignition source in a specified maximum length to diameter (L/D) ratio. Static flame arrestors have temperature and pressure limitations. If the waste gas contains particulate, redundant flame arrestors with offline cleaning capability are recommended. Any static flame arrestor for TO plants has to be equipped with thermocouples.[9,14] An increasing temperature recorded by the thermocouples indicates a flashback. If flashback is detected, the two safety shutoff valves in the waste line must close immediately within one second.[7,15] After an indication of flashback, the arrestor must be dismounted and the element has to be checked by the manufacturer or by certified personnel. Figure 9.18 shows a cutaway photograph of a typical static flame arrestor, and Figure 9.19 shows cross-sectional drawing of the static flame arrestor.

Liquid seal flame arrestors are designed to break the flow of an explosive mixture into defined small bubbles in a water column and thus prevent flame transmission. The waste gas will be injected via a bubble pipe into a horizontal vessel filled with water up to a certain level. The safety performance depends on the height of the water level above the bubble tubes, the bubble size, the bubble distance, and the bubble release frequency and distribution. The vessel is equipped with a thermocouple on the gas side. A flashback will be indicated by a temperature increase, which immediately stops the waste gas stream by closing the safety shutoff valves. Other safety parameters that are monitored in a liquid seal flame arrestor are the water level (low alarm), the waste gas flow rate (high alarm), and the water temperature (low alarm). If necessary, the water can be heated using steam.[14,16] Figure 9.20 shows an arrangement drawing of a liquid seal flame arrestor.

For *dynamic flame* arrestors, the waste gas flow rate must be continuously monitored to ensure that the discharge velocity at the nozzle is above the back flash velocity. If required, makeup air or nitrogen must be added to the waste to increase the velocity. If the flow rate does not reach the minimum requirement, the safety shutoff valves in the waste gas line must be closed.[14,17–24] The use of dynamic flame arrestors increases the reliability of the system by avoiding a flashback at the source and eliminating the need to shut down and inspect a static flame arrestor after a flashback incident.

The pipework and equipment between the ignition source and the flame arrestor must be capable of withstanding the maximum explosion pressure of the gas, typically 145 psi (10 bar). In some cases pipework and equipment may require higher pressure rating.[11] Valves and instrumentation used must be certified according to the required explosion zone classification.[11]

9.3.5.3 Operating Conditions

In general, the Combustor demonstrates good combustion performance and low NOx emissions. The intensive mixing allows operation at low reaction temperature levels. According to national and European regulations,[25–27] the CO emissions usually are required to stay below 50 mg/Nm^3 and VOC emissions have to stay below 10 mg/Nm^3.

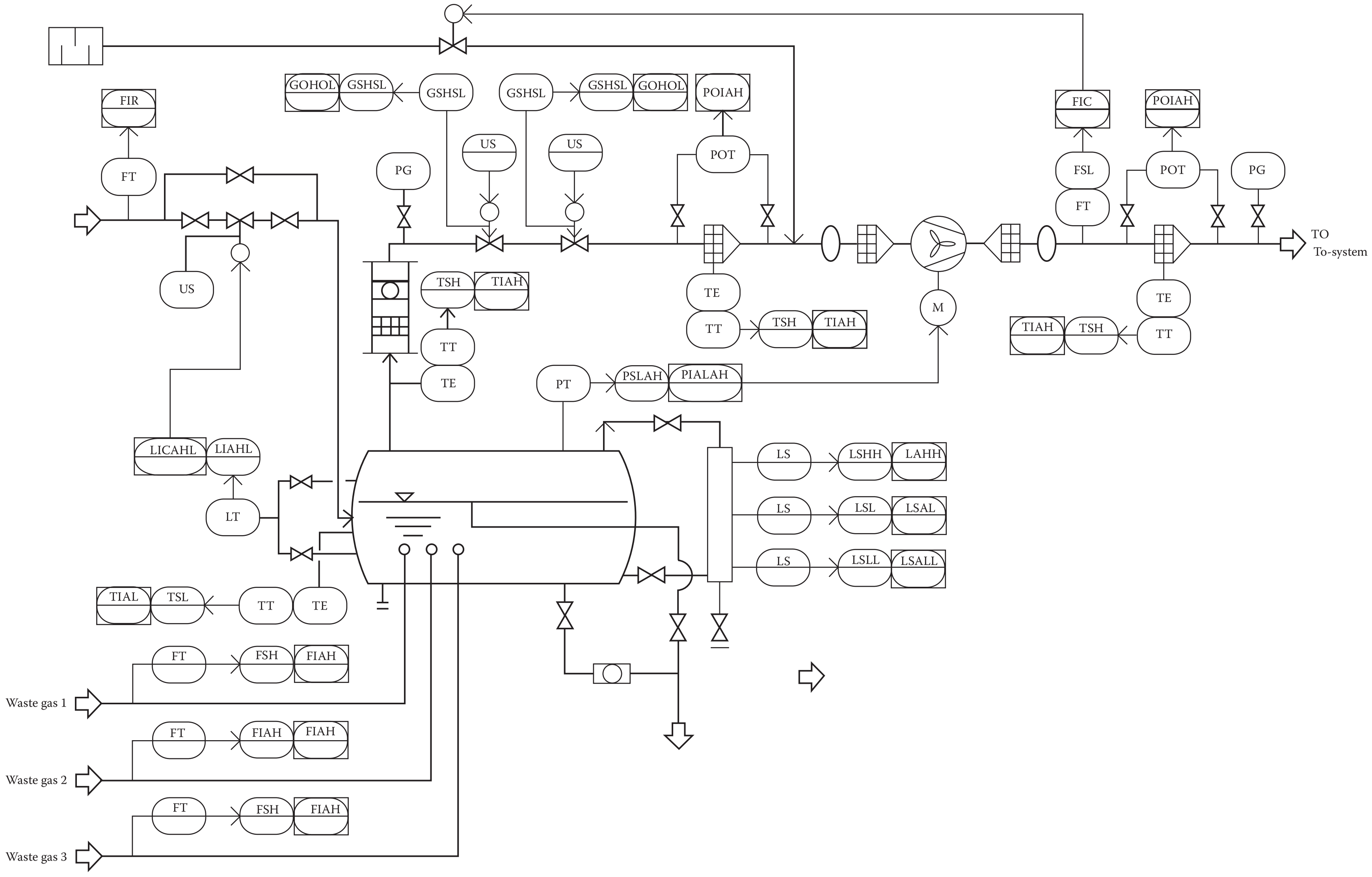


FIGURE 9.17
Schematic of explosion protection for Zone 0.

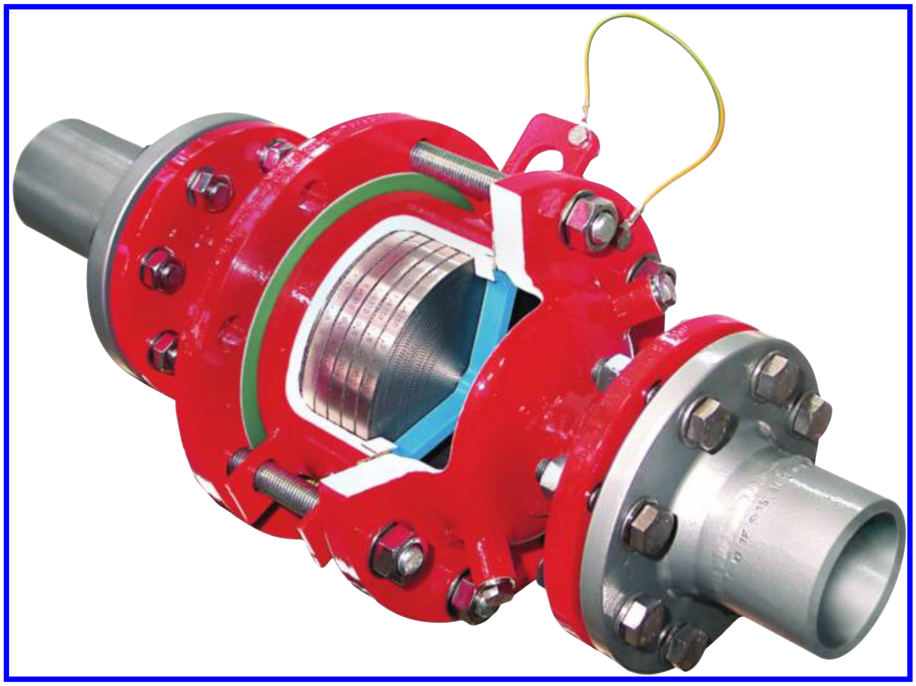


FIGURE 9.18
Cutaway photo of a static detonation arrestor.

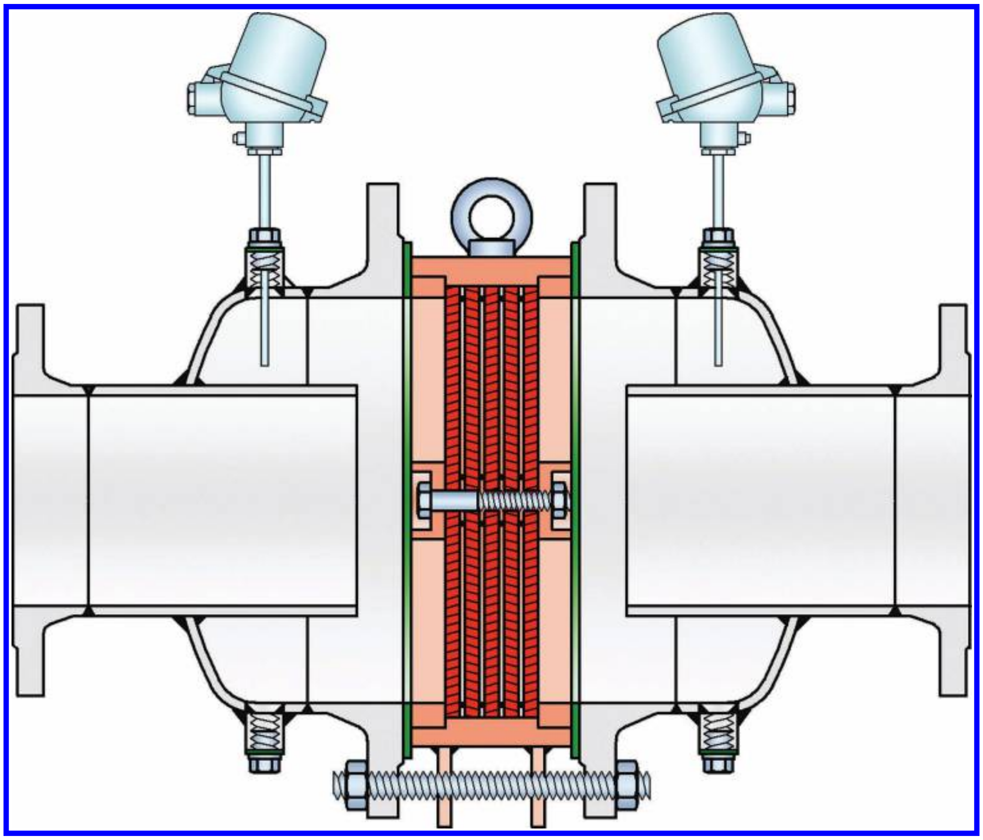


FIGURE 9.19
Sectional drawing of a static detonation arrestor.

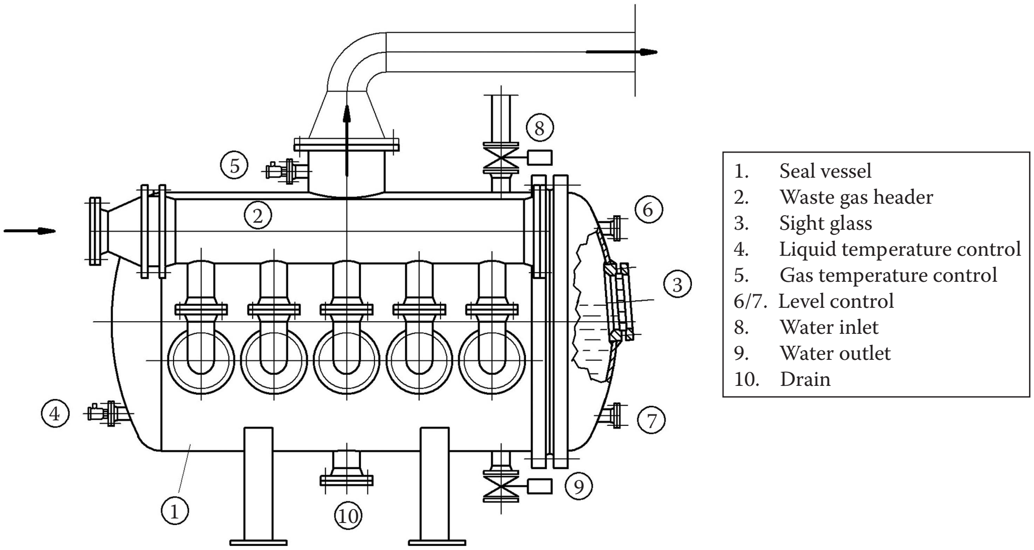


FIGURE 9.20
Arrangement drawing of a liquid seal vessel.

For the destruction of simple hydrocarbons to carbon dioxide and water vapor, the Combustor needs a reaction temperature between 1440°F and 1530°F (780°C–830°C), and a residence time of 0.6 s is sufficient. In challenging applications with high amounts of carbon monoxide, aromatic, or halogenated compounds, the Combustor needs slightly higher temperatures, between 1560°F and 1650°F (850°C–950°C), and slightly more residence time of 0.6–1 s. In cases where liquid wastes with halogenated compounds are fired in the TO, regulations usually require a minimum reaction temperature of 2000°F (1100°C) and a residence time of 2 s.[26,27]

9.3.6 Salts/Solids Systems

System configurations for handling salts/solids are determined by the condition of the material coming into and out of the combustion section. Either a liquid or a gas can carry the solid. If the material has a high melting point (>2400°F = 1300°C) and comes in with water, it will likely pass through the combustion section virtually unaffected. If the solid is part of an organic liquid, it may melt in the flame zone but will return to the solid phase very quickly after leaving the flame zone. If flow patterns and vessel orientation are wrong, the material will often collect in the equipment. Regardless, if the particulate emission exceeds the allowable amount, a particulate removal device must be employed. Examples of particulate are catalyst fines in organic or aqueous liquids, metal-machining dust in air, titanium dioxide (an opaque, white additive to many products) in water or organic liquid, some organic salts such as sodium acetate (which oxidizes to form organic combustion products and sodium oxides, which have high melting points), or inorganic salts such as sodium chloride or sodium sulfate.

If the material comes into the vessel as an organic solid, it may very well be completely oxidized before it exits. However, often, organic materials contain nonorganic compounds that may be high-melting-point inerts or lower-melting-point materials such as silica (~2000°F = 1100°C) or alkali metal inorganic salts such as NaCl (~1500°F = 820°C). If the solid is organic, certain configurations allow quicker, more complete oxidation than others. If lower-melting-point materials are present, the system must be designed more to handle molten material. Some examples of organic solids containing nonorganic material are sawdust and rice hulls. Both can be conveyed into a combustion system with air as the carrier and both have good heating value. However, both contain a large amount of silica. When both are burned, the silica forms very small molten particles of silicon dioxide in the flame zone. When cooled, these fine particles of "fume silica" have a tendency to collect inside equipment and are difficult to remove from the flue gas.

If the solid material is an alkali metal inorganic salt such as NaCl, Na_2SO_4, CaCl, or KCl, in water, for instance, the total system design must be based on molten material in the flue gas. In practice, virtually all cases involve molten material, so the inorganic salt case will be reviewed here.

Most of the wastes in this category are salt-contaminated liquids. The waste streams are both organic and water based. Because the waste is a hydrocarbon liquid or because the water-based waste liquid contains a hydrocarbon, thermal oxidation is often the best method for disposal.

Unfortunately, since most salt-containing wastes are liquid, they usually have to be oxidized at higher temperatures to rapidly achieve high destruction efficiency. This presents a problem with the refractory in the TO. Alkali material attacks the binder in refractory materials. Higher temperatures increase the rate of attack. The design must balance the need for higher temperature (i.e., higher destruction efficiency) with the need to prolong refractory life. Temperatures ranging from 1600°F to 2000°F (870°C–1100°C) and residence times of 1.0–1.5 s are commonly used.

The system best suited for molten salts is shown in Figure 9.21. The incinerator is usually vertical, with the burner mounted on the top and firing downward. The vertical design is highly desirable because it does not allow the molten salts to accumulate on the refractory, as it would in a horizontal oxidizer. This is extremely important as any accumulation of salts in the oxidizer can drastically reduce refractory life.

Basic (MgO) brick has very good resistance to salt attack. However, it is very expensive and highly susceptible to thermal shock and hydration and will quickly erode if hit by a stream of water. High-alumina brick has been used with varying results. In some cases, a 90% alumina brick, with its low porosity, has been found to be more effective against salt attack than a 60% alumina brick. Unfortunately, while the higher-alumina brick is two to three times the cost of the lower-alumina brick, it does not last two or three times longer. In general, using the lowest porosity, 60%–70% alumina brick and the proper burner/TO configuration provide the most cost-effective service. In any case, refractory lifetimes in salt service are much shorter than in other types of TOs. Refractory replacement in one to two years is to be expected in this service.

For organic wastes, moderate- to low-pressure-drop burners are preferred, since a high-pressure-drop burner, with its attendant turbulence, tends to centrifuge the salts in the waste toward the refractory, causing it to deteriorate more rapidly. A moderate-pressure-drop burner with no cyclonic action tends to keep the salts in suspension and away from the incinerator walls. The less the salt contacts the refractory, the slower the

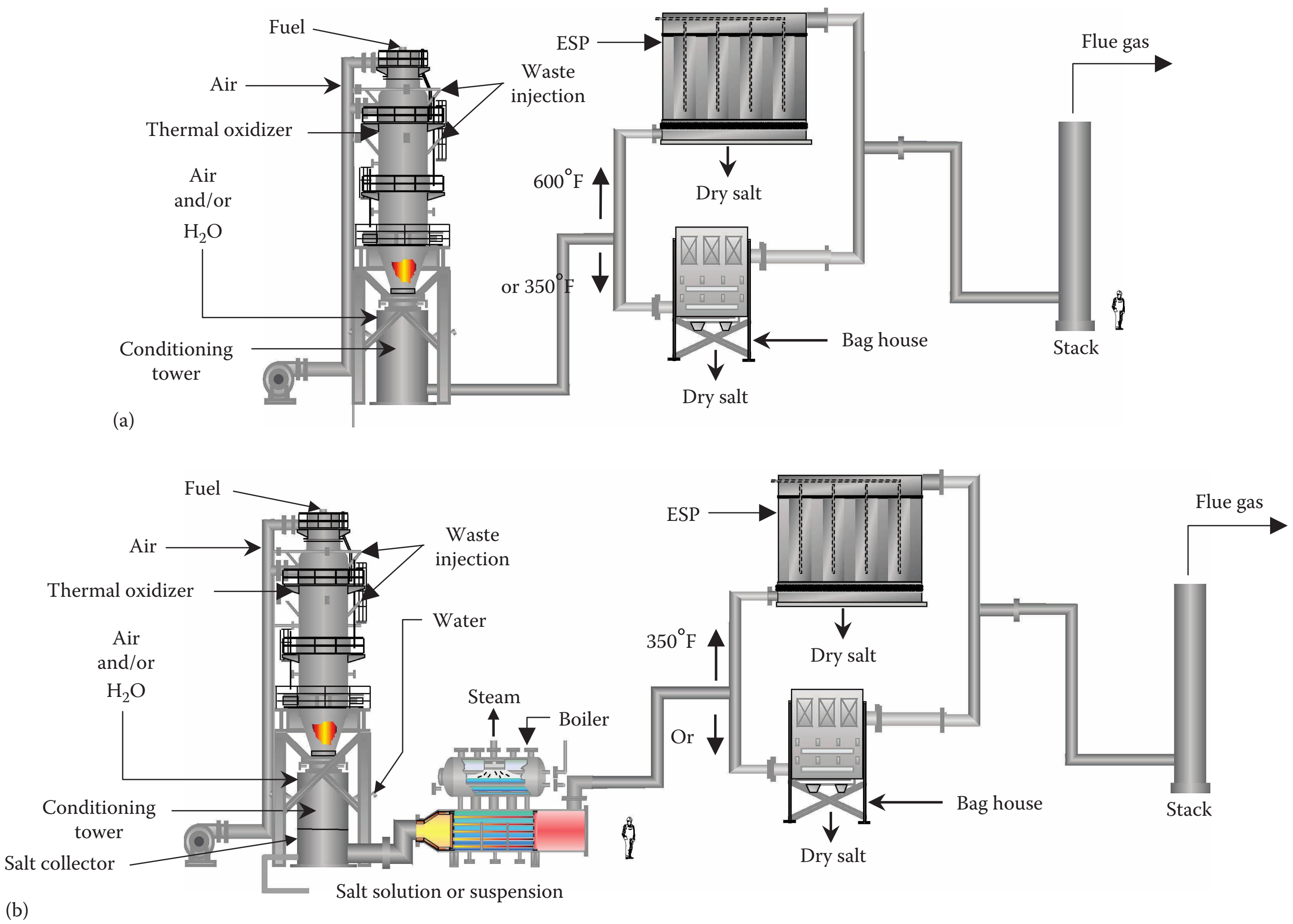


FIGURE 9.21
Molten salt system: (a) down-fired salts condition to low temperature and (b) down-fired salts conditioned to recover steam.

degradation of the refractory. Rich hydrocarbon liquids are fired through the burner.

Water-based wastes are injected into the TO downstream of the burner. Since the water in these wastes must be evaporated and the temperature of the whole mass be raised to near the oxidation point before combustion of organic materials can begin, it is necessary to burn some amount of auxiliary fuel. The amount of auxiliary fuel burned depends on the operating temperature and the waste composition. Again, a moderate-pressure-drop burner is used for these aqueous wastes. The auxiliary fuel must be allowed to burn to near completion before injecting the aqueous waste stream into the combustion products. Incomplete combustion of the auxiliary fuel can cause soot formation and incomplete combustion of the waste.

Although rare, waste heat recovery boilers have been used in some systems. To utilize heat recovery, the flue gas must be conditioned to a temperature below the melting point of the salt, freezing it before it contacts the cooler heat transfer surface. This makes the salt friable, so the accumulation is easier to remove from the heat transfer surface. A water-tube-type boiler is used to allow on-line cleaning by soot blowers. Figure 9.22 shows this configuration.

Before the flue gas can be vented to the atmosphere, particulate matter must be reduced to the concentration allowable by emission standards. This is accomplished with a baghouse, an ESP, a WESP, or a venturi scrubber. Each of these particle removal devices has been discussed in previous sections. As noted, proper flue gas conditioning or quenching must be accomplished upstream of any air pollution control (APC) equipment.

Excessive salt in the flue gas from a salts unit will form a non-dissipating visible plume. Consequently, the flue gas from the saturated system often receives intense public scrutiny. For this reason, some operators choose to install equipment to eliminate the visible water vapor plume. This is accomplished either by cooling and removing virtually all the water vapor in the saturated flue gas or by removing a substantial portion of it then heating the flue gas stream so that the plume becomes less visible. Depluming is costly, in terms of both capital and operating costs, and the results are not always satisfactory.

For both water- and organic-based liquid streams, 99.99% DRE is normally required. A particulate matter concentration of no more than 0.015 grains per dry standard cubic foot (DSCF) of flue gas, corrected to a specific excess O_2 level, is also normally required.

9.3.7 NOx Minimization or Reducing Systems

The theory behind the combustion-process-modification method of minimizing or reducing NOx was covered in previous sections. This section will review only the three-stage system configuration.

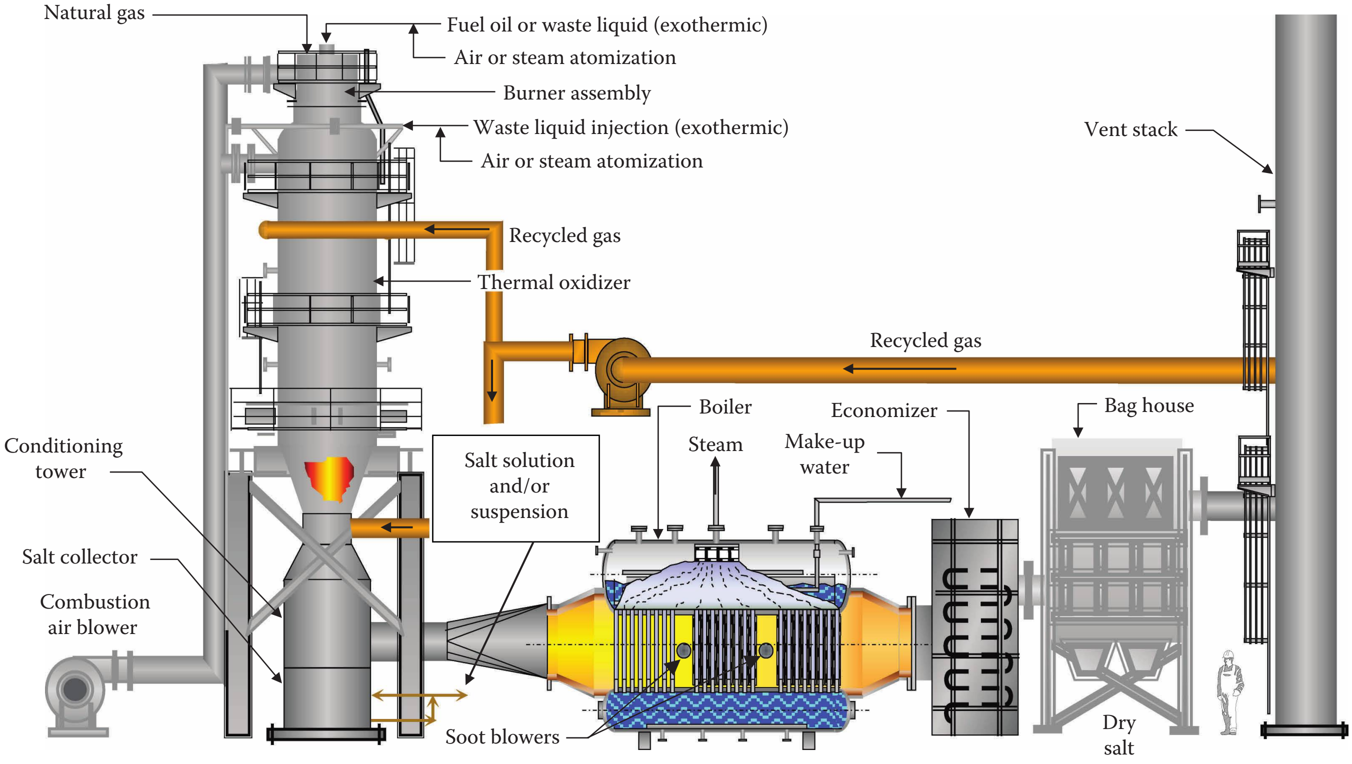


FIGURE 9.22
On-line cleaning with soot blowers.

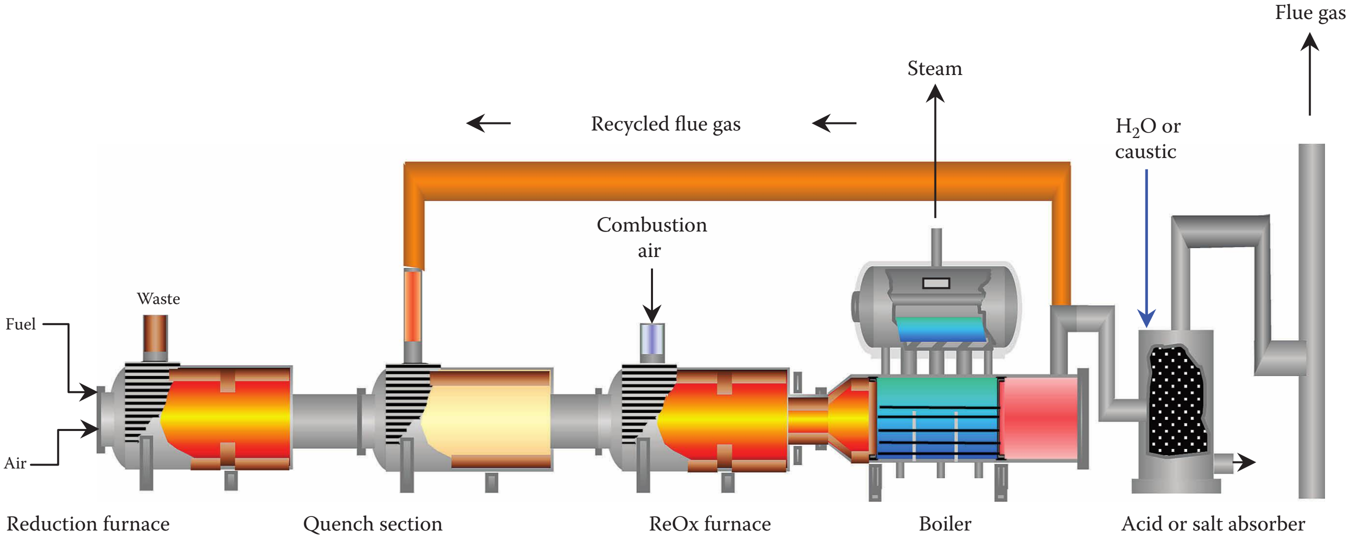


FIGURE 9.23
Three-stage NOx system with packed column scrubber.

Consider a waste stream consisting of chlorinated hydrocarbon, amine, and other hydrocarbons. Oxidizing this waste mixture in a standard excess-air TO would produce high NOx and HCl emissions. A three-stage reducing–oxidizing system NOx system would provide the NOx reduction needed, but the HCl generated would be an issue. Simply adding a packed column scrubber, which is capable of quenching the flue gas in the base of the scrubber, to the end of the NOx system would complete the system. Figure 9.23 illustrates such a configuration.

The combustion air blower, burner, reduction furnace, quench section, oxidation air blower, and oxidation section are very similar to those supplied for a “normal” three-stage reducing–oxidizing NOx system. Carbon steel construction is still acceptable. It is necessary to add a full-length rain shield or internal anticorrosion lining to the three vessels to protect against HCl corrosion. Also, the carbon steel duct between the boiler and the packed column and the entire recycle duct to the reduction furnace and quench section must be externally insulated and sealed from weather, so the unlined ductwork steel temperature will stay well above the HCl dewpoint. The recycle blower should be carbon steel with sealed external insulation. The packed column should be FRP.

The refractory is very similar to that in the “normal” NOx system with the exception that the presence of HCl must be considered. The castable refractory should be reviewed to ensure the material with the lowest CaO content is utilized without compromising the refractory strength or operating capability.

Instrumentation must be capable of operating in an HCl environment as are those associated with the chlorinated hydrocarbon systems.

Nomenclature

APC	Air pollution control
CSA	Collecting surface area
DRE	Destruction and removal efficiency
ESP	Electrostatic precipitator
FRP	Fiberglass reinforced plastic
LEL	Lower explosive (flammability) limit
MMBtu/h	Million Btu/h
PCB	Polychlorinated biphenyls
SCR	Selective catalytic reduction
SCS	Specific collector surface
SNCR	Selective non-catalytic reduction
TO	Thermal oxidizer
WESP	Wet electrostatic precipitator

References

1. H. L. Gutzwiller, *Fan Performance and Design*, Robinson Industries Inc., Coleman, MI, January 2000.
2. Verein deutscher Ingenieure VDI-Richtlinien VDI 2442, Page 18, 2006, *Waste Gas Cleaning, Methods of Thermal Waste Gas Cleaning*, Beuth Verlag GmbH, Berlin, Germany.
3. J. Hellwig, D. Müller, 1996/97, *Entsorgung auf der grünen Insel, Verbrennungsanlage für flüssige und gasförmige Abfälle in Irland*, Chemie Umwelt Technik.

4. D. Müller, 1995, *Thermische Abgasverbrennung entsorgt kritische Stoffströme. Jeder Zeit LRV konform*, Verfahrenstechnik 1995, Vol. 29, Vereinigte Fachverlage GmbH, Mainz, Germany.
5. Wolfgang Klaus, Bernd Reese, 1997, *Thermische Entsorgung von flüssigen Prozeßabwässern und Abgasen*, Chemie-Anlagen+ Verfahren, Special Umwelttechnik und Entsorgung, März 1997, Konradin Medien Gruppe, Leinfelden Echterdingen, Germany.
6. EUROPEAN COMMISSION, 2006, Integrated Pollution Prevention and Control. Reference Document on Best Available Techniques for the Manufacture of Organic Fine Chemicals, Chapter 4, p. 240, Website: http://www.ineris.fr/ippc/sites/default/files/files/ofc_bref_0806.pdf (accessed on March 19, 2013).
7. German Code DIN EN 746-2, 2010, *Industrielle Thermoprozeßanlagen Sicherheitsanforderungen an Feuerung und Brennstofführungssysteme*, Beuth Verlag GmbH, Berlin, Germany.
8. European Parliament, 1994, *Guideline 94/9/EG (ATEX 95)*; Website: http://www.maschinenrichtlinie.de/fileadmin/dokumente/94_9_EG%20ATEX-Richtlinie.pdf (accessed on March 19, 2013).
9. BArbBl. 4/2001 S. 60-Bundes Arbeitsblatt, 2001, *Technische Regeln für brennbare Flüssigkeiten TRbF 20 Läger*. http://www.netinform.de/Vorschriften/GW/pdf/trbf20.pdf (accessed on April 9, 2013).
10. Berufsgenossenschaftliche Regel für Sicherheit und Gesundheit bei der Arbeit BGR 104, 2007, *Explosionsschutz-Regeln, Regeln für das Vermeiden der Gefahren durch explosionsfähige Atmosphäre mit Beispielsammlung*. http://www.gasbehaelter.de/PDFs/BGR-104.pdf (accessed on March 20, 2013).
11. Bundesanstalt für Arbeitsschutz und Arbeitsmedizin, *Technische Regeln für Betriebssicherheit TRBS 2152 Tei4l, 2012*, http://www.baua.de/de/Themen-von-A-Z/Anlagenund-Betriebssicherheit/TRBS/pdf/TRBS-2152-Teil-4.pdf?__blob=publicationFile&v=4 (accessed on April 10, 2013).
12. German Code DIN EN 1227, 1.2009, *Explosionsfähige Atmosphären Explosionsschutz, Teil 1 Grundlagen und Methodik, Deutsche Fassung EN 1127–1. 2009*, Beuth Verlag GmbH, Berlin, Germany.
13. German Code DIN EN 12874, 2001, *Flammendurchschlagsicherungen Leistungsanforderungen, Prüfverfahren und Einsatzgrenzen*, Beuth Verlag GmbH, Berlin, Germany.
14. PROTEGO®-Braunschweiger Flammenfilter GmbH, Braunschweig, Germany. http://www.protego.com/en/products/technical-fundamentals/flame-arresters/ (accessed on April 10, 2013).
15. German Code DIN EN 161, 1996, *Automatische Absperrventile für Gasbrenner und Gasgeräte*, Beuth Verlag GmbH, Berlin, Germany.
16. W. Hüning, 1989, *VDI Berichte Nr. 730, Sicherheitsmaßnahmen bei der thermischen Abluftreinigung—zweite überarbeitete Fassung*, Verlag des Vereins Deutscher Ingenieure, Düsseldorf, Germany, p. 25.
17. IBExU Instititut für Sicherheitstechnik GmbH, 2004, *Prüfbericht IB-03–3–776/1. 2004. Über die Prüfung der strömungsüberwachten Einrichtung KEU EX-Organ IIB3 DN als Flammendurchschlagsicherung für brennbare Flüssigkeiten und Gase der Explosionsgruppe IIB3 mit einer Normspaltweite von ≥ 0,65 mm*, IBExU, Freiberg, Germany.
18. IBExU Instititut für Sicherheitstechnik GmbH. 2004. *Prüfbericht IB-03–3–776/2. 2004. Über die Prüfung der strömungsüberwachten Einrichtung KEU EX-Organ IIB3 DN als Flammendurchschlagsicherung für brennbare Flüssigkeiten und Gase der Explosionsgruppe IIC mit einer Normspaltweite von < 0,50 mm*, IBExU, Freiberg, Germany.
19. IBExU Instititut für Sicherheitstechnik GmbH, 2004, *Prüfbericht IB-03–3–776/3. 2004. Über die Prüfung der strömungsüberwachten Einrichtung KEU GPR-Organ IIB3 DN als Flammendurchschlagsicherung für brennbare Flüssigkeiten und Gase der Explosionsgruppe IIB3 mit einer Normspaltweite von ≥ 0,65 mm*, IBExU, Freiberg, Germany.
20. IBExU Instititut für Sicherheitstechnik GmbH, 2004, *Prüfbericht IB-03–3–776/4. 2004. Über die Prüfung der strömungsüberwachten Einrichtung KEU GPR-Organ IIB3 DN als Flammendurchschlagsicherung für brennbare Flüssigkeiten und Gase der Explosionsgruppe IIB3 mit einer Normspaltweite von < 0,50 mm*, IBExU, Freiberg, Germany.
21. IBExU Instititut für Sicherheitstechnik GmbH, 2004, *EG-Baumusterprüfbescheinigung gemäß Richtlinie 94/9/EG, Anhang III.2004, Nummer: IBxU04ATEX2227 X*, IBExU, Freiberg, Germany.
22. IBExU Instititut für Sicherheitstechnik GmbH, 2004, *EG-Baumusterprüfbescheinigung gemäß Richtlinie 94/9/EG, Anhang III.2004, Nummer: IBxU04ATEX2228 X*, IBExU, Freiberg, Germany.
23. IBExU Instititut für Sicherheitstechnik GmbH, 2004, *EG-Baumusterprüfbescheinigung gemäß Richtlinie 94/9/EG, Anhang III.2004, Nummer: IBxU04ATEX2229 X*, IBExU, D-09599 Freiberg.
24. IBExU Instititut für Sicherheitstechnik GmbH, 2004, *EG-Baumusterprüfbescheinigung gemäß Richtlinie 94/9/EG, Anhang III.2004, Nummer: IBxU04ATEX2230 X*, IBExU, D-09599 Freiberg.
25. GMBI Nr. 25–29 S. 511-Gemeinsames Ministerialblatt, 2002, *Erste Allgemeine Verwaltungsvorschrift zum Bundes-Immissionsschutzgesetz (Technische Anleitung zur Reinhaltung der Luft-TA Luft)*, Carl Heymanns Verlag KG, 50939 Köln.
26. BGBI Bundesgesetzblatt. 1990. *Siebzehnte Verordnung zur Durchführung des Bundes Immissionsschutzgesetzes (Verordnung über die Verbrennung und die Mitverbrennung von Abfällen – 17. BImSchV)*, http://www.gesetze-im-internet.de/bimschv_17/BJNR025450990.html (accessed on March 20, 2013).
27. Amtsblatt der Europäischen Gemeinschaften. 2000. *Richtlinie 2000/76/EG des europäischen Parlaments und des Rates über die Verbrennung von Abfällen, Veröffentlichungsbedürftige* Rechtsakte. http://eurlex.europa.eu/LexUriServ/LexUriServ.do?uri=OJ:L:2000:332:0091:0111:DE:PDF (accessed on March 20, 2013).

10

Selected Pollution Control Equipment

Klaus-Dieter Zschorsch

CONTENTS

10.1 Combination of Catalytic Oxidizer and Selective Catalytic Reduction

The selective catalytic reduction (SCR) process is based on a selective catalytic reduction of nitrogen oxides, NOx, by ammonia forming harmless nitrogen and water vapor. Due to a very high and cost-efficient removal of NOx, this technology has been successfully supplied to a variety of industries, ranging from the power generation sector through marine diesel engine propulsion to the chemical and pharmaceutical industries.

The catalytic oxidizer technologies for removal of volatile organic compounds (VOCs) or other combustible components from industrial off-gases offer high efficiencies and attractive economical aspects, including very low or zero consumption of energy, low investment costs, and equipment lifetimes twice as long compared to thermal oxidation units.

As legislation on air emissions is gradually being tightened within local legislation and elsewhere, the demand for flexible abatement technologies is increasing. The versatile catalyst-based technologies offer promising solutions when different catalysts are brought together to provide for simultaneous removal of various pollutants. The SCR and catalytic oxidizer technologies have been combined for reduction of emissions. The SCR and catalytic oxidizer technologies have been combined for reduction of emissions of VOCs, ammonia, CO, and NOx in the chemical industry, for removal of NOx, carbon monoxide, and ethylene in CO_2 fertilization plants for greenhouses and for removal of NOx and dioxin in waste incineration plants.

Legislators are gradually tightening regulations of emissions to the air. At the same time, new limits are being introduced on emissions, which were previously not regulated. This has led to an increasing requirement for flexible and efficient emission abatement technologies.

In the past, series of catalysts and processes for air cleaning were developed. These processes included the SCR and catalytic oxidizer processes for removing NOx and VOCs in flue and off-gases. These catalyst-based abatement technologies offer the required flexibility and have been combined and refined to fulfill an ever-increasing demand from various industries for treatment of more complex off-gases with emissions of NOx, VOCs, CO, dioxin, etc.[1] The SCR process is described in detail in Volume 1, Chapter 15. Catalytic combustion itself is described in Volume 1, Chapter 6.

In a number of industrial installations, the SCR and oxidation catalysts and technologies have been combined

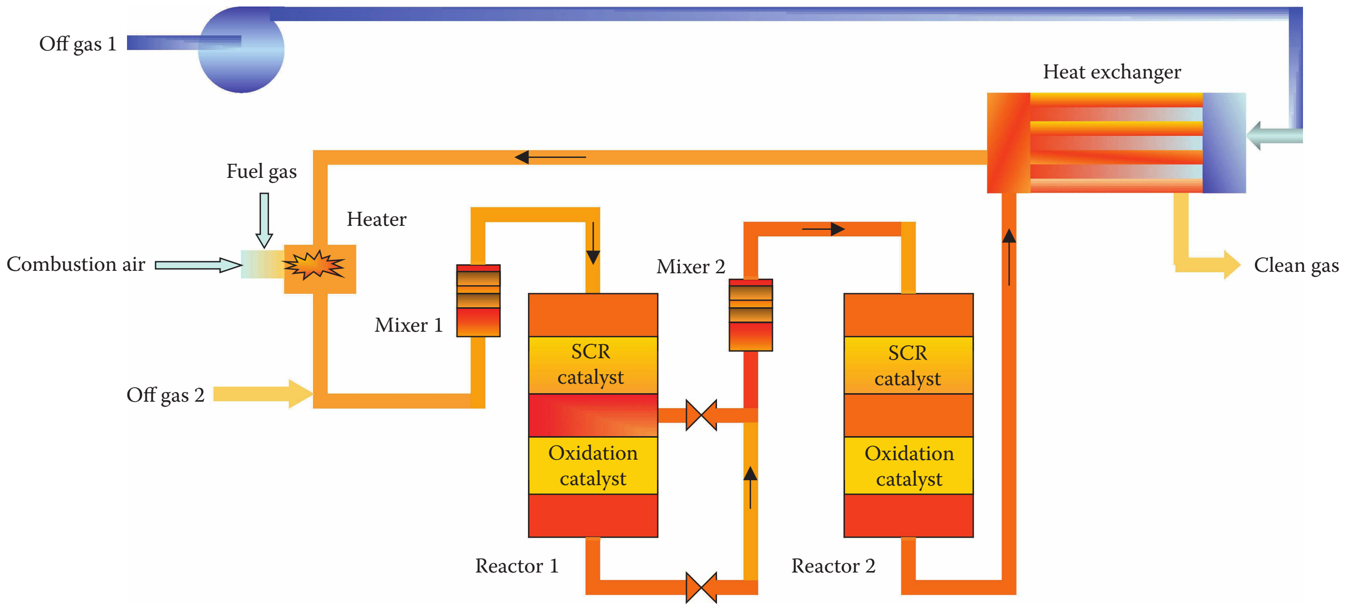


FIGURE 10.1
Combination of an SCR and a catalytic oxidizer for two streams (stream 1: containing high load of VOCs and NH_3, and stream 2: containing low NOx load).

for simultaneous removal of NOx, VOCs, CO, and/or dioxin from more complex off-gases or for special applications. A number of industrial examples exist for different applications. There are several solutions for using a combined system. For example, an SCR can be installed on the reactor inlet and behind the catalytic oxidizer. For this combination, some examples with individual technologies will be described in the following.

For example (Case 1), if one has a flue gas stream containing a high load of VOCs and ammonia and a stream containing a low load of NOx, the possible solution is shown in Figure 10.1.

For treatment, the VOC-containing off-gas stream 1 is preheated in the heat exchanger and then led through the start-up heater, after which the NOx-containing off-gas stream 2 is added. The gas streams are mixed in the static mixer element. The DNO (DN = DeNox, O = Oxidation) catalysts are specifically designed for abatement of carbon monoxide. They are prepared from a SCR catalyst base which is impregnated with an additional active component in the form of palladium. The synergy of vanadium and tungsten oxides and palladium provides the DNO catalysts with a uniquely high oxidation activity. The highly porous SCR base combined with deep-impregnation of palladium furnishes the DNO catalysts with an unsurpassed stability. In the first layer of DNO (a Haldor Topsøe trade name) catalyst in reactor 1, NO and NH_3 react according to the catalytic reactions described in Volume 1, Chapter 6 and methanol is oxidized to CO_2 and H_2O in a precious metal-doped SCR catalyst. After this catalyst, a fraction of the stream is led directly to a second mixer and the remaining part of the stream, fraction 2, is led through the oxidation catalyst in reactor 1, where the remaining methanol is oxidized to CO_2 and H_2O and the NH_3 is oxidized to NOx. The split streams are then reunited in the second mixer where uniform mixing of the NH_3- and NOx-containing streams is established. The homogeneous mixture is led to reactor 2, where NOx and NH_3 react on the SCR catalyst as described earlier and the remaining VOCs are oxidized across the oxidation catalyst. After the reactor, the cleaned off-gas delivers its heat to the cold incoming VOCs laden stream in the heat exchanger before it goes to the stack.

The actual required ratio between fraction 1 and fraction 2, which controls the position of the control valves between the two reactors, is controlled by a programmable logic controller (PLC), which is fed with information about the actual NH_3 content in the off-gas ("feedforward"), and the NOx concentration after the SCR catalyst in reactor 2 ("feedback"). The position of the heat exchanger bypass control valve and the duty of the heater are controlled by the temperature at the inlet of reactor 1.

Another example (Case 2) shows a flue gas stream containing a high load of NOx and CO and a low load of NOx and ammonia. The possible solution is shown in Figure 10.2. The provided cleaning system for treatment of the described off-gas shall handle various composition and flow rates.

The off-gas is taken from the production source to the treatment unit by means of the off-gas main or standby blowers. The off-gas is routed through the heat exchanger at the tube side, where the off-gas is preheated by heat exchange with the hot, cleaned gas. The off-gas is then routed through the electric heater, where

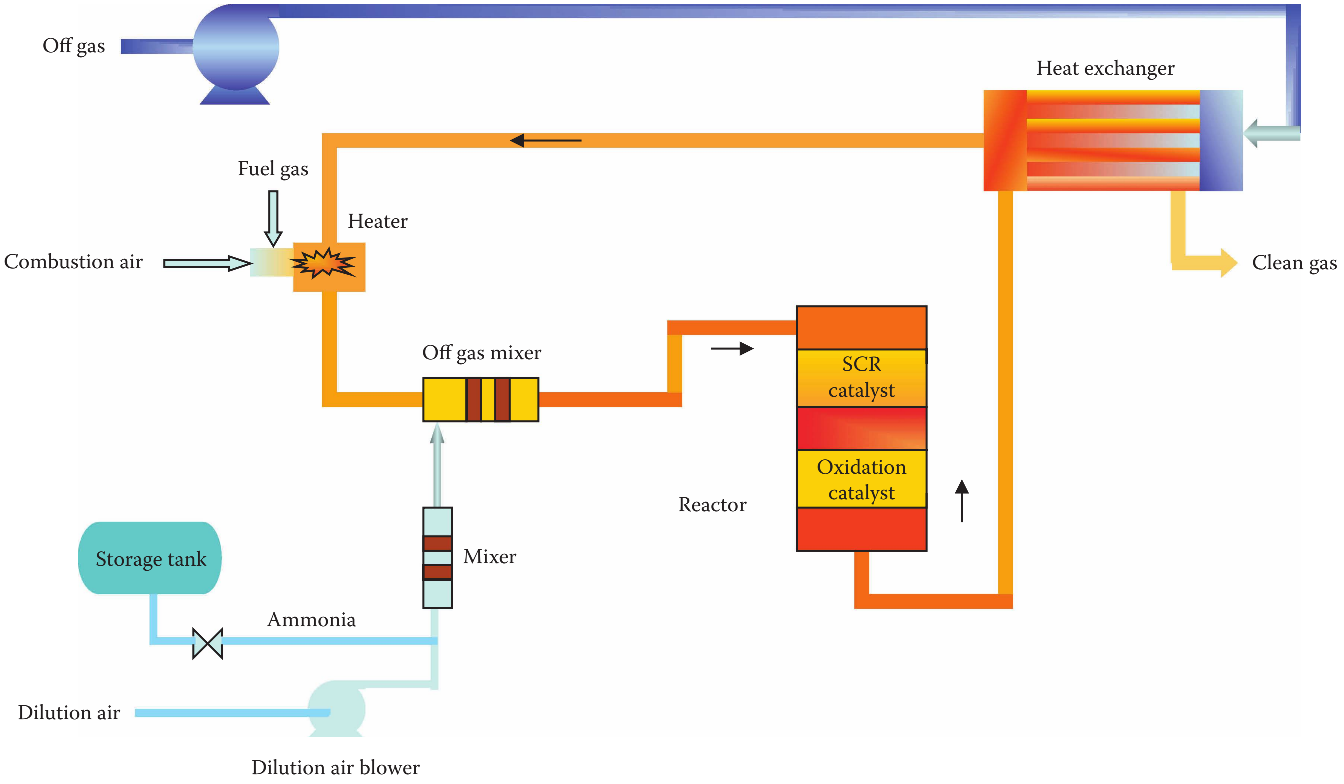


FIGURE 10.2
Combination of an SCR and a catalytic oxidizer for a flue gas stream containing a high load of NOx and CO and a low load of NOx and ammonia.

the temperature is increased, if necessary, to the process temperature. The electric heater is only in operation when the content of combustibles is lower than the specified content necessary for autothermal operation.

From the ammonia storage tank, the ammonia vapor is diluted with air and then injected and mixed into the hot off-gas stream as described previously. In the reactor, the homogeneous off-gas/ammonia mixture passes first through the SCR catalyst where the NOx is reduced as described earlier and secondly through a layer of oxidation catalyst where CO and VOCs and excess NH_3 are oxidized. The cleaned exit gas from the reactor is routed to the shell side of the heat exchanger and from there to the stack.

A typical application is the use of the combination of SCR and catalytic oxidizer for greenhouses (Case 3). Experience has proven that CO_2 fertilization in greenhouses is a feasible way to increase growth. It has for many years been common practice to add CO_2 to the atmosphere of the greenhouses, either in the form of purchased liquid CO_2 or by admitting cooled flue gas containing CO_2 from gas-fired boilers to the greenhouses. For example, power plants with lean-burning gas engines produce CO_2 and it is good practice to utilize the CO_2 in the exhaust gas as the CO_2 source for the growing of tomatoes after proper cleaning. A process has been developed for this specific purpose for the exhaust gas. The exhaust often contains ethylene, CO, and NOx.

The cleaning requirement for a CO_2-containing flue gas destined for a greenhouse is more stringent than for flue gases in stacks. A greenhouse is, to a large extent, considered a closed room. For both human beings and plants, stringent limit values exist that must be observed. For CO, the lower limit is defined by the allowable limit for human beings, whereas for NOx and ethylene, the lower limit is set by the plants. Ethylene is highly undesirable in the greenhouse, as it acts as a plant hormone that makes the tomatoes ripen prematurely.

The unit utilizes an aqueous solution of urea as a reducing agent for the process, as illustrated in Figure 10.3. By means of a frequency-controlled diaphragm metering pump, the urea is passed to a high-pressure injection nozzle from where it is injected into the hot exhaust gas. When in contact with the hot exhaust gas, the atomized urea solution is vaporized and decomposed into NH_3 and CO_2. The reduction of NOx takes place on the SCR catalyst, as described earlier. In the layer of oxidation catalyst, the CO and the ethylene are subsequently oxidized to CO_2 and H_2O.

At the outlet of the unit, the cleaned exhaust gas is continuously analyzed for NOx and CO in order to ensure that the gas fed to the crops in the greenhouses is of the required quality. During start-up of the unit, the exhaust gas is passed to the stack until the levels of NOx and CO are well below the setpoint values as specified earlier.

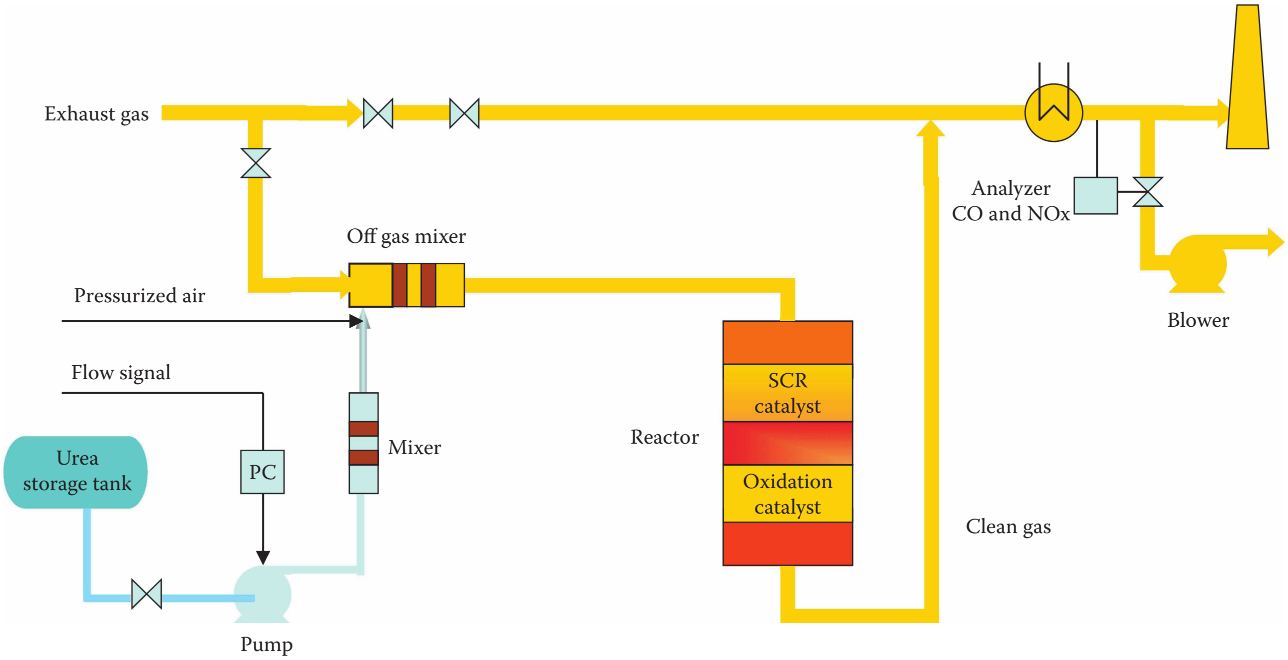


FIGURE 10.3
Combination of an SCR and a catalytic oxidizer for a greenhouse application.

When the set point values are obtained, the greenhouse control system informs the climate control system that the unit is ready to deliver CO_2 to the greenhouses if required. If, during operation of the unit, the content of CO or NOx exceeds the setpoint, the greenhouse distribution blowers will automatically stop and the exhaust will be passed to the stack.

However, when an off-gas containing combustible of nitrogen components (e.g., acetonitrile or ammonia) is treated in a catalytic oxidizer, then at the surface of the catalyst, the nitrogen components or ammonia is converted into CO_2, N_2, and H_2O. Some NO is also created, depending on the catalyst activity and the temperature level.

The ammonia reactions are

$$4NH_3 + 5O_2 \rightarrow 4NO + 6H_2O \qquad (10.1)$$

$$3NO + 2NH_3 \rightarrow 5N_2 + 3H_2O \qquad (10.2)$$

First, ammonia is partially converted into NO and water. Then, the NO reacts with the remaining ammonia to form nitrogen and water. For ammonia oxidation, a special type of oxidation catalyst is available. A small stream of SO_2 is added to optimize the conversion of ammonia over the oxidation catalyst.

If the level of NOx is higher than the allowed limit, an additional cleaning step will be required. The catalytic oxidizer can be combined with an SCR system. The SCR system reduces the NOx concentration up to the desired limit.

The SCR reactions occurring are

$$4NO + 4NH_3 + O_2 \rightarrow 4N_2 + 6H_2O \qquad (10.3)$$

$$6NO_2 + 8NH_3 \rightarrow 7N_2 + 12H_2O \qquad (10.4)$$

Details of the process are described in Volume 1, Chapter 15.

10.2 Combination of Condensation Unit and CO or TO

10.2.1 Introduction

Given the requirement to implement sustainable solutions to problems, the principle of emission reduction and prevention is accorded growing importance. This leads to new demands being made on waste gas cleaning facilities. Based on a wide variety of optimization approaches in production processes and production hardware (e.g., waste gas feedback into the process, enclosure of machines, control of suction units and stress relief units, gas return pipes), today's waste gas streams are normally of significantly lower volumes, often highly pollutant loaded, and due to the substances they contain, they have special properties that need to be taken into consideration. It is often the case that waste gas streams can no longer be cleaned in an ecologically sensible way through a single-stage cleaning method alone (e.g., formation of secondary pollutants). Further process stages may be needed. This depends on the conversion rate of polluting substances in the catalytic waste gas cleaning facility or thermal incinerator facilities, the resulting reaction products, and the utilizations of the released heat. In many cases, conditions imposed by regulatory authorities require the use of combined methods as the solution. Combined methods are multistage facilities where in every stage, depending on the volume flow and on pollutant loads, an adapted process needs to be used. Based on the design parameter volume flows, temperatures, pressure, pollutant load, type of pollutant, properties of component substances, and temporal flow of pollutant accumulation, normally there exist only rough indications for the appropriate area of application where a particular combined method would be used. To ensure optimal selection of a combined method, the operational circumstances and possibilities of the production system or of the entire plant should be included in the study. This preliminary study should be conducted at a very early project stage; otherwise, it is normally too late to set the course for economically and ecologically optimized combined methods. Before selecting a combined method, therefore, various possibilities should be considered. In special cases, in order to ensure operation within the permissible working range, the raw gas should undergo conditioning or precleaning. In order to prepare the final treatment reaction, protect this system, and ensure safe facility operation, it is generally necessary to condition the waste gas being treated. This conditioning can take place in one or several process steps and depends on the chemical and physical properties of the treated waste gas, the required clean gas parameters, and the required operational parameters at the final treatment system. A condensation unit can be combined with an upstream catalytic combustion (CO) or a thermal incinerator unit (TO) (see Chapter 8). Condensation systems are used when waste gas flows containing high organic loads have to be treated and solvents can be reused economically. The condensation unit is intended to clean the waste or vent gas and to recover the consisted solvent at the same time. A condensation stage may generally be useful where there are large quantities of condensable substances.

However, according to the forgoing explanation, one solution to treat a pollutant- or solvent-loaded waste stream in accordance with specified emission limits is the use of a two-stage process, consisting of a low-temperature condensation unit provided as precleaning system and a catalytic combustion or thermal incinerator provided as final cleaning.

The condensation units can be distinguished between direct condensation in a liquid bath of deep cooled solvents, here called FRIGOSOLVER® unit (see Figure 10.4), and condensation based on contact with a sprayed flow of deep cooled solvents, here called spray cooler.

FIGURE 10.4
FRIGOSOLVER® unit.

Advantages of such kinds of systems versus other recovery systems are as follows: continuous and batch operation are possible, high purity and often direct reuse of the recovered solvents reached by recovery process conditions, no fog formation, small temperature difference between liquid and gas phase, absorption by input of liquid phase is possible, operation at atmospheric pressure or overpressure is possible, little space required by compact design, safe and economical technology and operation with electricity as sole energy source is possible.

The condensation systems will be provided in applications such as the recovery of solvents from drying circuits (e.g., of coating plants and mixers), a volume reduction in gas displacement systems (e.g., tank farms and loading stations), or condensation as part of the production processes. Often, in the chemical and pharmaceutical industries, waste gas conditioning occurs upstream of a combined treatment plant.

10.2.2 Condensation System and Its Process

The condensation system and its process can be described in short form as follows.

10.2.2.1 Direct Condensation in a Liquid Bath of Deep Cooled Solvents (FRIGOSOLVER® Unit)

Principle:

- Dipped heat exchanger (cooling coil) in a liquid bath (see Figure 10.5)
- Rising bubbles divided by a sieve plate

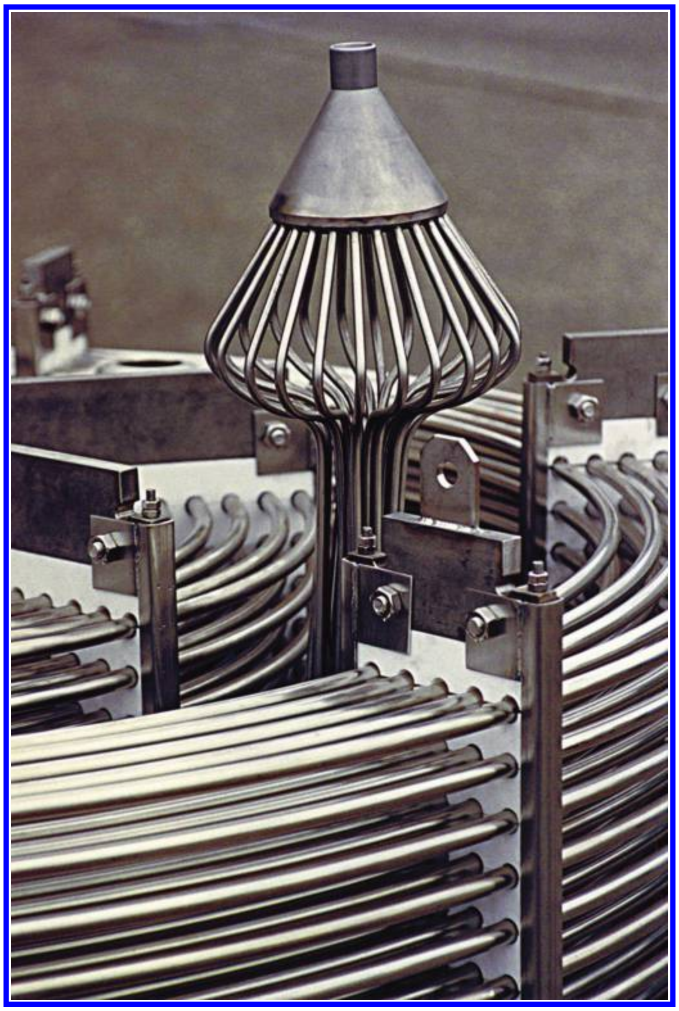


FIGURE 10.5
Cooling coil.

- Condensation at the bubbles' surface
- Throughput by a fan at the suction side or pressure side
- Cooling by conventional refrigeration system and then the cooling coil works as an evaporator or cooling by chilled solvents
- Removal of condensate by liquid-level control

Advantages:

- Condensation at temperatures down to −45°C (−45°F) or lower.
- Continuous processing.
- Pressure condensation possible.
- Low ΔT between liquid phase and gaseous phase.
- No outlet peaks depending on inlet peaks.
- High heat transfer efficiency.
- Large exchange surface.
- Condensation down to saturation at bath temperature with presence of uncondensable carrier gases.
- No dust is formed and existing dust will be separated.
- No plugging and blocking of the heat exchanger by crystallized substances if they crystallize at the bubbles' surface.
- Up to 10% solids content allowed. (Reason for solids is crystallization of components according to its melting point, e.g., water.)
- No aerosol formation.
- Installation of an additional defrosting system is possible (with electrical heater or with reverse of refrigerant to preheating function).
- Suitable for a wide range of organic compounds and solvents.

There are some possible disadvantages and cautions depending on the construction and the kind of operation. These include high pressure drop, a volume flow limit of approximately 1500 m^3/h (53,000 ft^3/h), a heat transfer limit of approximately 75 kW (260,000 Btu/h), and possible plugging of the exchanger surface if there is a partly water-soluble solvent present or if heterogeneous crystallization of the solvent mixture is possible.

10.2.2.2 Condensation Based on Contact with a Sprayed Flow of Deep Cooled Solvents (Spray Cooler)

Principle:

- Gas is cooled by contact with a downwardly sprayed liquid stream.
- The gas phase cools down close to the assumed liquid temperature with vapor pressure in accordance with the compound of the liquid mixture and the temperature.
- The gas is drawn by an injector driven by liquid.
- Accumulation of the condensate product in the bottom of the spray cooler.
- Cooling of the circulation flow in water chiller cooled or directly cooled heat exchanger.
- Pumping of the condensate back to the spraying nozzle via the precooled heat exchanger.
- Removal of condensate by liquid-level control.

TABLE 10.1

Plant Attribute

	FRIGOSOLVER®	Spray Cooler
Inlet temperature	0°C up to +80°C	0°C up to +300°C
Condensation temperature	0°C down to −45°C	+25°C down to −30°C
Condensation pressure	0 up to 10 bar	0 up to 10 bar
Condensation capacity	Up to app. 75 kW	Up to app. 1 MW
Throughput	20 Nm³/h up to1.500 Nm³/h	20 Nm³/h up to 5.000 Nm³/h or more
Operation	Continuously/discontinuously	
Inlet concentration	0%–100% saturation	
Inlet flow	0 to max concepted flow	

Advantages:

- Condensation cooled to approximately −35°C (−31°F) or lower.
- Continuous processing.
- Pressure condensation possible.
- Adsorption by feed of absorbance possible.
- Suitable for high-volume flows and heat transfers.
- Low ΔT between liquid phase and gaseous phase.
- No outlet peaks depending on inlet peaks.
- No separate blower is required.

There are several possible disadvantages and special cautions, depending on the construction and kind of operation that have to be taken into account. These include crystallizing of single substances or mixtures (eutecticum), insolubility of mixtures (if water is present, condensation requires a condensation temperature higher than 0°C [32°F]), and partly soluble mixtures and crystallizing of a substance out of the liquid phase when solubility is exceeded. Table 10.1 shows a comparison of the FRIGOSOLVER unit and a spray cooler.

10.2.2.3 Condensation Unit Based on Heat Exchanger System

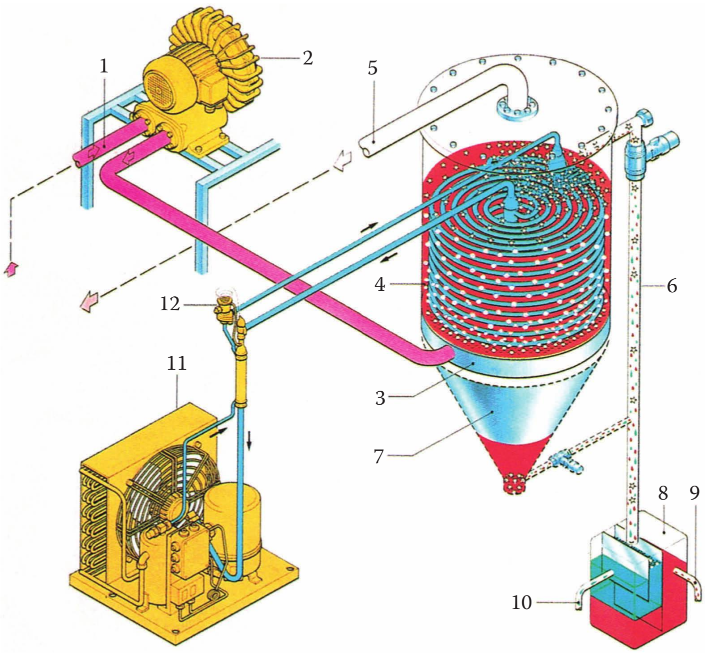
The condensation can also be provided in a suitable heat exchanger system. Here, a plate- or tube-type heat exchanger will be operated with a conventional refrigeration system, chilled water, or nitrogen evaporation for cooling. Initially, the waste gas consisting of condensable solvents or pollutants starts the condensation due to the saturation temperature direct on the surface of cooled plate or tubes. Depending on the application, often redundant heat exchangers for defrosting are required. The condensate has to be separated or sampled in a liquid separator. Such a system works well using pure vapors without incondensable carrier gases with the advantage of low pressure drop. The condensation in a heat exchanger is a normal and well-known system, but there are disadvantages and special precaution must be taken for cases of unsuccessful heat transfer characteristic; low exchange surface and dust may appear, which are difficult to separate. According to different vapor pressures of the solvent, fractional condensation of substances is possible. The heat exchanger can plug and become blocked by crystallized substances or damaged by freezing water. Also there is the risk of aerosol formation.

10.2.3 Methods

A two-stage treatment method designed as a combined process consisting of a condensation unit and a catalytic combustion (CO) or thermal incinerator (TO) is a proven technology. Depending on the treatment application and requirement of clean gas conditions, the first treatment stage is often a FRIGOSOLVER unit or a spray cooler. Both can be combined with a CO or TO. The following sections deal with the details of combination systems. The described information is only a guideline. It has to be adapted to the application based upon experience.

10.2.3.1 Direct Condensation in a Liquid Bath of Deep Cooled Solvents (FRIGOSOLVER® Unit)

The FRIGOSOLVER unit essentially consists of a vessel and a cooling coil inside the vessel, which utilizes a refrigerant unit to condense solvents from the process vent stream (for details, see Figure 10.5). The vessel typically can be operated between −45°C to 5°C and (−49°F to 41°F) depending on the pressure loss of liquid bath around 850 mbar (12.3 psia) at the vessel inlet. Condensable components are collected in the vessel and will be transferred to a slob or separator tank via a suitable pump. The vapor phase (process vents) will be drawn off the process by an induction fan, which forms part of the FRIGOSOLVER unit vent treatment package or unit. From the process, the vapor passes the vent gas inlet pipe. After passing the inlet, vent gases are drawn into the refrigerated FRIGOSOLVER unit vessel. The vents flow into the FRIGOSOLVER unit vessel



1. Solvent loaded exhaust air
2. Fan
3. Perforated bottom
4. Chilled, liquid solvent
5. Clean air outlet
6. Drain for condensate, ice and deposits
7. Ice hopper, condensate drained only in case of solvents of a low specific weight
8. Solvent/water separator
9. Solvent outlet
10. Water outlet
11. Refrigerating machine
12. Cutoff value

FIGURE 10.6
Direct condensation in a liquid bath of deep cooled solvents, here called the FRIGOSOLVER® unit.

under the sieve plate of the column by which it is divided into small bubbles. The solvent vapor in the rising bubbles condenses and absorbs at the bubbles' surface. The moisture in the vent air forms into small ice crystals, which will flow in the nonwater-soluble solvents and rise to the top or sink to the bottom of the cooling vessel, depending on specific weight. Solvents and ice remain in the cooling vessel and rise according to its liquid level. The condensable material (water and cycles) will start to condense as it passes into the vessel and the liquid will build up in the vessel (see Figure 10.6).

When the liquid in the vessel reaches a prescribed high level, a level control sequence will initiate the start of the transfer of condensed liquid from the vessel to slop tank, which will be transferred for off-site disposal of the liquid waste or if possible will be used as a recovered solvent. The transfer of liquid to the awaiting slop tank is accomplished by a suitable transfer pump. When the level drops to a prescribed low level, the level control will stop the transfer of liquid waste or recovered solvent to the slop tank.

The pressure in the separator vessel is controlled by adjusting a pressure-controlled recycle valve around the FRIGOSOLVER unit induction fan. If the pressure increases in the separator vessel, the recycle valve closes; if the pressure in the separator goes to low, the recycle valve will open.

The noncondensable material (air or nitrogen, plus small amounts of water vapor and solvents in accordance with their saturation concentration) passes from the refrigerated condensation vessel of the FRIGOSOLVER unit and is drawn off via an induction fan and routed to the next step of the treatment system such as a catalytic waste gas treatment system (CO) or thermal incinerator (TO).

The FRIGOSOLVER unit will require cooling water for cooling of the packaged refrigeration system, but also an air-cooled condenser can be provided to cool the refrigerant unit.

10.2.3.2 Condensation Based on Contact with a Sprayed Flow of Deep Cooled Solvents (Spray Cooler)

If the first treatment step (precleaning) is a spray cooler, the vapor phase (vent gas stream) is entrained by a spraying liquid at its nozzle system, which is designed as an ejector. This gas will flow by slightly increasing the pressure through the system. Therefore, the installation of a fan is not required. The mixture of solvents and water from the sump of the cooler is drawn by a pump and is supplied via heat exchanger to the ejector nozzle. On the other side, the heat exchanger is connected with a water-cooled refrigerator (air-cooled refrigerator or chilled system), which transfers the required condensation heat. The spray jet accelerates the gas throughput and also initiates and forms the surface of the condensation and acts as a heat transfer medium. The condensate is collected, together with the spray jet, in the sump of the unit. The cleaned gas flows through a droplet separator, which is composed of wire gauze material. The noncondensable material (air or nitrogen, plus small amounts of water vapor and solvents in accordance with their saturation concentration) will be led to the next step of treatment system such as a catalytic waste gas treatment system (CO) or thermal incinerator (TO). The condensate level is controlled by a level probe. The liquid level is maintained by periodical removal of the condensate.

10.2.3.3 In Combination Operated Catalytic Combustion or Thermal Incinerator

The vent or waste gas, which is precleaned in the condensation unit, will be led in the case of a two-stage treatment system to the final treatment stage. This stage can be a catalytic combustion or thermal incinerator (for an example of combined treatment system, see Figure 10.7).

Depending on the condensation process, the precleaned gas concentration is in accordance with the saturation temperature of the constituent components. In this regard, certain conditions have to be maintained (see Figure 10.8).

The design of the final cleaning step of a combined treatment system must be such that the gas is not present in explosive form or as an ignitable mixture. For safe operation, the concentration of pollutant must be kept below 25% of the lower explosion level (LEL). Refer to the calculation of lower explosion limit. Observe here that various temperatures are reached as a result of warming the waste gas. The prevention of explosive mixtures can be ensured through various methods, including gas concentration monitoring at the inlet to the catalytic cleaning facility, concentration-dependent limiting of waste gas preheating temperature, or in the event waste gas mixtures loaded with inerts.

The measuring equipment for concentration monitoring should be installed at the necessary distance from the inlet of the catalytic combustion or thermal incinerator cleaning facility taking into account the relevant response times. The monitoring device should be arranged such that the remaining path of the loaded gases from the device to the cleaning facility is adequate for safe switchover to the emergency discharge systems or safe shutoff of the discharge. After exceeding the ignition concentration, the pipeline system and the facility should be flushed. In addition, any electrostatic charges during transport of the organic components through synthetic pipes or lined systems should be prevented, e.g., by grounding the pipelines and by using electrically conducting materials.

Concentrations need to be monitored to avoid overloading or overheating the catalytic combustor or thermal incinerator. Depending on waste gas conditions, setting and monitoring of the oxygen content (min. O_2 content) is required. In any circumstance, the intake concentration of pollutants in waste gas or polluted gas, which is oxidizable, has to be at a certain limit before entry into the reaction zone.

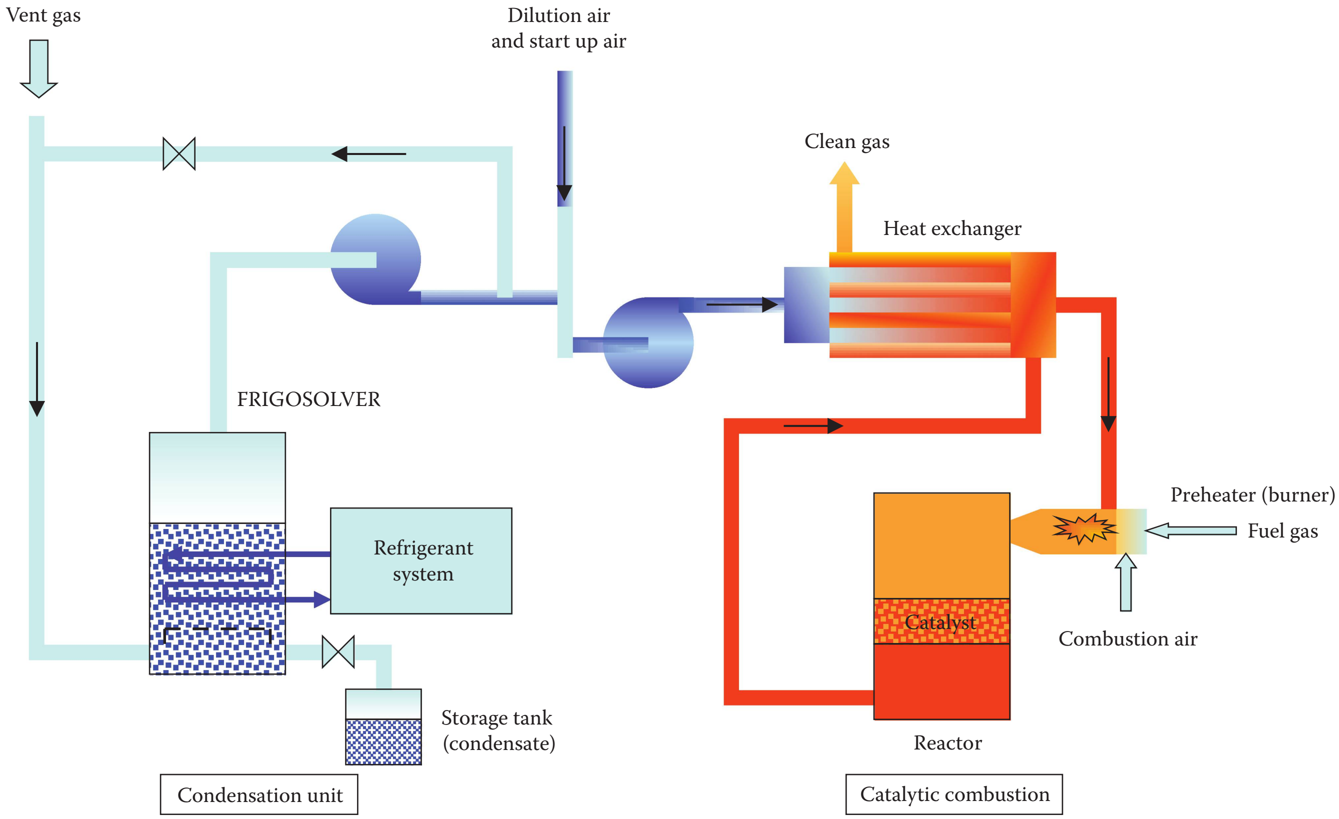


FIGURE 10.7
Combination of condensation unit and catalytic waste combustion.

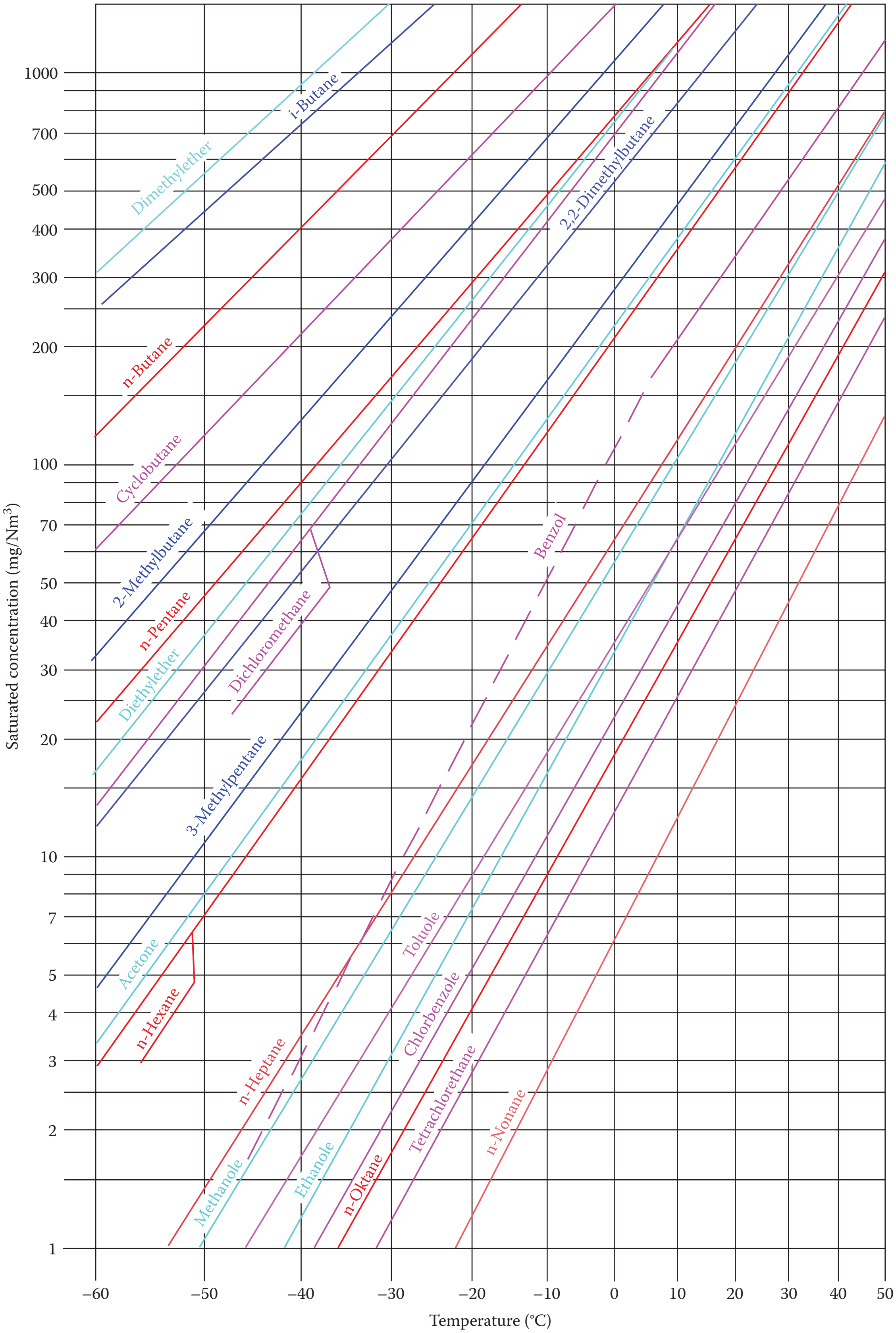


FIGURE 10.8
Saturated concentration in accordance with saturation temperature.

If the occurrence of an explosive waste gas stream cannot be ruled out due to the nature and the operation of the emitting processes and the physical conditions, then pursuant to the explosion-safety guidelines, this waste gas stream must be allocated to an explosion zone according to the occurrence probability of an explosive gas mixture. The allocation indicates the number of the independent explosion-safety systems required for separating the production process from the waste gas cleaning facility.

Upon start-up, the system must be heated with fresh air or circulated air. After a plant failure, the entire waste gas cleaning system must be cleaned. That means the flushing procedure has to be started. For a shutdown of the plant, the supply piping must be purged and the emission concentration of the organic components monitored.

Generally the final treatment, the precleaned waste or vent gas is fed to the inlet of the catalytic combustor or thermal incinerator. The condensation stage is diluted with fresh air to decrease the hydrocarbon concentration below 25% of the lower explosion limit. The diluted gas stream is drawn by the fan of the provided treatment system. It enters the heat recuperative exchanger where it is preheated by exchange with the cleaned gas from the combustion to the required reactor or catalyst inlet temperature. After the heat exchanger, the gas flow passes the electrical heater or fuel gas preheater and enters the reactor where the catalytic combustion takes place in presence of the catalyst. After combustion and heat exchange with the incoming off-gas, the now cleaned air is led to the atmosphere via the stack. The increase in the temperature in the catalyst bed depends on the solvent or combustible concentration and its net calorific value. As the resulted temperature increases, which depends on the efficiency of heat exchanger, the plant works without any additional heating from an electrical heater or burner.

Reference

1. A. B. Jensen, H. Jensen-Holm, and M. Schröter, Catalytic removal of NOx, VOC and dioxin, Originally presented at *Pollutec 2000*, October 17–20, Lyon, France.

11

Flares

Robert E. Schwartz, Jeff White, and Wes Bussman

CONTENTS

11.1 Flare Systems

During the operation of many hydrocarbon industry plants, there is the need to control process conditions by venting gases and/or liquids. In emergency circumstances, relief valves act automatically to limit equipment overpressure. For many decades of the last century, process vents and pressure relief flows were directed, individually or collectively, to the atmosphere unburned. Gases separated from produced oil were also vented to the atmosphere unburned. The custom of unburned venting began to change in the late 1940s when increased environmental awareness and safety concerns created the desire to convert vents to continuously burning flares.

Burning brought about the need for pilots and pilot ignitors and the need for awareness of the design factors and considerations imposed on a system by a flame at the exit. In many cases, the desirable flaring of the gases was accompanied by objectionable dense black smoke as shown in Figure 11.1. In addition to their development of flare pilots and ignition systems, industry pioneers John Steele Zink and Robert Reed[1] invented the first successful smokeless flare burner (Figure 11.2) in the early 1950s. This invention was an important point in the transition from unburned vents to flaring and from vent pipes to burners specifically designed for flare applications.

While the combustion fundamentals discussed in Volume 1 continue to apply, flare burners differ from process and boiler burners in several respects, including

FIGURE 11.1
Typical early 1950s flare performance.

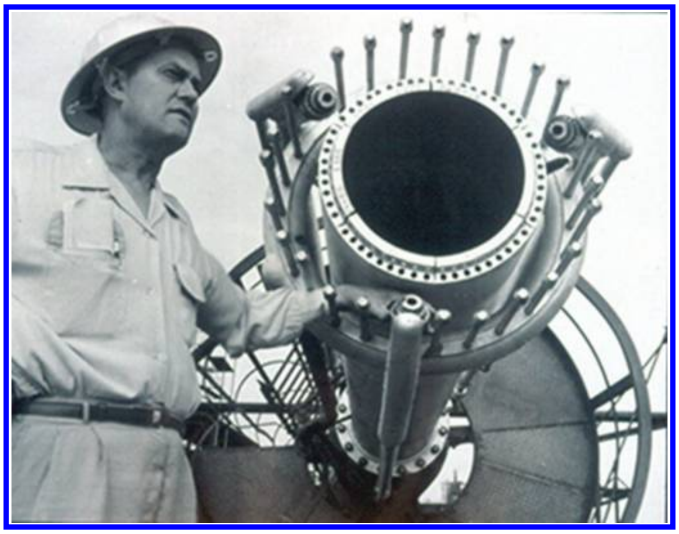


FIGURE 11.2
An early model smokeless flare.

1. The composition of the gases handled by a flare often varies over a much wider range.
2. Flares are required to operate over a very large turndown ratio (maximum emergency flow down to the purge flow rate).
3. A flare burner must operate over long periods of time without maintenance.
4. Flare burners operate at high levels of excess air as compared to other burners.
5. Many flare burners have an emergency relief flow rate that produces a flame hundreds of feet long with a heat release of billions of Btu per hour (Figure 11.3).

11.1.1 Purpose

The wide range of applications for flares throughout the hydrocarbon and petrochemical industries challenges plant owners and designers as well as the flare equipment designer. The purpose of this chapter is to provide an understanding of the design considerations and factors influencing flare system and equipment design. The most frequently used flaring techniques and associated equipment are also discussed.

11.1.2 Objective of Flaring

All flare systems have a common prime objective: safe, effective disposal of gases and liquids … at an affordable cost. There should be constant awareness that flare system design and operation must never compromise the prime objective.

FIGURE 11.3
Major flaring event. (Note that the stack height is over 300 ft [90 m].)

11.1.3 Applications

Within the hydrocarbon and petrochemical industries, from the drilling site to the downstream petrochemical plant and at many facilities in between, flares are utilized to achieve the prime objective. Individual flare design capacity can range from less than 100 to more than 10 million lb/h (4.5 to more than 4.5 million kg/h). Material released into a flare system is often a mixture of several constituents that can vary from hydrogen to heavy hydrocarbons and may at times include inert gases. Some of the heavy hydrocarbons may be in the gaseous state when released into the system, but can condense as they cool.

While this chapter focuses on flares and flaring in the hydrocarbon industry, many of the subjects discussed also relate to flare applications in other industries.

The design requirements for a given facility are seldom identical to those of any other facility. This variation, plus the wide range of flare applications and site conditions, often requires that the flare system be custom designed.

11.1.4 Flare System Types

Flares for service in the hydrocarbon and petrochemical industries are generally of the following types or combinations thereof:

- Single point
- Multipoint
- Enclosed

11.1.4.1 Single-Point Flares

Single-point flares can be designed with or without smoke-suppression equipment and are generally oriented to fire upward, with the discharge point at an elevated position relative to the surrounding grade and/or nearby equipment (Figure 11.4). Occasionally, a single-point flare is positioned to fire more or less horizontally, usually over a pit or excavation (Figure 11.5). Horizontal flares are generally limited to drilling and production applications where there is a high probability of nonrecoverable liquids.

FIGURE 11.4
Typical elevated single-point flare.

FIGURE 11.5
Typical pit flare installation.

FIGURE 11.6
A grade-mounted, multipoint LRGO flare system.

FIGURE 11.7
Elevated multipoint LRGO flare system.

11.1.4.2 Multipoint Flares

Multipoint flares are used to achieve improved burning by routing the gas stream to a number of burning points. For refinery or petrochemical plant applications, multipoint flares are usually designed to achieve smokeless burning of all flows. Such flares often divide the multiple burning points into stages to facilitate better burning. The multiple burning points can be arranged in arrays located at or near grade (Figure 11.6) or at an elevated position (Figure 11.7).

11.1.4.3 Enclosed Flares

Enclosed flares are constructed so as to conceal the flame from direct view. Additional benefits can be the reduction in noise level in the surrounding community and

FIGURE 11.8
Multiple ZTOF installation in an ethylene plant.

FIGURE 11.9
Combination elevated flare system (left) and ZTOF (right).

minimization of thermal radiation. Capacity can be the system maximum, but is often limited to a flow rate that will allow the connected facility to start up, shut down, and operate on a day-to-day basis without exposed flame flaring. Multiple enclosed flares are sometimes used to achieve the desired hidden flame capacity. Each of the two units shown in Figure 11.8 is designed for 100 metric tons/h (110 tons/h) of waste gas from ethylene furnaces during start-ups.

11.1.4.4 Combination Systems

A common combination is an enclosed flare of limited capacity paired with an elevated flare (Figure 11.9) that is sized for the maximum anticipated flow to the system. Such a pairing results in a flare system that only has an exposed flame during major upset or failure events. Other pairings, such as an elevated flare with a multipoint flare, have also been used (Figure 11.10).

11.1.5 Major System Components

Each flare system type has its own set of required components. In addition, systems may include components that are noted in the following as being optional. These optional components address special needs, such as smoke suppression or liquid removal. Optional equipment can also help reduce cost or aid system operation. It should be noted that there is a difference between the flare burner, or burners, required for each type of flare system.

FIGURE 11.10
Combination elevated LRGO and utility flare system.

11.1.5.1 Single-Point Flares

For a single-point flare, the major required and optional components are

1. Flare burner, with or without smoke-suppression capability
    a. One or more pilots
    b. Pilot ignitor(s)
    c. Pilot flame detector(s)
2. Support structure, piping, and ancillary equipment
3. Purge reduction device (optional)
4. Knockout drum (optional)
5. Liquid seal (optional)
6. Auxiliary equipment
    a. Smoke-suppression control (optional)
    b. Blower(s) (optional)
    c. Flow, composition, heating value, or video monitor (optional)

11.1.5.2 Multipoint Flares

For a multipoint flare, the major required and optional components are

1. Two or more multipoint flare burners
2. Pilot(s), pilot ignitor(s), and pilot flame detector(s)
3. If elevated, support structure and ancillary equipment
4. A fence to limit access and reduce flame radiation and visibility (optional)
5. Knockout drum (optional)
6. Liquid seal (optional)
7. Piping
8. Auxiliary equipment
    a. Staging equipment and instrumentation (optional)
    b. Smoke-suppression means where very large turndown is required
    c. Flow, composition, heating value, or video monitoring (optional)

11.1.5.3 Enclosed Flares

For an enclosed flare, the major required and optional components are

1. Flare burners, with or without smoke-suppression capability
2. Pilot(s), pilot ignitor(s), and pilot flame detector(s)
3. Enclosure/structure with protective refractory lining
4. A fence to limit access
5. Knockout drum (optional)
6. Liquid seal (optional)
7. Piping and optional heat shielding
8. Auxiliary equipment
    a. Staging equipment and instrumentation (optional)
    b. Flow, composition, heating value, or video monitoring (optional)

11.2 Factors Influencing Flare Design

As one approaches the specification of a flare system, there must be an awareness of certain factors that influence size, safety, environmental compliance, and cost. Major factors influencing flare system design[2] include

- Flow rate
- Gas composition
- Gas temperature
- Gas pressure available
- Utility costs and availability
- Safety requirements
- Environmental requirements
- Social requirements

Information regarding each of these factors is normally available to the plant designer and/or the plant owner. These factors define the requirements of the flare system and should be made available to the flare designer as early in the design process as possible.

In reviewing the list of factors, it can be seen that the first four factors are all determined by the source(s) of the gas being vented into the flare header. The next factor is related to the design of the facility itself and its location. Safety, environmental, and social requirements all relate to regulatory mandates, the owner's basic practices, and the relationship between the facility and its neighbors. A discussion of each factor is provided in the following.

11.2.1 Flow Rate

The flare system designer relies heavily on the flow data provided. Therefore, the data must realistically reflect the various flow scenarios. Overstatement of the flows will lead to oversized equipment, which increases both capital and operating costs, and can lead to shorter service life. Understatement can result in an ineffective or unsafe system.

Flow rate obviously affects such things as the mechanical size of flare equipment. Its influence, however, is much broader. For example, increased flow generally results in an increase in thermal radiation from an elevated flare flame (see Volume 1, Chapter 8), which in turn will have a direct impact on the height and location of a flare stack.

The maximum emergency flow rate can occur during a major plant upset such as the total loss of electrical power or cooling water. However, some processes have their maximum flow rate under less obvious emergency conditions such as partial loss of electrical power whereby, for example, pumps continue to supply feedstock to a disabled section of the plant.

The duration of the maximum flow rate can affect flare system design in a number of ways. For example, the length of time a worker is exposed to heat from the flare flame can affect the choice of allowable heat flux. Usually, a very short-duration relief into a flare system can result in a relatively high allowable radiation. In contrast, a very long-duration, high-flow relief may require a lower design allowable radiation level.

In the past, the maximum flow rate was sometimes determined by summing the flow rates of each of the connected relief devices. This approach resulted in an unrealistically large maximum flow rate because the assumption that all the connected devices relieve simultaneously is often false. Modern plant design and analysis tools such as dynamic simulation allow the process designer to define more appropriately the maximum flow rate to the flare. Careful attention to the design of control and electrical power systems can significantly reduce flare loads as well.[3]

In addition to the maximum flow conditions, it is also important to explicitly define any flow conditions under which the flare is expected to burn without smoke. These flow conditions can come from process upsets, from incidents such as a compressor trip-out, or from various operations of the plant, including start-up, shutdown, and blowdown of certain equipment. Attempts to shortcut the establishment of factually based smokeless burning scenarios by setting the smokeless flow rate as a percentage of the maximum emergency flow rate can lead to disappointment or needless expense.

11.2.2 Gas Composition

Gas composition can influence flare design in a number of ways. The designer should be given the gas composition for each of the flow conditions identified earlier and for any special gases that may be in use, such as pilot fuel or purge gas. By studying the gas composition, its combustion characteristics and the identity of potential flue gas components can be determined.

For example, the composition reveals the relative presence of hydrogen and carbon. The weight ratio of hydrogen to carbon in a gas is one of the parameters that can indicate the smoking tendency of the gas.[4] The influence of the weight ratio of hydrogen to carbon, often referred to as the H/C ratio, on the smoking tendency is illustrated in Figure 11.11. Figure 11.11 shows the flame produced by burning three different gases using the same flare equipment and operating conditions. The flame produced by burning a 25 molecular weight (MW) wellhead natural gas (H/C = 0.27) is clean, as shown in Figure 11.11a. The flame in Figure 11.11b reflects the burning of propane (H/C = 0.22). Note that the propane flame has some trailing smoke, has a yellow color much closer to the flare burner, and is more opaque when compared to the natural gas flame. The dense black smoke and dark flame shown in Figure 11.11c were produced by burning propylene (H/C = 0.17). Note that a portion of the flame is being cloaked or shrouded by the smoke. The fact that the smoke hides part of the flame must be taken into account when calculating the radiation from the flare flame.

Composition analysis also will reveal the presence of nonhydrocarbons such as hydrogen sulfide or inerts. Such gases might require special metallurgies or design considerations such as ground-level concentration (GLC) analysis. Composition combined with the flow rate allows determination of the volume flow or mass flow of gases to be handled by the flare system.

The practice of defining a stream by its bulk properties alone (molecular weight [MW], lower heating value [LHV], upper/lower explosive limits [UEL/LEL], etc.) can disguise safety hazards or prevent equipment and operating cost reductions that would otherwise be recognized by the flare designer. For example, a 28–30 MW gas could be ethane, ethylene, nitrogen, carbon monoxide, air, or even a mixture of hydrogen and xylene. A flare system to handle some fixed amount of 28 MW gas would have very different design and operating characteristics, depending on the actual gas composition involved. Ethylene may tend to smoke, but remains lit and stable at high exit velocities. Carbon monoxide, on the other hand, will not smoke, but can be difficult to keep lit even at moderate to low exit velocities. Radiation, possible relief gas enrichment, purge requirements, and the potential for condensation at ambient temperatures are other examples of the impact that gas composition can have on flare design.

11.2.3 Gas Temperature

In addition to the impact of relief gas temperature on thermal expansion, gas volume, and metallurgical requirements, there is the more subtle effect of gas temperature

(a)

(b)

(c)

FIGURE 11.11
Comparison of the flame produced by burning (a) 25 MW wellhead natural gas, (b) propane, and (c) propylene.

on the potential for some components of the gas to condense. Possible condensation or two-phase flow necessitates liquid removal equipment to avoid a higher smoking tendency and/or the possibility of a burning liquid rain. Condensation at low or no flow conditions will result in the formation of a vacuum condition in the flare header and the resulting potential to draw air in through the flare tip or through piping leaks. Liquid seals are sometimes used to address this hazard. However, gas temperature can affect liquid seal design and operation. Hot gases will tend to boil off the seal fluid, sometimes very suddenly. On the other hand, extremely cold gases present a freezing scenario that could completely block waste gas flow.

While a flare stack may appear to be unrestrained and therefore free to expand, there can be mechanical design challenges as a result of large gas temperature variations. Header piping growth, relative movement of utility piping, and stack guy wire tensioning are just three areas where problems can arise. Both high and low temperatures have the potential to create issues that affect the design of the stack. In cases where the relief gas source pressure is extremely high, the plant designer should account for cooling by expansion across the relief or vent valve. Where the gas temperature at the source is significantly different from ambient, it is advisable to estimate the heat loss or gain through the flare header walls from the source to the flare stack and to determine the resulting gas temperature at the flare. Attention to such details can result in a reduced cost for the stack.

11.2.4 Gas Pressure Available

The gas pressure available for the flare is determined by hydraulic analysis of the complete pressure relief system from the vent or pressure-relieving devices to the flare burner. Each major flowing condition should be analyzed to determine the pressure at each relief or vent in each branch of the flare header. This pressure is usually limited by the lowest allowable back pressure on any relief valve in the system. The limit applies to all flowing conditions, regardless of whether or not the limiting relief valve contributes to the flowing condition under study.

In most flare systems, much of the system pressure drop is due to flare header piping losses with little pressure drop remaining for the flare burner. Such system designs may not maximize the value of the gas pressure in promoting smokeless burning. Smokeless burning can be enhanced by converting as much of the gas pressure available as possible into gas momentum. In addition, redistributing the system pressure drop to provide more pressure at the flare tip can reduce the overall system cost.

Another benefit of taking a greater pressure drop across the flare burner is the increase in gas density in the flare header, which can lead to a smaller flare header size and reduced piping cost. More pressure at the flare tip generally means a smaller flare burner and, consequently, lower purge flows. The enhancement of smokeless burning and the decrease in purge gas requirements both reduce the day-to-day operating cost. Both capital and operating costs can be reduced in this manner.

Available gas pressure at the flare can be defined as total pressure at the flare inlet or as static pressure in a specific size inlet pipe. Static pressure is the pressure applied by the gas to the walls of the pipe. That is, this is the pressure sensed by a pressure gauge mounted on a simple nozzle in the side of the pipe. This is also the pressure that determines gas density. Total pressure is the sum of the static pressure and the velocity pressure at a given point in the piping (e.g., the stack inlet). When static pressure is used to define available gas pressure, the plant designer should also specify the anticipated inlet size.

11.2.5 Utility Costs and Availability

In many cases, the momentum of the gas stream alone is not sufficient to provide smokeless burning. In such cases, it is necessary to add an assist medium to increase the overall momentum to the smokeless burning level. The most common medium is steam, which is injected through one or more groups of nozzles. An alternative to steam is the use of a large volume of low-pressure air furnished by a blower. Local energy costs, availability, reliability, and weather conditions must be taken into account in selecting the smoke-suppression medium.

Purge and pilot gas must be supplied to the flare at all times. The amount of each gas required is related to the size of the flare system. The composition of the purge gas and/or the composition of the waste gas can also influence the purge gas requirement. Pilot gas consumption can be affected by the combustion characteristics of the waste gases. The gases used for purge gas and to fuel the flare pilots should come from the most reliable source available.

Purge gas can, in principle, be any noncorrosive gas that does not contain oxygen and does not go to dewpoint at any expected conditions. An attractive option for purge gas may be a mixture of nitrogen and a non-hydrogen-containing fuel gas such as natural gas or propane. For example, a 300 Btu/scf (11 MJ/Nm^3) mixture of nitrogen and propane can be effective as a purge medium. Such a mixture presents a number of benefits when compared to fuel gas alone, including

- Reduced CO_2 emissions
- Potential cost savings if nitrogen is less expensive
- Higher reliability because either supply alone can function as purge gas
- Reduced wear and tear on the flare burner

11.2.6 Safety Requirements

Almost every aspect of flare design involves some safety concerns. Safety concerns (see Volume 2, Chapter 1) include thermal radiation from the flare flame (see Volume 1, Chapter 8), reliable ignition, hydraulic capacity, air infiltration, and flue gas dispersion. Certain aspects of safety are dictated by the basic practices of the owner. For example, the owner's safety practices usually set the allowable radiation from the flare flame to people or equipment. Therefore, it is not surprising that the allowable thermal radiation level will vary from owner to owner.

A common point of variation involves the treatment of solar radiation relative to the allowable level. Experience has shown that solar radiation need not be considered in the majority of designs. In practice, consideration of solar radiation is a complex issue that does not lend itself to a simple solution. The solar radiation question involves a number of variables and is site specific. By way of example, it would be appropriate to include solar radiation in the design basis if there is substantial likelihood that a worker can become exposed to the maximum flare radiation and the sun's radiation in an additive manner.

There are several sources for guidance on the allowable radiation level. The most widely referenced is American Petroleum Institute (API) Recommended Practice (RP) 521.[5] Most specifications call for a maximum radiation level of 1500 Btu/h/ft^2 (4.73 kW/m^2) for emergency flaring conditions. Some specifications define an additional radiation level limit of 500 Btu/h/ft^2 (1.58 kW/m^2) for unprotected individuals during long-duration flaring events. Special consideration should be given to radiation limits for flares located close to potential public access areas along the plant boundary where public exposure could occur (see Section 11.3.5 for more detail on flame radiation).

Reliable ignition at the flare tip is one of the most fundamental safety requirements, ensuring that gases released to the flare are burned in a defined location.

Dependable burning also ensures destruction of potentially toxic releases. The prime objective (Section 11.1.2) demands reliable burning of the flare. The subject is covered in more detail in Sections 11.3.1 and 11.4.2.

Hydraulics of the flare system determines back pressure on relief valves. Improper initial system sizing or subsequent additions to the flare relief loads can prevent a unit from achieving its maximum relief rate when necessary and create an overpressure risk in the plant. Section 11.3.2 provides further discussion on the effect of relief valve selection on the design of the flare system.

Prevention of air infiltration should be a consideration when developing operations and maintenance plans for the flare system and connected equipment. Air sources include the flare tip exit, loop seals on vessels, low-point drains, high-point vents, and flanges. These issues are discussed in Section 11.3.4.

11.2.7 Environmental Requirements

Flares can affect their environment by generating smoke, noise, or combustion products. Regulatory agencies sometimes define limits in some or all of these areas. In many cases, it is necessary to inject an assist medium such as steam in order to achieve smokeless burning and to meet smoke emission regulations. The injection of the steam and the turbulence created by the mixing of steam, air, and gas cause the emission of sound. The sound level at various points inside and outside the plant boundary is often subject to regulation.

Other environmental concerns are the reaction efficiency and flue gas emissions. Pioneering tests conducted by the John Zink Company established that a properly designed and operated flare burner will have a combustion efficiency of more than 98%.[6] Emissions (see Volume 1, Chapters 14 and 15) of NOx, CO, and unburned hydrocarbons (UHCs) were also determined during these tests. NOx, CO, and UHC emission factors for flares are available in AP-42.[7] These emission factors are widely accepted by regulatory authorities as a basis for flare permit emission estimates. For emission estimates of SOx, it is often assumed that 100% of the available sulfur is converted to SO_2.

11.2.8 Social Requirements

Most emergency flare systems include a flare stack that is the tallest, or one of the tallest, structures in the plant. As a result, the flare flame is visible for great distances. Although the plant owner has complied with all environmental regulations, the flare system may not meet the expectations of the plant's neighbors. Public perception of the purpose and performance of the flare can place more stringent requirements on the flare design. For example, a smokeless flame may meet the regulatory requirements, but might be objectionable to the neighbors due to light or noise.

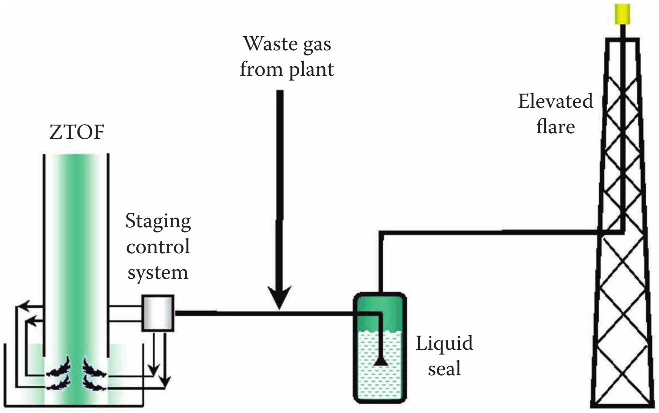


FIGURE 11.12
General arrangement of a staged flare system, including a ZTOF and an elevated flare.

The John Zink Company recognized the need to reduce elevated flaring more than 30 years ago and invented the world's first successful enclosed ground flare for the elimination of day-to-day visible flaring. Since that time, many plants have included a Zink Thermal Oxidizer Flare (ZTOF) in their flare system. The general arrangement of a flare system that incorporates a ZTOF for day-to-day flaring rates and an elevated flare for emergency rates is shown in Figure 11.12. The liquid seal in the system acts to divert flow to the ZTOF until it reaches its maximum capacity. Any additional flow will pass through the liquid seal and be burned at the elevated flare tip. An installation of such a system is shown in Figure 11.9.

11.3 Flare Design Considerations

Having received information on the system defining factors set forth earlier, the flare designer must now apply his/her expertise to the following design considerations. A given project may require inclusion of all or only a few of these points, depending on the nature of the system information disclosed and the scope of the project. The flare designer must consider how decisions relating to each factor will affect the entire flare system as well as all the other factors. The prime objective of safe, effective disposal of gases and liquids should be used as a guiding principle as each appropriate consideration is incorporated into the overall flare design. The design considerations are

1. Reliable burning
2. Hydraulics
3. Liquid removal

FIGURE 11.13
Flare test at the John Zink Co. test facility in Tulsa, Oklahoma.

4. Air infiltration
5. Flame radiation
6. Smoke suppression
7. Noise/visible flame
8. Air/gas mixtures

Successful selection and operation of flare equipment require a clear understanding of these design considerations. The success and cost-effectiveness of a flare design are dependent on the skill and experience of the flare expert and his/her access to the latest state-of-the-art design tools and equipment. A key development tool is the ability to conduct flare tests at the high flow rates experienced in real plant operations.[8] Facilities capable of conducting tests of this magnitude (Figure 11.13) represent a substantial capital investment and take on the characteristics of a complex process plant flare system. Insight into each consideration is set forth in the following.

11.3.1 Reliable Burning

Venting of waste gases can happen anytime during plant operation. Therefore, an integrated ignition system (see Chapter 12) is required that can immediately initiate *and* maintain stable burning throughout the period of waste gas flow. Stable burning must be ensured at all flow conditions. An integrated ignition system includes one or more pilot(s), a pilot ignitor, pilot monitor(s), and a means to stabilize the flame.

In principle, all flares should have a continuous pilot flame to ensure reliable burning. This is especially true of refinery, petrochemical, and production field flares because they cannot be shut down unless and until the entire plant shuts down. In addition, such flares may be online for weeks or months before an unpredictable event that creates an immediate need for reliable ignition. Notable exceptions are landfill flares or biogas flares (see Chapter 13) that operate continuously at substantial rates and include flame monitoring systems that automatically shut off waste gas flow in case of flame failure. Discontinuous pilots should only be considered in cases where *all* of the following conditions apply:

- The main flame remains lit and stable without a pilot at all design conditions.
- The main flame is monitored.
- The flare is shut down automatically on main flame failure.
- The flare shutdown does not create a safety hazard in the plant.

The number of pilots required can vary, depending on the size and type of the flare burner and its intended use. Flare pilots are usually premixed burners designed such that pilot gas and air are mixed together at a point remote from the flare burner exit and delivered through a pipe to the pilot tip for combustion. This ensures that the pilot flame is not affected by conditions at the flare burner exit (e.g., the presence of flue gas, inert gas, or steam). Pilot gas consumption varies according to the specific flaring requirements. However, there is a practical lower limit to the pilot gas consumption.

A pilot monitor is often required to verify the pilot flame. As a safety consideration, pilot ignition is usually initiated from a position remote from the flare stack. Either a flame-front generator (FFG) or direct spark pilot ignition can be used, depending on the system requirements. Further discussion of this important safety aspect is provided in Chapter 12.

There is a complex relationship between flare tip exit velocity, gas composition, tip design, and the maintenance of stable burning. There are a number of advantages in using the highest exit velocity possible, including minimum equipment size and optimal flame shape. In addition, because high discharge velocity tends to improve air mixing with a resultant reduction in soot formation, one can see that maximizing discharge velocity can help improve smokeless performance. It is important to note that discharge velocity can be constrained by the gas pressure available or concerns about flame stability. In some circumstances, such as VOC control, the discharge velocity may be limited by regulation. Early research on flare system design suggested limiting discharge velocity to 0.2 Mach due to stability concerns. It was later suggested that a discharge velocity of 0.5 Mach or higher could be used if proper flame stabilization techniques were employed. Flame stabilization techniques have been successfully employed for exit velocities of Mach 1 or greater.

Waste gas composition can significantly affect the allowable exit velocity. For example, a properly designed flare burner can maintain stable burning of propane at Mach 1 or greater. On the other hand, if the propane is mixed with a large quantity of inert gas, the maximum exit velocity must be limited to a much lower Mach number in order to ensure stable burning.

11.3.2 Hydraulics

Most flare systems consist of multiple relief valves discharging into a common flare manifold or header system. A key item influencing the flare system design is the allowable relief valve back pressure. The system pressure drop from each relief valve discharge through the flare tip must not exceed the allowable relief valve back pressure for all system flow conditions. The allowable back pressure is typically limited to about 10% of the minimum relief valve upstream set pressure for conventional relief valves. The allowable relief valve back pressure can be increased by the use of balanced pressure relief valves. Balanced valves can accept a back pressure of about 30% of upstream set pressure in most cases. Where there is a wide variation in the allowable relief valve back pressures, it may be economical to use separate high- and low-pressure flare headers.

Increasing the allowable relief valve back pressure can have several effects on the flare system components, including

- Smaller manifold and header piping
- Smaller knockout and liquid seal drums
- Smaller flare size, giving lower purge rates and enhanced operating life
- Significant reduction or elimination of utilities required for smokeless burning through the utilization of increased pressure energy at the flare tip

As mentioned in Section 11.2.4, each major flowing condition should be analyzed to verify that no relief source is overpressured. In some applications, a large number of different flowing conditions can occur. To simplify the process of identifying those cases that are likely to govern the hydraulics, a comparative measure of flow rates is useful. The volumetric equivalent, or V_{eq}, is one measure used to identify the hydraulically controlling case:

$$V_{eq} = Q\sqrt{\frac{\mathrm{MW}}{29} \times \frac{T}{520}} \qquad (11.1)$$

where

V_{eq} is the volumetric equivalent, SCFH

Q is the waste gas flow, SCFH

MW is the waste gas molecular weight

T is the waste gas temperature, °R

V_{eq} is the volumetric flow of air that would produce the same velocity pressure as the waste gas flow in the same size line

While this method gives general guidance, it should not replace a more thorough hydraulic analysis.

Properly utilized, a higher allowable pressure drop for the flare system provides an opportunity for capital cost savings, operating cost savings, and reduced downtime due to longer equipment life. While the capital cost savings are most apparent on entirely new flare systems, all of these savings can be realized on existing systems as well.

11.3.3 Liquid Removal

Inherent in many flare systems is the potential for either liquid introduction into or the formation of hydrocarbon or water vapor condensate in the flare header. Allowing this liquid-phase material to reach the combustion zone may make operation more difficult. For example, hydrocarbon droplets small enough to be entrained by waste gas and carried into the flame usually burn incompletely, forming soot, and, as a result, reduce the smokeless capacity of the flare. If the droplets become larger, they may be able to fall out of the main flame envelope. In addition, events have been reported where a mostly liquid stream has been discharged from the flare.

Figure 11.14 shows an offshore flare that received liquids in the manner described earlier. The last two

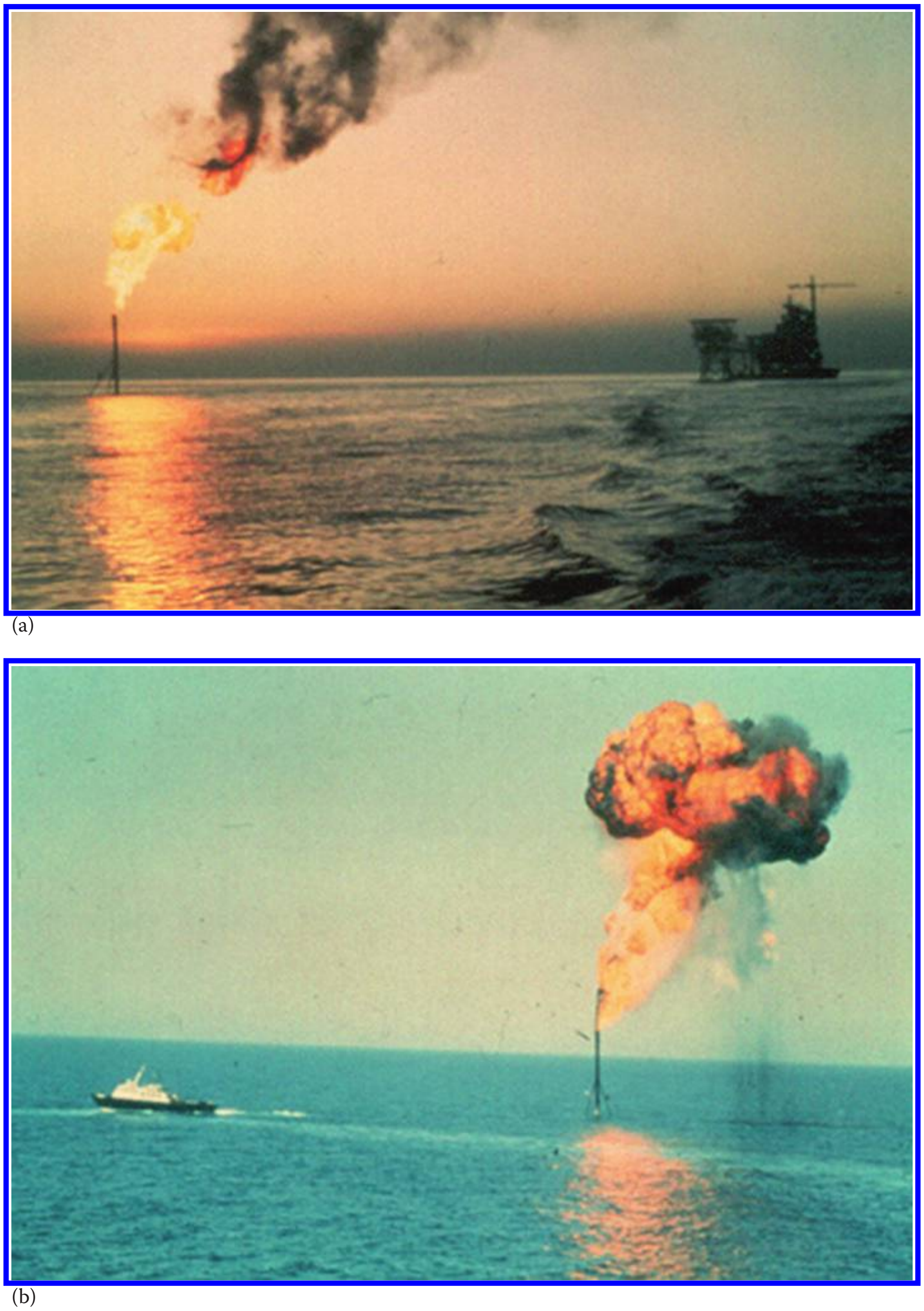


FIGURE 11.14
Liquid carryover from an elevated flare: (a) start of flaring event, (b) liquid fallout and flaming rain from flare flame.

(*continued*)


(c)

FIGURE 11.14 (continued)
Liquid carryover from an elevated flare. (c) Flaming liquid engulfs flare stack.

photos, (b) and (c), were taken only a few minutes apart and illustrate how rapidly this situation can deteriorate.

Incorporation of a properly designed and operated knockout drum into the flare system can minimize these problems. There are three basic types of knockout drums that can be incorporated into a flare system: a horizontal settling drum, a vertical settling drum, and a cyclone separator. For more information on each of these types of knockout drums, refer to Section 11.4.3.

Regardless of the knockout drum concept, the holding capacity of the drum should be carefully considered. An overfilled knockout drum can obstruct gas flow to the flare, resulting in overpressure to upstream systems. In the extreme case, an overfilled knockout drum can result in blowing large volumes of liquids up the flare stack. Liquid draw-off capacity must be adequate to prevent overfilling of the drum. In addition, a backup pump and drive means should be considered. Liquid recovered from the knockout drum must be carefully disposed or stored. Flare header piping must be sloped properly to prevent low-point pockets where liquids can accumulate.

11.3.4 Air Infiltration

Infiltration of air into a flare system can lead to flame burnback, which in turn could initiate a destructive detonation in the system. Often, burnback can only be observed at night. Air can enter the flare system by one or more of the following scenarios:

- Through stack exit by buoyant exchange, wind action, or contraction
- Through leaks in piping connections
- As a component of the waste gas

Prevention measures are available to address each of the air infiltration mechanisms.

Purge gas is often injected into the flare system to prevent air ingression through the stack exit. The quantity of purge gas required is dependent on the size and design of the flare, the composition of the purge gas, and the composition of any waste gas that could be present in the system following a vent or relief event. In general, the lower the density of the gas in the flare stack, the greater the quantity of purge gas necessary for the safety of the system. The purge gas requirement can be reduced using a conservation device such as a John Zink AIRRESTOR or Molecular Seal (see Section 11.4.4). The cost and availability of the purge gas will guide the choice of such a device.

Contraction of gas in the flare system occurs due to the cooldown following the flaring of hot gases. The rate of contraction will accelerate dramatically if the cooling leads to condensation of components of the contained gas. Contraction risk can be minimized by use of the Tempurge system.[9] Tempurge senses conditions in the flare header and initiates the introduction of extra purge gas to offset contraction.

An elevated flare stack filled with lighter than air gas will have a negative pressure at the base created by the difference in density between the stack gas and the ambient air. The gas density in the stack is related to the MW of the gas and its temperature. Equation 11.2 defines the pressure at the base of a flare stack at very low flow conditions such as purge:

$$P_{\text{base}} = \frac{27.7H\left(\rho_{\text{gas}} - \rho_{\text{amb}}\right)}{144} \tag{11.2}$$

where

P_{base} is the static pressure at base of stack, in W.C.
H is the height of stack above inlet, ft
ρ_{gas} is the gas density in stack, lb-m/ft^3
ρ_{amb} is the density of atmospheric air, lb-m/ft^3

If a negative static pressure exists at the base of the stack, then at low flows the entire flare header system will be under negative pressure. Operation of the flare system under negative pressure greatly increases the potential of air infiltration into the header system through leaks, open valves, or flanges or through the tip exit by decanting in the stack. Such leakage is known to have occurred during the servicing of relief valves.

Installation of a liquid seal in the system can produce positive flare header pressure although the pressure downstream of the seal is negative. This greatly reduces the potential of air leakage into the system. Because a liquid seal can also be a barrier to air entering the header from the flare stack, locating the liquid seal in the base of the stack offers maximum protection of the header system. In this position, the liquid seal can also be designed to isolate the flare ignition source from the flare header and the process units.

Oxygen-containing gases should be segregated from the main flare system. Waste gases that contain oxygen present a special design challenge. The risk of flashback in systems handling such gases can be minimized through the use of flame/detonation arrestors, special liquid seals, and/or the use of specialized flare burners. The presence of more than a trace amount of oxygen (more than 1% by volume) in a waste gas stream creates a separate design consideration discussed in Section 11.3.8.

11.3.5 Flame Radiation

As the waste gases are burned, a certain portion of the heat produced is transferred to the surroundings by thermal radiation (see Volume 1, Chapter 8). Safe design of a flare requires careful consideration of the thermal radiation. The radiation limits discussed in Section 11.2.6 can become the basis for determining the height of the flare stack and its location. For a given set of flare flow conditions, the radiation limits can usually be met by adjustment of the flare stack overall height and/or the use of a limited access area around the flare. The flare height and/or size of the limited access area can affect the economics of the plant. For plants with limited plot area (or for ships), an enclosed flare can be employed to meet radiation restrictions. Water spray curtains have also been used to control radiation on offshore platforms.

Radiation from a flame to another object is determined by

- Flame temperature
- Concentrations of radiant emitters in the flame (e.g., CO_2, H_2O, and soot)
- Size, shape, and position of the flame
- Location and orientation of the target object relative to the flame
- Characteristics of the intervening space between the flame and the object

Calculations based on theory may be feasible within the well-defined confines of a furnace operating at a steady condition. Unfortunately, most of these factors cannot be accurately defined for a flare flame in the open air.

Because temperature appears in the radiation equation to the fourth power, it is clearly a dominant factor. Despite its importance, the temperature of a flare flame is extremely difficult to measure or estimate. An error of only 10% in absolute temperature affects the calculated radiant heat transfer by over 40%. Observers have noted variations in local flame temperature as high as 1000°C (1800°F) between the core and the cooler outer surface of an open burning flame. Figure 11.15 shows a thermogram of a flare flame. In the thermogram, white represents the highest temperature and dark blue the lowest. Thus, only the small, bright yellow zone is at a high temperature. The temperature falls rapidly as one approaches the outer edge of the flame envelope.

The temperature of a flame is influenced by its interaction with its surroundings. The availability of ambient air causes the outer portions of the flame envelope to cool. In addition, the flame will radiate both to cold outer space and to relatively warmer objects on Earth. Therefore, it is not surprising that observations indicate peak flare flame temperatures far less than the calculated adiabatic flame temperature. To approach flare radiation from a theoretical basis, local flame temperature, which varies substantially throughout the flare flame, would need to be predicted with greater accuracy than present tools allow.

The other factors listed are also very difficult to determine. The concentrations of substances that are radiant emitters vary a great deal from point to point within a flare flame, which creates problems in predicting

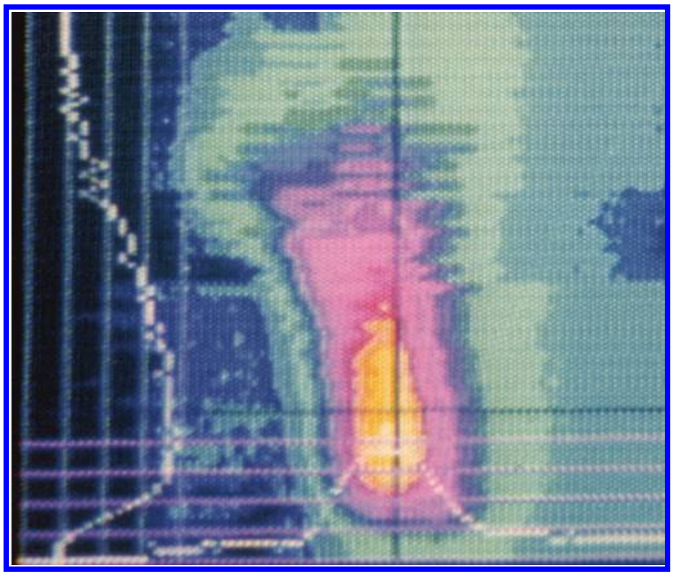


FIGURE 11.15
Thermogram of a flare flame.

beam lengths and emissivity. The detailed shape of a flare flame is much more convoluted and chaotic than any simple geometric approximation can represent. Wind fluctuations cause the flame to move constantly, so concentrations, temperature, and relative positions are always changing. Atmospheric absorption and scattering depend on transient and unpredictable weather conditions such as ambient temperature, humidity, fog, and rain.

To overcome these difficulties, engineers have historically estimated radiation by treating flare flames as point sources, using heat release as a basis for emissive power and empirical radiant fractions in lieu of true radiant emissivity. Figure 11.16 shows the general geometry assumptions that affect the point source approach. The classic API radiation equation represents this approach in its simplest form:

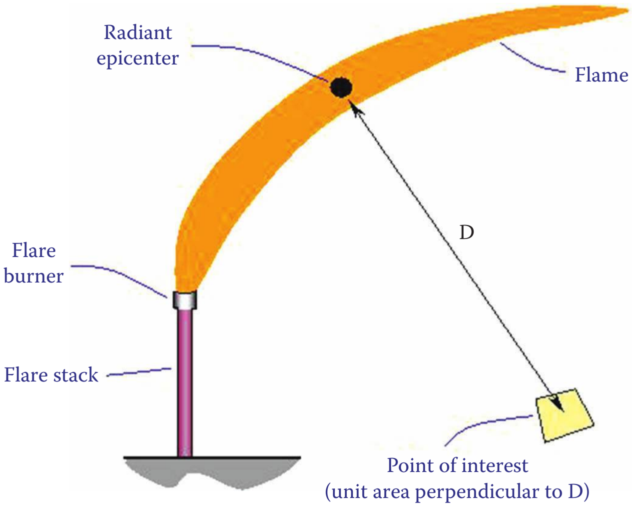


FIGURE 11.16
API radiation geometry.

$$K = \frac{\tau F Q}{4\pi D^2} \tag{11.3}$$

where

K is the radiation, Btu/h-ft^2
τ is the atmospheric transmissivity
F is the radiant fraction
Q is the heat release, Btu/h
D is the distance from the heat epicenter to the object, ft

Many of the complexities of the full theoretical treatment are lumped into the empirically determined radiant fraction. This factor includes flame temperature effects, gas and soot emissivity, mean beam length, and other flame shape issues. The distance factor disguises a number of subtleties that arise as a result of flame shape prediction, including flame length, flame trajectory, and position of the heat epicenter. Nevertheless, this type of simplified approach has been used in one form or another to estimate radiation from flare flames for many years.

Several published methods are available for preliminary estimation of flare radiation and stack heights. An article by Schwartz and White[10] presents a detailed discussion of flare radiation prediction and a critical review of published methods. Based on Example 2 in the referenced paper, Figure 11.17 provides a visual comparison of the stack heights determined by each of several radiation methods and the relative equipment cost associated with each stack height. Plant designers and users alike must be cognizant that traditional methods of calculating radiant heat intensities are neither consistently conservative nor consistently optimistic. Long ago, the John Zink Company recognized the limitations and risks associated with the traditional methods and undertook the development of proprietary methods for radiation prediction. The latest prediction methods capture the effect of flare burner design, gas quantity and composition, various momenta, smokeless burning rate and smoke formation on the flame shape, and radiant characteristics.

11.3.6 Smoke Suppression

Smokeless burning is a complex issue that involves many of the system defining factors discussed in Section 11.2. In choosing the best smoke-suppression method, the flare designer is guided by his/her experience in interpreting the job-specific information received relative to each of these factors.

Smokeless burning, in general, occurs when the momentum produced by all of the employed energy sources educts and mixes sufficient air with the waste gas. For smokeless burning, a key issue is the momentum of the waste gas as it exits the flare burner. In some cases, the waste gas stream

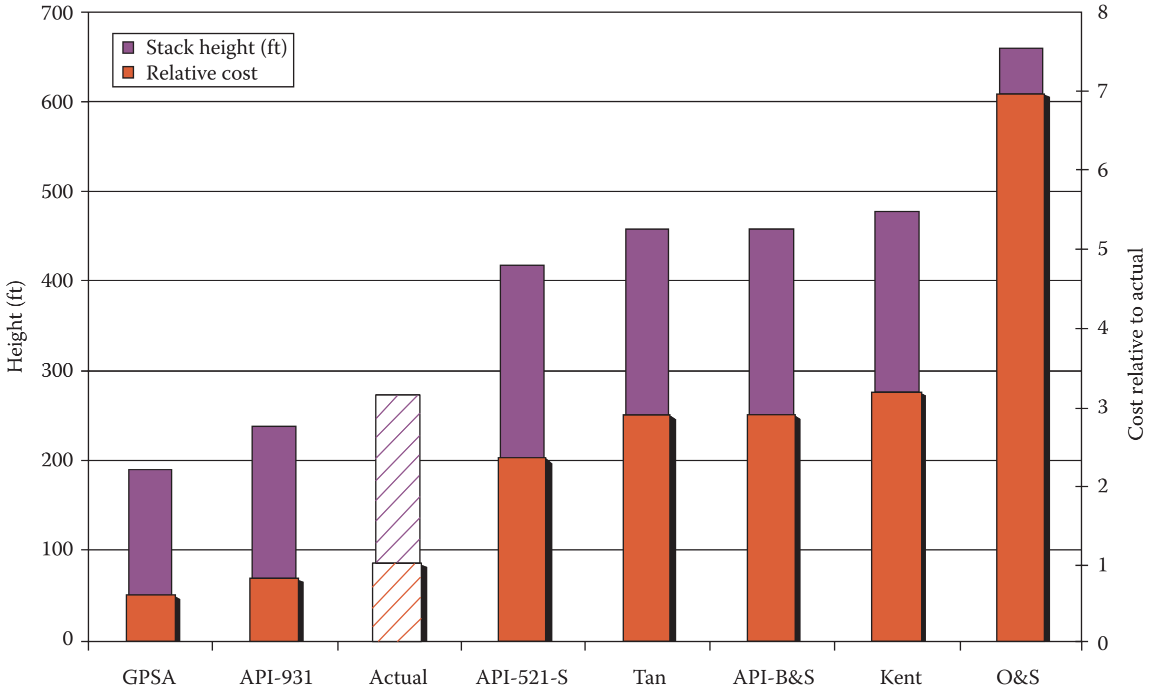


FIGURE 11.17
Comparison of stack height and relative cost for various radiation calculation methods.

is available at a pressure that, if properly transformed, can provide the required momentum. If the waste gas pressure (momentum) is not adequate for smokeless burning, the flare designer must enlist assistance from another energy source (e.g., steam or low-pressure air). In some cases, a combination of energy sources can be effective.

Briefly, energy transformation entails conversion of the internal energy (pressure) of the waste gas to kinetic energy (velocity). Designs for high-pressure flares (5–10 psig [0.3–0.7 barg] or more) exist that require no supplemental assist medium. Systems employing this technique have been very successful and enjoy low operating cost and an excellent service life.

Steam injection is the most common technique for adding momentum to low-pressure gases. In addition to adding momentum, steam also provides the smoke-suppression benefits of gas dilution and participation in the chemistry of the combustion process. The effectiveness of steam is demonstrated in the series of photographs shown in Figure 11.18. In frame (a), there is no steam injection; in frame (b), steam injection has just begun; and in frame (c), steam injection has achieved smokeless burning.

Some plants have steam available at several different pressure levels. There is often an operating cost advantage to using low-pressure steam (30–50 psig = 2–3.4 barg). The plant designer must balance this operating cost advantage against the increased piping costs associated with low-pressure steam. Also, while the flare may achieve the design smokeless rate at the maximum steam pressure, steam consumption at turndown firing rates below the maximum may be higher than expected. Because most flaring events involve relatively low flow rates, performance under these turndown conditions must be carefully considered.

Low-pressure (0.25–1.0 psig = 1.7–6.9 kPa) air is utilized in cases where gas pressure is low and steam is not available. The supplied air adds momentum and is a portion of the combustion air required. Figure 11.19 shows another series of photographs that illustrates the effectiveness of air assistance. Frame (a) shows the flare with no assist air. The blower is turned on in frame (b), but because the blower requires some time to reach full speed, the complete effect of air injection is not seen until frame (d). The gas being flared in this case is propylene.

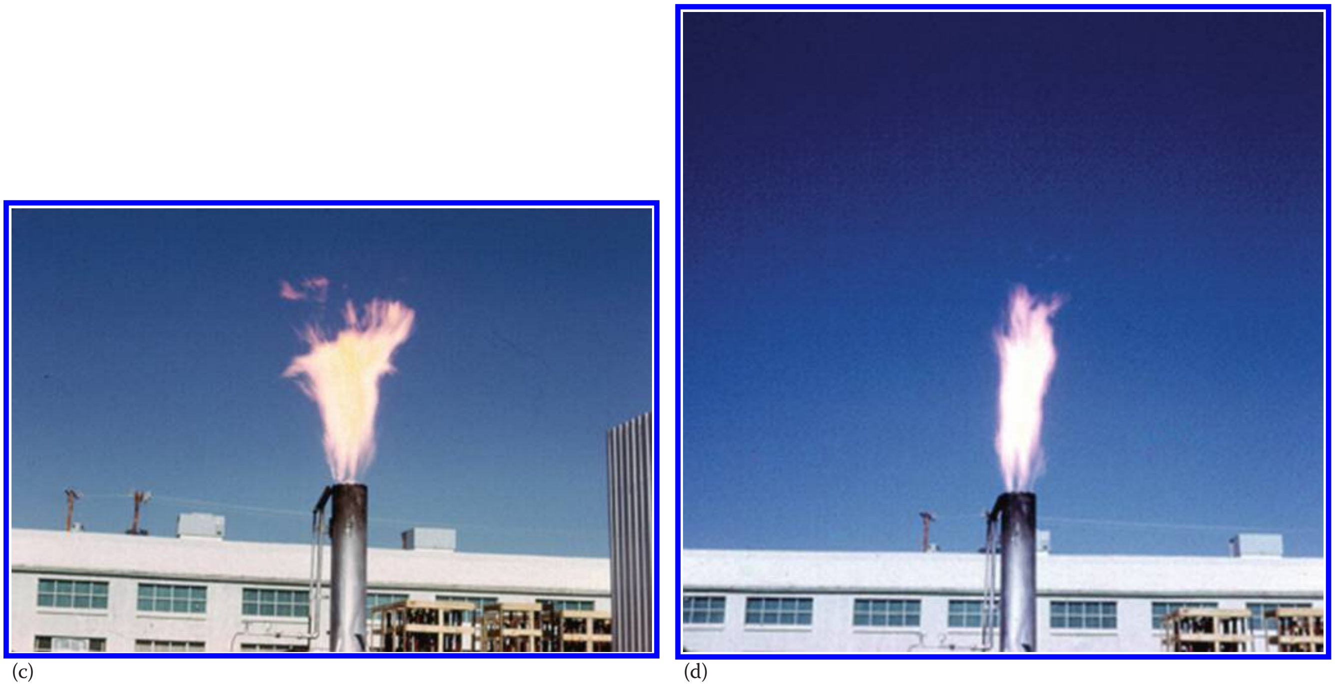
Generally, the blower supplies only a fraction of the combustion air required by the smokeless flow condition. For most designs, 15%–50% of the stoichiometric air requirement is blown into the flame by the blower. The remainder of the required air is entrained along the length of the flare flame.

11.3.7 Noise/Visible Flame

The energy released in flare combustion produces heat, ionized gas, light, and sound. Most plants are equipped with elevated flares that by their nature broadcast flaring sound to the plant and to the surrounding neighborhood. In some cases, the sound level becomes objectionable and

FIGURE 11.18
Effectiveness of steam in smoke suppression: (a) no steam, (b) starting steam, and (c) full steam flow.

FIGURE 11.19
Effectiveness of air in smoke suppression: (a) no blower air and (b) start blower.



(c) (d)

FIGURE 11.19 (continued)
Effectiveness of air in smoke suppression: (c) air flow increasing and (d) full air flow.

is considered to constitute noise (see Volume 1, Chapter 16). Flaring sound is generated by at least three mechanisms:

1. By the gas jet as it exits the flare burner and mixes with surrounding air
2. By any smoke-suppressant injection and associated mixing
3. By combustion

Upstream piping and valves associated with the source of the relief gas may also create substantial noise levels that are carried along the flare header and exit through the flare tip.

At the maximum smokeless flaring rate of a steam- or air-assisted flare burner, gas jet noise is usually a minor contributor. The noise generated by the second mechanism can be mitigated by use of low-noise injectors, mufflers, and careful distribution of suppressant. The Steamizer™ flare burner shown in Figure 11.20 is of low-noise design with additional noise reduction coming from a muffler concept first developed for use on enclosed ground flares. Careful design can reduce flaring noise levels by a factor of 75% or more (6 dB or more).

Where the light from a flare flame is objectionable, an enclosed flare is a good selection. A properly sized enclosed flare can eliminate a visible flame for all cases except emergencies. An equal benefit of an enclosed flare is the reduction of flaring noise.

FIGURE 11.20
Steamizer™ steam-assisted smokeless flare.

11.3.8 Air/Gas Mixtures

Waste gas streams containing air/gas mixtures can generally be divided into two types. The first type is comprised of systems that are expected to contain air/gas mixtures. Examples include landfills (see Chapter 13), gasoline loading terminals (see Chapter 15), and medical equipment sterilization facilities. The second type is potentially more dangerous in that air is not expected in the composition of vents and reliefs. An example is venting air from a vessel or tank at the beginning of a prestart-up purging cycle.

Flare systems that handle air/gas mixtures usually involve a number of special safety considerations. The special considerations, which relate primarily to the increased risk of flashback, include

1. Safety interlocks to prove purge gas and pilots on start-up
2. Automatic shutdown on loss of purge gas or pilots
3. Higher than normal purge rates to maintain burner exit velocity and prevent burnback
4. Limited turndown range to maintain burner exit velocity and prevent burnback
5. Use of detonation and/or flame arrestors
6. Special operational practices

11.4 Flare Equipment

While evaluating the general design considerations set forth earlier, the flare designer must also begin equipment selection. Overall design considerations and specific equipment selections are interrelated aspects of the system design process. The various major system components listed in Section 11.1.5 are discussed here in more detail.

11.4.1 Flare Burners

Although they are installed at the end of the flare system, flare burners are among the first items considered in the system design. At this point, it should be clear that substantial benefits are attached to the use of most of the available pressure at the flare burner. The exit of the flare burner is the point where the flow determining pressure drop usually occurs. Designs range from simple utility flares to enclosed multipoint staged systems and from nonassisted to multiple steam injectors to multiple blower air-assisted designs. Regardless of the tip design, adequate ignition means must be available to ensure that the prime objective is achieved.

11.4.1.1 Nonassisted or Utility

The simplest types are the nonassisted, or utility, flare burners. These burners consist essentially of a cylindrical barrel with attachments for enhanced flame stability (flame retention means) and pilots to initiate and maintain ignition of the relief gases. A typical nonassisted flare is shown in Figure 11.21. Utility flares are usually flanged for ease of replacement. The horizontally mounted versions are usually referred to as burn pit flare tips (Figure 11.5). In both cases, a turbulent diffusion flame is produced. The flame may be an attached or detached stable flame. The exit velocity of a flare burner is dependent on the waste gas composition, the specific design of the flare burner, and the allowable pressure drop. In some cases, the exit velocity can safely reach Mach 1. It should be noted that some flares or flare relief cases may be subject to regulations that limit the exit velocity.

Optional features that can extend equipment service life include windshields and refractory lining. The Zink double refractory (ZDR) severe service flare tips (Figure 11.22)

FIGURE 11.21
Typical nonassisted flare.

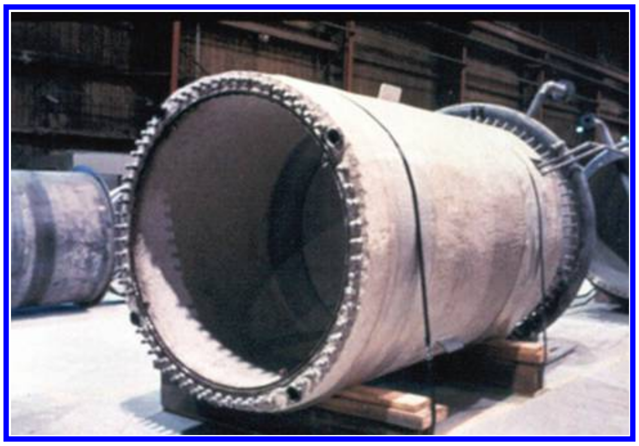


FIGURE 11.22
ZDR severe service flare tip.

FIGURE 11.23
Simple steam-assisted flare.

use refractory linings internally and externally to protect the tip against both internal burning and flame pulldown outside the tip. Alternatively, center steam is sometimes used to help avoid internal burning instead of an internal refractory lining to extend tip life. This approach is most effective in climates where freezing is not an issue. Center steam is a relatively inefficient means to control smoke because it does not entrain air, which is normally an essential part of any smoke-suppression strategy.

11.4.1.2 Simple Steam Assisted

The first smokeless flares were adaptations of simple utility flares. This basic design has been improved over the years with multiport nozzles to reduce steam injection noise, optimized injection patterns to improve steam efficiency, and optional center steam injection to reduce damaging internal burnback. Figure 11.23 shows a modern example of this design. A steam manifold, often referred to as the upper steam manifold or ring, is mounted near the exit of the flare tip. The steam ring can be designed for steam supply pressures normally ranging from 30 to 150 psig (2 to 10 barg). Several steam injectors extend from the manifold and direct jets of steam into the waste gas as it exits the flare tip. The steam jets inspirate air from the surrounding atmosphere and inject it into the gas with high levels of turbulence. These jets also act to gather, contain, and guide the gases exiting the flare tip. This prevents wind from causing flame pulldown around the flare tip. Injected steam, educted air, and relief gas combine to form a mixture that burns relief gas without smoke. The maximum smokeless rate achievable with a given flare burner depends on a number of factors, including the gas composition, the amount of steam available, and the gas and steam pressures.

However, there are inherent design limitations in this type of flare tip. The steam injectors are located close to the exit of the flare tip, so it becomes very difficult to muffle the steam noise produced by the high-pressure jets. Any muffler for the upper steam noise would need to be able to withstand direct flame impingement in adverse winds, and it could tend to interfere with air being drawn in by the steam jets. Furthermore, as the tip size increases, it becomes more difficult for the steam/air mixtures to reach the center of the flame. Finally, the perimeter of the flare tip only increases linearly with tip size, while the flow area of the flare tip increases with the square of tip size, as shown in Figure 11.24. Therefore,

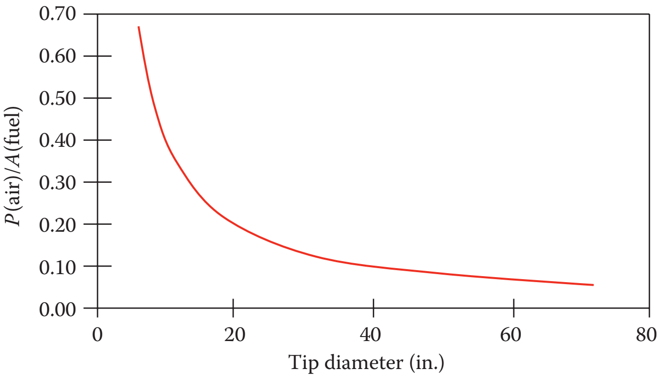


FIGURE 11.24
Perimeter/area ratio as a function of tip size for a simple steam-assisted flare.

as flare tip size increases, the need for air (a function of the flow area) quickly outstrips the ability to educt air (a function of the perimeter.) This fundamental characteristic of a simple steam-assisted flare limits the maximum effective size of such a flare burner.

11.4.1.3 Improved Steam Assisted

To overcome the limitations and other shortcomings of the simple steam-injected design, an improved steam-assisted flare was invented that uses multiple steam injection points.[11] In addition to the upper steam manifold, a set of external–internal tubes is utilized to deliver a steam/air mixture to the core of the waste gas exit (Figure 11.25). The presence of these tubes, properly distributed across the tip exit, increases the effective perimeter available for air access to the waste gas (Figure 11.26).

The external–internal tubes start outside the wall or barrel of the flare tip, pass through openings in the wall, and turn upward, terminating at the tip exit. Welding seals the point where the tube penetrates the wall. Steam jets inject steam into the inlet of these tubes, inspirating large amounts of air. The steam/air mixture exits the tubes at high velocity, delivering momentum, dilution steam, combustion air, and turbulence into the base of the flame. Innovations have increased the effectiveness of the tubes allowing greater smokeless capacities. Enhanced steam injectors have increased the air eduction efficiency. Upper steam injection further enhances smokeless combustion through increased turbulence and mixing and by mitigating adverse wind effects.

The improved steam-assisted flare design incorporates several smoke-suppression strategies: increased perimeter, higher momentum, more combustion air, greater turbulence for mixing, dilution and chemical interaction by steam, and molding of the flame to resist wind effects. Each of these strategies helps to reduce smoke; in combination, they produce some of the highest smokeless rates available in single-point flares.

New flare systems can achieve smokeless rates of more than 500,000 lb/h (230,000 kg/h) of gases that are generally considered difficult to burn cleanly. An example of an improved flare burner design is shown in Figure 11.27. Improved muffler designs and redistribution of steam can give noise levels much lower than earlier models. In some cases, the steam jet noise can be totally neutralized through injector design and the use of muffling techniques.

As the capacity, size, and smokeless capability of flare burners have increased, more emphasis has been placed on sophisticated design tools that can predict noise and smokeless performance. Design tools of this type can be developed and validated using large-scale tests in a facility such as shown in Figure 11.13.

11.4.1.4 Advanced Steam Assisted

Improved understanding of combustion mechanics has led to the development of an advanced steam-assisted tip design. This design, as shown in Figure 11.28, breaks up the gas discharge into multiple smaller discharges.

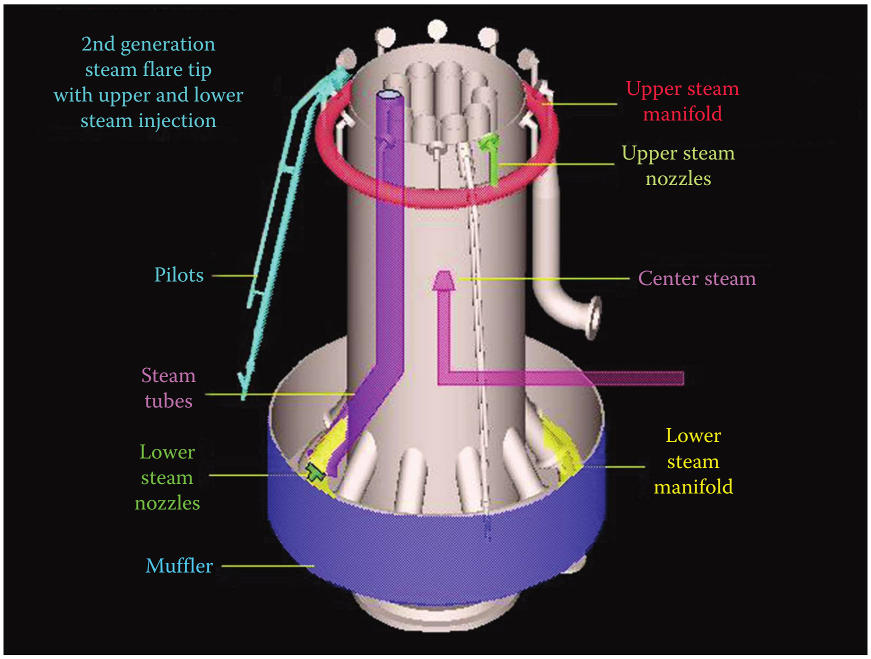


FIGURE 11.25
Schematic of an advanced steam-assisted flare.

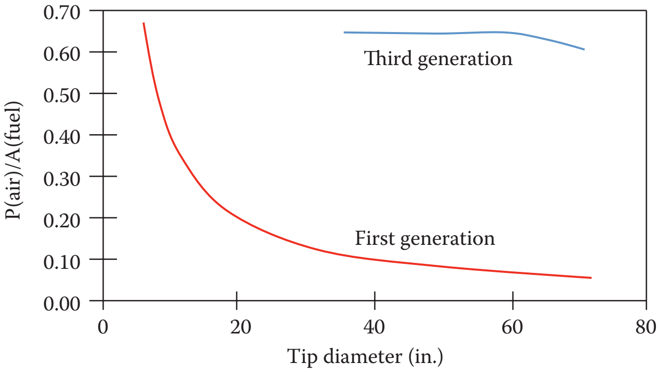


FIGURE 11.26
A comparison of the perimeter/area ratio for simple and improved steam-assisted flares.

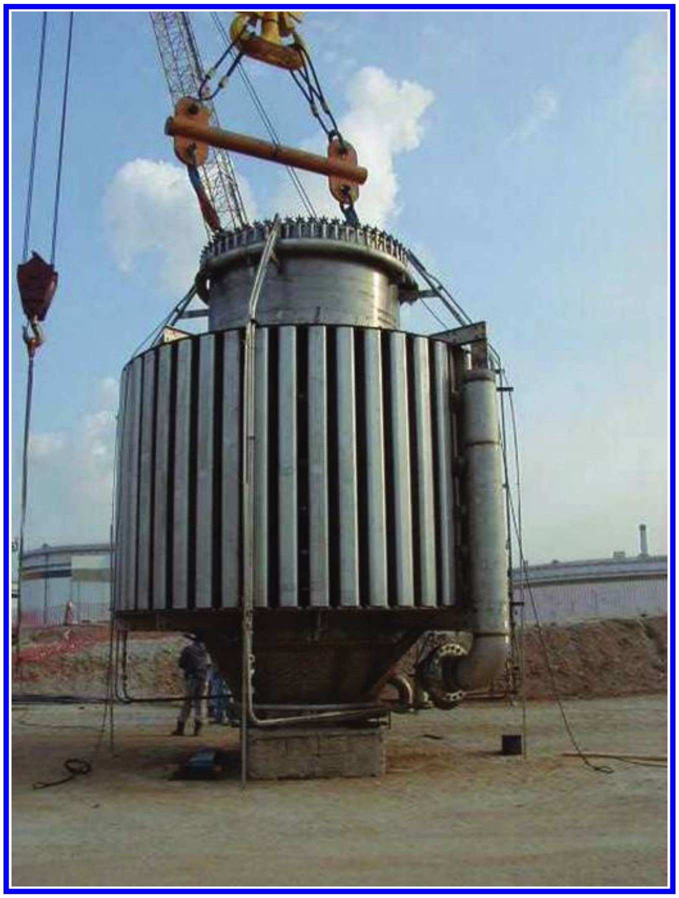


FIGURE 11.27
Steamizer® flare burner and muffler.

The smaller the discharge, the greater the perimeter to area ratio and thus the quicker air can penetrate the interior of the gas bundle promoting smokeless combustion. Each discharge has an associated external–internal tube, similar to those present in the improved steam-assisted design. The difference between the external–internal tubes of the advanced design from those of the improved design is that the advanced design tubes are straight. Straight tubes are more efficient educting air than bent tubes since there are no turning losses. Higher eduction efficiency allows for more air to be drawn in with the same amount of steam or the same amount of air with less steam. Another feature of the advanced design is the inclusion of a premix zone where the steam/air mixture from the external–internal tube is mixed with the vent gas prior to discharge and ignition. Premixing of air with a vent gas prior to combustion greatly improves the smokeless performance. The advanced steam-assisted flare design can provide the same smokeless performance as an improved design while only requiring 70% of the steam.

FIGURE 11.28
State-of-the-art Steamizer® XP™.

Operation of the advanced design is much simpler than that of the improved design. While an improved design typically has three different steam lines and careful coordination is required between the three steam flows, the advanced design uses a single steam line. The advanced design brings together the best aspects of the simple design (ease of operation) with the best aspects of the improved design (improved smokeless performance) while using less steam.

11.4.1.5 Low-Pressure Air Assisted

Not all plants have large amounts of steam available for use by the flare. Some plants prefer not to use steam to avoid freezing problems, others cannot commit water to make steam for smoke control, and still others choose

FIGURE 11.29
Air-assisted smokeless flare with two blowers in a refinery.

not to install a boiler. To meet this need, a series of air-assisted flare designs was invented.[12]

Generally, the air-assisted flare burner consists of a gas burner mounted in an air plenum at the top of the flare stack (Figure 11.29). Relief gas is delivered to the burner by a gas riser pipe running coaxially up the center of the flare stack. Low-pressure air is delivered to the burner from one or more blowers located near the base of the flare stack. The air flows upward through the annular space between the flare stack and the gas riser.

The first air-assisted flare applications were associated with operations some distance from the main plant or totally remote from plant utilities support. Early air flares were often designed to flare small to moderate flow rates. The success of these flares led to the use of air assist on flares of greater capacity. More recently, air-assisted flares have come into use as the flare for large process facilities. The flame similarity method and the related near-field mixing region models are examples of the design tools necessary for cost-effective application of air-assisted flares. Today, air flare designs are available with demonstrated tip life spans of 5–10 years. Smokeless rates above 150 × 10^6 standard cubic feet per day (SCFD) (4.2 × 10^6 standard cubic meters per day) are available for saturated hydrocarbons such as production facility reliefs.

FIGURE 11.30
Annular air flare.

Figure 11.30 shows an example of a more recent air flare design. Waste gas exits the burner in one or more narrow annular jets, each surrounded by assist air. This design makes good use of the perimeter/area ratio concept discussed earlier in the context of steam-assisted flares.

11.4.1.6 Energy Conversion

In the smokeless flaring discussions earlier, the focus centered on adding energy from an outside source to boost the overall energy level high enough to achieve smokeless burning. An advantage is gained if an outside source is not required. This is the case with energy conversion flare burners. Such burners are also referred to as high-pressure flare burners or multipoint flare burners. Where they can be employed, the use of energy conversion flare burners can provide a significant reduction in flare operating cost.

There are two distinct groups of energy conversion burners. The first group is distinguished by having a single inlet and relatively close grouping of the waste gas discharge points. The other group employs a means beyond energy conversion to achieve smokeless flaring. In both groups, an underlying principle is the conversion of the static pressure of the waste gas, at the burner, to jet velocity and ultimately into momentum.

Another concept employed by both groups is the division of the incoming gas stream into multiple burning points. This concept is sometimes referred to as the "firewood principle" because of an illustrative analogy. The "firewood principle" considers the situation where a tree has been cut down and is to be used as fireplace fuel. If a large section of the tree trunk is placed in the fireplace, it will be difficult to ignite and burn because the ratio of fuel to surface (air exposure) is very large. This situation can be improved by splitting the trunk section into smaller pieces, thus gaining a better fuel/surface ratio. Obviously, a balance must be reached between the effort expended to split the wood (think of it as cost) and the improved ability to burn (think of this as smokeless burning ability).

Applications of flare burners that employ energy conversion to achieve smokeless burning require special attention to flow turndown cases. Some flares, usually those associated with oil production activities, operate at or near their maximum design flow most of the time. Energy conversion flares for production applications, such as the commercially available Hydra flare shown in Figure 11.31, offer a controlled flame shape and reduced flame radiation. These features are particularly attractive for platform-mounted flares.

FIGURE 11.31
Hydra flare burner in an offshore location.

The opposite is normally true of flare burners serving refineries or petrochemical plants. These flares usually have a very small load (purge gas plus any leakage) with an occasional short-duration intermediate load. In energy conversion flares, the gas pressure vs. flow characteristic follows the same rules as an orifice. Above critical pressure drop, the flow/pressure characteristic varies with the ratio of the absolute pressures. At or below the critical pressure drop, the flow/pressure characteristic follows a square relationship. For example, a flow reduction of 50% will reduce the gas pressure drop to 25% of the full flow value.

The turndown ratios experienced by refinery and petrochemical flares would reduce the gas pressure to such a degree that energy for smokeless burning would not be present. This problem has been overcome using a staging concept.[13] The staging concept divides the burners into stages or groups of burners, with the first stages having a smaller number of burners than the last stages. The flow to each stage or group of burners is controlled by a valve that operates in an on/off manner as directed by a control system. The valve to the first stage is usually open all the time. The control system principle is to proportion the number of burners in service to the gas flow. In effect, this allows the burners to operate within a certain pressure range (see Figure 11.32) so that at least the minimum energy level for smokeless burning is always present.

The operation of a multipoint flare can be spectacular, as shown in Figure 11.32. This series of photographs illustrates the addition of burner stages as the gas flow to the flare increases. In the final frame, the flare achieves a smokeless burning rate of more than 550,000 lb/h (250,000 kg/h). The commercially available linear relief gas oxidizer (LRGO) flare system shown is surrounded by a fence to exclude personnel and animals from the flare area. In cases where land space is limited, an enclosing fence, such as shown in Figure 11.33, can be employed to reduce radiation to the surroundings and to reduce visibility and noise. The largest flare system in the world employs an LRGO system to handle more than 10 million pounds of waste gas per hour (>4.5 million kg/h), smokelessly.

Design of an energy conversion flare system, either single point or multipoint, involves issues that are not a consideration for other flare burner types. Questions such as burner or gas jet spacing must be resolved. For example, the ability of a given burner to be lit and to light neighboring burners is of paramount importance. Figure 11.32f captures this crosslighting feature in progress. However, spacing burners solely on the basis of crosslighting may restrict airflow and hinder smokeless burning. Gas properties are even more critical when the design depends on energy conversion alone to achieve smokeless burning.

11.4.1.7 Endothermic

Some flare applications involve gases with a high inert gas content. When the inert content is high enough, the combustion reaction becomes endothermic, meaning that some external source of heat is required to sustain the reaction. Crude oil recovery by CO_2 injection, incinerator bypasses, coke ovens, and steel mills are examples of activities that generate gases that require additional fuel to maintain the main burner flame. Such gases often contain significant amounts of toxic materials such as H_2S, NH_3, CO, or various gases normally sent to an incinerator. Flaring has been recognized for many years as an adequate method of disposing of such gases. Substantially complete destruction of such gases protects the community and the environment.


(a)

(b)

FIGURE 11.32
LRGO staging sequence during a flaring event from inception (a) to full load (g).

The earliest endothermic flares consisted of simple nonassisted flare tips with fuel gas enrichment of the waste gas upstream of the flare to ensure that the mixture arriving at the tip was burnable. This system was simple, but imposed a high fuel cost on the facility. An alternative design supplied a premixed supplemental fuel/air mixture to an annulus around the flare tip. Combustion of this mixture supplied heat and ignition to the waste gas as it exited the flare tip. This design had a limited supplemental fuel gas turndown before burnback occurred in the annulus, thus requiring a full on or off operation of the supplemental gas.

(c)

(d)

FIGURE 11.32 (continued)
LRGO staging sequence during a flaring event from inception (a) to full load (g).

(*continued*)

Today's high energy costs provide an incentive to reduce such fuel usage. The RIMFIRE® flare burner (Figure 11.34) is a modern endothermic flare with an air-assisted supplemental gas burner surrounding the waste gas exit.[14] Using supplemental fuel to build a strong "forced-draft" ignition flame, the amount of enrichment gas required to sustain ignition is reduced. The amount of supplemental fuel required depends on the flare burner size and service.

The RIMFIRE® flare was originally developed for the CO_2 injection fields in the western United States. Vents and reliefs from the oil recovery system sent highly inert materials to the flare. In this service, a nonassisted tip would require enrichment of the relief gases to an LHV of about 300 Btu/scf (11 MJ/Nm^3). Using the RIMFIRE® flare burner, the LHV requirement was reduced to approximately 180 Btu/scf (6.7 MJ/Nm^3).


(e)


(f)

FIGURE 11.32 (continued)
LRGO staging sequence during a flaring event from inception (a) to full load (g).


(g)

FIGURE 11.32 (continued)
LRGO staging sequence during a flaring event from inception (a) to full load (g).

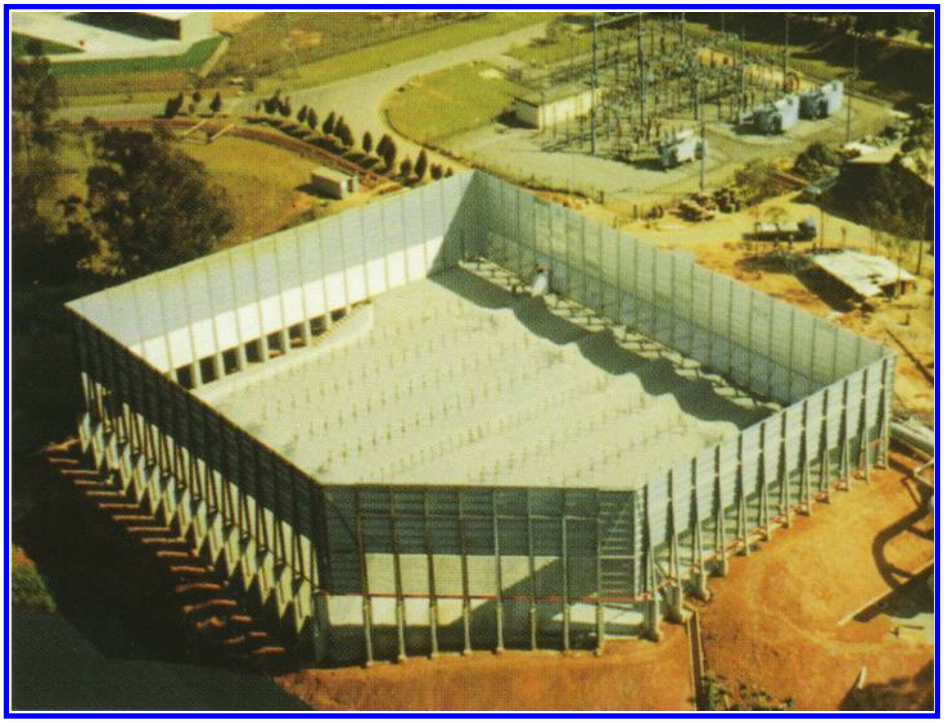


FIGURE 11.33
Multipoint LRGO system with a radiation fence. (Courtesy of Maria Celia, Setal Engineering, Sao Paulo, Brazil.)

FIGURE 11.34
A RIMFIRE® endothermic flare.

The total fuel requirement, enrichment plus supplemental fuel, for the RIMFIRE® is substantially lower than the fuel requirement for other designs. Today, the RIMFIRE® burner is used in a broad range of applications where enrichment of the waste gas is required.

11.4.1.8 Special Types

Sections 11.4.1.1 through 11.4.1.6 have described the wide variety of flare burners available to the flare designer today. In general, these burners are used to burn gaseous waste material and utilize certain smoke-suppression techniques. There are, however, other flares that are designed to burn liquid hydrocarbons or to use a liquid as the smoke suppressant. Some examples are given next.

Figures 11.35 and 11.36 show two examples of liquid flares having quite different designs. The oil water burner (OWB) flare (Figure 11.35) has multiple burners combined into a single unit. At the time of the photograph, the flare was burning oil with a heat release of about 1 billion Btu/h (290 MW). The OWB tips allow a modular approach to design for a specific capacity. In addition, a very wide turndown can be obtained by simply turning off flow to some of the tips. Typical applications are oil well testing and spill cleanup.

The Dragon flare (Figure 11.36) uses one or more burners and is equipped with a blower to improve

FIGURE 11.35
OWB liquid flare test firing 150 gpm (570 L/min).

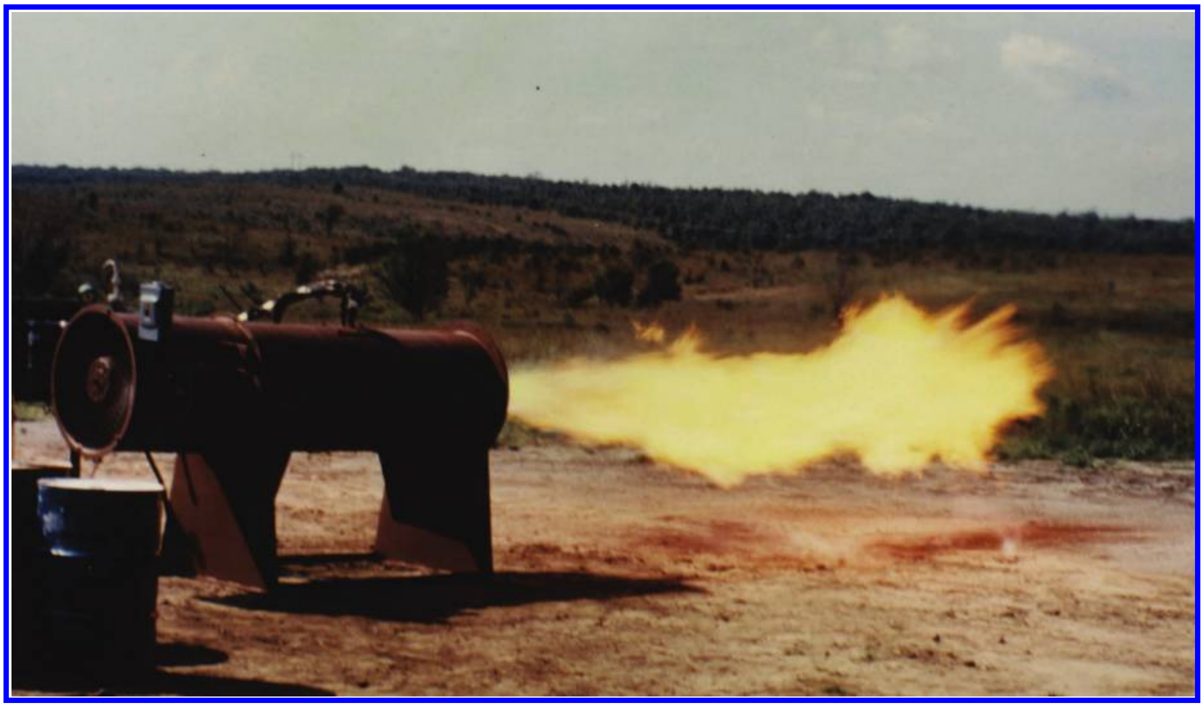


FIGURE 11.36
Forced-draft Dragon liquid flare.

mixing for smokeless burning. This flare is employed in destruction of surplus or off-spec product or waste oil.

The success of steam as a smoke suppressant has prompted the use of water to prevent smoke. Over the years, a number of designs using water for smoke suppression have been invented. In certain situations, water can be used with great benefit. A case in point is the offshore installation of a Poseidon flare shown in Figure 11.37. This flare utilizes water to enhance smokeless burning, reduce thermal radiation, and decrease noise. The installation shown in the photograph achieved a 13-dBA reduction in noise and a 50% drop in radiation compared to the previous conventional flare.[15]

11.4.2 Knockout Drums

There are three basic types of knockout drums that can be incorporated into a flare system: a horizontal settling drum, a vertical settling drum, and a cyclone separator.

Horizontal settling drums are large drums in which droplets are allowed sufficient residence time to separate from the gas by gravity. API RP-521[5] provides detailed design guidelines for this type of drum. Figure 11.38 shows a typical example of this design. The pressure drop across these drums is relatively low. Drums of this type are particularly useful for removing liquids within or near the process units that may send liquids to the flare header. It is common for the maximum liquid level to be at the drum centerline, thus allowing 50% of the total vessel volume to be used for temporary liquid storage during a relief.

Vertical settling drums work in a similar fashion. In designing vertical settling drums, careful attention must be focused on droplet terminal velocity because this velocity determines the drum diameter. Also, the volume available for storage of liquid during a relief is limited by the elevation of the flare header piping.

Any small droplets that pass through the knockout drum can agglomerate to form larger droplets in the flare system downstream of the knockout drum. Locating the knockout drum very near the base of the flare stack, or incorporating it into the stack base, can minimize this problem. Although the pressure drop required for the settling drums is generally low, the required drum diameter can become impractical to shop fabricate if the flow rate is high.

Elimination of very small liquid droplets cannot be accomplished through a simple reduction in gas stream velocity. Cyclone separation is best for small droplet removal. Mist eliminators, utilizing centrifugal force, can be very effective when incorporated into the base of the flare stack. They are smaller in diameter than horizontal or vertical settling drums and usually provide high liquid removal efficiency at the expense of a greater pressure drop. The frost on the outside of the drum in Figure 11.39 vividly illustrates the liquid flow pattern.

FIGURE 11.37
Poseidon flare: water-assisted Hydra.

Agglomeration of droplets downstream of a cyclone separator is generally less of a problem than it is for the settling drum designs. The typical settling drum is designed to remove droplets larger than 300–600 μm at the smokeless flow rate. Droplet sizes at the maximum flow rate can be over 1000 μm in some cases. By comparison, the droplets exiting the cyclone are much smaller, typically 20–40 μm, and the droplet size remains low throughout the operating range.

However, available volume for liquid storage in a cyclone separator is generally small compared to the vertical settling drum because a substantial length is required for the vapor space in this design. When substantial liquid loads are anticipated, horizontal settling drums are usually provided at the upstream end of the flare header to catch most of the liquid volume. The mist eliminator in the base of the flare stack removes the remaining liquid and minimizes problems with agglomeration.

11.4.3 Liquid Seals

A liquid seal is a device that uses a liquid, such as water or a glycol/water mix, as a means of providing separation of a gas (or vapor) conduit into an upstream section and a downstream section. The physical arrangement of a typical liquid seal is shown in Figure 11.40. Gas flows through the seal when the gas pressure on the upstream side of the seal is equal to or greater than the pressure represented by the seal leg submergence plus any downstream back pressure. The submergence depth used depends on the purpose of the seal. Present practice involves submergence depths ranging from about 2 in. (5 cm) to over 120 in. (300 cm).

Liquid seals are found in many types of combustion systems, including flares, due to the fact that liquid seals can be used to accomplish any of a variety of goals:

- Prevent downstream fluids from contaminating the upstream section
- Pressurize the upstream section
- Divert gas flow
- Provide a safe relief bypass around a control valve
- Arrest a flame front or detonation

FIGURE 11.38
Horizontal settling drum at the base of an air-assisted flare.

11.4.3.1 Prevent Upstream Contamination

One of the most frequent uses of liquid seals is to prevent air infiltration into the downstream section from propagating to the upstream section. Properly operated liquid seals provide a safeguard against the formation of an explosive mixture in the flare header by acting as a barrier to backflow.

11.4.3.2 Pressurize Upstream Section

As discussed in Section 11.3.4, a negative pressure can exist at the base of the flare stack at low flow conditions. By injecting purge gas upstream of the liquid seal, the upstream section is pressurized to a level related to the submergence depth. As a result, any leaks in the upstream section will flow gas out of the flare header rather than air into the header.

11.4.3.3 Diverting Gas Flow

Liquid seals are often used to divert gas flow in a preferred direction. An example is a staged flare system involving an enclosed flare and an elevated flare such as the one shown in Figure 11.12. The liquid seal in the line to the elevated flare diverts all flows to the enclosed flare until the pressure drop caused by gas flow through the enclosed flare system exceeds the submergence depth.

FIGURE 11.39
Cyclone separator. (Note the frost indicating the flow path of the low-temperature liquid as it is removed.)

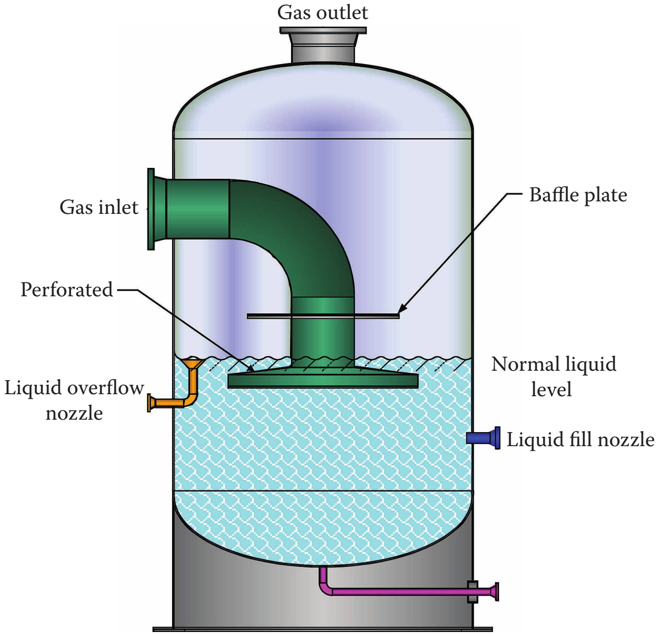


FIGURE 11.40
Schematic of a vertical liquid seal.

11.4.3.4 Control Valve Bypass

A control valve in flare header service can represent a safety hazard should it fail to open when required. A liquid seal bypass around the control valve protects the plant against possible failure of the control valve to open during a relief. When the upstream pressure reaches the submergence depth plus any back pressure from the elevated flare, waste gas will begin to flow through the liquid seal to the elevated flare—whether the control valve is open or not.

11.4.3.5 Liquid Seals as Arrestors

Liquid seals that are used as flame arrestors generally fall into three categories as follows:

1. Seals designed to handle incoming combustible gases that do not contain air (or oxygen), for example, a refinery flare
2. Seals designed to handle incoming vapor streams that contain a mixture of combustible gases and air (or oxygen), for example, the fuel vapor/air stream produced during tank truck or barge loading operations
3. Seals designed to handle either ethylene oxide (ETO) or acetylene, for example, the vent seal for the ETO gases from a sterilizer used for treating medical supplies

11.4.3.6 Design Factors

Once the purpose of the liquid seal has been established, the designer can produce a suitable liquid seal design. Several factors influence general liquid seal design, including

- Seal vessel orientation (horizontal or vertical)
- Seal vessel diameter
- Seal leg submergence depth
- Configuration of the end portion of the seal leg (seal tip or seal head)
- Space above the liquid level
- Type and size of outlet
- Seal fluid selection

A complete discussion of all these factors is beyond the scope of this text.

Liquid seals have been known to cause pulsating flows to flares. Pulsating flow, in turn, makes smoke control difficult and creates a fluctuating noise and light source that can become a nuisance to neighbors. Figure 11.41 shows the “smoke signal” effect that can occur in extreme cases of such flow patterns. The fundamental cause of this pulsating flow is the bidirectional flow that occurs when the displaced liquid in the vessel moves to replace gas bubbles released at the gas exit of the seal head or inlet pipe. As the submergence depth increases, the buoyant forces acting on the liquid increase and the potential for violent movement by the seal fluid grows. Larger vessel diameters also increase the potential for liquid sloshing, which is another driver of pulsation.

Properly designed internals can reduce such pulsations by controlling the bidirectional gas flow and movement of the liquid. Dr. Robert Reed, who was the Technical Director for John Zink Co. for many years, produced some of the earliest internals for liquid seals in the late 1950s. Since that time, a number of alternative designs have been developed[16] that improve on the basic idea. Several of the designs that have been used over the years are shown in Figure 11.42. Today, designs are available that can convert an existing horizontal knockout drum into a combination liquid seal/knockout drum with enough submergence depth to enable flare gas recovery.

The two fluids normally used for a liquid seal are water and glycol/water mix. Whenever possible, water is preferred. The use of hydrocarbon liquids (other than glycol) is strongly discouraged.

11.4.4 Purge Reduction Seals

Air infiltration into a flare system through the flare burner was discussed in Section 11.3.4. Most systems are designed to combat air infiltration into the tip and riser

FIGURE 11.41
"Smoke signals" from a surging liquid seal.

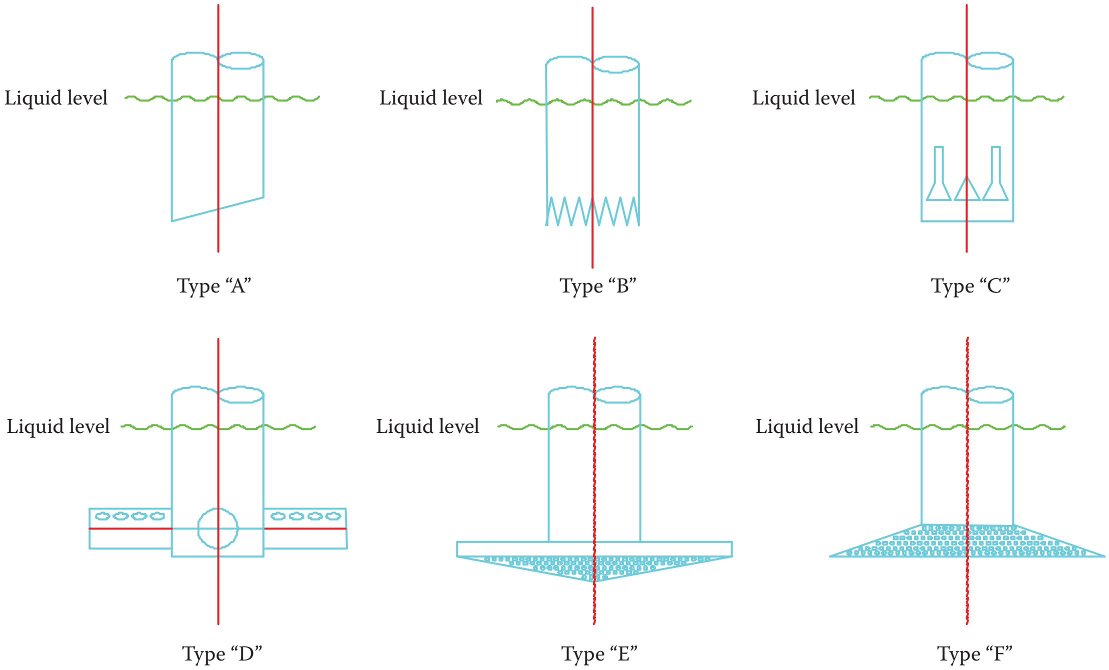


FIGURE 11.42
Various liquid seal head types. Seal tip design: "A" beveled end, "B" sawtooth, "C" slot and triangle (After API RP-521, 4th edn., American Petroleum Institute, Washington, DC, March 1997), "D" arms with ports on upper surfaces, An alternative design uses ports on the lower surfaces: "E" downward facing perforated cone, "F" upward facing perforated cone.

by purging. The amount of purge gas required to prevent air from entering into the system can be quite large, especially in the case where light gases are present. A high purge rate may pose several disadvantages. First is the cost of the purge gas; second, the heat from the combustion of the purge gas can be damaging to the flare burner; and third, burning more gas than is absolutely necessary increases the emission level of the plant. Adding a purge reduction device to the flare system can mitigate these disadvantages. Normally, such devices are installed just above or immediately below the flare burner-mounting flange so as to maximize the air exclusion zone.

Purge reduction devices are intended to improve the effectiveness of the purge gas so that the amount required to protect the system can be reduced. Purge reduction devices, often referred to as seals, are based on the use of either of two basic strategies: (1) density difference (sometimes called a density seal) or (2) trap and accelerate (sometimes called a velocity seal). The discussion in the following addresses the principle of each seal type and its advantages and disadvantages.

A velocity seal is shown in Figure 11.43. The principle of the velocity seal is to trap air as it enters the flare tip, reverse its direction, and carry it out of the tip with accelerated purge gas. Tests on flare stacks, large and small, have demonstrated that air enters a flare tip along the inner wall of the tip. In the velocity seal, a shaped trap is placed on the inner wall of the tip. The trap intercepts the incoming air and turns it back toward the tip exit. At the same time, the shape of the trap acts to accelerate the purge gas. The accelerated purge gas and outflowing air meet at the exit of the seal device and flow out the tip. Without the accelerated purge gas, the trap will only delay air entry, not reduce it.

Compared to a density seal, a velocity seal is relatively small and has low capital cost. The velocity seal will reduce the purge gas requirement, but the reduction is tempered by the amount of oxygen allowed below the seal. A velocity seal requires more purge gas than a density seal. An additional disadvantage of a velocity seal occurs when purge gas flow is interrupted. In this event, the oxygen level in the riser begins to increase almost immediately.

The arrangement of a density-type purge reduction device is illustrated in Figure 11.44. As the gas flows upward through the riser, it is directed through two annular 180° turns, thus forming spaces where lighter- or heavier-than-air gases are trapped. The density difference between the trapped purge gas and air forms a barrier to air movement. Only diffusion will allow the air to work its way through the barrier. Thus, a purge rate sufficient to constantly refresh the gas at the gas/air interface is all that is required. This purge rate is much lower than the rate required for a velocity seal (which will have some level of oxygen below it). A density difference as small as nitrogen to air is sufficient for the seal to function. The lighter (or heavier) the purge gas, the more effective the seal becomes. Tests have shown that the oxygen level below a properly purged density-type seal will be zero. If the purge gas flow to a density seal is interrupted, air will begin to penetrate the gas by diffusion. However, the diffusion process is slow and a significant time will pass before air enters the riser.

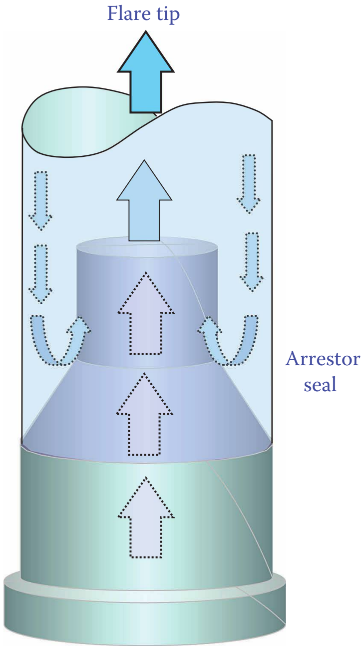


FIGURE 11.43
AIRRESTOR velocity-type purge reduction seal.

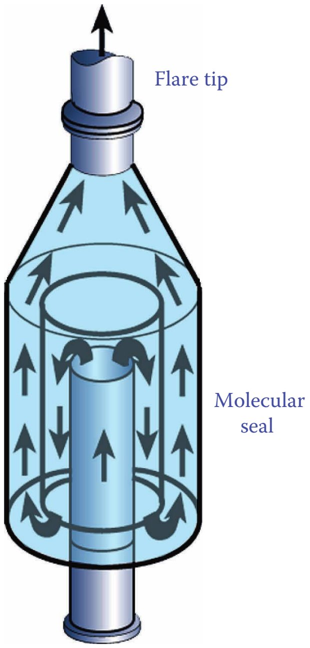


FIGURE 11.44
Molecular seal density-type purge reduction seal.

The density seal requires the smaller purge gas rate and enjoys the lower operating cost. A lower purge gas requirement also means less heat around the flare tip and lower emissions. However, the physical size of the density seal makes its capital cost larger.

11.4.5 Enclosed Flares

The desire to hide flaring activities dates back to the 1950s. Flare vendors and users tried for several years to design enclosed ground flares and failed, sometimes spectacularly. In one case, the smoke generated by a ground flare was so dense that it forced a plant shutdown. In another case, smoking caused the shutdown of a major highway, and noise broke windows at a great distance.

A different design concept yielded the first successful enclosed ground flares in 1968 when two units designated as ZTOFs were placed in service by Caltex.[17] These enclosed ground flares and dozens of additional units constructed over the ensuing years led to the development of the modern enclosed ground flare.

Enclosed ground flares use a refractory-lined combustion chamber to contain the entire flame, rendering it invisible to neighbors. A schematic diagram of an enclosed ground flare is shown in Figure 11.45. The combustion chamber is generally cylindrical, but can be rectangular, hexagonal, or other shapes formed from flat panels. Cylindrical sections are generally favored. Flat panels are used in some cases to reduce shipping costs or to optimize field assembly.

A ZTOF is essentially a giant, direct-fired air heater. Air required for combustion and for temperature control enters the combustion chamber by natural draft after passing through a burner opening. Elevated temperatures in the combustion chamber reduce the density of the flue gases inside and produce draft according to Equation 11.2. This draft is the motive force that drives combustion products out the top of the stack and draws air in through the burner openings. Optimizing the use of the available energy is an essential part of the proper design of a ZTOF.

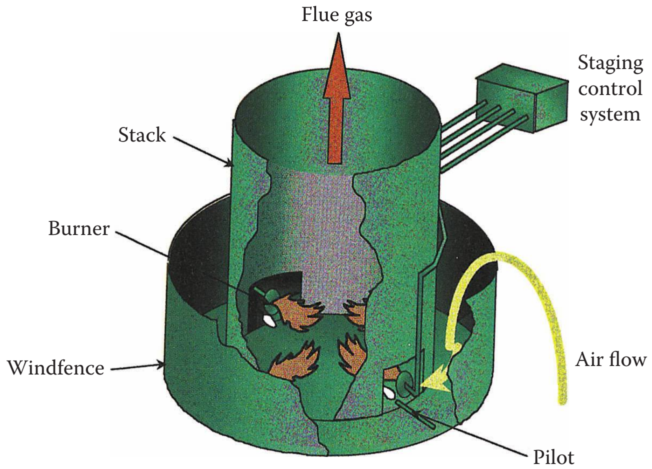


FIGURE 11.45
Schematic of a ZTOF.

Most ZTOFs are designed to handle substantially more than the stoichiometric air requirement. The excess air is used to quench the flame temperature. This reduces the required temperature rating for the refractory lining, which is a significant part of the overall cost of the system. Although the quenched flue gas temperature may be 1600°F (900°C) or lower, the refractory lining—at least in the lower section of the stack—should be selected for a higher service temperature because local temperatures may be higher than the final flue gas temperature.

When a ZTOF is used as the first stage of a flare system, there is the potential to deliver more waste gas to the ZTOF than it is designed to handle. Overfiring the ZTOF can result in flame and/or smoke out the top of the stack. There are usually two safeguards to prevent this from happening. First, when the pressure drop created by the gas flow through the ZTOF system exceeds the set point of the diversion device (whether liquid seal or valve), excess gas automatically flows to the elevated flare. Second, most ZTOFs are equipped with thermocouples to monitor the stack temperature. When the stack exit temperature exceeds the design level, a temperature switch initiates an automatic shutdown, either partial or total, of the ZTOF burner system. Gas flow is sent to the other parts of the plant flare system until the cause of the overfiring condition can be identified and corrected.

ZTOFs have been designed with capacities ranging from less than 100 lb/h (45 kg/h) to more than 100 metric tons/h (110 ton/h). Combustion chambers vary from 3 ft (1 m) to more than 50 ft (15 m) across and may be over 100 ft (30 m) tall.

To maximize the benefit of the available combustion volume, ZTOFs are usually equipped with multipoint burner systems. As discussed in Section 11.4.1.5, breaking up the gas flow into many small flames improves burner performance. ZTOF systems frequently operate at pressure levels consistent with liquid seal depths. As a result, the available energy from the waste gas is reduced. Thus, staging the ZTOF burner systems is even more important to maintain good performance at turndown conditions. When steam-assisted burners are used in ZTOFs, steam efficiencies are substantially higher than open air flares, resulting in lower day-to-day steam consumption.

On small units, adjusting the air openings feeding air into the combustion chamber can control temperature in the combustion chamber. Temperature control is common in landfill flares, biogas flares, and vapor combustors in gasoline loading terminals. Proper temperature control minimizes emissions from these units, which in some cases run continuously.

The wind fence used to manage airflow into the combustion chamber can be designed to muffle combustion and steam noise generated by the burners. The refractory-lined combustion chamber may also absorb high-frequency noise and serves to block the direct line-of-sight path for noise transmission from the flames in the chamber. By providing clean, quiet, invisible disposal of day-to-day reliefs, use of a ZTOF allows plant operation in harmony with its neighbors.

11.4.6 Flare Support Structures

The combination of the heat released at maximum design flow and the owner's instructions on allowable incident radiation poses a design challenge that is often solved by elevating the flare burner (see Sections 11.2.6 and 11.3.5 for discussion of the factors and design considerations that influence the determination of the required height). Once the height has been calculated, the design focus turns to the selection of the type of flare structure to use.

In principle, there are three basic support structure concepts plus a variant that can be very useful in certain circumstances. The concepts and the variant are

- Self supported
- Guy wire supported
- Derrick supported
- Derrick with provision for lowering the riser and flare burner

A self-supported structure (Figure 11.46) requires the least land space and can easily accommodate a liquid seal or knockout drum, or both, in the base section. Varying the diameter and thickness of the structure at various elevations absorbs wind loads. Potential undamped vibration is avoided by varying the length and diameter of sections of the structure. Generally, self-supported structures are not cost-effective at heights above about 250 ft (76 m).

Perhaps, the most common means of supporting an elevated flare burner is a riser that is held in line by guy wires (Figure 11.47). Usually, there are three sets of guy wires spaced 120° apart. The number of guy wires arranged vertically at a given location is dependent on the height of the structure, wind loads, and the diameter of the riser. Guy wire–supported structures require the greatest land space commitment. Overall heights can reach 600 ft (180 m) or more.

Where land area is of high value or limited availability, a derrick structure can be employed. The derrick itself (Figure 11.48) acts as a guide to keep the riser in line. In general, derricks are designed with three or four sides and have been utilized at heights greater than 650 ft (200 m).

FIGURE 11.46
Self-supported flare.

Flare burners on very tall support structures and flares located in remote areas are difficult to maintain or replace due to the limited number of cranes that can service the required elevation. In such cases, a derrick variation often referred to as a demountable derrick is employed. The design of the demountable derrick (Figure 11.49) allows the riser and attached flare burner to be lowered to the ground, either as a single piece or in multiple sections. An additional advantage of a demountable derrick is its ability to support more than one full-size riser.

With the exception of demountable derricks, any flare burner support structure with a height of more than about 50 ft (15 m) above grade typically includes a 360° platform for use during maintenance. If a 360° platform is used, it is often located just below the flare burner-mounting flange. The structure will also provide support for ladders, required step-off platforms, and utilities piping. Some flare structures, due to their height and location, may require aircraft warning markings such as paint or lights.

FIGURE 11.47
Guy wire–supported flare.

FIGURE 11.48
Derrick-supported flare.

A number of factors enter into the selection of the structure: physical loads, process conditions, land space available, cost of land, availability of cranes, and the number of risers to be considered. The selection process can be simplified by using the guide shown in Figure 11.50. The guide asks a series of questions that can be answered "yes" or "no," with the answer influencing the next question. While the yes/no answers appear to lead to an absolute answer, there are subtleties that can promote an alternative. For example, the desire to locate a liquid seal or knockout drum, or both, in the base of the stack may make a self-supported design attractive.

The guide refers to situations in which there will be one waste gas riser (R^1) or two waste gas risers (R^1 and R^2). If there are two risers, the guide questions the size of the second riser as compared to $R^1/3$. Typically, a second riser with a size of $R^1/3$ or less will be a small-capacity flare serving a vent system or incinerator bypass. Such a small flare, often referred to as a "piggyback flare," will be supported by the main flare or its support structure. If the second riser is greater than $R^1/3$, it will be treated, for structural design purposes, as a second major flare. Cases involving more than two risers are good candidates for a demountable derrick structure.

11.4.7 Flare Controls

Flare systems are often associated with flare headers that collect gases discharged from relief valves and other sources. A flare is called upon to operate properly during upset and malfunction conditions that impact control systems throughout the plant, including power failure and instrument air failure. Therefore, controls on flare systems must be used with discretion to ensure that the flare will continue to operate safely even if its controls fail. Flare controls can help provide effective smokeless performance, low-noise operation, and other desirable characteristics during normal day-to-day operation.

FIGURE 11.49
Demountable derrick.

Many of the controls used in flare systems are associated with pilots, ignition, and pilot monitoring and have already been discussed in Chapter 12. This section discusses steam control, burner staging, level controls, and purge control.

11.4.7.1 Typical Steam Control Valve

Reliable steam control is an important part of the smoke-suppression strategy for steam-assisted flares. The simplest steam control system consists of a manual valve that an operator uses to adjust steam flow to the flare tip. Most plants prefer not to dedicate an operator to manage the steam use of their flares. Instead, steam control valves are equipped with remote positioning equipment that allows an operator in the control room to adjust steam flow while performing other, more profitable duties. Figure 11.51 depicts a typical steam control valve station.

The steam control valve on a flare can operate almost completely closed for extended periods of time. As a result, wear on the valve seat becomes a maintenance issue. To allow for removal and maintenance while the flare is in operation, block valves are recommended both upstream and downstream of the control valve. To operate the flare smokelessly during control valve maintenance, a bypass line with a manual valve is installed around the control valve and its block valves. A pressure gauge should be installed downstream of the control valve station to provide the operator with a tool for diagnosing control issues and a guide for manual control, when needed.

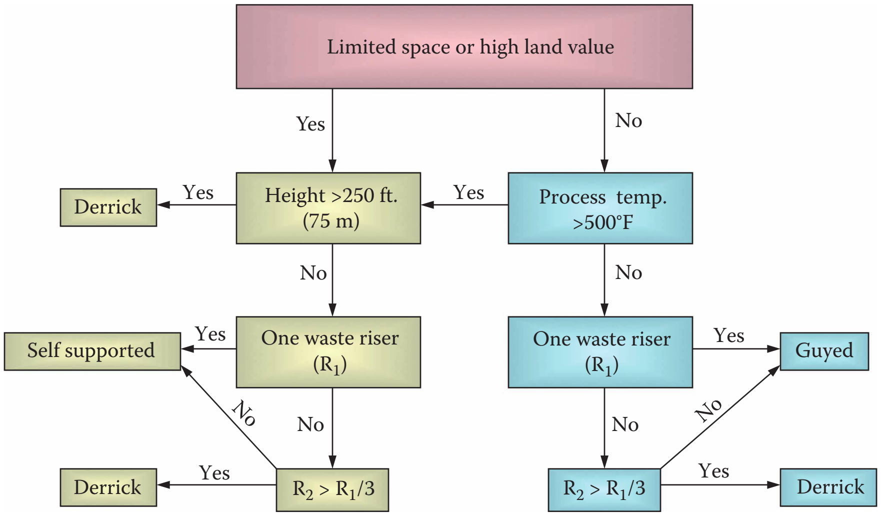


FIGURE 11.50
Flare support structure selection guide.

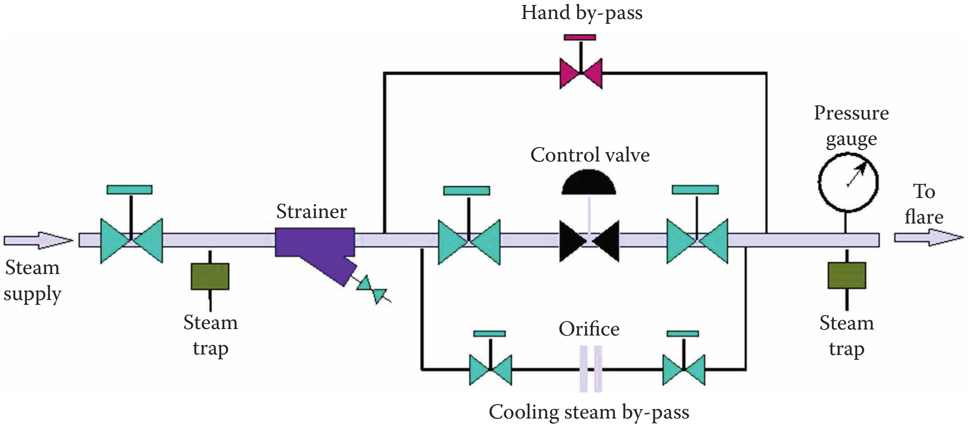


FIGURE 11.51
Steam control valve station.

Most steam-assisted flares require a minimum steam flow for two reasons. First, a minimum steam flow keeps the steam line from the control valve to the flare burner warm and ready for use. It also minimizes problems with condensate in that line. Second, a minimum steam flow keeps the steam manifold on the flare burner cool ("cooling steam") in case a low flow flame attaches to the steam equipment. To maintain the minimum steam flow, a second bypass line is installed with a metering orifice sized for the minimum flow and a pair of block valves for maintenance of the orifice.

Steam traps are mandatory wherever condensate can accumulate in the steam piping. Many steam injector designs use relatively small orifices, at least in part to reduce audible noise. Therefore, a steam line strainer is recommended. If the orifices are very small, all stainless steel steam piping may be appropriate.

11.4.7.2 *Automatic Steam Control*

As the flow or composition of waste gas sent to the flare varies, the amount of steam required for smoke-suppression changes. Many plants adjust the steam requirement based on periodic observations by an operator in the control room looking at a video image from a camera aimed at the flare. Any smoking condition will be quickly corrected by an increase in steam flow to the flare. However, when the gas flow begins to subside, the flare flame continues to look "clean" to the operator. Therefore, some time may pass before the operator reduces the steam flow. As a result, this method of smoke control tends to result in oversteaming of the flare, which in turn produces excessive noise and unnecessary steam consumption.

Optical sensing systems are available to monitor the condition of the flare flame and adjust the steam flow continuously. Automatic optical sensing equipment can effectively control steam flow to maintain a consistent flame appearance with minimum steam usage and minimum noise.

11.4.7.3 *Typical Staging Control Valve*

Energy conversion flare types such as the LRGO discussed in Section 11.4.1.5 are designed to operate smokelessly when the gas pressure is above a certain level. Two key operational goals for such systems using energy conversion burners are to (1) maintain the gas pressure above the minimum level required for smokeless performance and (2) prevent back pressure from exceeding the allowable design level. To achieve these goals, a staging control system is used that starts and stops flow to various groups of burners based on the incoming gas flow. Depending on the application, the staging valve used to accomplish this can be installed in any of a number of possible configurations, ranging from a single valve to a complex system of bypasses and block valves. Figure 11.52 shows a typical staging valve assembly.

The main staging valve can be either fail closed or fail open, depending on the safety considerations governing the system as a whole. Generally, the staging valves for an enclosed ground flare are fail closed when another flare is available to handle emergency relief loads. Such staging valve assemblies usually do not include a bypass device.

Staging valves, especially on the last stages, of a multipoint staged plant flare system are usually designed to fail open when the bypass device is a rupture disk. If the bypass device is easily reclosed or resets automatically, the staging valve can be designed to fail closed. The bypass device, shown in the figure as a rupture disk, can also be a relief valve or a liquid seal.

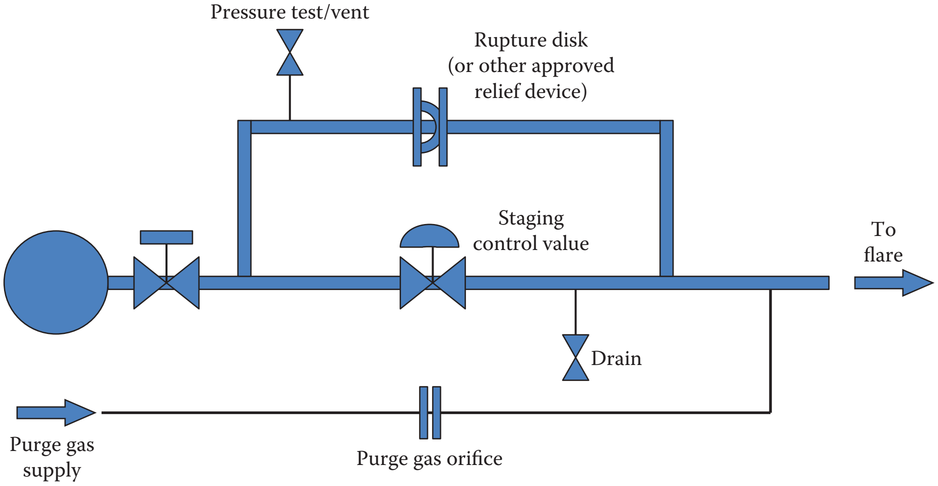


FIGURE 11.52
Staging control valve assembly.

11.4.7.4 Level Controls

Flare systems often include vessels such as knockout drums or liquid seals that can contain liquid levels that must be monitored and/or controlled for safe operation. Liquid level is controlled in knockout drums to prevent overfilling, as discussed in Section 11.3.3. In some cases, it is also important to prevent too low a level. When all the liquid in a knockout drum is removed, it becomes possible for waste gases in the flare header to migrate into the drain system, creating a possibly explosive mixture and a serious safety hazard. Instrumentation generally consists of one or more level switches or transmitters often mounted together with gauge glasses to simplify setpoint adjustments and to allow visual monitoring or manual control.

Liquid seal-level control presents a number of challenges not found in other level control applications. In normal operation, when gas is flowing through the liquid seal, the surface of the liquid is violently agitated. Small-scale wave action, spraying, and foam generation also create special requirements for liquid seal-level control systems.

Liquid seals in flare service can accumulate a certain amount of hydrocarbon condensate. Such condensates are generally lighter than water and affect level control and safety. The presence of such condensates in the liquid seal creates the potential to generate hydrocarbon droplets in the waste gas flowing to the flare tip. As discussed in Section 11.3.3, this can become a safety hazard. To protect against this hazard, flare liquid seals are often equipped with hydrocarbon skimming systems that remove accumulated condensate from the liquid surface, in some cases, automatically.

Loop seals are used to prevent gases from escaping a vessel while allowing liquids to be removed automatically. Hydrocarbon skimming systems on flare liquid seals often utilize loop seals, such as shown in Figure 11.53, to provide constant removal of liquids. Loop seal design guidance is provided by API RP-521.[5] Some additional concerns include

- Loss of loop seal fluid by evaporation or overpressure can result in waste gas entering the sewer system and/or escaping to atmosphere through the antisiphon break at the top of the outboard loop.
- Freezing of loop seal fluid can result in overfilling the vessel and/or accumulation of hydrocarbon condensate in the vessel.

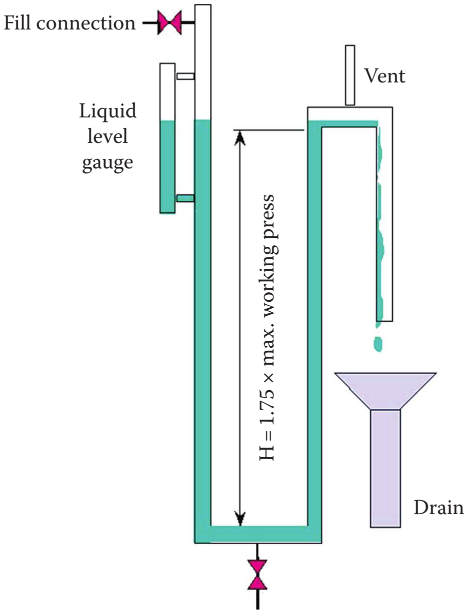


FIGURE 11.53
Loop seal.

- Elevation of the top of the outboard loop must be at or below the controlled liquid level.
- If liquids of differing densities are anticipated, the elevation of the outboard loop must be low enough to allow the lightest liquid to push out the heaviest liquid.

11.4.7.5 Purge Controls

Purge gas injection is one of the most important safety features in a flare system. A common method for controlling the flow of purge gas is the use of a metering orifice and a supply of purge gas with regulated pressure. A typical arrangement is shown in Figure 11.54. Safety features should include an effective strainer to prevent plugging of the metering orifice, a flow transmitter with an alarm for low flow condition, and a supply pressure substantially higher than any anticipated flare header pressure.

11.4.8 Arrestors

Dry arrestors such as flame arrestors or detonation arrestors have limited application in plant or production flare systems. This is due to the concern that the small passages of an arrestor could become plugged, leading to increased back pressure on the relieving or venting source. In the worst case, the source could become overpressured. A few of the sources of concern regarding plugging are

- Scale or debris carried by high waste gas velocities in the flare header
- Gas compositions that include compounds prone to polymerize
- Two-phase flow carrying liquids or condensate into the arrestor

There are some limited circumstances where the use of a dry arrestor may be acceptable. For example, an arrestor may be used on systems that handle relatively clean, dry material or systems that can be easily shut down for maintenance should an overpressure be detected. As with all flare system issues, careful attention to the safety aspect of the prime objective is required.

11.5 Flare Combustion Products

An industrial flare is the most suitable and widely used technology for disposing of large quantities of organic vapor releases. In the 1980s, the U.S. Environmental Protection Agency (U.S. EPA) conducted several tests to determine the destruction and combustion efficiency of an industrial flare operating under normal conditions. Based in part on these tests, the U.S. EPA made several rulings concerning the design of a flare. The purpose of this section is to discuss the results of these tests and U.S. EPA rulings on flare design.

11.5.1 Reaction Efficiency

The terms *combustion efficiency* and *destruction efficiency* have frequently and mistakenly been considered synonymous. In fact, these two concepts are quite different. A flare operating with a combustion efficiency of 98% can achieve a destruction efficiency in excess of 99.5%.

11.5.1.1 Definition of Destruction and Combustion Efficiency

Destruction efficiency is a measure of how much of the original hydrocarbon is destroyed, that is, broken down into nonhydrocarbon forms, specifically carbon

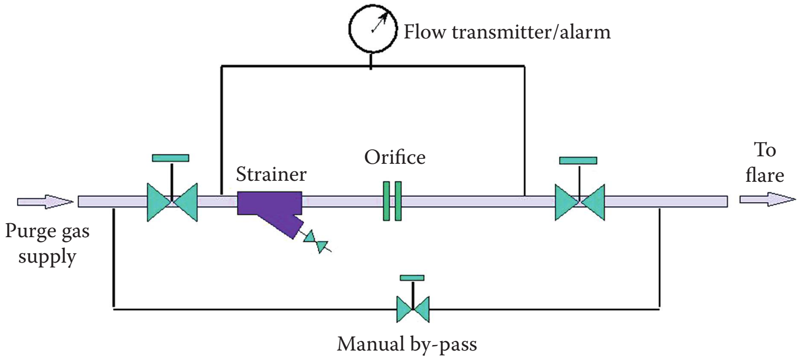


FIGURE 11.54
Purge control station.

monoxide (CO), carbon dioxide (CO_2), and water vapor (H_2O). The destruction efficiency can be calculated by a carbon balance as follows:

$$\%DE = \frac{CO_2 + CO}{CO_2 + CO + UHC} \times 100 \tag{11.4}$$

The term %DE is the percent destruction efficiency, and CO_2, CO, and UHC are the volume concentrations of carbon dioxide, carbon monoxide, and UHCs (as methane) in the plume at the end of the flare flame, respectively. Notice that if no UHCs escaped the flame, the destruction efficiency would be 100%.

Combustion efficiency is a measure of how much of the original hydrocarbon burns completely to carbon dioxide and water vapor. Using the carbon balance approach, combustion efficiency can be calculated as

$$\%CE = \frac{CO_2}{CO_2 + CO + UHC} \times 100 \tag{11.5}$$

where %CE is the percent combustion efficiency. Notice that even if no UHCs escape the flame, the combustion efficiency can be less than 100% because CO represents incomplete combustion. It is evident from Equations 11.4 and 11.5 that the combustion efficiency will always be less than or equal to the destruction efficiency.

11.5.1.2 Technical Review of Industrial Flare Combustion Efficiency

In 1983, the John Zink Company and the Chemical Manufacturers Association (CMA) jointly funded a research project aimed at determining the emissions of flares operating under normal, real-world conditions.[18] Various mixtures of crude propylene and nitrogen were used as the primary fuel, with waste gas LHVs varying from approximately 80 to 2200 Btu/scf (3–82 MJ/Nm^3) and flow rates up to 3000 lb/h (1400 kg/h). Tests were conducted using steam-assisted, air-assisted, and nonassisted flares. These tests concluded that flares operating with a normal, stable flame achieve combustion efficiencies greater than or equal to other available control technologies.

The U.S. EPA ruled that a flare can achieve a combustion efficiency of 98% or greater if the exit velocity of the organic waste stream, at the flare tip, is within the following limits[19]:

1. Nonassisted and steam-assisted flares
   a. If 200 Btu/scf < LHV < 300 Btu/scf (7 MJ/Nm^3 < LHV < 11 MJ/Nm^3),

$$V_{\max}(\text{ft/s}) = 60 \times \left(\frac{T(°\text{F}) + 460}{528}\right) \tag{11.6}$$

   b. If 300 Btu/scf < LHV < 1000 Btu/scf (11 MJ/Nm^3 < LHV < 37 MJ/Nm^3),

$$V_{\max}(\text{ft/s}) = \text{anti}\log_{10}\left(\frac{\text{LHV} + 1209.6}{849.1}\right) \times \left(\frac{T(°\text{F}) + 460}{528}\right) \tag{11.7}$$

   c. If LHV > 1000 Btu/scf (37 MJ/Nm^3),

$$V_{\max}\ (\text{ft/s}) = 400 \times \left(\frac{T(°\text{F}) + 460}{528}\right) \tag{11.8}$$

2. Air-assisted flare
   a. If LHV > 300 Btu/scf (11 MJ/Nm^3),

$$V_{\max}\ (\text{ft/s}) = \frac{(329.5 + LHV)}{11.53} \times \left(\frac{T(°\text{F}) + 460}{528}\right) \tag{11.9}$$

NOTE: For Equations 11.6 through 11.9, LHV is based on standard conditions of 68°F (20°C) and 1 atm.

11.5.1.2.1 Hydrogen Enrichment

Because hydrogen has a lower volumetric LHV than organic gases commonly combusted in flares, the U.S. EPA amended 40 CFR 60 to include an allowance for hydrogen content. The EPA believes that hydrogen-fueled flares, meeting the maximum velocity limitation as shown in the following, will achieve a combustion efficiency of 98% or greater:

$$V_{\max}\ (\text{m/s}) = (X_{H_2} - K_1) \times K_2 \times \left(\frac{T(°\text{C}) + 273}{293}\right) \tag{11.10}$$

where

$V_{\max}$ is the maximum permitted velocity, m/s
K_1 is a constant, 6.0 vol% hydrogen
K_2 is a constant, 3.9 (m/s)/vol% hydrogen
XH_2 is vol% hydrogen, on a wet basis

Equation 11.10 should only be used for flares that have a diameter of 3 in. (7.6 cm) or greater, are nonassisted, and have a hydrogen content of 8.0% (by volume) or greater.

Example 11.1

Given: A steam-assisted flare is burning a gas with an LHV of 450 Btu/scf and 160°F (71°C).

Find: The maximum exit velocity of the gas at the flare tip to achieve a combustion efficiency of 98% or greater according to the U.S. EPA ruling

Solution: Because the LHV of the fuel is 450 Btu/scf, Equation 11.7 is used to determine the maximum velocity:

$$V_{\max}(\text{ft/s}) = \left[\text{anti}\log_{10}\left(\frac{450+1209.6}{849.1}\right)\right] \times \left(\frac{160+460}{528}\right) = 106 \text{ ft/s} \qquad (11.11)$$

11.5.2 Emissions

Industrial flares have been endorsed by the Clean Air Act Amendments to be one of the acceptable control technologies that can effectively destroy organic vapors. The U.S. EPA AP-42 guidance document[7] suggests that a properly operated flare, with a combustion efficiency of 98% or greater, will emit UHC, CO, and NOx at the following rates:

- UHC = 0.14 lb/MMBtu fired
- CO = 0.37 lb/MMBtu fired
- NOx = 0.068 lb/MMBtu fired

In 40 CFR 60.8c, the regulations indicate that during periods of start-up, shutdown, or malfunction, emissions above the regulated limit may not be considered a violation. This includes UHC, CO, and NOx. This could also be interpreted to mean that exit velocity limits do not apply under these conditions. However, according to 40 CFR 60.10(c), states can make their own rules regarding flare operations as long as they are more stringent than the U.S. EPA ruling. As regulations are in a constant state of flux, the reader should determine the current regulations for the plant site in question.

Example 11.2

Given: Following the conditions of Example 11.1, assume that the flare is properly designed and operated according to the U.S. EPA ruling.

Find: The pounds of UHC, CO, and NOx emitted in 1 year if the flare is burning the waste gas at a rate of 10,000 lb/h (4500 kg/h), 50 times per year, for 1 h during each event (neglect the emission contribution from the pilots and purge gas). The density of the gas is 0.05 lb/scf.

Solution: First determine how many Btus are released in 1 year (year):

$$\frac{HR}{\text{year}} = 450\frac{\text{Btu}}{\text{scf}} \times \frac{1 \text{ scf}}{0.05 \text{ lb}} \times \frac{10{,}000 \text{ lb}}{\text{h}} \times \frac{1 \text{ h}}{\text{Event}} \times \frac{50 \text{ Events}}{\text{year}} = 4500\times 10^6 \frac{\text{Btu}}{\text{year}} \qquad (11.12)$$

The pounds of UHC, CO, and NOx emitted in 1 year are then calculated as follows:

$$\text{UHC} = 4500\times 10^6 \frac{\text{Btu}}{\text{year}} \times \frac{0.14 \text{ lb}}{1\times 10^6 \text{ Btu}} = 630\frac{\text{lb}}{\text{year}} \qquad (11.13)$$

$$\text{CO} = 4500\times 10^6 \frac{\text{Btu}}{\text{year}} \times \frac{0.37 \text{ lb}}{1\times 10^6 \text{ Btu}} = 1670\frac{\text{lb}}{\text{year}} \qquad (11.14)$$

$$\text{NOx} = 4500\times 10^6 \frac{\text{Btu}}{\text{year}} \times \frac{0.068 \text{ lb}}{1\times 10^6 \text{ Btu}} = 310\frac{\text{lb}}{\text{year}} \qquad (11.15)$$

11.5.3 Dispersion

If a flare fails to properly dispose of toxic, corrosive, or flammable vapors, it could pose a serious health hazard to personnel in the vicinity and the community downwind of the release. Sax[20] provides extensive information on many compounds that are sometimes sent to a flare. It is important for owners to have a dispersion analysis of their flare performed. A dispersion analysis is a statistical method used to estimate the downwind concentration of a gas vented to the atmosphere or emitted from a flare flame. Dispersion models are widely used in the industry and have been used in the past to size flare stack heights, estimate worst-case scenarios from emergency releases, and determine potential odor problems.

Mathematical modeling of stack gas dispersion began in the 1930s. At that time, these models were somewhat simplified. Today, however, through the advent of computers, these models have become more sophisticated and able to capture much more detail of the dispersion problem. The purpose of this section is to discuss the general concepts used for estimating the ground level concentration (GLC) of a pollutant emitted from a flare.

When a pollutant is emitted from a flare, it is dispersed as it moves downwind by atmospheric turbulence and, to a lesser extent, by molecular diffusion, as illustrated in Figure 11.55. The GLC of a pollutant downwind of the flare depends on how fast the pollutant is spreading perpendicular to the direction of the wind and on the height of the plume above the ground.[21] The rate at which a pollutant is dispersing, in turn, depends on such factors as the wind speed, time of day, cloudiness, and type of terrain. The Gaussian dispersion model was one of the first models developed to estimate GLC. This model assumes that the concentration of the pollutants, in both the crosswind

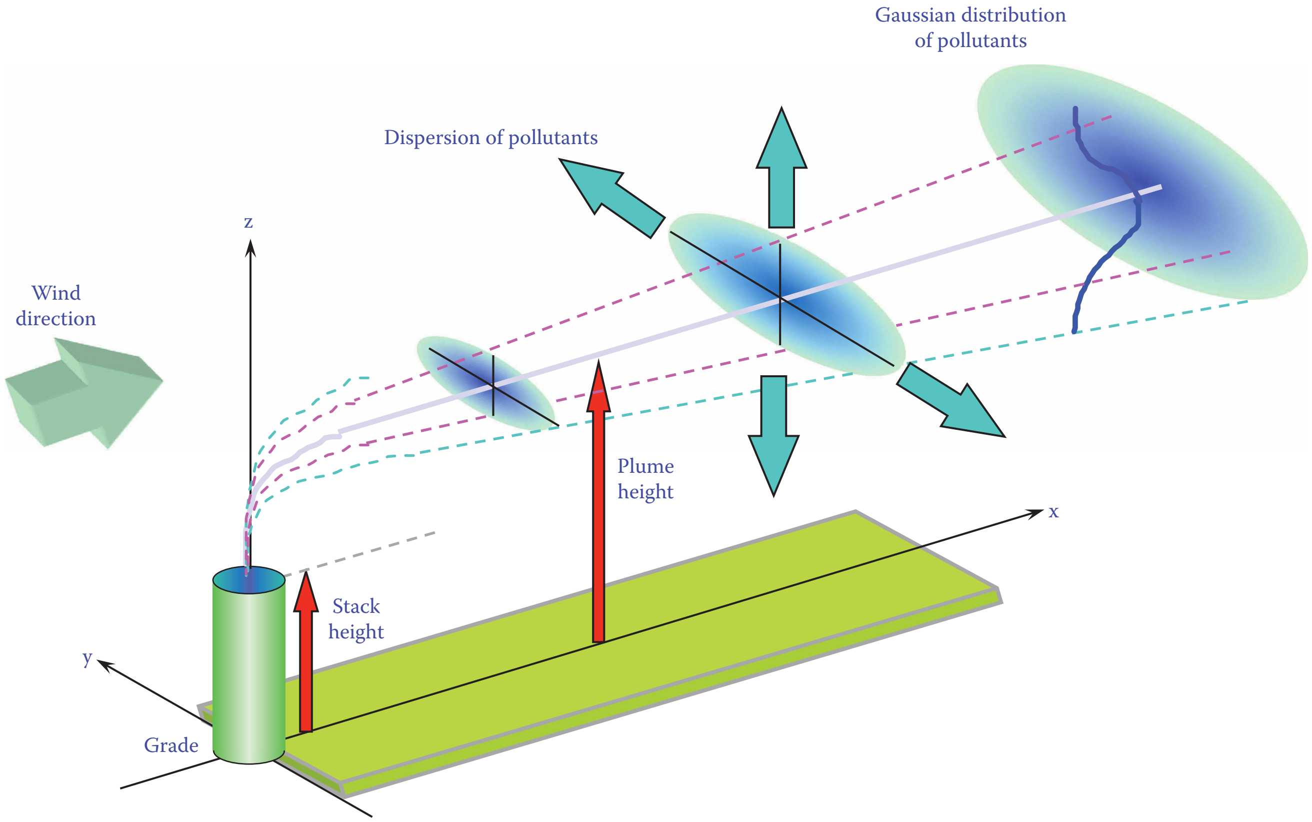


FIGURE 11.55
Geometry for plume dispersion calculations.

and vertical directions, takes the form of a Gaussian distribution about the centerline of the plume and is written as follows:

$$C = \left(\frac{Q}{U\sigma_z\sigma_y\pi}\right)\exp\left(\frac{-H^2}{2\sigma_z^2}\right)\exp\left(\frac{-y^2}{2\sigma_y^2}\right) \qquad (11.16)$$

where

C is the predicted GLC concentration, g/m^3
Q is the source emission rate, g/s
U is the horizontal wind speed at the plume centerline height, m/s
H is the plume centerline height above ground, m
σ_y and σ_z are the standard deviations of the concentration distributions in the crosswind and vertical directions, respectively, m
y is the crosswind distance, m (see Figure 11.55)

This Gaussian dispersion model was derived assuming a continuous buoyant plume, single-point source, and flat terrain. Beychok[22] discusses the shortcomings of Gaussian dispersion models. Beychok suggests that it is realistic to expect Gaussian dispersion models to consistently predict real-world dispersion plume concentrations within a factor that may be as high as 10. Gaussian dispersion models, however, are useful in that they can give a rough and fairly quick estimation and comparison of pollutant levels from elevated point sources.

The accuracy of a Gaussian dispersion model depends on how well one can determine the plume rise, H, at any given downwind distance and dispersion coefficients, σ_y and σ_z. A standard atmospheric stability classification method, known as the Pasquill–Gifford–Turner classification, is widely used in GLC models. This method categorizes the stability of the atmosphere into six classes that vary from very unstable (class A) to very stable (class F). An atmosphere that is stable has low levels of turbulence and will disperse a pollutant more slowly than an unstable atmosphere. The dispersion coefficients, σ_y and σ_z, are dependent on the amount of turbulence in the atmosphere and are, therefore, related to the atmospheric stability class. For more information on the equations describing the dispersion coefficients, see Turner.[23]

The plume height is defined as the vertical distance from the plume centerline to grade, as illustrated in Figure 11.55. There are several variables that can affect the plume height. These variables are divided into two categories: emission factors and meteorological factors. The emission factors include the (1) stack gas exit velocity, (2) stack exit diameter, (3) stack height, and (4) temperature of the emitted gas. The meteorological factors include the (1) wind speed, (2) air temperature with height, (3) shear of the wind with height, (4) atmospheric stability, and (5) terrain. None of the equations reported in the literature for estimating plume heights, however, take into account all the emission and meteorological factors. For a review of these equations, see Moses et al.[24]

GLC analysis is very complex because the results can depend on so many variables, as briefly discussed earlier. In the past, engineers and scientists have described GLC modeling as an art rather than a science. However, this paradigm is shifting due to more sophisticated computer models.[25] Due to the complexity of these models, one should consult an expert when requiring GLC analysis.

References

1. J.S. Zink and R.D. Reed, Flare stack gas burner, U.S. Pat. 2,779,399, issued January 29, 1957.
2. R.E. Schwartz and S.G. Kang, Effective design of emergency flaring systems, *Hydrocarbon Engineering*, 3(2), 57–62, 1998.
3. S.H. Kwon, D.I. Shin, D.D. Cobb, D.K. Kang, and C. Stacklin, Improve flare management, *Hydrocarbon Processing*, 76(7), 105–111, 1997.
4. R.D. Reed, *Furnace Operations*, 3rd edn., Gulf Publishing, Houston, TX, 1981.
5. API Standard 521, *Pressure-relieving and Depressuring Systems*, 5th edn., American Petroleum Institute, Washington DC, May 2008.
6. M.R. Keller and R.K. Noble, RACT for VOC—A Burning Issue, *Pollution Engineering*, July 1983.
7. U.S. EPA AP-42, *Compilation of Air Pollutant Emissions Factors*, 5th edn., Vol. 1, Section 13.5, Industrial Flares, Washington DC.
8. C. Baukal, J. Hong, R. Poe, and R. Schwartz, Large-scale flare testing, in *Industrial Combustion Testing*, C. Baukal (ed.), Chapter 28, CRC Press, Boca Raton, FL, 2011.
9. J.S. Zink, R.D. Reed, and R.E. Schwartz, Temperature-pressure activated purge gas flow system for flares, U.S. Pat. 3,901,643, issued August 26, 1975.
10. R.E. Schwartz and J.W. White, *Flare Radiation Prediction: A Critical Review*, John Zink Company, Tulsa, OK, 1996.
11. R.D. Reed, Flare stack burner, U.S. Pat. 3,429,645, issued February 25, 1969.
12. R.D. Reed, R.K. Noble, and R.E. Schwartz, Air powered smokeless flare, U.S. Pat. 3,954,385, issued May 4, 1976.
13. R.D. Reed, J.S. Zink, and H.E. Goodnight, Smokeless flare pit burner and method, U.S. Pat. 3,749,546, issued July 31, 1973.
14. R.E. Schwartz and R.K. Noble, Method and apparatus for flaring inert vitiated waste gases, U.S. Pat. 4,664,617, issued May 12, 1987.
15. W.R. Bussman and D. Knott, Unique concept for noise and radiation reduction in high-pressure flaring, *Offshore Technology Conference*, Houston, TX, May 2000.
16. J.S. Zink, R.D. Reed, and R.E. Schwartz, Apparatus for controlling the flow of gases, U.S. Pat. 3,802,455, issued April 9, 1974.
17. H. Glomm, Anordnung und Betrieb von Notabblasesystemen (blow down systems), *Rohrleitungstechnik in der Chemishen Industrie*, 199, 18–28, 1967.
18. McDaniel, Marc. Flare Efficiency Study. Rep. no. EPA-600/2-83-052. Washington DC, U.S. EPA, 1983.
19. U.S. EPA, Code of Federal Regulations, Title 40, Part 60, *Standards of Performance for New Stationary Sources*.
20. N.I. Sax and R.J. Lewis Sr., *Dangerous Properties of Industrial Materials*, 7th edn., Van Nostrand Reinhold, New York, 1989.
21. M. Miller and R. Liles, Air modeling, *Environmental Protection*, 6(9), 34–37, 1995.
22. M. Beychok, Error propagation in stack gas dispersion models, *The National Environmental Journal*, 6(1), 33–37, 1996.
23. B. Turner, U.S. EPA Office of Air Programs. Workbook of Atmospheric Dispersion Estimates. Washington DC. U.S. EPA, 1970.
24. H. Moses, G.H. Strom, and J.E. Carson, Effects of meteorological and engineering factors on stack plume rise, *Nuclear Safety*, 6(1), 1–19, 1964.
25. C. Seigneur, Understanding the basics of air quality modeling, *Chemical Engineering Progress*, 88(3), 68–74, 1992.

12

Pilot, Ignition, and Monitoring Systems

Adam Bader and John Bellovich

CONTENTS

12.1 Introduction

For most flare operations a release of waste gases can happen at any time and with little warning. As a result of these sudden release events, an integrated ignition system that can immediately and reliably ignite the flare is required. Many flares must be operated with a flame present at all times.[1] A typical integrated ignition system includes one or more pilots, pilot igniters, pilot monitors, and a means to stabilize the flame. A reliable ignition system is one of the single most important aspects of a flare tip's safe operation.

Before 1947, venting of unburned hydrocarbons (UHCs) to the atmosphere was an industry practice. When combustion of waste gases was required it was not uncommon to light flares with flare guns, flaming arrows, burning oily rags, or incendiary pellets. These methods were not very reliable and were potentially dangerous to other processes and personnel in the plant. In addition, they were not very effective at relighting flares under adverse weather conditions. In 1949 the John Zink Company developed the first pilot for use as a continually lit ignition source for combustion of released gases from a flare. Since then there have been numerous designs and improvements of pilot, ignition, and monitoring systems.[2] This chapter discusses pilots and the typical methods used for pilot ignition and monitoring.

12.2 Pilots

In principle, all flares should have a continuous pilot flame to ensure reliable burning. This is especially true of refinery, petrochemical, and production field flares because they are usually a safety device required for operation at all times. Also, such flares may be online for extended periods of time that often exceed years of operation before there is an immediate need for reliable ignition. Intermittent pilots should only be considered in special applications where each release is controlled and there is no risk of any unplanned venting. Notable exceptions to continuous pilots are landfill flares or biogas flares and flares used in batch processes.

12.2.1 Premixed Pilots

A flare pilot is typically a premixed burner designed to operate over a specific heat release range. In operation, the pressure energy of the pilot fuel is used to aspirate ambient air into the mixer inlet, mix the fuel and air, and propel the mixture through the downstream section and out the pilot tip. The key goals for a properly designed pilot system are to

1. Provide reliable ignition of the main flare flame
2. Be capable of reliable ignition of the pilot(s)
3. Provide high pilot flame stability (difficult to extinguish)
4. Provide a long service life

To achieve these goals, a pilot must (1) meter the fuel and air, (2) mix the fuel with the air, (3) mold the desired flame shape, and (4) have a means of stabilizing the pilot flame to protect the system from external interference.

Typically, pilots consist of four fundamental parts: a gas orifice, a mixer or venturi, a downstream section that connects the mixer and the tip, and a tip (see Figure 12.1). All components of a pilot are carefully designed to work together as a system to achieve proper performance. A change in any component can affect the balance of the system and hence the operation of the pilot.[3]

Pilot gas consumption varies according to the specific flaring requirements. However, there is a practical lower limit to the pilot gas consumption. If the volume of air aspirated into the pilot falls outside of the flammability limits of the fuel gas, the pilot will not operate properly. For example, methane requires 5.7–19 volumes of air per volume of fuel in order to burn. If a pilot is operating below this volumetric air/fuel ratio limit, air external to the pilot must be available in order for the fuel to burn. If the pilot tip is surrounded with inert gas from the flare, then the pilot cannot be lit nor maintain combustion. Conversely, if the pilot is operating with a volumetric air/fuel ratio above 19, the pilot will not ignite since the air/fuel mixture is too lean. Operation very near the maximum allowable air/fuel ratio can make the pilot difficult to light, and the pilot may be unstable in windy conditions.

The number of pilots required for a given system can vary, depending on the size and type of flare burner and

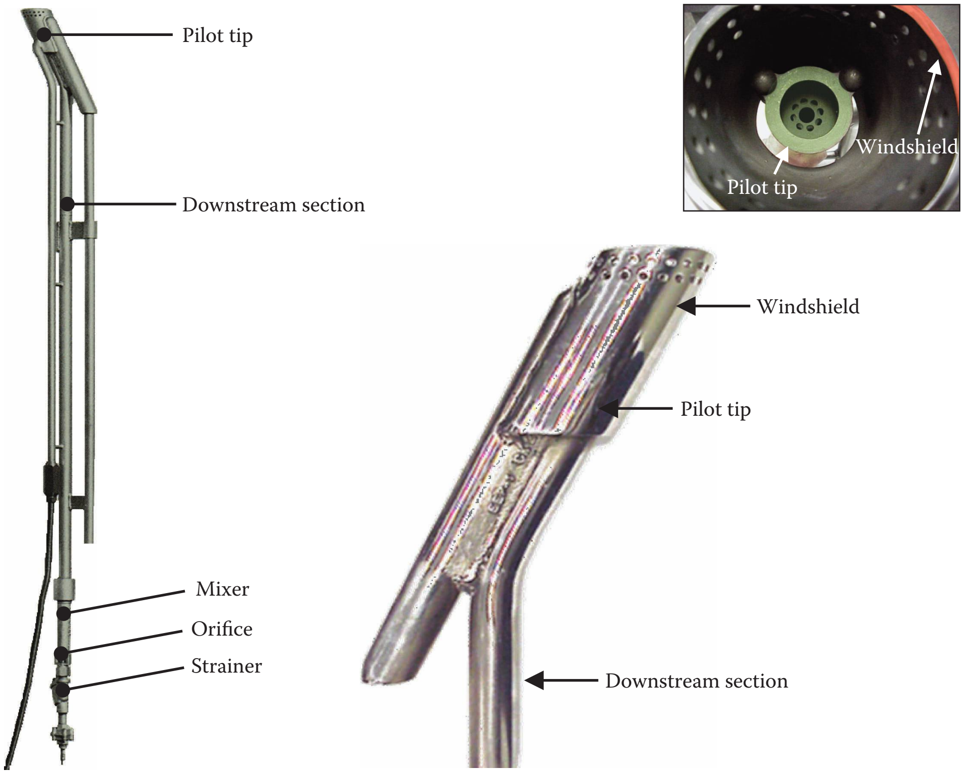


FIGURE 12.1
Pilot schematic.

TABLE 12.1

Recommendation for Minimum Number of Pilots from API SP 537 Document

Minimum Number of Pilots Recommended	Nominal Tip Diameter (in.)
1	From 2 to 8
2	From 10 to 24
3	From 30 to 42
4	From 48 to 60

its intended use. Premixed pilot burners are designed such that pilot fuel and air are mixed together at a point remote from the flare burner exit and delivered through a pipe (downstream section) to the pilot tip for combustion. This helps isolate the pilot from the flare gas conditions at the flare burner exit (e.g., the presence of flue gas, inert gas, or steam). Premixed pilots also allow for a more wind/rain resistant design, which is why some designs can operate in over 150 mph wind conditions. A good industry standard for the proper number of pilots, fuel flow heat release, and recommended fuels can be found in API 537[4] (see Table 12.1).

A steady fuel flow from a reliable source is required for pilots in continuous operation. Fuel filtration systems, such as strainers, are recommended to ensure the small fuel port in the pilot does not plug from contaminates in fuel or from scaling inside pipes. Typical pilots include a strainer just upstream of the orifice; often it is located near a platform for ease of access. Because the pilot is only accessible during shutdown, it is often recommended to locate an additional strainer at the base of a flare system. The strainer at grade will allow for routine access and cleaning and reduce the amount of particles that can reach the upper strainer.

12.2.2 Slipstream Pilots

Some pilots utilize a flow of combustible gas from the main flare; in this arrangement the flare gases become the pilot fuel. These gases are typically lit with a type of electronic sparking device. While this is not common in the industry, there are instances where this arrangement can produce a flame capable of igniting the main process tip. The drawback to these types of systems is that many flaring events are uncontrolled and vary in process flow and composition, and thus the pilot fuel is difficult to maintain and control. This can limit the effectiveness of such a pilot and result in an inability to maintain a pilot flame or ignite the main flare when the situation is required.

12.2.3 Pilot Requirements and Maintenance

A flare pilot must be able to withstand rain, wind, heat from the flare flame, and direct flame contact (see Figure 12.2). Each of these operating conditions can be extremely variable and is unique to each system. Common pilot problems are failure to light and burn with a stable flame, flashback, fuel line plugging, and poor operation in windy conditions. Often times the cause of these pilot issues is the result of poor maintenance. Routine inspection and review of a system is required for good service life.

FIGURE 12.2
Pilot operating in high wind and rain conditions.

12.3 Ignition

Reliable combustion of a pilot begins with the ignition system. As a safety consideration, pilot ignition is usually initiated from a position remote from the flare tip, often at the base of the stack or a set distance away based on the expected heat release from a major flaring event. There are various methods of achieving ignition.

12.3.1 Flame Front Generator

A flame front generator (FFG) ignition system is the most common method of lighting flare pilots. An FFG combines ignition fuel and compressed air in a mixing tee, and the air/fuel mixture flows through an ignition line to the pilot tip. The ignition line can exceed 5000 ft (1500 m) in length. FFG ignition is initiated by purging the ignition line with the air/fuel mixture. The air/fuel mixture is regulated to a proper ratio with a typical air/fuel mixture for natural gas of 10:1. An ignition chamber containing a spark plug is located immediately downstream of the mixing zone. Once a spark has been initiated, the air/fuel mixture will ignite and the flame front will travel the length of the piping until it exits at the pilot tip and ignites the pilot fuel. This flame front can travel at speeds of up to 70 ft/s (21 m/s) and is unimpeded by pipe directional changes.

For proper ignition with an FFG system, it is important to allow the ignition line to completely fill with the air/fuel mixture before a spark is generated. If the FFG ignition line is not properly filled with the air/fuel mixture, the flame front may extinguish before it reaches the pilot exit. Flexible tubing lines are not recommended for this service. Exceptionally long FFG ignition lines may take minutes to properly fill with the flammable mixture depending on the air/fuel flow rates, line size, and distance. For example, if the velocity of a typical air/fuel mixture in the FFG line is 50 ft/s (15 m/s) and the FFG line is 1000 ft long (300 m), the air/fuel mixture should flow for about 20–30 s before attempting to light the pilot. Each attempt to light the pilot in this example should allow 20–30 s for the line to refill with the air/fuel mixture. It is also important to note that the spark should only last for the brief duration required to initiate the flame front. Continuous sparking can cause a stable flame within the mixing chamber producing excessive heat on the downstream piping and valves that can damage the system.

12.3.2 Self-Inspirating FFG

The self-inspirating FFG is a variation of the standard FFG system in which an air/fuel mixture is generated at grade using an eductor system. This eductor is separate from the main pilot venturi mixer. A spark, generated just downstream of the igniter eductor, creates a flame front inside the ignition line that leads to the pilot tip. The main advantage this system has over the conventional FFG system is that compressed air is not required. In addition there is a variation of this design referred to as a piezoelectric sparking system that does not require a power supply. The disadvantage of these systems is that the maximum distance of the ignition line is limited to approximately 200 ft (60 m). The exact distance, however, can vary, depending on the fuel pressure, composition of the fuel, diameter and wall roughness of the ignition line, and ambient air density. In addition, these systems can be more susceptible to ignition failures caused by ambient conditions due to the method of air induction.

12.3.3 Slipstream FFG

The slipstream FFG is another version of the standard FFG system where ignition takes place on the pilot itself. A portion of the air/fuel mixture generated by the pilot venturi is diverted to a tube located adjacent to the pilot, as shown in Figure 12.3. This slipstream travels through the tube and exits near the pilot tip. A high-energy discharge ignition probe is used to ignite the mixture, generating a flame front within the slipstream line that in turn ignites the pilot. The main advantages that this system has over the conventional FFG system are quick pilot relight, no flame front ignition lines, and no compressed air required. Another advantage of this design is the location of the ignition probe as it is located away from the combustion zone while also being cooled by the flowing air/fuel mixture. The main disadvantage is that critical components are located at the flare tip and therefore inaccessible without a flare shutdown. The distance between the transformer and the ignition probe is limited to approximately 1000 ft (300 m).

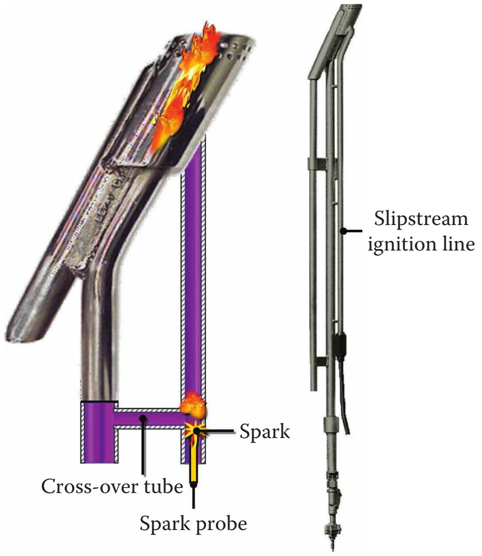


FIGURE 12.3
Slipstream-type ignition.

12.3.4 Direct Spark

Direct spark ignition systems operate very similar to the slipstream FFG, in that an ignition probe is used to energize fuel gas at the pilot tip. The difference is the direct spark ignition occurs at the immediate exit of the pilot. This allows for a very quick ignition response time. Direct spark ignition shares a similar disadvantage as the slipstream FFG in that critical components are located at the flare tip and therefore inaccessible without a flare shutdown. The distance between the ignition transformer and the spark probe is limited to approximately 1000 ft (300 m). In addition, direct spark ignition may expose a portion of the ignition probe to flame impingement from the pilot or main flare flame.

This impingement can damage the ignition probe over time, whereas the slipstream FFG ignition probe is protected inside the guide tube and shielded from direct flame contact. An advantage of this type of ignition system is that the wire used for ignition purposes can also be used for monitoring via flame ionization.

12.3.5 Ballistic Ignition

A ballistic ignition system utilizes an incendiary type of device that travels along a guide tube to detonate at a position near the flare tip. The maximum distance between the load/launch apparatus and the flare tip is approximately 500 ft (150 m). When ignition of the flare is required, a mechanism activates the incendiary pellet and compressed air propels it along polished tubing toward the flare tip. Upon exiting the guide tube, the incendiary device detonates and creates hot particles that ignite the flare. These types of systems have been used on production facilities and batch processes when on-demand flaring can be controlled. As these systems only provide one ignition attempt per incendiary pellet, proper installation is critical to ensure the device is able to traverse the length of tubing without becoming lodged in place. During ignition the incendiary device produces debris. This debris must be properly contained or the flare must be in a remote location in order to not harm personnel or equipment. Another area of concern with ballistic systems is the timing of the pellet launch. If the ballistic pellet is launched too early, the flare gases will not have reached the flare tip, and thus the hot particles will not ignite the flare. If the pellet is launched too late, a vapor cloud can form that can produce a detonation upon ignition.

12.3.6 Multiple Ignition Systems

All of the noted types of ignition can be employed on a single pilot or any combination as required. For example, many of the pilots equipped with a slipstream FFG or direct spark ignition are also equipped with a conventional FFG. In this case, the conventional FFG is used as an installed backup ignition system. The advantage of multiple ignition systems on a given pilot is redundancy, which improves safety and reliability.

12.4 Monitoring

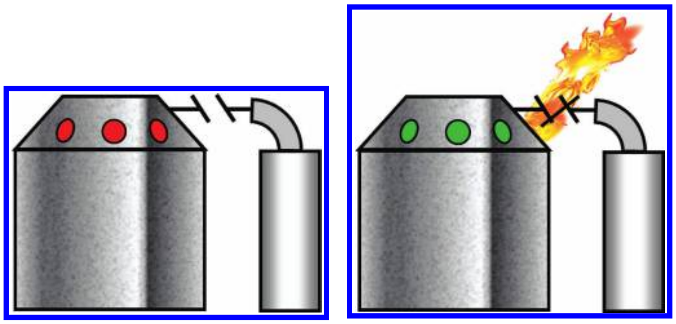
A pilot monitor is always recommended and often required to verify the pilot flame. The pilot's remote location and inaccessibility during flare operation make flame verification difficult. A pilot flame produces four by-products during combustion: heat, ionized gas, light, and sound—all of which present a means of detection.

12.4.1 Thermocouples

By nature, one immediately associates flame with heat. Heat can be measured at the end of a pilot tip with a thermocouple, which is the most common detection method. The thermocouple is connected to a temperature switch or a control system that indicates pilot temperature. Flame failure is indicated if the temperature drops below a given set point. Typical pilot failure set points range from 300°F–600°F (150°C–320°C). A balance must be struck by the manufacturer when setting the position of the element on the pilot between a high exposure to heat with possible rapid thermocouple failure and a lower exposure with a slower response time. A thermocouple's expected life span is limited based on the conditions to which it is exposed. A downside to fixed thermocouples is that a shutdown is required to replace a failed element.

12.4.2 Ionization

Ionized gases are typically measured with a type of flame rod. Ions from the flame are released during combustion and act as electrical conductors. An AC current is produced by a transformer at grade and connected to a flame rod on the pilot. When a flame is present, the ions act as a conductor from the flame rod to the pilot and create a rectified AC signal to indicate the presence of a flame. When the flame is lost, the circuit opens and the control system recognizes the pilot failure, prompting an alarm for relighting (see Figure 12.4). Like a thermocouple, these elements cannot be maintained during flare operation. In addition, the required location for measuring of flame ions requires a portion of the flame rod to be in the combustion zone that can damage the system over time.


Open circuit = Flame out

Closed circuit = Flame on

FIGURE 12.4
Flame ionization.

FIGURE 12.5
John Zink infrared grade-mounted camera.

FIGURE 12.6
Pilot operating at night.

12.4.3 Light: Infrared

When a pilot flame is burning, both visible light and infrared energy are emitted. A specially designed infrared camera mounted at ground level can be used for flame monitoring. However, optical methods may be unable to distinguish pilots from the main flame or one pilot from another. In addition, the optical path can be obscured by heavy rain, fog, snow, or the movement of the top of the flare stack may move the flame out of the sensor's field of view. An advantage of this type of system is its maintainability while the flare is in service (see Figure 12.5).

12.4.4 Light: Visible

Most pilot fuels produce a low-luminosity flame because the gas mixture at the pilot tip contains close to 100% of the air required by the fuel. As a result it can be very difficult to see a pilot flame during the day with the human eye or standard optical system/camera. Viewing at night is generally more successful. If the pilot is ignited using a conventional FFG, opening the fuel valve of the FFG can enhance visual sighting, day or night. The added fuel will produce a larger and more luminous flame at the pilot. After the pilot flame has been sighted, the extra fuel should be shut off. While this is a very manual means of detection, it is a good option when all other methods fail. It should be noted that this means of detection is not formally approved when monitoring is required by regulation (see Figure 12.6).

12.4.5 Acoustic

Sound can also be used as a monitoring method. These types of systems use an ignition line as a waveguide to carry sound from the pilot to grade where a special sensor is installed to detect a specific acoustic signature produced by the pilot when burning. Acoustic data are conveyed from the sensor to a signal processor via

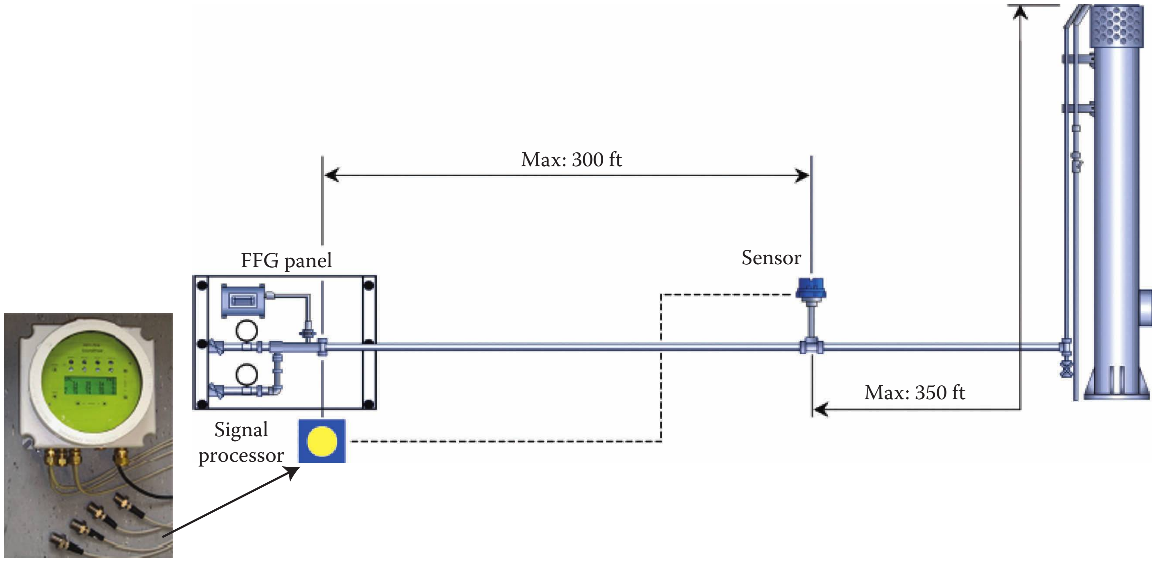


FIGURE 12.7
Acoustic pilot monitoring.

a cable. The signal processor analyzes the acoustic data and indicates the pilot flame status (see Figure 12.7). An acoustic pilot monitor can distinguish its connected pilot from nearby sound sources such as other pilots, steam injection, and combustion of the flare. Advantages of this design are that weather conditions do not adversely affect the monitor and the equipment can be mounted at grade for easy access and maintenance. Table 12.2 compares various flare pilot detection systems.

12.5 Retractable Pilot and Thermocouple Systems

As access to a flare pilot is usually limited to shutdowns, it can be years before a pilot can be removed or repaired. However, there can be situations where replacement or repairs are needed for a pilot system before a planned shutdown. For this reason retractable pilot systems

TABLE 12.2
Comparison of Flare Pilot Detection Systems

Method	Thermocouple	Flame Ionization	IR Sensor	Acoustic
Sensed phenomenon	Heat	Electrical current	Light	Sound
Able to distinguish between individual pilot flames?	Yes	Yes	No	Yes
Able to distinguish pilot flame from main flare flame?	Partially	Yes	No	Yes
Average response time (s)	100–300	<5	<5	5–10
Location of key system components	Pilot tip	Pilot tip and grade	Grade	Grade
Installation while flare is operational?	No*	No	Yes	Yes
Serviceable while flare is operational?	No*	No	Yes	Yes
Resistant to				
Wind	No	Yes	Yes	Yes
Rain	No	Yes	No	Yes
Fog	Yes	Yes	No	Yes
Snow	Yes	Yes	No	Yes
Steam	No	Yes	Yes	Yes
Sunlight (direct or reflected)	Yes	Yes	No	Yes

*No, fixed; Yes, retractable.

have been developed that allow for complete removal and reinstallation of a pilot on a flare structure while the flare is in service. These systems typically consist of a custom pilot bracket that is raised and lowered with a configuration of pulleys and cables. Retractable systems typically accommodate stack heights up to 600 ft (180 m) in height. A retractable system can be designed to accommodate all pilots on a flare tip or only one to ensure the capability of at least one functional pilot.

Some flare operators may not desire a fully retractable pilot system; however proving pilot flame is still a priority. In these cases a retractable thermocouple system may be used. A retractable thermocouple system uses a thermocouple that is approximately the same height as the stack. The thermocouple is run in polished tubing that acts as a guide from grade to the thermowell located on the pilot. This system allows a thermocouple to be removed and reinstalled from grade or a safe access point while a flare system is in service. The inserting process typically uses a device that straightens the thermocouple before entry into the guide tube then mechanically pushes the thermocouple through the guide tube. The size of the guide tube to the thermocouple diameter is engineered to ensure the thermocouple remains straight once inserted. As with other guide tube systems, having a properly installed guide tube is critical in order to prevent binding.

References

1. Code of Federal Regulations, General control device requirements, 40 CFR 60.18, CFR, Washington, DC (1986).
2. Bellovich, J., Franklin, J., and Baukal, C., The last line of defense, *Hydrocarbon Engineering*, 11(4), 47–54 (2006).
3. Bader, A., Baukal, C., and Bussman, W., Selection the proper flare systems, *Chemical Engineering Progress*, 107(7), 45–50 (2011).
4. American Petroleum Institute, *Flare Details for General Refinery and Petrochemical Service*, 2nd edn., API Standard 537, API, Washington, DC (December 2008).

13

Biogas Flaring

Tim W. Locke, Brandy S. Johnson, and Jason P. Rolf

CONTENTS

13.1 Biogas Fundamentals

13.1.1 Introduction

Biogas generally is considered a gas with primary constituents of methane and carbon dioxide resulting from the decomposition of organic matter, which occurs in an anaerobic (without oxygen) atmosphere. Typical sources for biogas production include biological degradation of refuse, or municipal solid waste (MSW) in landfills along with agricultural waste, livestock waste, and organic waste from sewage in a digestion process. While the landfill process generally develops methane concentrations averaging 50%, the digestion process generates higher methane levels ranging from 50 to 75%. For either landfill gas or digester gas, enough methane is produced to sustain thermal oxidation or combustion.

13.1.2 Biogas Formation

The generation of biogas in landfill applications is first dependent on the depletion of oxygen within the refuse, and further optimum levels of moisture, temperature, acidity, and alkalinity in the deposited waste. Once these levels are established methanogenic microorganisms, or bacteria, begin decomposing the waste in an anaerobic atmosphere producing methane (CH_4) and carbon dioxide (CO_2) as by-products. The amounts and composition of biogas produced in a landfill are dependent on the type and composition of refuse. The methane CH_4 concentration in landfill biogas typically ranges between 45 to 55% with the remaining gas being mostly carbon dioxide (CO_2).

Similarly, the production of biogas from sewage, agricultural waste, or food waste commonly is referred to as digester gas because it is produced in an airtight container or "digester." This digester may be a container, storage tank, or a lagoon with a membrane cover. The digestion process producing digester biogas is similar to the biogas production in a landfill, but within more controlled conditions. The biodegradable waste is collected in the digester along with methanogenic microorganisms, or bacteria, and maintained at an ideal temperature in an anaerobic atmosphere with appropriate amounts of agitation. The microorganisms decompose the waste and produce the methane and carbon dioxide mixture. Methane (CH_4) concentrations in digester gas typically range between 60 to 70% with the remaining gas being mostly carbon dioxide (CO_2).

13.1.3 Biogas Characteristics

Biogas mainly is comprised of methane (CH_4) and carbon dioxide (CO_2), but will include trace amounts of other organic and nonorganic substances and could even have small percentages of oxygen (O_2) and nitrogen (N_2). The amount and composition of substances other than methane and carbon dioxide will be determined by the type of the waste being decomposed and the decomposition process. If biogas is being collected for use or flaring, then oxygen and nitrogen may be introduced during the collection process.

Other substances in the gas are generally a direct result of what is being decomposed. For example, if the waste material contains sulfur such as in the case of construction drywall (calcium sulfate), the sulfur in the molecule will attach to a hydrogen radical and form hydrogen sulfide (H_2S). It is such hazardous air pollutants (HAP) and non-methane organic compounds (NMOC) which become dangerous and even deadly at higher concentrations. For landfills, the Environmental Protection Agency (EPA) established NMOC criteria in the Code of Federal Regulations (40 CFR 60, subpart WWW-Standards of Performance for MSW Landfills). In certain instances, if landfill gas is collected, NMOC levels must be reduced by 98 weight percent only when an elevated flare is operated. When operating an enclosed flare, either the same 98% reduction applies, or NMOC concentrations must be less than 20 ppm when measured on a dry basis as hexane and corrected to 3% oxygen.

13.1.4 Collection of Biogas

Different methods are available for collecting biogas, depending on whether it is digester gas or landfill gas. For digester gas, collection is relatively easy since the biogas is produced in an confined volume such as a storage tank, container, or even a covered lagoon. Because of the fixed volume available, the release of gases during the digestion process will create a pressure in the storage atmosphere. The amount of pressure generated in a storage tank is generally low, around 10–20 in. water column (25–50 mbar); however, a lagoon with a membrane cover will sustain higher pressures around 5–10 psig (0.3–0.7 bar), sometimes even more. With the elevated pressure, collection becomes simplified. Only a header pipe with a pressure or flow control device is needed to transport the gas to an end device.

Landfill gas collection is more complicated than digester gas collection. With landfills, the biogas is produced underground, so creating a pressurized environment becomes both an advantage and disadvantage. As the pressure builds, it will travel in the path of least resistance, which may be upward, downward, laterally, or any other direction possible. If biogas travels laterally, migration into adjacent houses, basements, or businesses becomes a concern. If it migrates downward, ground water contamination may occur.

Two basic methods for collecting landfill biogas involve either a passive collection system or an active collection system. The passive collection system simply utilizes pressure generated by decomposition in the landfill to promote the collection process. Various techniques are available for accomplishing passive collection, but often involve drilling wells either around the perimeter of the landfill itself or throughout the actual property. A perforated polyvinyl chloride (PVC) or high-density polyethylene (HDPE) pipe is located in the well with aggregate surrounding the pipe to allow gas flow into the pipe. Pressurized gas in the landfill will enter the perforated pipe and escape to the surface for either venting to atmosphere or combustion with a passive flare.

The active collection system introduces some driving force to extract landfill gas. Most commonly, vertical, wells are drilled throughout the landfill, and perforated PVC or HDPE piping is installed similar to the passive system. Once vertical piping is in place, a cap with a valve, referred to as a wellhead, is attached to the top of each vertical pipe. Each wellhead is then connected together with horizontal PVC or HDPE header piping. This pipe header is then routed to a flare station. At the flare station, a gas blower or blowers will generate a vacuum on the header, and the valve on each wellhead is then adjusted to control the flow through each well. Because the decomposition process is anaerobic, it is crucial to control the vacuum on the system to avoid pulling outside air into the landfill. Not only might intrusion of ambient air diminish the decomposition process or kill the bacteria, it creates a scenario with heat, oxygen, and methane, which occassionally produces an underground fire and is extremely difficult to extinguish. Many situations contribute to daily, weekly, or monthly problems for extracting biogas from the landfill, including atmospheric conditions such as barometric pressure, precipitation, freezing weather, or heat. Fluctuating conditions influence gas production in an area of the landfill for various reasons, so periodic monitoring at, and adjusting of, wellhead valves becomes critical for collecting the gas.

In addition to collecting landfill gas, proper collection of the liquids produced in the landfill is also important. These liquids include both condensate and leachate. Because the anaerobic decomposition process is a chemical reaction, heat is generated to temperatures of and exceeding 150°F (66°C). This hot subsurface gas is saturated with water, so when it rises in the collection system to the surface and cools, water condenses and collects in the header pipes. This condensate must then be drained into sumps, traps, or tanks to prevent slugs of water from accumulating in piping and restricting or obstructing gas flow. Fluctuating, or surging, gas flow often results when enough condensate unknowingly collects in a gas pipe.

Leachate is formed when surface water drains into the landfill waste and absorbs particulates, organics, and soluable inorganics. This leachate will accumulate at the bottom of the landfill, and if enough is formed, it may restrict biogas production by filling the wells with liquid causing extraction and collection issues. Frequently, leachate is very corrosive and odorous and must be treated with special disposal techniques.

13.2 Biogas Flares

13.2.1 Purpose and Incentives

For the majority of biogas users in the United States, controlled combustion of biogas is required. Sites within the United States are commonly regulated by the EPA or even state and local codes to actively collect, treat, dispose of, or use biogas. Whether a biogas user is subject to rules and regulations largely depends on their location and size. For smaller sites that fall below a certain threshold, implementing a biogas flare system can become an incentive. These sites may also install an active collection and flare system for odor control or because of their proximity to neighborhoods, schools, or parks.

Biogas is generated by from human-related activities including landfills, anaerobic digesters, sludge pits, livestock manure management systems, and composting. Most biogas in the United States is produced in landfills; as of 2009, this was 59%.[1] Landfills in the United States are subject to EPA New Source Performance Standards (NSPS), including 40 CFR 60, Subpart WWW—Standards of Performance for Municipal Solid Waste Landfills and 40 CFR 60, Subpart Cc—Emission Guidelines and Compliance Times for Municipal Solid Waste Landfills. If a landfill has a design capacity that exceeds 2.5 million cubic meters (88 million cubic feet), then the landfill owner is required calculate the NMOC levels to determine if it exceeds 50 Mg/year (110,000 lb/year). If this limit is reached, then the landfill must collect and control landfill gas rather than venting it to the atmosphere. Controlling landfill gas largely consists of an active extraction system paired with a flare device or a device for beneficial use (e.g., turbines, engines, boilers).

For biogas sites not subject to national, state, and local regulations, owners are able to implement biogas systems for incentives such as carbon credits or tax credits. The United States has a few regulations that offer voluntary emission reduction programs. There are also programs outside of the United States that allow developing countries to earn credits by reducing their emissions through biogas flare systems to trade or sell to industrialized countries. These industrialized countries then can use these credits as offsets to reduce their overall emissions. These types of programs give larger biogas emission producers the ability and flexibility to meet their emission limits.

Credits are typically based on the amount of methane destroyed. One ton of methane is equal to more than 20 tons of carbon dioxide. This conversion is called

CO_2e or carbon dioxide equivalent. Because methane has a value of 21, biogas is an excellent candidate for incentivized projects. Furthermore, each program will give various values depending on the type of flare system used. The majority of the time, enclosed flare devices will offer a larger earning of credits. The reason for the variation is largely dependent on the ability to measure methane destruction efficiency, which will be discussed later in this chapter.

13.2.2 Types

The four main types of biogas flares are passive, elevated, enclosed, and low emission flares. These flares are applicable for all biogas applications including landfill, agriculture, industrial wastewater, and municipal wastewater. Various regulations, incentives, policies, or the site location can dictate the type of flare that can be installed. Benefits, cost, maintenance, and design requirements vary greatly between the types of flare systems.

13.2.2.1 Passive Flares

Passive flares are widely used for small, unregulated biogas applications. These flares are generally designed for a gas flow rate of 100 SCFM (2.8 SCMM) or less. They offer the least expensive option for burning biogas or eliminating odors. They are attached directly to gas wells within the waste. Because they typically employ a battery-powered ignition mechanism that sparks at a constant interval to ensure that venting gas is combusted, they do not require power or other controls in order to operate (see Figure 13.1).

Most passive flares are used within the landfill industry. These flares are designed and normally used to combust gas that passively vents from a vent pipe, such as a gas vent within the waste, leachate collection pipe in which gas may accumulate, or leachate collection manholes. Per EPA 40 CFR 60.752(b)(2)(ii)(B), landfill owners do have the option of controlling emissions using a passive collection system. If they choose to install any type of flare, the flare must meet the rules established within this regulation. The rule stipulates that the flare meet the following three criteria:

- Include a heat sensing device, such as an ultraviolet (UV) beam sensor or thermocouple, at the pilot light or the flame itself to indicate the continuous presence of a flame (40 CFR 60.756(c)).
- Operate in a manner such that if the flare is inoperable, the system must shut down to prevent the free venting of gas within 1 h of the flare shutdown (40 CFR 60.753(d)).
- Operate whenever gas is routed to the flare (40 CFR 60.753(f)).

FIGURE 13.1
Solar powered passive flare. (Courtesy of Solar-Spark by LSC Environmental Products, LLC.)

Passive flares do not typically include monitoring equipment (UV sensors or thermocouples), recording equipment (chart recorders), or a programmable logic controller (PLC) that would ensure the flare operates only when gas is routed to it. The majority of passive flares do not contain automatically closing valves or isolation valves that close when the operation discontinues of faults to prevent free venting of gas out the flare.

For a passive flare to comply with the operational requirements of the previously mentioned regulations, it would be necessary to modify the flare so that it includes the additional control equipment. Because of the cost and effort required to add this equipment and demonstrate to regulatory agencies that the flares will meet these requirements, passive flares are rarely used for NSPS compliance.

13.2.2.2 Elevated (Open) Flares

Elevated flares have many different names including open (see Figure 13.2), candlestick, utility, or pipe flares. These flares have an exposed flame, are less expensive than other flare systems, and are usually approved for most applications or regulations. Specific regions in the United States will not allow elevated flares based on their local regulations. Elevated flares are considerably cheaper than other types of flares; therefore, they are typically chosen for their cost savings.

Elevated flares do not offer controlled combustion, meaning the air required for proper combustion for various gas flows is not automatically adjusted. In addition, the flame is exposed to outside conditions (e.g., rain, wind) and not contained in an enclosure. Both of these factors contribute to a less stable flame, which is more susceptible to flame loss.

Elevated flares are not source tested or regulated. In the United States, two EPA documents are most commonly used as the guideline for emissions and destruction efficiency. Destruction efficiencies based on maximum exit velocity and gas heating value are defined by EPA 40 CFR 60.18. If flares are designed with an exit velocity not exceeding what is listed in this regulation, then the flare will yield a 98% destruction efficiency. The second document from the EPA that is commonly used is AP-42, *Compilation of Air Pollutant Emission Factors*, 5th edition, Section 13.5, for industrial flares. As listed in table 13.5-1, emissions are 0.068 lb/MMBtu for nitrogen oxides (NOx) and 0.37 lb/MMBtu for carbon monoxide (CO) where MMBtu represents million Btu. These factors were developed by actual laboratory testing by the EPA on open flares.

FIGURE 13.2
Elevated flare.

There is another AP-42 document, for MSW Landfills (5th edition, Section 2.4), which is published but not commonly used for guidelines on emissions on elevated flares. This document states that NOx is 0.04 lb/MMBtu and CO is 0.74 lb/MMBtu, converting the values given in table 2.4-5. While these two documents conflict each other, the AP-42 for industrial flares is considered more accurate; the main reason being the source of the data used for the MSW (Section 2.4). As indicated in the footnotes of table 2.4-5, these data were developed from actual source tests on *enclosed flares*. Many of the source tests found in the reference section are from the 1980s, which was a time where combustion was not as controlled or efficient as it is today. High CO values reflect these inefficiencies. CO increases with lower operating temperatures and inefficient burning. If one assumes that elevated flares have similar combustion as enclosed flares that are tested today, then the AP-42 used for industrial flares is more representative of the values used for enclosed flares (Section 13.2.2.3).

Since they are not able to be tested and regulated, various carbon credit programs will give a reduced amount of available credits for elevated flares due to the fact that their destruction efficiency is either believed to be less or cannot be proven to meet or exceed 98% destruction. Elevated flares cannot maintain a specific operating temperature; therefore, it is difficult to prove a constant destruction rate (see Figure 13.3).

Even if local regulations allow the use of elevated flares, they may not be chosen due to the fact they produce a visible flame and produce radiation. Both of these factors will limit the location of the flare, which may disqualify the option for installing this type of flare. Examples include locations by highways, residential areas, or other locations where power lines or structures are within close proximity.

13.2.2.3 Enclosed Flares

Enclosed flares, as shown in Figure 13.4, offer controlled combustion of the biogas and a hidden flame. They offer air dampers with actuated louvers which modulate to maintain a specific operating temperature to ensure proper destruction of the biogas. This operating temperature can be selected by the user and modified to achieve various emissions. Enclosed flares have sample ports toward the top of the stack for emission testing. With the ability to test emissions and measure the operating temperature, these devices are believed to offer superior performance and

FIGURE 13.3
Unconfined flame extending beyond the windshield.

destruction efficiencies. A higher operating temperature yields higher methane destruction; therefore, many carbon credit programs offer more earnings for these devices.

Standard enclosed flares operating within 1400°F–1800°F (760°C–980°C) yield 98%–99% destruction rates for NOCs, 99% or greater expected destruction rates for methane, NOx of 0.06 lb/MMBtu, and CO at 0.2 lb/MMBtu. These values exceed those of an elevated flare. If lower emissions are required, it is possible to modify the enclosed flare to achieve slightly better results or implement a low emission enclosed flare, which is discussed in the next section.

Since the combustion is taking place within a stack or chamber, the flame is stable and controlled and offers no radiation.

Enclosed flares offer an option for injecting various waste streams. This is an advantage discussed in more detail in Section 13.4.2.

13.2.2.4 Low Emission Flares

Low emission flares are a type of enclosed flare with a modified design to achieve lower emissions. These flares are typically used in the United States where regulations put more emphasis on NOx and CO emissions. They can offer NOx and CO at a 60%–70% reduction over standard enclosed flares.[2] There are specific regions in the United States that mandate the use of low emission flares because they are considered the best available control technology (BACT). Sites outside of this requirement may implement a low emission flare depending on the facility's cumulative emissions from all of the stationary emission sources on site, turbines, or engines as an example. For facilities in areas designated as "attainment," meaning the regional air quality meets or exceeds the National Ambient Air Quality Standards (NAAQS), facilities can emit 250 ton (230 metric-ton) per year of NOx or CO before more rigorous permitting, known as prevention of significant deterioration (PSD), is required. A low emission flare may be required as a result of the PSD permitting process or may be voluntarily installed if a site wishes to avoid the cost of PSD permitting and to ensure that the flare's emissions do not affect the permitting of future emission sources at the site or facility expansions that might increase emissions.

FIGURE 13.4
Enclosed flare.

In geographical areas where the regional air quality is below the NAAQS, the area will be designated as marginal, moderate, serious, severe, or extreme nonattainment, with significantly lower emission thresholds as they move toward the "extreme" classification. The EPA establishes how an area will be designated based on its ambient air monitoring data. Emission thresholds for these nonattainment designations are 100 ton (91 metric-ton) per year (tpy) for marginal and moderate, 50 tpy (45 metric-ton/y) for serious, 25 tpy (23 metric-ton/y) for severe, and 10 tpy (9.1 metric-ton/y) for extreme.[3] Because no area of the United States is designated as nonattainment for CO,[4] the nonattainment status only applies to NOx emissions. For a facility whose NOx emissions exceed the applicable threshold, a low emission flare is commonly required.

The modifications required to make an enclosed flare low emissions are discussed further in Section 13.2.3.3.

13.2.3 Design Considerations

The major considerations used for designing flares are heat release, gas composition, gas flow rate, gas temperature, radiation, flame stability, combustion, operating temperature, velocities, and turndown. Some of these considerations only apply to certain types of flares.

The main design point, especially for enclosed flares, that determines the size of the flare is not necessarily flow rate, but heat release. Heat release takes into account not only flow rate, but also methane content. Heat release can be calculated using the following equation:

$$\text{Heat release }(\text{Btu/h}) = \text{Flow rate }(\text{SCFH}) \times \text{Heating value }(\text{Btu/SCF})$$

- SCF is standard cubic foot.
- SCFH is standard cubic feet per hour.
- Heating value for biogas is the lower heating value of methane (LHV = 910 Btu/ft^3 = 34 MJ/m^3) multiplied by the volume percentage of methane in the gas stream.

Turndown is another common design consideration in all types of flares. Turndown is expressed by a ratio of the maximum design point to the minimum design point. For example, a flare that offers a 10:1 turndown will be able to operate between 50 and 5 MMBtu/h (15–1.5 MW). It is important to consider turndown ratio based on heat release and not on flow rate alone. A flare that offers a 10:1 turndown and a maximum of 5000 SCFM (7900 Nm^3/h) may not be able to turn down to 500 SCFM (790 Nm^3/h) if the methane content was reduced at the lower flow rates. Methane content plays a key role in turndown and flare design.

13.2.3.1 Elevated Flares

Elevated flares are designed based on heat release, turndown, radiation, flame stability, and exit velocities. Typical sizes of elevated flares can range from 4 to 16 in. (10 to 41 cm) in diameter and 15 to 45 ft (5 to 14 m) tall, where the diameter is typically determined by exit velocities and height by radiation. Elevated flares consist of a riser section, which is the longer portion typically made of carbon steel (see Figure 13.5), and the flare tip, which is stainless steel to withstand the heat of combustion.

FIGURE 13.5
Elevated flare inlet and riser.

Within the flare tip is the windshield, pilot, flame detection device (thermocouples), and flame stability devices. The windshield plays the major role in flare operation. It protects the flame and pilot from wind conditions, offers a stable flame so that the thermocouples can detect the flame, and is the major factor in the flare's turndown capabilities (see Figure 13.6). Exit velocities for elevated flares are limited by 40 CFR 60.18 as previously discussed. Beyond this limitation, flame stability is another consideration for sizing the diameter of the flare. A manufacturer may choose to design a flare with a lower exit velocity than what is governed by 40 CFR 60.18. If exit velocities are too high, the flame may lift off the flare tip and could burn toward the top or above the windshield. Combustion is proven within elevated flares by detecting a flame, which is the function of the thermocouples. Since

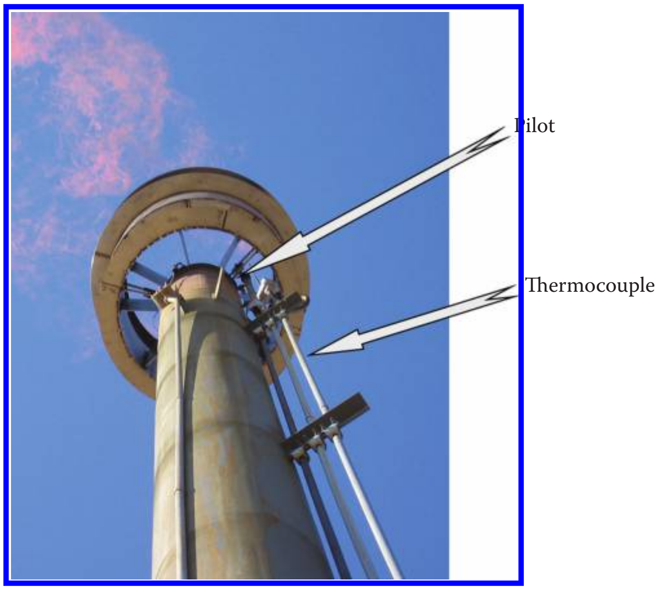


FIGURE 13.6
Elevated flare pilot and main flame thermocouples.

the thermocouples are located within the windshield, the thermocouple may not detect the presence of a flame, even though it exists. It is important to design a flare with proper considerations on limiting exit velocities and to include a windshield that keeps the flame within the main combustion zone throughout all flow rate conditions. Many windshields will have flame stabilization devices, such as rings or ledges, to create flame recirculation and back pressure to keep stable combustion within close proximity of the thermocouples.

Flame stability is also a function of the heating value of the gas and the mixture of outside air with the biogas. Lower heating values, or lower methane content, decreases flame stability. The flame will become lazy and then is more susceptible to blow out from wind. One of the disadvantages of elevated flares is the inability to control the amount of air used for combustion. Air is entrained through the bottom of the windshield, which is a fixed open area, with no provisions for automatic air adjustments based on gas flows. Since the air used for combustion does not vary automatically, it is important to have a properly design windshield for increased turndown.

Radiation is the major design factor when determining flare height (see Volume 1, Chapter 8). Guidelines for acceptable radiation levels are given by the American Petroleum Institute (API). A radiation level of 500 Btu/h-ft^2 (160 W/m^2) is acceptable for people working around the flare on a continuous basis with proper attire and limited skin exposure. Therefore, most flare systems are designed to have 500 Btu/h-ft^2 (160 W/m^2) at grade or less. A simple isoplot, such as the one shown in Figure 13.7, can show various radiation levels at different distances. This figure does not take into account solar radiation a human may experience, which can equate to 250 Btu/h-ft^2 (80 W/m^2) on a hot summer day. Because the flame is exposed on elevated flares, structures, power lines, and trees are required to be a certain distance away from the flame. A general guideline for distances between the flare and other structures is two times the height of the flare stack.

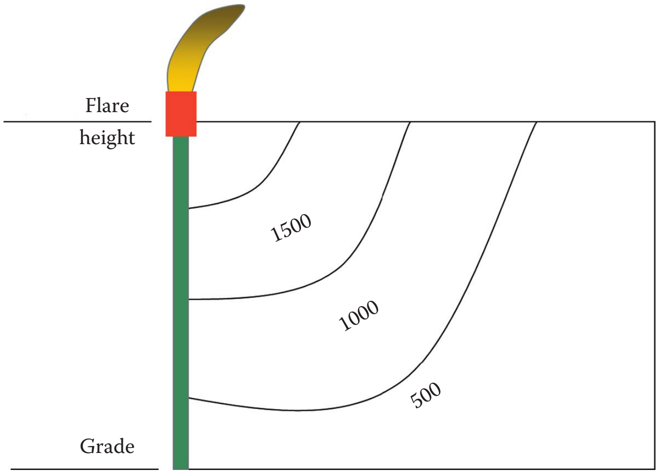


FIGURE 13.7
Radiation isoplot (Btu/h-ft^2).

13.2.3.2 Enclosed Flares

Enclosed flares are designed based on heat release, turndown, operating temperatures, flame length, and retention time. Typical sizes of enclosed biogas flares range from 4 to 13 ft (1 to 4 m) in diameter and 20 to 70 ft (6 to 21 m) tall. The number of flare tips will also vary and increase for larger stacks with higher flow rates. The pictures shown in Figures 13.8 and 13.9 demonstrate

FIGURE 13.8
Enclosed flare gas manifold.

FIGURE 13.9
Enclosed flare gas manifold with flanged flare tips.

multiple burners with a single gas manifold. Enclosed flares utilize natural draft for combustion air with the use of dampers (see Figure 13.10). With increased gas flow rates, the required air for combustion will also increase. Enclosed flares may have one to four air dampers.

Heat release is the key component on designing enclosed flares. Equation 13.1 applies here also. Since heat release takes into account the percentage of methane, the size of a flare for a landfill application, which typically contains 50% CH_4, will not be the same as a digester application with methane around 65% CH_4 even when both of these applications have the same flow rate. Heat release will determine the required stack volume, influences the size and quantity of flare tips, and influences the flame length, which affects flare height. It is important to reemphasize that enclosed flares are not sized on flow rate alone.

Operating temperatures for enclosed flares typically range from 1500°F to 1800°F (820°C to 980°C). Most flare systems will have a low temperature alarm or shut down at 1400°F (760°C). From research and testing, a minimum of 1400°F (760°C) is required for proper destruction efficiencies. In addition to a low temperature alarm, the flare will also have a high temperature shutdown around 2000°F (1100°C). This setpoint is to ensure that the flare is not exceeding an operating temperature considered safe for the materials used to manufacture the flare. Also, with increased temperatures, NOx may increase beyond most permit limitations. With standard flares and operating temperatures as the operating temperature increases, the NOx increases. The opposite is true with CO. Since both of these emissions are regulated, it is important to maintain an operating temperature that balances these two factors. This is typically between 1600°F and 1800°F (870°C and 980°C). The operating temperature is regulated by the air dampers. The air dampers receive a signal from thermocouples mounted at various elevations in the stack, which will prompt them to adjust to maintain a temperature set point. Enclosed flares will have multiple thermocouples within the stack that are used for temperature control. A flare may have 3–4 temperature monitoring thermocouples (see Figure 13.11), but only one is selected to control the air dampers. The thermocouple should be selected based on flow rate. As flows and flame length increase, the retention time is affected; therefore, a high elevation thermocouple should be used for air damper control. The number and size of the air dampers will be determined by the heat release. If the operating temperature is 1600°F (870°C), then 505 SCFM (800 Nm^3/h) of air is required for every 1 MMBtu/h (0.3 MW) of biogas.

Flame length (see Figure 13.12) will determine the height of the stack and is a function of the heating value of the gas, tip velocities, and operating temperature.

FIGURE 13.10
Enclosed flare air damper louvers.

FIGURE 13.11
Enclosed flare interior insulation and thermocouples.

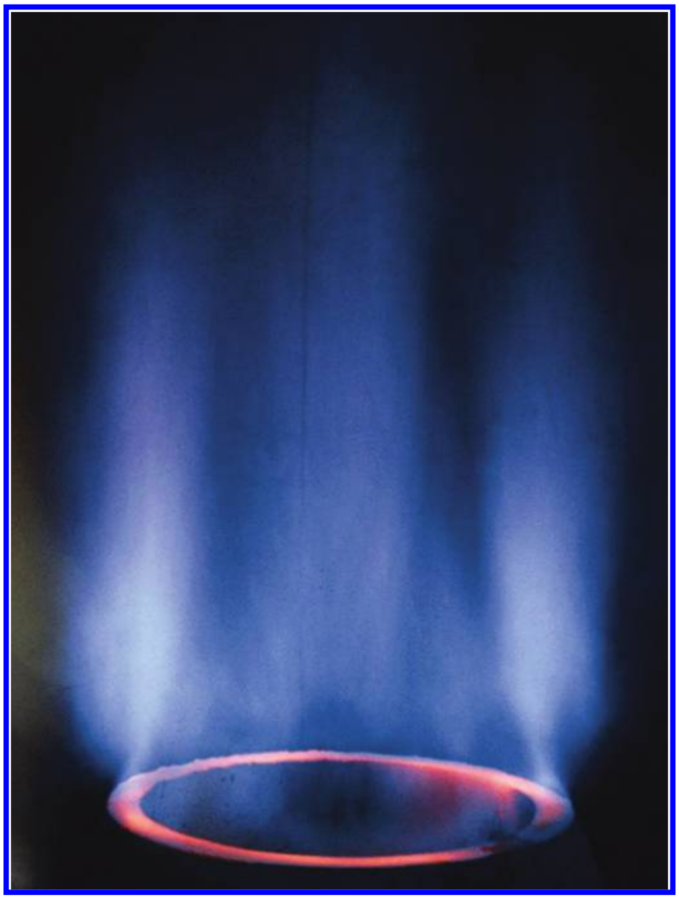


FIGURE 13.12
Landfill gas flame.

As seen in Figure 13.13, flame length will increase as the heating value of the gas increases, which in biogas would be the increase in methane. Flame length will decrease as gas velocities increase, and it will increase if operating temperatures increase. Because flame lengths increase as the heating value of the gas increases, applications for digesters, which have a higher methane concentration, generally have taller stacks.

Proper retention time allows for more adequate combustion of the biogas and is a factor in determining the destruction efficiency and emission levels. The industry standard in the United States is 0.7 s of retention time. Retention time is measured by the flare volume divided by the volumetric flow rate. The flare volume is measured from the flame ignition point, commonly at the flare tip, to the sample ports. It is not the volume of the entire stack. Sample ports, as seen in Figure 13.14, are regulated by the EPA and are required to be a half a stack diameter below the top of the stack as per 40 CFR (chapter 1, part 60, app. A, method 1). Retention time will determine the required flare volume to achieve a desired time for optimal efficiencies and emissions.

The last major factor in designing enclosed flares is turndown. As previously mentioned, turndown is expressed by a ratio of the maximum design point to the minimum design point. Enclosed flares can be rated for a turndown of 10:1. However, it is important to note that a minimum heat release is required to maintain operating temperatures for proper destruction efficiencies and emissions. The typical rule of thumb is that 2.5 MMBtu/h (0.73 MW) is the lower limiting factor. For example, a flare sized for 500 SCFM (790 Nm^3/h) may not be able to operate at the 10:1 turndown or 50 SCFM (79 Nm^3/h). For larger existing stacks where the owner is trying to achieve a greater turndown than the manufacturer's original guarantee, there are a few modifications that can be made. One can increase turndown by limiting the amount of air flow through the dampers, blinding flare tips, and relocating or lowering the thermocouples.

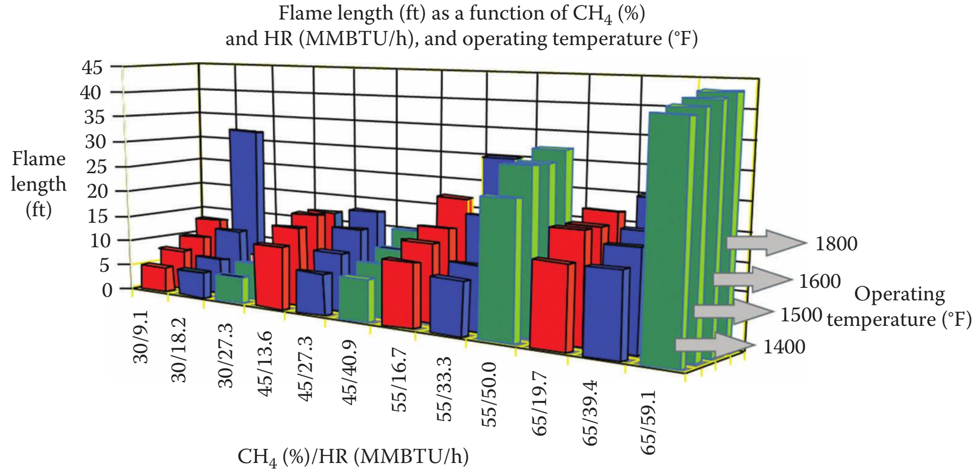


FIGURE 13.13
Flame lengths.

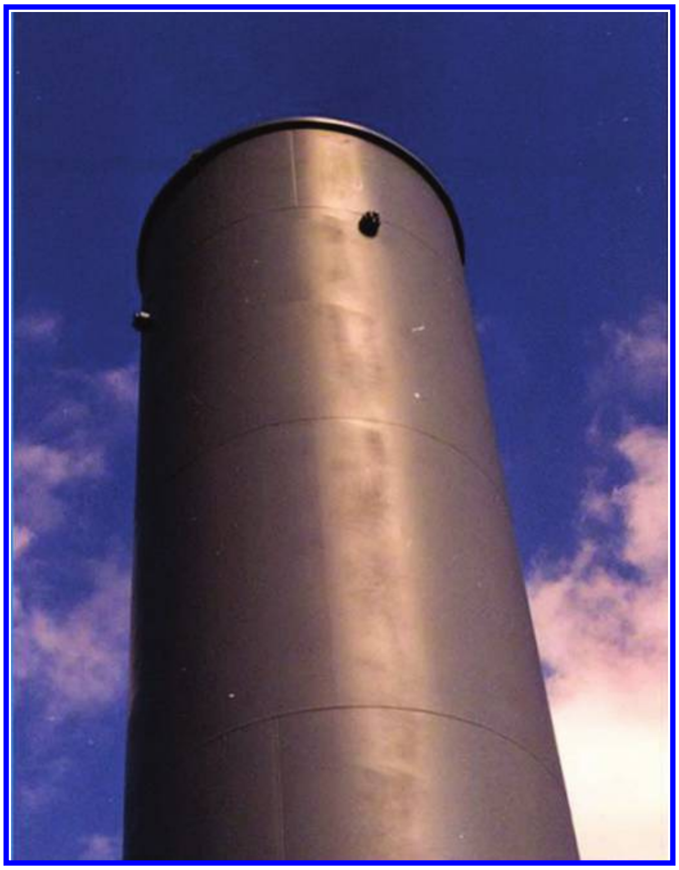


FIGURE 13.14
Sample ports.

13.2.3.3 *Low Emission Flares*

Various modifications can be made to standard enclosed flares to reduce emissions. Slightly lower emissions can be achieved through altering flame lengths, increasing retention time, and modifying operating temperatures. These modifications may only improve emissions slightly. For ultralow emissions, additional design modifications are required. For this design, it is important to understand how NOx (see Volume 1, Chapter 15) and CO (see Volume 1, Chapter 14) are formed. NOx can be further broken down into three categories, prompt NOx, thermal NOx, and fuel NOx. Thermal NOx is defined as the NOx produced from the combustion air, which contains atmospheric nitrogen and oxygen.[2] The way to reduce thermal NOx is to reduce flame temperature due to the fact that N_2 and O_2 are further broken down as combustion temperatures increase. Fuel NOx forms due to the nitrogen that is organically-bound in the fuel gas, for example, ammonia (NH_3). And finally, prompt NOx is defined as NOx that is formed in the initial portion of the flame zone when fuel and air quickly react. There are three mechanisms or designs that alter NOx: a staged fuel design, staged air design, and dilution of the fuel or air. Dilution is the preferred method for reducing both thermal and prompt NOx.

Dilution is achieved by premixing the gas with air. By premixing biogas with air before combustion, CO emissions are lowered. Since the air is premixed with the gas, at ignition, the conversation of CO begins as opposed to standard flares where the gas is mixed with O_2 once it leaves the flare tips. The faster the O_2 is mixed with the gas stream, the more time the combustion process has to convert the CO to CO_2. From extensive testing, there is a key amount of excess air needed to mix with the gas. Too much or too little excess air will actually make the emissions higher. An air blower is used to extract outside air and send to a static mixer where it blends with the gas (see Figures 13.15 and 13.16). Required silencers, filters, and weather protection are important for the air blower. The air blower is operated with a variable frequency drive so that it controls the amount of air mixed with the gas based on gas composition and flow rate. This ensures a constant percent of excess air for the gas. Once the air and gas are blended in the static mixer, the mixture then enters the enclosed flare chamber. It is important to note that by blending the air and gas before combustion, one has a flammable mixture of gas before ignition (see Figure 13.17). Therefore, special considerations are required for the combustion chamber and ignition process. The biogas piping and configuration remain the same design as for a conventional system.

13.3 System Components

As with combustion devices in general, a biogas flare system contains various components included specifically to ensure safe, efficient, and reliable operation. Along with the flare, other components, such as control systems, gas blowers, separator vessels, valves, or instruments, are involved with managing and directing the flow of a biogas stream.

The composition of biogas is predominately methane, a combustible hydrocarbon, so special safety features are required to mitigate a potential explosion and avoid an explosive atmosphere. Since biogas often is corrosive and combustion generates elevated operating temperatures, material selection criteria also become important for certain components to maintain adequate corrosion or temperature resistance.

13.3.1 Safety Devices

Although similar safety devices are common for many biogas flare systems, design of certain flares may dictate incorporating different safety features.

FIGURE 13.15
Zink Ultra Low Emission (ZULE®) enclosed flare.

FIGURE 13.16
ZULE static mixing chamber.

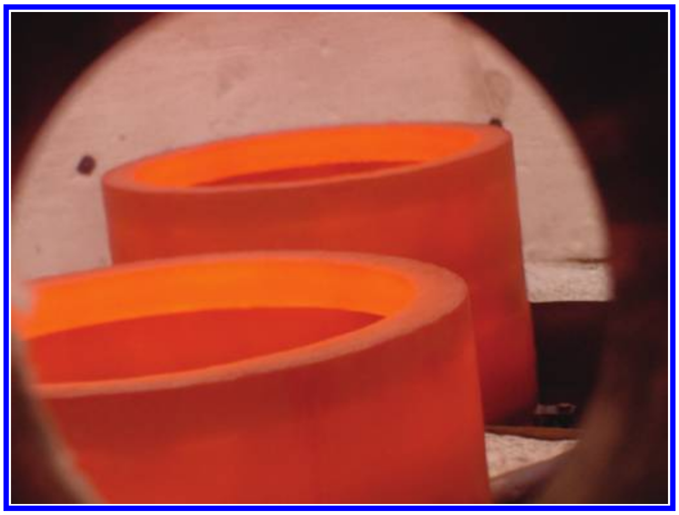


FIGURE 13.17
View inside ZULE enclosed flare.

To address explosion and equipment damage concerns, a variety of equipment is available to perform some combination of flame detection, temperature monitoring, flame suppression, or composition monitoring (see Volume 2, Chapter 1 for a general discussion of combustion safety).

13.3.1.1 Flame Detection

Many products are available to perform flame detection, also referred to as flame sensing or flame monitoring. The premise of flame detection is to recognize the presence or absence of a flame.[5] A flame scanner and a burner management system together are the customary approach for flame detection in the combustion industry, as described in Section 2.2.1.7 of Volume 2, and Section 4.8.1.1 of this book. Because combustion generates elevated temperatures, a simple temperature sensing device may also monitor the flame. In the biogas industry, flame scanners are popular for enclosed flares, but thermocouples frequently are installed rather than scanners for elevated flares.

A scanner performs optical detection as either ultraviolet (UV) or infrared (IR) sensors react to radiation emitted from a flame.[5] More sophisticated scanners are capable of sensing both radiation forms. Content of the fuel stream influences the characteristics of emitted radiation and dictates whether UV or IR sensing is appropriate. Products of combustion vary as fuel streams vary. Combustion products may absorb both forms of radiation to some extent or even partially obstruct radiation from view of the scanner. For biogas applications, with fairly consistent gas compositions, a UV flame scanner alone performs well, since flames generating significant UV levels are quickly detected.[5] An IR scanner may be less effective because it tends to detect radiation from hot, glowing surfaces inside the enclosed flare combustion chamber.[5]

Since a flame scanner is best suited to monitor a flame within a confined volume, a scanner, or viewing head, is mounted to view inside the enclosed flare. Because enclosed flares often operate continuously, around the clock a flame scanner with a self-checking feature is recommended. This feature allows the scanner to perform internal diagnostics periodically by obstructing the view of the flame source momentarily and confirming an intentional flame loss is recognized. If detected properly, scanner operation continues, but if not detected, a scanner fault condition occurs. Without this feature, a scanner might fail in a flame proven state, compromising any safety assurance. Early scanner models were designed with an electromechanical shutter, but recent innovations achieve the same self-checking performance without this shutter motion.

Some scanners only function when connected to a separate, dedicated scanner module, which interprets and responds to the scanner signal. Other scanners offer an integral design, capable of interpreting and responding to the scanner signal independently. Regardless of product, either the scanner or scanner module generates a discrete or analog output, which interacts with some operating system to assure safe conditions are maintained.

Unfortunately, scanners are complicated and sensitive instruments susceptible to many negative environmental influences, including excess heat, electrical interference, and lightning. Scanners are less suitable for elevated flares because of accessibility, positioning, and nuisance concerns. For an elevated flare, the flame is located approximately 15 ft (4.6 m) or more above grade and is influenced by wind direction since it is not contained entirely within a confined combustion chamber. Monitoring the main flame and the pilot flame with a thermocouple becomes more practical compared to a flame scanner.

Because scanners contain complex electronics, thermocouples are less expensive and require less maintenance. To accommodate varying wind direction and any resulting flame location, multiple scanners are necessary to view the main flame. Installing multiple, properly positioned thermocouples is less expensive. Locating scanners near the flame source is necessary to view the flame properly, but the flame is elevated above grade, so accessibility becomes a concern when performing routine maintenance. When the scanner is positioned upward to view the flame, UV radiation from sunlight may simulate a false flame condition as well.

Normal operating temperatures range from 200 to nearly 2000°F (90 to nearly 1100°C) for a biogas elevated flare and pilot operating with natural gas or propane, even though the adiabatic flame temperature for methane, natural gas, and propane each exceeds 3500°F (1900°C). Numerous thermocouple types exist. Each type defines a specific metal material combination and calibration range. A thermocouple, as described in Section 2.2.2.2 of Volume 2), relies on physical flame detection.[5]

Type K thermocouples, typically utilized for biogas applications, contain a positive wire composed of nickel–chromium material and a negative wire composed of nickel–aluminum material, both of which are protected with a magnesium oxide (MgO) compound inside a metal thermocouple sheath. For a given wire combination, the millivolt (mV) signal generated by the thermocouple correlates to a known temperature. This signal is connected to, and interpreted by, some operating system to assure safe conditions are maintained.

Standard calibration values published for type K thermocouples range from approximately −400°F to 2500°F (−200°C to 1400°C). Standard type K identification colors are yellow for the positive wire and red for the negative wire. The sheath material is selected to endure anticipated operating temperatures. A sheath consisting of 310 stainless steel material generally is adequate, since it will resist maximum operating temperatures approaching 2000°F (1100°C). For additional protection, exotic alloy sheath materials, such as Inconel®, provide slightly better temperature resistance.

13.3.1.2 Temperature Monitoring

Combustion generates heat, so equipment supporting combustion is designed to endure the resulting elevated temperatures. In addition to performing critical flame detection functions, temperature monitoring is necessary to protect equipment from the potential damage of excess temperatures. Just as thermocouples perform flame detection for elevated flares, similar thermocouples monitor operating temperatures to protect materials of construction for an enclosed flare.

Steel is an adequate structural material for fabricating enclosed flares. But carbon steel materials, which are less expensive and commonly available, generally do not withstand temperatures beyond 1000°F (540°C). Since enclosed flares frequently operate between 1500°F and 1800°F (820°C and 980°C) to achieve proper emission performance, carbon steel alone is not suitable. Although stainless steel or other exotic alloy materials are appropriate for higher temperatures, increased production expense and decreased availability are significant disadvantages (see Volume 2, Chapter 4 for a more detailed discussion of metallurgy).

A practical solution for biogas enclosed flares involves fabricating a carbon steel shell structure and protecting the interior combustion chamber with fibrous blanket insulation (see Volume 2, Chapter 5 on refractory). Other options exist for protecting the carbon steel shell, as described in Section 8.3.3 of this book, but blanket insulation is economical when considering performance and installation costs.

Blanket insulation is commonly referred to as ceramic fiber. It is produced by combining silica (SiO_2), alumina (Al_2O_3), or MgO fibers to achieve a flexible, lightweight material with low thermal conductivity, excellent corrosion resistance, and temperature stability of approximately 2200°F (1200°C). Even if this insulation becomes saturated with moisture from precipitation, the original properties are maintained after drying. Blankets are available from multiple manufacturers with a variety of thickness and density options. Biogas enclosed flares usually contain layers of 1 or 2 in. (2.5 or 5 cm) thick blankets with densities ranging from 6 to 8 lb/ft^3 (96 to 130 kg/m^3).

The insulation is secured inside the combustion chamber with metal anchors welded to the carbon steel shell. Each straight anchor, or stud, penetrates the layers of blanket, and then a locking washer is installed over the end of the anchor. Because the end of the anchor and washer both are exposed to flame inside the combustion chamber, 310 stainless steel or Inconel materials again are necessary to endure the elevated operating temperatures. This anchoring system, consisting of a stud with washer, is also referred to as a pin and keeper design.

Even with internal insulation, operating temperatures inside enclosed flares are monitored for equipment protection, to ensure structural integrity is maintained. Temperature monitoring again is performed with a type K thermocouple inserted into the combustion chamber. Whereas main flame and pilot flame thermocouples for elevated flares are subject to less intense or intermittent flame exposure, enclosed flare thermocouples receive direct, sustained, and intense flame exposure.

To compensate for these conditions, a thermocouple extends less than 2 ft (0.6 m) into the combustion chamber and is protected by a thermowell. A thermowell also is described in Section 2.2.2.2 of Volume 2. The thermowell material, either 310 stainless steel or Inconel, again must endure high temperature conditions. Ideally, it is positioned near the bottom of the chamber where the flame originates, to detect an extreme temperature quickly. The thermocouple signal is connected to, and interpreted by, some operating system to assure safe conditions are maintained. A temperature switch, for instance, may contain a limiting value of 2000°F (1100°C). If the thermocouple detects a temperature above this value, the switch initiates a command discontinuing operation immediately to avoid potential equipment damage. When temperatures remain below this value, normal operation continues.

13.3.1.3 Flame Suppression

A primary safety concern during the transfer and combustion of hydrocarbon gases is reducing or avoiding flashback potential. A flashback simply is ignition of a flammable gas and air mixture inside the piping of a biogas flare system. It may produce an undesirable fire or explosion, as flame travels backward through the pipe away from the ignition source, in a direction opposite of normal gas flow. Conditions for creating a flashback are dependent upon gas composition, ignition temperature, flame speed, and flammability limits as explained in Volume 1, Chapter 4.

Consider a pipe containing a flammable mixture of fuel and air. If the mixture at the end of the pipe is ignited, the flame may flashback into the pipe. This burning rapidly expands and consequently compresses the unburned mixture ahead of it. The pressure and velocity inside the pipe when a flashback initially occurs may be manageable. As the flashback continues through the pipe, though, the pressure and velocity increase tremendously and may eventually reach 5000 mph and 300 psig (2200 m/s, 21 barg).[6] Flashbacks of this magnitude are referred to as detonations. The results may be disastrous, as expressed in Volume 2, Chapter 1.

To minimize a flashback, consider the fire tetrahedron, which also was described in Volume 2, Chapter 1. By increasing the amount of fuel, a mixture may become too rich for combustion. By increasing the amount of air, a mixture may become too lean for combustion. Since the gas stream must remain flammable for proper combustion, removing the heat source becomes a practical option.

A flame arrester is a passive device designed simply to eliminate the heat source. It relies upon quench distance, which was described in Section 4.11.5 of Volume 1, to extinguish the flame. The flame arrester contains a metal element with tiny passages for the gas stream to flow through. Different gas compositions may influence the passage design. Element types and construction differ too, but most elements are larger than the diameter of the connecting pipe to increase the surface area and reduce the pressure differential. The mass of this element is much larger than the mass of the gas within it, and all of the passages create a large surface area. If a flashback enters the arrester, the flame is separated and the heat is transferred to the element quickly, extinguishing the flame.

As previously mentioned, flame arresters are designed specifically for a flashback condition, when velocities and pressures remain below that of a detonation. As a flashback travels farther through a pipe, the velocity and pressure increase as it transitions into a detonation. Certain stipulations dictate proper placement of a flame arrester, so it will never be exposed to a detonation. The manufacturer may require installation within a limited distance from the ignition source.

The flame arrester body is designed either concentrically or eccentrically. Usually the concentric design is available for vertical pipe, and the eccentric design is available for horizontal pipe. Although the eccentric design often is more expensive, it prevents liquid from collecting in the housing, which may reduce effectiveness. The most common housing and element material is aluminum because it is less expensive and weighs less. When necessary, stainless steel and other exotic metals also are available.

A plug in the bottom of the flame arrester body is recommended for draining liquids. Also, threaded connections installed in the body on either side of the element are convenient for monitoring differential pressure. A differential pressure less than 5 in. H_2O (12 mbar) is normal even at maximum flow conditions. If a differential pressure gauge or transmitter is installed between these threaded connections, at least one manual valve is required downstream of the element. This valve must only be opened while viewing the measuring device. Otherwise, a flashback may bypass the element through the device. Any flame path bypassing the element compromises the safety of the system.

Any element will fail if a flame burns against it for a long enough duration. This failure occurs with flame stabilization. When gas velocity decreases, the flame may flash back and remain burning against the element. The element eventually will heat above the ignition temperature, and the flashback propagates through the arrestor. To indicate flame stabilization, a temperature sensor is installed between the element and the ignition source.

If a high temperature condition is detected, system operation discontinues immediately. Often temperature sensing is accomplished with a type K thermocouple installed near the downstream surface of the flame arrester element. The thermocouple signal is connected to, and interpreted by, an operating system, which instructs an actuated valve to close if this flame arrester thermocouple detects a flashback.

By closing a valve, the gas source is eliminated, which extinguishes the flashback. Although various valve styles are available, actuating a high-performance butterfly valve is considered both economical and effective. A high-performance design minimizes the amount of actuator torque required for opening and closing the valve compared to a resilient seat design. Incorporating a spring return pneumatic actuator ensures the valve closes quickly if a flashback occurs. A spring return electric actuator is considered a suitable alternative for installation in smaller diameter pipe, even if it closes slightly slower. As pipe increases beyond 12 in. (31 cm) diameter, though, spring return electric actuators become less common and less viable.

Regardless of the valve or actuator selected, including a fail close valve is critical. The fail close designation ensures the valve will close to a safe position if some failure condition occurs. The spring mechanism in either actuator will close the valve within seconds if power is disrupted. The spring in a pneumatic actuator also closes the valve if the supply of compressed air or nitrogen is removed.

A liquid seal vessel, as described in Chapter 11, once was utilized frequently for flashback protection. Flame arresters are more prevalent now because they generally are less expensive to install and involve less maintenance.

13.3.1.4 Composition Monitoring

The composition of biogas will remain relatively constant as long as the environmental conditions of the decomposing waste stay relatively constant. However, outside influences to the decomposing waste cannot only change the make-up of the biogas, they can slow down or stop process altogether. By using an analyzer in the biogas being collected, it is possible to monitor composition changes as they occur. The two main constituents that are generally measured are methane and oxygen, and they can be measured at each individual landfill well head or in the piping around the gas blowers. For digester gas applications, the sample will be extracted from the main header or storage.

When measuring the methane and oxygen concentrations at a well head, a portable or handheld analyzer is used. This type of analyzer will take an instant sample, and then it is manually recorded or saved. The handheld or portable analyzers are not made to operate continuously and for this reason can go for long periods of time with very little maintenance or calibration required.

If an analyzer samples landfill gas around the gas blowers, it will be a stationary device for taking a continuous sample and can be set up to deliver either a digital or analog signal to a control system for alarms or continuous recording. The stationary analyzer will have a sample conditioning system to remove water, particulates, and hazardous compounds. In cold weather climates, the analyzer and sample lines must be protected from freezing due to the condensate that will be collected. Because of the continuous operation and conditioning system, this analyzer will require periodic maintenance and routine calibration.

Knowing the composition of the biogas is important for mainly two reasons, one is it will give an indication of the biogas production efficiency and the other is to indicate if the biogas is becoming or is in the flammable range, which is a dangerous scenario. If the oxygen level is too high or the methane is too low, it is likely that something is having an impact on the biogas production and it should be investigated. If the oxygen concentration exceeds 15%, then the biogas is getting close to the flammable range and the flow should be ceased and investigated.

13.3.2 Control Systems

Control systems (see Volume 2, Chapter 2) for biogas flaring operations are generally accomplished with a programmable logic controller (PLC). The PLC will interface with various instrumentations as needed to start, stop, monitor, and generally control the parameters needed to safely operate the system.

13.3.2.1 Permissives

Permissives are those items that must be proven in a safe mode before a flare can start or will cause a flare to shut down if a failure occurs. Typical permissives for flares are

- Biogas shutoff valve is proven closed. This is accomplished with a limit switch on the shutoff valve that makes an electrical contact when the valve is in the closed position and sends a digital signal to the PLC.
- Flame not present. A flame scanner is utilized in enclosed flares, and a thermocouple is utilized in candlestick flares to verify that a flame is not present before sending biogas to the unit.
- General faults. If any fault is present according to the PLC, the system will not be allowed to start up. Typical faults in the system include high stack temperature, low stack temperature, purge failure, blower failure, thermocouple failure, and pilot failure.

Each of these permissives will be programmed in the PLC to prevent the flare from operating unless they are satisfied to be in the correct mode.

13.3.2.2 Purge Cycle

For enclosed flares, it is a requirement that they be purged prior to the ignition sequence to ensure that there is no hydrocarbon buildup in the enclosure. This is commonly done with a purge air blower mounted on the base of the stack. The air blower will generally run for 5 min (less time can be used with large flow rate blowers) to evacuate any residual hydrocarbons from the stack or generate an atmosphere substantially below the flammability limits. In order to prove that the blower is in operation, a pressure switch is placed at the blower discharge. This pressure switch closes a set of contacts when a blower pressure is sensed and sends a digital signal to the PLC to confirm blower operation. If the pressure switch does not properly sense the pressure, the PLC will cease the flare operation.

13.3.2.3 Ignition Sequence

The ignition sequence is the portion of the flare start-up that ignites the flare. This sequence begins immediately after the purge cycle is complete and any residual hydrocarbons have been substantially eliminated from the stack. At this point, the pilot gas automatic valve will open allowing fuel gas to the pilot. A transformer mounted on the stack will energize and send a high voltage current to the ignitor in the pilot assembly. The high voltage at the pilot tip will create an electrical arc that will light the fuel gas. A flame scanner mounted on the stack and pointed at the pilot tip will verify if flame is present. On a candlestick flare, a thermocouple will verify the presence of a flame.

Once the scanner or thermocouple verifies that a flame is present on the pilot, the safety shutoff valve will open and a blower, if needed, will start supplying biogas to the flare. After the open limit switch on the shutoff valve proves the valve open, the pilot fuel gas valve will close. If the flame scanner or thermocouple verifies a flame is still present, system operation continues.

13.3.2.4 Temperature Control

For an enclosed flare, the operating temperature in the stack can be controlled and is generally done so to maintain low emissions and high destruction efficiencies. The typical operating temperature for an enclosed flare is 1400°F–1800°F (760°C–980°C). The operating temperature, or temperature maintained inside the flare enclosure, is controlled by varying the ambient air available through the air damper louvers. The elevation of the temperature control thermocouple is selected depending on the operating retention time as defined by the gas flow rate and methane concentration. The operating temperature is maintained by adjusting automatically the position of the louvers. Closing the louvers reduces the amount of air available and increases the operating temperature, while opening the louvers increases the amount of air available and decreases the operating temperature. The louvers are maintained open initially, before beginning automatic temperature control modulation, which introduces air inside the flare enclosure, while the enclosure is cold and lacking draft, to minimize smoke during initial operation. After this short period of time, a thermocouple in the stack will monitor the temperature through the PLC or a transducer, and then an analog signal will be sent to the modulating air damper in order to maintain the required temperature.

13.3.3 Gas Blowers

Blowers (see Volume 2, Chapter 3) are a major component of biogas systems, specifically landfill applications. The landfill is designed to be under vacuum (negative pressure), therefore requiring mechanical means to pull the gas off the landfill and send to a combustion device. Other biogas systems, in particular digesters and lagoons, where the gas is contained under pressure, typically produce enough pressure to deliver the gas to a combustion device. The amount of pressure needed to pull the gas off the landfill is a function of how the landfill collection system (well field) is designed. It will depend on many factors including, but not limited to, footprint of the well field, depth of waste, diameter and depth of each well, quantity of landfill gas, and distance between the well field and blowers. A medium-size landfill may require approximately 60 in. of water column (15 kPa) on the inlet side of the blower. The outlet of the blower is a function of delivering the gas from the blower to the combustion device, which is usually 10–15 in. of water column (2.3–3.7 kPa). Since the total pressure of this blower in most cases is only 70–75 in. of water column (17–19 kPa), a blower for low-pressure applications can be used.

The majority of blowers used for landfill applications are centrifugal machines, either single (see Figure 13.18) or multistage. These types of blowers offer the required pressure, a wide range of flows, and appropriate turndown (high flow to low flow). Single-stage blowers, often referred to as fans, are for lower-pressure applications. These fans tend to be restricted to 60 in. of water column (15 kPa) total pressure or less. This number will vary depending on the model of the fan and gas characteristics. Fans are less expensive than multistage blowers and offer a higher turndown. Both belt drive and direct drive models are used.

For higher-pressure applications, a multistage blower (see Figure 13.19) is used in lieu of a single stage. This blower has multiple impellers, referred to as stages, which increase the pressure through the machine (see Figure 13.20). As the number of impellers increases, the pressure produced increases. Depending on the model of multistage blower used, available pressure can reach up to 15–20 psig (1–1.4 barg). It is important to emphasize that as the pressure increases through the machine, gas temperature also increases due to the heat produced during compression (see Figure 13.21). This may effect what piping is downstream of the blower, as high-density polyethylene (HDPE) piping discussed in the Section 14.3.5 is rated for maximum temperature of 160°F (71°C).

Where multistage blowers offer greater pressure, they are limited on their turndown capabilities. Turndown is the ratio of maximum flow to minimum flow and in main instances can be less than 3:1 for multistage blowers. It is common practice to use more than one blower in parallel to achieve the full range of flows over the life of the landfill. For example, a landfill designed for 1000–5000 standard cubic feet per minute (SCFM) (28–140 standard cubic meters per minute, SCMM) may have two blowers rated for 2500 SCFM (71 SCMM) so that one blower will operate alone to achieve the desired turndown of 1000 SCFM (28 SCMM). It is also recommended to have a multistage blower or single-stage fan installed as a

FIGURE 13.18
Single-stage fans in a landfill application.

FIGURE 13.19
Multistage blowers.

backup device, in standby mode, in case one of the other machines is required to go offline for maintenance. In the previous example, we would now have three blowers rated for 2500 SCFM (71 SCMM), where only two of the three are operating together to achieve the design flow of 5000 SCFM (140 SCMM). Since single-stage fans offer higher turndown, often unlimited turndown, this same example with fans would now only require two at 5000 SCFM (140 SCMM) each. In any application where you have rotating equipment in standby mode, it is important to rotate the machines in operation every few weeks.

Again, look at the previous example designed to have three blowers at 2500 SCFM (71 SCMM) each to achieve the maximum flow of 5000 SCFM (140 SCMM). If the system is designed in the beginning life of the landfill and gas flows are under 2500 SCFM (71 SCMM), it is beneficial for cost and maintenance to only install two machines, leaving space and control

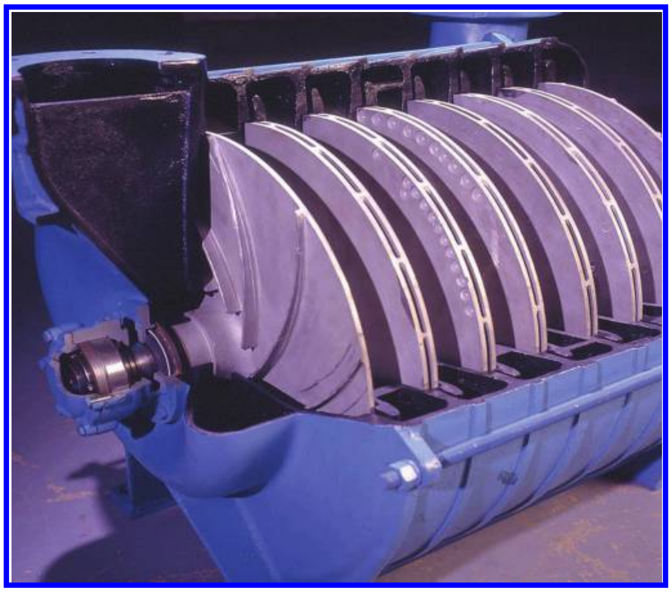


FIGURE 13.20
A cutaway of a multistage blower showing eight impellers. This blower is an eight-stage machine.

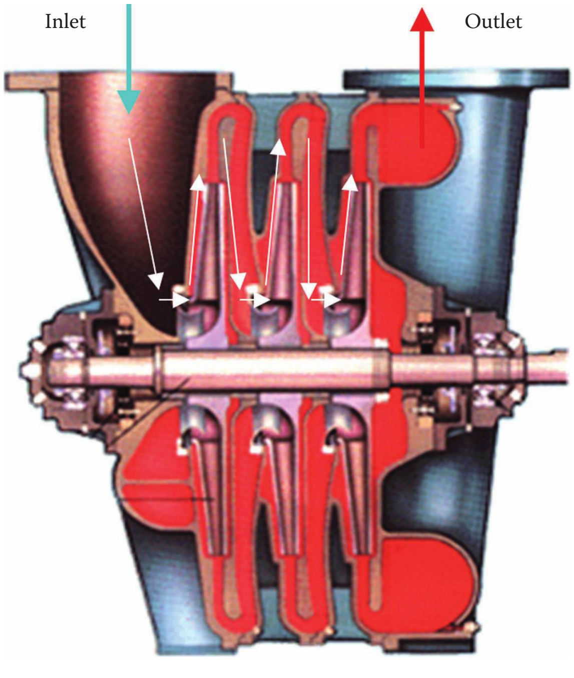


FIGURE 13.21
As the gas travels through each impeller, it increases temperature.

capabilities for the future third blower. Once gas production exceeds 2500 SCFM (71 SCMM), the third blower would be installed.

To size the multistage blowers or fans, the gas composition, elevation of where the machine will be installed, inlet gas temperature, flow rate, and required pressure are all required data. It is also important to verify all cast iron parts in contact with the biogas have special coatings to protect against corrosion. Once the blowers or fans have been sized, note the outlet temperature, horsepower, inlet and outlet connections, footprint, turndown, and efficiencies to size the rest of your components.

Most blowers or fans used in biogas applications are 3600 RPM machines. They can be powered through motor starters or variable frequency drives (VFDs). VFDs can change the rotational speed of the blower, therefore offering many advantages. Even though VFDs are more expensive than motor starters, in most cases, they pay for themselves within the first few years if used to run the blower at reduced speeds, therefore saving on energy costs by reducing energy demand. Since the traditional direct drive blower is a fixed speed machine, if demand or inlet condition changes, the most often used blower control method is inlet throttling. Not only does this create pressure losses at blower inlet by closing a valve, which is not energy efficient, but the blower is operating the same as it would be if it had full flow and pressure conditions.[7] The effects of reducing the blower speed are known as the 1-2-3 law (see Figure 13.22). When the motor speed is increased, flow will increase proportionally, and pressure increases proportionally to the square and horsepower to the cube.

By reducing the blower speed, the required horsepower is greatly reduced. Referring to Figure 13.22, a 25% speed reduction, or operating the blower at 75% flow and speed, reduces your required horsepower to 42%.

VFDs can also save initial cost in blower selection and motor horsepower. By increasing the speed of a blower from the standard 3600 RPM, flow and pressure are increased. Typically, a 10%–20% speed increase is readily available for most of blowers designed for 60 Hz. With this increase in speed, a smaller machine can be used since it can handle bigger flows and pressures.

The reduced speed from the VFD also offers lower discharge temperature and lower vibration and noise levels. The absence of a partly closed butterfly valve further reduces noise. A VFD also acts as a "soft start" in which the motor comes up to speed slowly instead of abruptly. This reduces the stress on the blower and motor, which also reduces maintenance and repair costs and extends machinery life.[7] If the VFD is used in conjunction with a vacuum transmitter, they are helpful in the vacuum control of the well field to decrease operational labor. The VFD and vacuum transmitter operate off a typical PID control loop to change the speed of the blower to constantly supply the desired vacuum.

13.3.4 Moisture Separator

A moisture separator is a vessel that is installed immediately upstream of the biogas blowers, and its function is to remove both slugs of liquid and liquid droplets from the biogas to protect the blowers and promote flare

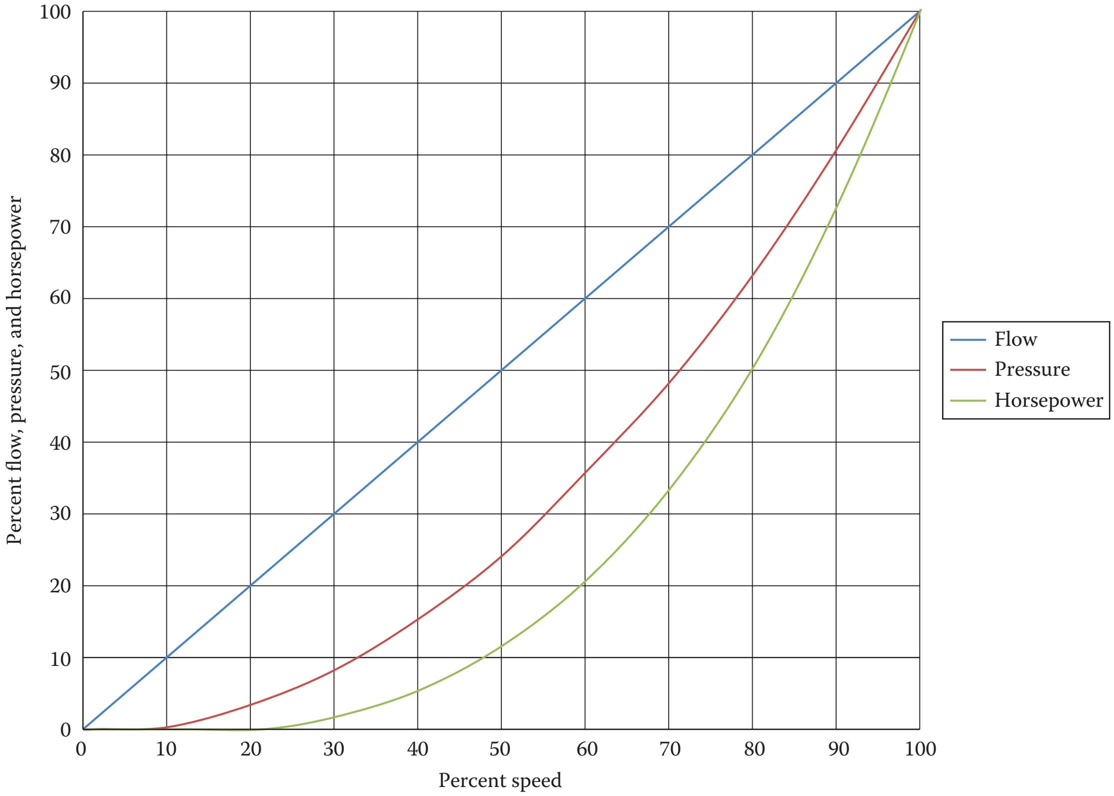


FIGURE 13.22
1-2-3 Fan law.

flame stability. If slugs or droplets of liquid were to enter a blower while it is in operation, it will decrease the efficiency and could do permanent damage. If this liquid were to enter the flare, it could cause unstable combustion, which is associated with a rumbling noise, or it could cause a flame out.

Since biogas is generally corrosive, all wetted parts of the moisture separator are corrosion resistant. Typical moisture separators are constructed of HDPE, stainless steel, or internally coated carbon steel. The preferred method of moisture removal is a stainless steel or polypropylene mesh pad with removal efficiencies of 99% for liquid droplets 10 μm and larger. These vessels will have removable tops for ease of cleaning or removing the mesh pad and will be large enough to hold a small amount of liquid in the base. Recommended instrumentation includes a differential pressure gauge mounted across the mesh pad for determining when it needs to be cleaned and a level gauge on the bottom of the vessel to determine when it needs to be drained. Automated instrumentation such as level switches and pumps can also be added.

13.3.5 Miscellaneous Components

In this section, some miscellaneous electrical and mechanical components associated with the blowers and surrounding equipment are discussed. Each blower should have isolation valves on the inlet and outlet locations. These valves are usually manual butterfly valves with an appropriate seat material suitable for biogas. In addition to having butterfly valves on the inlet and outlet, the blower should also have expansion joints to absorb the blower vibration and protect surrounding piping connections. Hypalon is typically specified unless there are contaminates in the gas that are not suitable for this material, for example, high hydrogen sulfide. On the outlet of the blowers, it is recommended to install check valves. These valves are a wafer-type valve that prevents backflow when that particular blower is not in operation while others are operating. Some installations may include the placement of various pressure and temperature gauges to monitoring system conditions. These gauges are usually placed before and after each blower and upstream and downstream of other major components.

For electrical components, in addition to what has been discussed in Sections 13.3.2 and 13.3.3, blower systems may include bearing thermocouples, vibration sensors, and transmitters used for monitoring or alarm conditions. Blower bearing failures can be detected by measuring temperature though the use of thermocouples mounted at both the inboard and outboard bearing housing. An increase in temperature equals an increase in friction. This friction is typically caused from shaft imbalance or bearing corrosion. A set point for a temperature shutdown is recommended for 200°F–250°F (93°C–120°C) and is monitored through the control panel by receiving an analog signal. Vibration monitors play a key role in early detection of blower problems caused from an imbalance in the seals, bearings, or direct couple motor. Vibration monitors utilize a transmitter mounted in the blower inboard bearing housing, which sends an analog signal to the control panel. The use of pressure or temperature transmitters throughout your system can be used for recording operational parameters or for any alarm conditions.

13.4 Alternate Applications

13.4.1 Gas Utilization

Using biogas for electricity or direct use is becoming more prevalent. Currently, direct use that can include boilers, high heating value or pipeline gas, or alternative fuel only makes up 28% of the gas utilization projects. Electrical projects are far more common, and within those projects, 78% of them send the gas to internal combustion engines to generate electricity. Most engines require an inlet pressure of 1.5–5 psig (10–34 kPa) and dry gas; therefore, the biogas needs to be compressed and conditioned prior to reaching the engine. Depending on the type of engine being used and the landfill gas characteristics, conditioning may include more than just removing water. Hydrogen sulfides and siloxanes are common constituents that require removal from the biogas. Siloxanes are any organic compound with Si–O–Si linkage, commonly used in cosmetics, soaps, lubricants, and many other products. They form a SiO_2 layer when combusted, therefore shortening equipment life and increasing maintenance for engines, turbines, and other equipment that are used for utilization projects. Siloxanes can be removed through temperature swing adsorption (TSA) or pressure swing adsorption (PSA), which produces other waste streams that need to be sent to a combustion device or flare (see Figure 13.23).

For landfill applications or other biogas applications that require blowers to extract the gas, the conditioning and compression equipment is typically installed

FIGURE 13.23
Siloxane removal system.

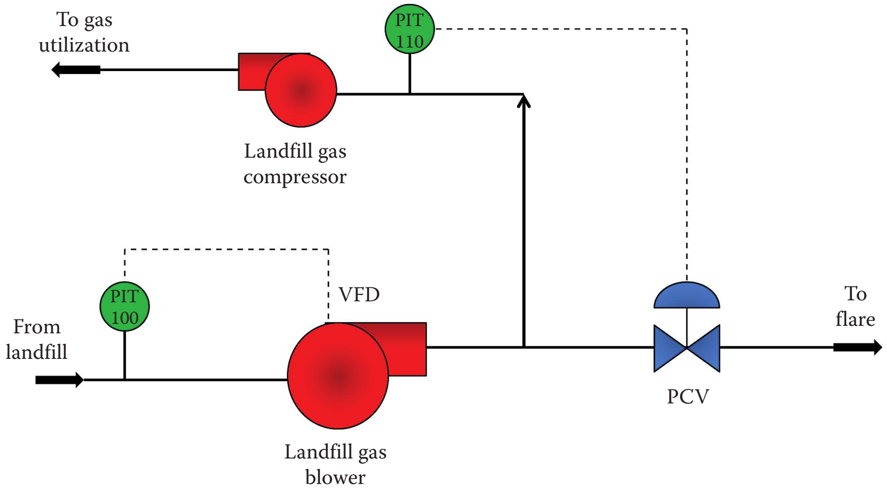


FIGURE 13.24
P&ID of gas utilization and a single flare.

downstream. For landfills in particular, it is common practice to have one vacuum source to maintain balance on the well field to lower maintenance and well field tuning. And in most cases it is required by local regulations to have a combustion device or flare as a backup to the gas utilization facility.

Figure 13.24 shows a simple process and instrumentation diagram (P&ID) and how the gas is collected from the landfill and sent to a gas utilization facility and a single flare. The biogas is pulled off the landfill by the main landfill gas blowers and using a pressure transmitter (PIT-100) to control the VFDs off vacuum control. The gas is then sent to the gas utilization facility or to the flare. The compression and conditioning equipment needed for utilization is separate from the main landfill blowers. One of the benefits of this arrangement is that when the facility is offline, the conditioning and compression equipment is also offline since it is associated with that side of the process. The flare system does not require higher pressures nor conditioned or dry gas. When establishing gas distribution priorities, the preference is given to the utilization facility over the flare system since it is typically generating revenue. The facility will accept as much gas as it is able to receive, and the balance goes to the flare system. This is controlled by a PID loop using a pressure transmitter (PIT-110) and a pressure control valve (PCV). The main landfill gas blower is responsible for delivering the gas to the gas compressors at a minimal positive pressure, usually 20 in. of water column (5 kPa) or less. If the facility loses capacity, for example, one of the engines is offline, the pressure in this line will increase, and PIT-110 will read a higher pressure. Once this pressure exceeds the programmed set point, it will then continue to open PCV to send more gas to the flare system until the pressure in the facility line is reduced. This is achieved through a standard PID control loop. If the flare system is offline, it is common practice to change the landfill gas blower VFD control to operate off PIT-110 instead of PIT-100 so that the landfill is sending more gas to the facility that it can receive.

In the instance the landfill has two flares installed as well as a gas utilization facility, the gas distribution between devices becomes slightly more complicated (refer to Figure 13.25). The landfill gas blowers operate off vacuum control (PIT-100) as noted in the previous example as well as the interaction between PIT-110 and PCV. Downstream of PCV, another PID control loop is introduced. This loop is controlling the flow between the two flares by using a flowmeter (FM) and a flow control valve (FCV). If only one flare is in operation along with the facility, this control loop is not active, and FCV is driven full open. Likewise, if both flares are operating and the facility is offline, PCV is driven full open. In the case where both flares are offline and the facility is receiving gas, the VFDs can operate off discharge control (PIT-110) to avoid sending excess gas to the facility.

In either situation, most control valves will have pressure drop across them, and the landfill gas blowers should have enough discharge pressure to compensate.

13.4.2 Enclosed Flare Use

Enclosed flares operating with biogas offer a beneficial purpose since they can function as a thermal oxidizer or incinerator for small or lightly contaminated liquid or gas streams. Since the enclosed flare has a defined volume and a relatively constant high temperature being controlled, vapors or atomized liquids can be

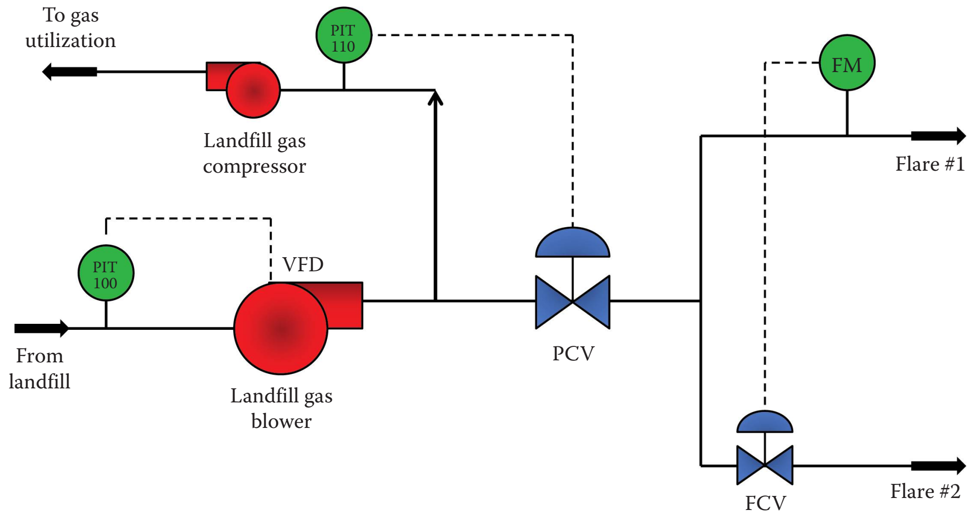


FIGURE 13.25
P&ID of gas utilization and dual flares.

injected and thermally destroyed. Typical streams that are injected into enclosed flares are condensate, leachate, miscellaneous volatile organic compounds (VOC) or internal combustion engine exhaust.

13.4.2.1 Condensate Injection

For landfill sites, condensate from biogas piping is collected and then pumped or gravity fed to a storage container. Each site is then responsible for properly disposing of this liquid. Depending on the circumstances, if the condensate is tested and meets a certain standard, it can be placed back in the landfill or used as dust control at the site. If it is odorous or has a higher quantity of contaminants, it is likely to be locally treated or it can be injected and evaporated in an enclosed flare.

The two most common forms of condensate injection are mechanical atomization and air atomization. Mechanical atomization uses high pressure over 200 psig (14 barg) in a spray nozzle to mechanically create a fine mist from the condensate. Air atomization uses a lower-pressure condensate of about 80 psig (5.5 barg), and force mixes it in a spray nozzle with a volume of air at the same pressure to create a fine mist. Both of these methods use an apparatus to transform the condensate into the fine mist and then inject it into the enclosed flare when it is operating at a minimum temperature, typically 1400°F (760°C) or greater. When the fine mist contacts the hot flue gases in the stack, it vaporizes and the water turns into steam and any contaminants are oxidized at a high rate of efficiency. Test reports have shown that the overall emissions from the stack remain relatively unchanged when condensate is being injected into an enclosed flare.

A potential negative consequence from condensate injection results if damage to internal insulation and stack occurs when the system is not properly maintained. In some cases, the injection nozzle can get plugged or corroded causing the atomization to be minimized. When this happens, large droplets to a solid stream of condensate will form, making it hard to vaporize inside the enclosed flare. This stream will impinge on the insulation causing it to fail and possibly exposing the stack shell to high temperatures, both of which can lead to costly repairs.

13.4.2.2 Leachate Evaporation

Leachate is a problem in many landfills especially when it cannot be put back in the landfill and must be treated. Many methods for treating leachate exist, including evaporation ponds or lagoons, aeration, filtration, biological treatment, and thermal evaporation. Since leachate is generally pumped from the bottom of the landfill, it can include many of the organic and inorganic compounds of the landfill, many of which may be in the form of dissolved or as suspended solids. When these items are thermally evaporated, the solids will drop out and concentrate, but the liquid compounds will boil off, and the concentration of these may be too high to emit to the atmosphere. If this is the case, then this contaminated evaporate must be further oxidized or combusted.

An enclosed landfill gas flare is the most common way to thermally oxidize leachate evaporate. When the evaporate exits the evaporator, it will be piped or ducted to the enclosed flare and injected into the stack. Since the enclosed flare is operating at a high excess oxygen level and a high temperature, any organic or inorganic compounds will be converted to carbon dioxide and water at a rate of 98%. Special care is needed in the design to protect the refractory from premature failure and to promote mixing without impacting flame stability.

13.4.2.3 Miscellaneous Vapor Injections

Because the enclosed flare has a confined space, high temperature, and elevated oxygen levels, it has a high flexibility of having a variety of other vapors injected to thermally oxidize them. Some examples of these vapor streams are soil vapor extraction, odorous compounds from buildings, off-gas from siloxane removal vessels, PSA off-gas, low heating value streams, or exhaust from internal combustion engines. The injection philosophy depends on the content of the vapors being combusted. If the vapor is 99% or greater air, then this is generally introduced similar to ambient air entering through air dampers, while all other streams are generally introduced higher in the actual combustion section of the stack. The expected destruction efficiency of these vapors is 98% or above.

References

1. U.S. EPA, Inventory of U.S. greenhouse gas emissions and sinks: 1990–2009, April 2011. http://www.epa.gov/climatechange/emissions/downloads11/US-GHG-Inventory-2011-Executive-Summary.pdf
2. *Ultra-Low Emission Enclosed Landfill Gas Flare—A Full Scale Factory Test*, presented at SWANA 21st Annual Landfill Gas Symposium, Austin, Texas, March 1998.
3. Virginia Department of Environmental Quality website, October 13, 2011, http://www.deq.state.va.us/air/permitting/permitprocess.html
4. U.S. EPA, Criteria pollutant area summary report, August 30, 2011, http://www.epa.gov/airquality/greenbk/ancl2.html
5. C. Filoon and C.E. Baukal, *Flame Safety, Hydrocarbon Engineering*, Palladian Publications Limited, U.K., 2009.
6. W.E. Baker, P.A. Cox, P.S. Westine, J.J. Kulesz, and R.A. Strehlow, *Explosion Hazards and Evaluation*, Elsevier Scientific Publishing Company, Amsterdam, the Netherlands, 1983.
7. B.S. Johnson, Specifying a cost effective landfill flare system, Presented at *SWANA's 28th Annual Landfill Gas Symposium*, San Diego, California, March 8, 2005.

14

Flare Gas Recovery

Jeff Peterson, Nick Tuttle, Harley Cooper, and Charles E. Baukal, Jr.

CONTENTS

14.1 Introduction

To the casual observer, minimizing the flaring from a refinery or petrochemical facility may seem easy. Remember that flares are safety systems designed to protect site employees, the public, and the facility (see Chapter 11). New strategies can be applied to find cost-effective methods that safely minimize if not eliminate the need for flaring.

14.2 Designed to Protect

Flares are combustion devices designed to safely and efficiently destroy waste gases generated in a plant (see Figure 14.1). In refinery operations, flammable waste gases are vented from processing units during normal operation and process upset conditions. These waste gases are collected in piping headers and delivered to a flare system for safe disposal. A flare system often has multiple flares to treat various sources for waste gases. There may be several different flare types used in a system, depending on site requirements. Flares are primarily safety devices that prevent the release of unburned gas to the atmosphere; these gases could burn or even explode if they reached an ignition source outside the plant.

Two levels of flaring are of interest. The first level of flaring relates to its hydraulic capacity that occurs during a plant emergency. This can be a very large flow of gases that must be destroyed, where safety is the primary consideration. These flows can be more than one million pounds an hour (450,000 kg/h) or more, depending on the application. The second level of flaring is waste gases generated during normal operation, including planned decommissioning of equipment. While safety is still imperative, emission control is also important. The actual waste-gas flow rate and composition may vary significantly during normal operation, but the flare should still be capable of safely destroying the waste gases while minimizing emissions. The American Petroleum Institute has developed guidelines for handling these waste gases.[1,2]

Traditionally, there have been three important performance parameters of interest for most flares.[3] The first is the so-called *smokeless capacity*. This is the maximum flow of waste gases that can be sent to the flare without producing significant levels of smoke. A flare is typically sized so that the smokeless capacity is at least as much as the maximum waste-gas flow rate expected during normal operation. The second performance parameter is the *thermal radiation* generated by the flare (see Volume 1, Chapter 8) as a function of the waste-gas flow rate and composition.[4] The radiation levels at ground level are typically limited to avoid injuring personnel and damaging equipment. After choosing the most remote, practical flare location, the height of the flare stack is determined so the acceptable thermal radiation levels are not exceeded at ground level. The third parameter is *noise* (see Volume 1, Chapter 16).

FIGURE 14.1
Example of an air-assisted flare during testing.

Excessive noise can injure personnel, equipment, and property both inside and outside the plant.

While the primary function of flares is to protect the facility, employees, and the surrounding environment, flaring gases create emissions such as nitrogen oxides (NOx) (see Volume 1, Chapter 15), sulfur oxides (SOx), greenhouse gases (CO_2 and CO), and volatile organic compounds (VOCs) (see Volume 1, Chapter 14 for a general discussion of combustion pollution emissions). These emissions, in combination with any unburned hydrocarbons, contribute to the total facility emissions.

Historically, flare emissions have not specifically been a parameter of interest because they are very difficult to measure. Since nearly all flares burn in the open, there is no enclosure or combustion chamber with a well-contained exhaust stream to insert probes into for extractive or in situ emission measurements. Research is currently being done on using remote monitoring analyzers to measure flare emissions, but this technique is still under development.[5]

The size of flare flames and elevation above the ground make it very difficult to use some type of hood to collect exhaust gases and measure emissions. Another very challenging problem is the weather conditions, the waste gas flow rate, and composition are highly variable and not generally controllable. For example, wind plays a very significant role in the performance of a flare.[6] High waste gas flow rates, such as those that could occur during emergency conditions, are generally impossible to test in an operating plant because fortunately they rarely occur. There are some flare test facilities capable of simulating very high flow rates, but even these can rarely test the maximum flow rate that could occur at a plant (see Volume 2, Chapter 9).[7]

There is growing interest in reducing the pollutant emissions from flaring. For example, the Bay Area Air Quality Management District in California established Regulation 12, Rule 12, entitled "Flares at Petroleum Refineries" on July 20, 2005. The rule requires flare minimization projects and studies for area refineries. There is growing concern that emissions of VOCs from flares may be much higher than previously thought.[8] One possible reason is that wind effects can reduce flare destruction efficiency.[9] The estimated emissions from flares are often based on measurements made with little or no wind. Accordingly, the emissions may be much higher under windy conditions.

Another possible reason is improper operation of flares. Many flares use steam as an assist medium to increase air entrainment into the flame to increase the smokeless capacity. However, oversteaming (see Figure 14.2), or providing too much steam to a flare compared to the waste-gas flow rate, can actually reduce the destruction efficiency. The cooling effect of excessive steam may inhibit dispersion of flared gases, particularly during weather inversions. In the extreme case, oversteaming can actually snuff out the flame and allow the waste gases to go into the atmosphere unburned.

There is growing concern that many flares are being oversteamed to minimize smoking over a wide range of waste-gas flow rates. In most steam-assisted flares, the steam flow rate is manually controlled and sometimes set for the maximum expected waste-gas flow during normal operation. However, this means the flare could be severely oversteamed during periods where the waste gas flow is much lower.

14.3 Problem

To the casual observer, it may seem relatively easy to minimize and even eliminate routine flaring from refineries and petrochemical. It appears that these plants are unnecessarily wasting energy and generating pollution. The main challenge is that it can be uneconomical to recover the gases, either for use in the plant or to sell as energy, for a variety of reasons.

FIGURE 14.2
Example of a flare being oversteamed during testing.

The flow rate and composition of the waste gases going to the flare are often highly variable. The unsteady flow (see Figure 14.3) and variable composition (see Table 14.1) make it difficult to use the waste gases elsewhere in the plant where the energy demand is normally steady. The variable composition makes it difficult to sell, unless a purification process is added to produce a more consistent composition.

The waste gases may have a low heating value, which means that equipment such as burners must be properly designed for the low heating value. The waste gases may be off-spec product that is being flared because it cannot be sold and is not easily reprocessed to produce on-spec product.[10] Off-spec flaring may occur for some time during start-up until the product is within specification.

The waste gas pressure is low; thus, a compressor is needed to aid transporting the gases. In most refineries and petrochemical plants, the fuel gas is at a high enough pressure that it can be used to entrain the air needed for combustion so that the burners do not need a fan or blower.[11] Additional piping may also be needed to connect the waste gas to the fuel-gas system.

14.4 Potential Solutions

A variety of strategies for minimizing flaring is possible and can be grouped into two broad categories: *plant practices* and *new equipment*. *Plant practices* involve

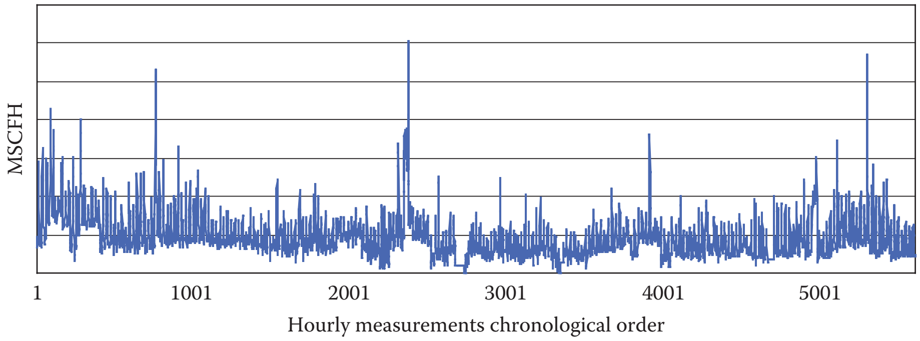


FIGURE 14.3
Example of waste-gas flows to a flare in a typical refinery over approximately an 8-month period.

TABLE 14.1

Example of Waste–Gas Compositions at a Typical Plant

Flare Gas Constituent		Gas Composition Range (%) Minimum	Maximum	Flare Gas (%) Average
Methane	CH_4	7.17	82.0	43.6
Ethane	C_2H_6	0.55	13.1	3.66
Propane	C_3H_8	2.04	64.2	20.3
n-Butane	C_4H_{10}	0.199	28.3	2.78
Isobutane	C_4H_{10}	1.33	57.6	14.3
n-Pentane	C_5H_{12}	0.008	3.39	0.266
Isopentane	C_5H_{12}	0.096	4.71	0.530
Neopentane	C_5H_{12}	0.000	0.342	0.017
n-Hexane	C_6H_{14}	0.026	3.53	0.635
Ethylene	C_2H_4	0.081	3.20	1.05
Propylene	C_3H_6	0.000	42.5	2.73
1-Butene	C_4H_8	0.000	14.7	0.696
Carbon monoxide	CO	0.000	0.932	0.186
Carbon dioxide	CO_2	0.023	2.85	0.713
Hydrogen sulfide	H_2S	0.000	3.80	0.256
Hydrogen	H_2	0.000	37.6	5.54
Oxygen	O_2	0.019	5.43	0.357
Nitrogen	N_2	0.073	32.2	1.30
Water	H_2O	0.000	14.7	1.14

controlling the processes producing waste gases using existing equipment in the plant. One example is simply ensuring that equipment is properly maintained to minimize leaks into the waste-gas header. Another example might be improved understanding of what waste gases are produced under a given set of conditions so those conditions can be avoided.

New equipment refers to adding hardware that reduces the amount of waste gases going to the flare. One example might be redesigning plant processes to minimize waste-gas production. This might mean recycling waste gases back into the process or using alternative technologies that produce less waste. Another example is flare gas recovery systems (FGRSs) that can capture waste gases that would have been flared, either for use in the plant or for sale.[12]

There are many potential benefits of using an FGRS. As an example, the Reliance Industries refinery in Jamnagar, India, received the following benefits: coker flaring was reduced by 95.4%, flare SOx emissions were reduced by 95%, and fuel-gas yield increased by 0.17 wt% of the coker feed.[13] At the Khangiran refinery in Iran, the gases saved from flaring replaced a more expensive fuel gas, pollution emissions were reduced by 90%, and thermal radiation and noise from flaring were reduced.[14] The U.S. Environmental Protection Agency (EPA) identified flare-gas recovery as a technology for reducing greenhouse gases in the petroleum refining industry.[15]

14.5 Flare Gas Recovery Systems

A FGRS is designed to capture waste gases that would normally go to the flare system.[16] The FGRS is located upstream of the flare to capture some or all of the waste gases before they are flared. There are many potential benefits of an FGRS. The flare gas may have a substantial heating value and could be used as a fuel within the plant to reduce the amount of purchased fuel. In certain applications, it may be possible to use the recovered flare gas as feedstock or product instead of purchased fuel.

The FGRS reduces the continuous flare operation, which subsequently reduces the flare visible flame, thermal radiation, noise, and pollutant emissions associated with flaring. It also reduces the negative public attention drawn to the facility. Capturing waste gases may reduce odor levels. Reduced flaring also reduces steam consumption for steam-assisted flares and can extend the life of the flare tips.

A schematic of a typical FGRS is shown in Figure 14.4. When the recovered flare gas is to be utilized as a fuel and the flow is less than or equal to the capacity of the FGRS, the flare gas will be recovered and directed to the refinery-fuel-gas header. During these periods, there will be little or no visible flame at the flare, although the flare pilot(s) may be visible. When the

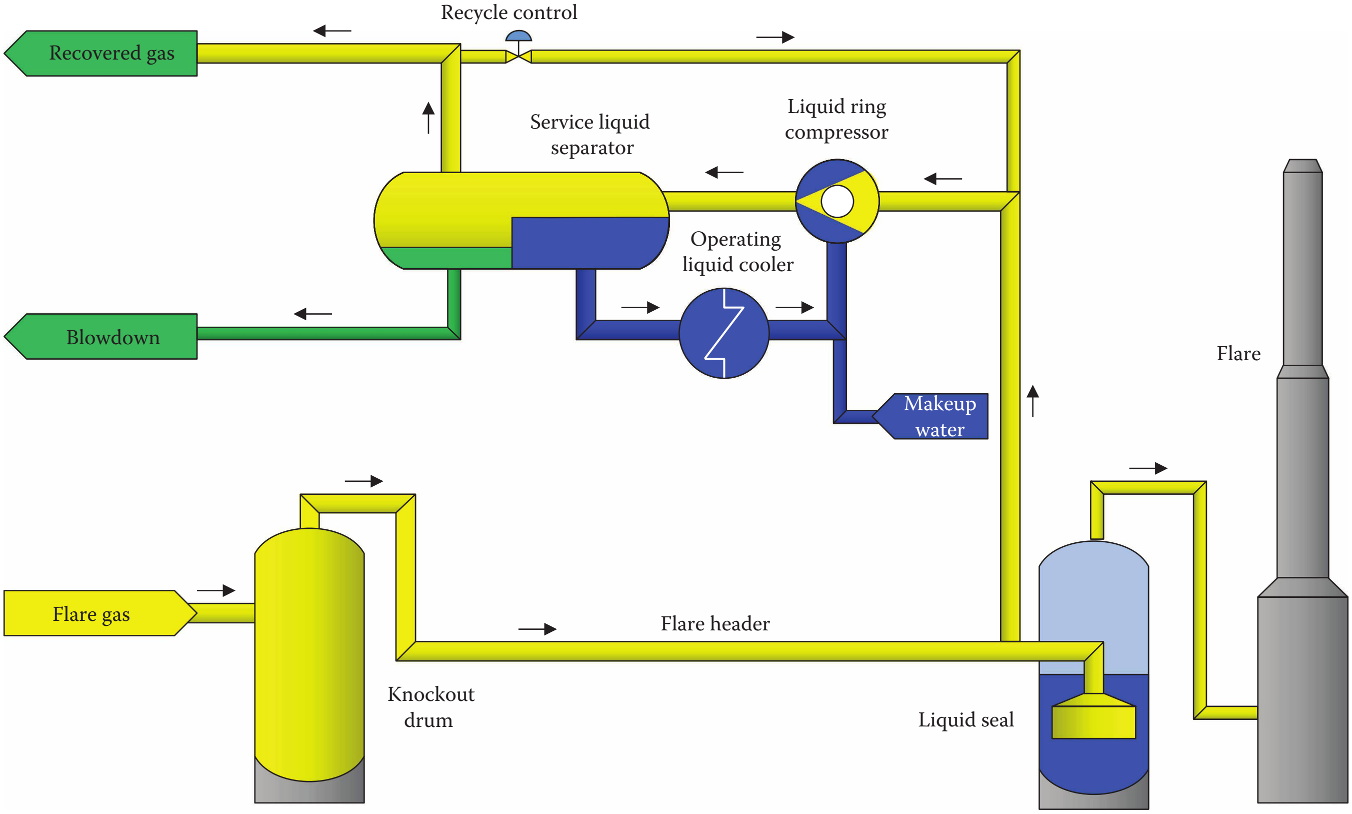


FIGURE 14.4
Generalized flare-gas recovery process schematic.

flare-gas flow rate is greater than the capacity of the FGRS, the excess flare gas will flow through the liquid seal drum and to the flare tip where it will be combusted. From flaring rates just above the FGRS capacity to a maximum flaring episode, the liquid seal drum will promote a smooth, safe operation of the flare tip. The FGRS is typically operated at a slight positive pressure to prevent air infiltration into the system that could create a flammable mixture.

The basic processes employed in the FGRS are compression and physical separation. The basic operation of the FGRS is

- Process vent gases are recovered from the flare header.
- Gas compressors boost the pressure of this gas.
- Recovered gas is discharged to a service liquid separator.
- Separated gas may pass through a condenser where the easily condensed constituents may be returned as liquid feedstock, while the components that do not easily condense are returned for use as fuel gas after scrubbing for contaminant removal, such as H_2S.

Gas compression is performed by compressors selected for the specific application. For example, if a liquid-ring compressor is used, then separating recovered vapor phase from a mixed liquid is accomplished via a horizontal separator vessel. As flare gas flows into the header, an established hydrostatic head in the liquid seal drum will prevent flare gas from flowing to the flare. This causes a slight increase of pressure in the flare gas header, but not enough to significantly affect the capacity of the overpressure protection devices in the refinery. When the flare gas header pressure reaches the gas recovery initialization setpoint in a batch operation plant, the compression system will begin to compress the flare gas. The FGRS will start and stop with control signals from the programmable logic controller programmable logic controller (PLC). In plants with varying flare loads, additional parallel compressors can be automatically staged on or off to augment the capacity of the base-load compressor as needed. Based on the inlet pressure of the flare-gas header, fine-tuning of FGRS capacity control is by the spillback (recycle) of recovered gas from the service liquid separator back to the suction.

Discharge of the liquid-ring compressors will flow into the service liquid separator vessel where gas and service liquid are disengaged and the compressed recovered flare gas is delivered to the facility fuel-gas scrubbing and distribution system. The compressor service liquid, usually water, is used in the compressor as a seal between the rotor and the compressor case. The service liquid is separated from the recovered gas stream, cooled, and recirculated to the gas compressor train for reuse.

The gas processing capacity of the FGRS adjusts to maintain a positive pressure on the flare header upstream from the existing liquid seal drum. This positive pressure will ensure that air will not be drawn into either the flare system or the FGRS. If the volume of flare gas that is relieved into the flare system exceeds the capacity of the FGRS, the pressure in the flare header will increase until it exceeds the back pressure exerted on the header by the liquid seal. In this event, excess gas volume will pass through the liquid seal drum and onto the flare where it will be burned. This will be the case when there is a rapid increase in flare-gas flow due to an emergency release. Since the liquid seal serves as a back-pressure control device for the FGRS, a properly designed deep-liquid seal is critical to the stable operation of the FGRS and flare. A deep-liquid seal, typically 30 in. W.C. (75 mbar) minimum, is required to permit a suitable control range for the capacity control of the FGRS. As the flow transitions to the flare, this must be done with a very stable liquid level or else unstable flare header pressure could result, affecting FGRS control and proper flare operation.

If the volume of flare gas relieved into the flare header is less than the total capacity of the FGRS, the capacity of the FGRS adjusts to a turndown condition. This is accomplished by turning off compressors and/or by diverting discharged gas back to the suction header through a recycle control valve. The compressor speed can also be varied. Control of the FGRS is automated with minimal requirement for direct operator intervention.

14.6 Flint Hills Resources' Experience

Many FGRSs have been installed based primarily on economics, where the payback on the equipment was short enough to justify the capital cost. Such systems were sized to collect most, but not all, of the waste gases. The transient spikes of high gas flows are typically very infrequent, meaning normally it is not economically justified to collect the highest flows of waste gas because they are so sporadic. However, there is increasing interest in reducing flaring not based strictly on economics, but on environmental stewardship.

Flint Hills Resources (FHR) has made a strong commitment to dramatically reduce flaring at all of its facilities.[17] Overall, flaring at FHR facilities has been reduced by more than 95% since 1997. This is part of the company's commitment to strive to be the operator of choice within its communities. The company won a Clean Air Award from the U.S. EPA in 2004 for its efforts to reduce refinery flaring and thus the emissions created during flaring. FHR has worked with the EPA in a consent decree to minimize all pollution emissions from FHR plants.[18] The specific focus is on start-ups, shutdowns, and malfunctions (SSMs), which often lead to significant flaring events. An example of a flaring event caused by an unplanned shutdown occurred in Wilmington, California, in September 2005 when brown and yellow smoke was emitted from several refineries (none of which were FHR facilities) for more than 8 hours after an area power outage.[19] FHR provides the EPA and state regulators with information on its SSM practices across the regulated community to minimize such emissions.

FHR's refining complex in Corpus Christi, Texas, has dramatically reduced its flaring from the refinery. The West Plant set a plant record in 2007 for going 155 days without flaring. A combination of equipment and operating practices was required to achieve this record. The West Plant has a FGRS system that was installed in the early 1980s (see Figure 14.5). As shown in Figure 14.5, three parallel compressors are used to accommodate the wide range of flow rates. The system was originally installed based on economics, where most, but not all, of the waste gases were recovered.

After the decision was made to dramatically reduce flaring at the refinery, plant engineers analyzed all processes venting waste gases into the flare header. This aided in determining ways to reduce the waste-gas base load so the volume of gases could be handled by the existing FGRS. For example, an improved coker blowdown process minimizes vapor generation with no resultant flaring. Nonroutine waste flows to the FGRS are ceased during coker blowdown operations. Operators began tracking flaring time to identify processes that needed to be modified.

A daily report was reviewed to continually monitor flare events. In some cases, hardware changes were needed to repair or replace leaking equipment. In other cases, this meant procedural changes to plant practices. This took a coordinated effort of operators, engineers, and management to make the changes necessary so that no additional capacity was required in the existing FGRS system. Refinery computer controls were upgraded and centralized, which significantly improved communication and management of the flare system ensuring FGRS capacity availability if a significant flow of waste gas was going to be generated. Alarms were added to alert operators when potential flaring conditions may occur to give them time to adjust operations. Root-cause

FIGURE 14.5
FGRS at FHR West Plant in Corpus Christi, Texas.

analysis was also used to analyze significant unplanned emission events to eliminate future occurrences.

As an example of FHR's commitment to reducing flaring, only 1.77 h of flaring was required in the first half of 2006. Most of that flaring occurred during a planned event. Typically, during an outage at a plant, there would be significant flaring to de-inventory and decommission (purge) the process equipment so that maintenance can be safely performed. To minimize flaring during outages, FHR developed a comprehensive plan to bring down certain equipment at different times so that nearly all of the waste gases could be captured by the FGRS. Most of the 1.77 h of flaring occurred when the FGRS itself was shut down for flare line maintenance.

14.7 Conclusions

There is growing interest in minimizing flaring, in part due to the pollution emissions generated by flaring and potentially significant emission sources within a plant. FHR has approached this problem through equipment modifications and new operating practices, in combination with an existing FGRS. The FHR West Plant in Corpus Christi, Texas, has operated for long stretches of time without any flaring. FHR partnered with the U.S. EPA to help develop best practices that can be applied at other plants to minimize flaring and the associated pollutant emissions that come with flaring.

References

1. API, *Guide for Pressure-Relieving and Depressuring Systems*, Recommended Practice RP 521, 4th edn., American Petroleum Institute, Washington, DC, March 1977.
2. API, *Flare Details for General Refinery and Petrochemical Service*, Standard 537, American Petroleum Institute, Washington, DC, September 2003.
3. R. Schwartz, J. White, and W. Bussman, Flares, in *The John Zink Combustion Handbook*, ed. C. Baukal, CRC Press, Boca Raton, FL, 2001, Chapter 20.
4. J. Hong, J. White, and C. Baukal, Accurately predict radiation from flare stacks, *Hydrocarbon Processing*, 85(6), 79–81, 2006.
5. URS Corporation, *Passive FTIR Phase I Testing of Simulated and Controlled Flare Systems—Final Report*, Prepared for the Texas Commission on Environmental Quality, Austin, TX, June 2004.
6. P.E. Gogolek and A.C. Hayden, Performance of flare flames in a crosswind with nitrogen dilution, *Journal of Canadian Petroleum Technology*, 43(8), 43–47, 2004.
7. J. Hong, C. Baukal, R. Schwartz, and M. Fleifil, Flare testing, *Chemical Engineering Progress*, 102(5), 39, 2006.

8. R. Levy, L. Randel, M. Healy, and D. Weaver, Reducing emissions from plant flares, *Proceedings of the Air and Waste Management Association Conference and Exhibition*, New Orleans, LA, June 2006, Paper # 61.
9. M. McDaniel, Flare efficiency study, U.S. Environmental Protection Agency report: EPA-600/2-83/052, 1983.
10. D. Chenevert, C. Harry, J.H. Walker, B. Unterbrink, and M. Cain, Flare minimization practices improve olefins plant start-ups, shutdowns, *Oil & Gas Journal*, 1039(33), 54–60, 2005.
11. C. Baukal (ed.), *The John Zink Combustion Handbook*, CRC Press, Boca Raton, FL, 2001.
12. P.W. Fisher and D. Brennan, Minimize flaring with flare gas recovery, *Hydrocarbon Processing*, 81(5), 83–85, 2002.
13. R.K. Sharma, Y.B. Prasad, and V. Harishbabu, Minimize your refinery flaring, *Hydrocarbon Processing*, 86(2), 105–106, 2007.
14. O. Zadakbar, A. Vatani, and S. Mokhatab, Gas refineries can benefit from installing a flare gas recovery system, *Hydrocarbon Processing*, 89(8), 51–54, 2010.
15. U.S. Environmental Protection Agency, *Available and Emerging Technologies for Reducing Greenhouse Gas Emissions from the Petroleum Refining Industry*, Office of Air and Radiation, Research Triangle Park, NC, October 2010.
16. J. Peterson, N. Tuttle, H. Cooper, and C. Baukal, Minimize facility flaring, *Hydrocarbon Processing*, 86(6), 111–115, 2007.
17. R. Gough, Flint hills resources shows flare for not flaring, *World Refining*, 14(6), 36–39, 2004.
18. Anon., Pact with oil company may help EPA develop guide on "upset" emissions, *Clean Air Report*, 15(19), September 9, 2004.
19. J. Wilson, Environmental groups sue EPA over refinery emission standards, *The Los Angeles Times, Part B*, June 21, 2006, p. 3.

15

Hydrocarbon Vapor Control Technology

Roger E. Blanton

CONTENTS

15.1 Introduction

Currently, there are two generally accepted methods for controlling unwanted vapor emissions from liquid hydrocarbon and chemical loading, unloading, and storage operations: vapor recovery and vapor combustion. The vapor recovery operation is primarily achieved through carbon adsorption, where the vapors are processed and volatile organic compound (VOC) emissions are reduced by taking the hydrocarbon from the vapor stream and emitting air containing very low levels of hydrocarbon. Carbon adsorption vapor recovery cannot handle as wide a range of hydrocarbons or chemicals as vapor combustion can handle, and it is a more expensive solution. However, it does have the advantage of recovering hydrocarbon (the product) providing a return on investment that will pay off the initial capital investment. Vapor recovery also has the advantage of not producing products of combustion as by-product air pollutants.

Vapor combustion is a more brute force approach. In comparison to vapor recovery, vapor combustion is a less expensive solution, but it does not have a return on investment and has additional operating costs such as electricity, assist gas, and compressed air depending on the design of the system. Vapor combustion also has the disadvantage that while reducing VOC emissions, it will produce NOx and COx (CO and CO_2) emissions as products of combustion. The main advantage of vapor combustion is that it can handle a wide range of hydrocarbons with no significant effect on the design of the combustor.

15.2 Pressure and Vacuum

The term "pressure" is used to describe either negative (below atmospheric) or positive (above atmospheric) pressure (see Volume 1, Chapter 9). Positive pressure is "gauge pressure." The term "vacuum" describes the region of pressure below one atmosphere of pressure, also referred to as negative pressure. When speaking of vacuum, one must remember it as the opposite of pressure: high vacuum means low pressure. A vacuum (in. HgV) reading is similar to gauge pressure (psig), in that the gauge reading references the current atmospheric or barometric pressure (which changes over time and place to place). When the reading references the theoretical absolute zero for a unit of measure, the reading is called an absolute value (psia, in.HgA). Standard atmospheric pressure is 29.92 in.HgA, 760 torr (mmHgA), or 14.7 psia.

When using a vacuum or gauge pressure reading in a calculation, the reading must be converted to an absolute value. This requires taking the current barometric pressure and subtracting the vacuum reading or adding the gauge pressure reading. If a current barometric pressure is unknown, then assume the standard atmospheric pressure (29.92 in.HgA).

Convert a vacuum reading of 22 in.HgV as follows:

$$\text{Standard pressure} - \text{Vacuum reading}$$
$$= \text{Absolute pressure}$$

$$29.92 \text{ in.HgA} - 22 \text{ in.HgV} = 7.92 \text{ in.HgA}$$

When using a gauge pressure reading in a calculation, convert the gauge pressure reading to an absolute value. With a positive gauge reading, add the value to the barometric pressure.

Convert a gauge pressure reading of 22 psig as follows:

$$\text{Standard pressure} + \text{Pressure reading}$$
$$= \text{Absolute pressure}$$

$$14.7 \text{ psia} + 22 \text{ psig} = 36.7 \text{ psia}$$

There are a number of different units of measure used to describe a level of pressure. At standard atmospheric pressure, all the values are equivalent:

$$14.7 \text{ psia} = 29.92 \text{ in.HgA} = 760 \text{ mmHgA} = 760 \text{ torr}$$
$$= 760{,}000\ \mu\text{m} = 1{,}013 \text{ millibar (mbar)}$$

Always use units of torr, microns, and millibar as absolute values. Refer to Table 15.1 for a comparison of these units.

15.3 Hydrocarbon Vapor Control

A series of steps defines the petroleum processing industry whereby various volatile organic liquids are stored and transported. For example, crude oil is pumped from the ground, stored in oil field storage tanks, and transported to refineries via marine vessels, tank trucks, railcars, and pipelines. A refinery processes the crude oil and produces various volatile refined products, such as gasoline. These refined products are then shipped via pipeline, trucks, railcars, and marine vessels to various marketing and distribution terminals where they again are stored and subjected to further transportation before ultimately reaching their end use.

These storage and transportation steps are often accomplished in vessels that are maintained at or close to atmospheric pressure. These vessels can be sources of VOC vapor emissions because they communicate to the atmosphere via simple vents or via low pressure/vacuum relief valves. During the cooler parts of the day, at night, and during vessel unloading, atmospheric air is drawn into the vessels.[1] When the air contacts a volatile organic liquid, the liquid is partially evaporated forming a hydrocarbon vapor–air mixture. As long as the resulting vapor–air mixture is contained within the storage or transport vessel, no emissions occur. However, because of filling operations, tank out-breathing during warmer periods of the day, and vapor growth caused by the evaporation process itself, air containing evaporative VOCs is vented from these vessels. If not controlled, these VOCs are emitted to the atmosphere.

Evaporative VOCs, if emitted to the atmosphere, can be a health hazard in one of two ways. First, many of these chemicals themselves are detrimental to human health and subsequently are classified as hazardous air pollutants (HAPs). Examples of HAPs include such chemicals as benzene, hexane, toluene, carbon tetrachloride, and methyl ethyl ketone. Secondly, all VOCs, even if not classified as HAPs, can be detrimental to human health if emitted to the atmosphere as a vapor. This is because VOCs, often called reactive organic compounds (ROCs), in the presence of the sun's ultraviolet radiation, react with combustion by-product oxides of nitrogen to form ozone. Ozone produced in this manner is formed at relatively low altitudes and is detected by a brown haze often referred to as smog. It can lead to human respiratory problems and other negative health effects.

The extent to which the evaporation of volatile organic liquids occurs and the resultant mix of vapor and air vented from storage and transportation vessels depends on several factors including the vapor pressure of the liquid being handled, temperature, pressure, the method of loading, and the method of unloading. The volatility

TABLE 15.1

Vacuum Conversion Chart

	Torr (mmHg)	mbar	in.HgV	in.HgA	psig	psia	% Vacuum
[a]	760	1013.25	0.000	29.921	0.000	14.697	0.00
	750	999.92	0.394	29.528	−0.193	14.504	1.32
	700	933.26	2.362	27.559	−1.160	13.537	7.89
	600	799.93	6.299	23.622	−3.094	11.603	21.05
	500	666.61	10.236	19.685	−5.028	9.669	34.21
	400	533.29	14.173	15.748	−6.962	7.735	47.37
	380	506.62	14.961	14.961	−7.348	7.348	50.00
	300	399.97	18.110	11.811	−8.896	5.801	60.53
	200	266.64	22.047	7.874	−10.829	3.868	73.68
	150	199.98	24.016	5.906	−11.796	2.901	80.26
	100	133.32	25.984	3.937	−12.763	1.934	86.84
	90	119.99	26.378	3.543	−12.957	1.740	88.16
	80	106.66	26.772	3.150	−13.150	1.547	89.47
[b]	74.2	98.92	27.000	2.921	−13.262	1.435	90.24
	70	93.33	27.165	2.756	−13.343	1.354	90.79
	60	79.99	27.559	2.362	−13.537	1.160	92.11
	50	66.66	27.953	1.969	−13.730	0.967	93.42
	40	53.33	28.346	1.575	−13.923	0.774	94.74
	30	40.00	28.740	1.181	−14.117	0.580	96.05
	25.4	33.86	28.921	1.0000	−14.206	0.491	96.66
[c]	23.4	31.20	29.000	0.9213	−14.244	0.453	96.92
	20	26.66	29.134	0.7874	−14.310	0.387	97.37
	10	13.33	29.528	0.3937	−14.504	0.19338	98.68
	7.6	10.13	29.622	0.2992	−14.550	0.14697	99.00
	1.00 (1000 μm)	1.3332	29.882	0.0394	−14.678	0.01934	99.87
	0.75	0.9999	29.892	0.0295	−14.682	0.01450	99.90
	0.50	0.6666	29.9016	0.01969	−14.687	0.00967	99.9342
	0.10	0.1333	29.9173	0.00394	−14.695	0.00193	99.9868
	0.01	0.0133	29.9209	0.00039	−14.6967	0.00019	99.9987
[d]	0.00	0.0000	29.9213	0.00000	−14.6969	0.00000	100.0000

[a] Atmospheric pressure.
[b] Typical vacuum level of standard efficiency VRU.
[c] Typical vacuum level of high efficiency VRU.
[d] Perfect vacuum (unobtainable).

of the liquid and the operating pressure of the vessel limit the maximum theoretical hydrocarbon vapor concentration that can exist in an air stream vented from a storage or transportation vessel. In general, the more volatile the liquid product is, the greater the potential for the liquid to vaporize when it is contacted with air. Volatility is determined by the liquid's vapor pressure, which, in turn, is determined by the liquid's chemical composition and temperature. For example, gasoline has a higher vapor pressure than diesel fuel at the same temperature. Therefore, air vented from storage or transportation vessels containing gasoline will have a much higher theoretical hydrocarbon vapor concentration than if, for example, the vessel contained diesel. Because increasing temperature increases vapor pressure of the liquid, air vented from storage or transportation vessels will contain higher theoretical hydrocarbon concentrations at higher liquid temperatures than at lower liquid temperatures. Furthermore, vessel pressure has an inverse relationship with theoretical hydrocarbon vapor concentration in the air vented from the vessel; the higher the pressure, the lower the theoretical hydrocarbon vapor concentration. As a guideline, the theoretical hydrocarbon vapor concentration in volume percent can be determined by dividing the liquid's true vapor pressure by the vessel's operating pressure in equivalent absolute units and multiplying by 100. For example, the theoretical gasoline vapor concentration in air for a gasoline with a true vapor pressure of 375 mmHg (200 in. H_2O) being loaded into a tanker truck at atmospheric pressure of 760 mmHg (410 in. H_2O) can be estimated to be (375/760)(100) = 49.3 vol%. In most cases, however, the actual hydrocarbon vapor concentration in the vented air stream exists at something less than the theoretical saturation value. For example, typical

TABLE 15.2

Typical Chemical Composition of Gasoline Vapors

Component	Vol%	Weight%
Air	58.1	37.6
Propane	0.6	0.6
i-Butane	2.9	3.8
Butene	3.2	4.0
n-Butane	17.4	22.5
i-Pentane	7.7	12.4
Pentene	5.1	8.0
n-Pentane	2.0	3.1
Hexane+	3.0	8.0
	100.0	100.0

hydrocarbon vapor concentrations when loading gasoline range from 15 to 45 vol%.

Gasoline is a multicomponent mixture, which varies in composition from region to region, from refiner to refiner, and from season to season. Further, because gasoline is a multicomponent mixture, evaporative hydrocarbon vapor generated by contact of gasoline with air results in a vapor composition different from the parent liquid with a much higher proportion of lighter, more volatile, components.

Table 15.2 lists a typical composition that is generally accepted as representative of gasoline vapor.

The method of filling the vessel influences the volume of vapor vented and the extent to which the hydrocarbon vapor concentration approaches the theoretical value. This can be illustrated by the example of loading a tank truck with a volatile product such as gasoline. In general, the greater the amount of liquid to air contact when loading the tank truck, the greater the evaporation of the liquid product.

The greatest amount of liquid to air contact occurs during splash loading operations (see Figure 15.1). When splash loading, as illustrated by Figure 15.1, the liquid loading hose is simply held over the top of the truck above an open loading hatch allowing the liquid product to fall freely into the truck compartment, which causes splashing and allows a great deal of air-to-liquid contact. Splash loading, because of the high product evaporation rates, causes a close approach to theoretical vapor concentration in the vented air and a larger volume of air–hydrocarbon vapor to be vented than the volume of liquid being loaded. The vapor growth factor is the ratio of the volume of air–vapor vented to the volume of product loaded. With splash gasoline loading techniques, the vapor growth factor can be as high as 1.5. Splash loading techniques are rarely used in conjunction with vapor recovery due to the large vapor growth and the difficulty of collecting the vapors for processing.

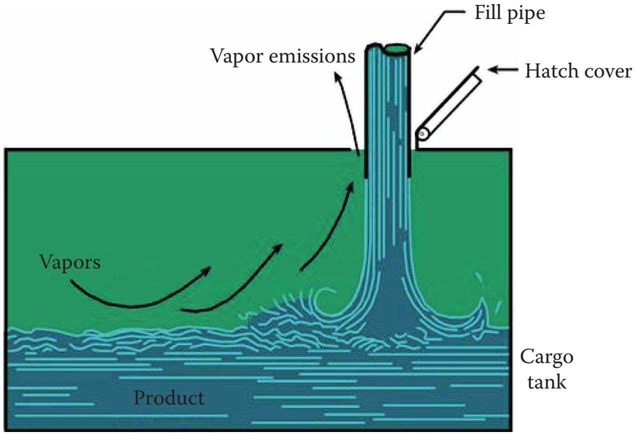


FIGURE 15.1
Splash loading method.

A better loading method, illustrated by Figure 15.2, to minimize evaporation and vapor growth is to employ a technique called submerged loading. With submerged loading, the loading is from the top of the truck, but a liquid fill tube is placed in the open truck hatch, which extends almost to the bottom of the truck. With this method, liquid loading proceeds slowly until the fill tube end is covered with liquid; after which, the rate of loading is increased, but splashing and air-to-liquid contact are minimized. Air and hydrocarbon vapor are displaced from the top of the vessel being filled either through an open hatch or through a dedicated vapor connection. Submerged loading is normally used with marine transport vessels where permanent sealed submerged loading tubes and separate vapor space connections are installed on the vessel. This method is not normally used for trucks because of the difficulty of collecting the displaced vapors and the need to get on top of the trucks to handle liquid loading and vapor collection equipment.

The preferred truck loading method, illustrated by Figure 15.3, for vapor recovery is bottom loading. With this method, the compartments are filled with liquid product from the bottom through bottom loading

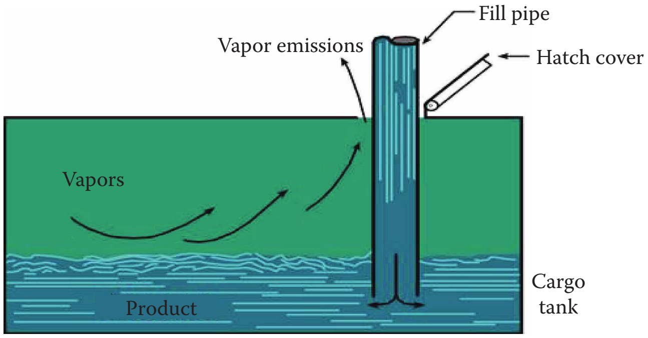


FIGURE 15.2
Submerged fill pipe.

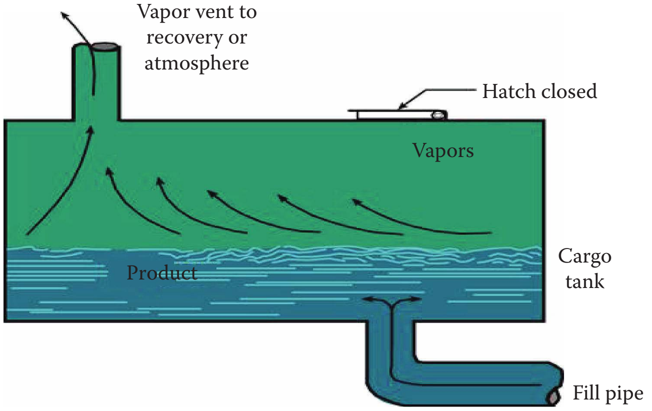
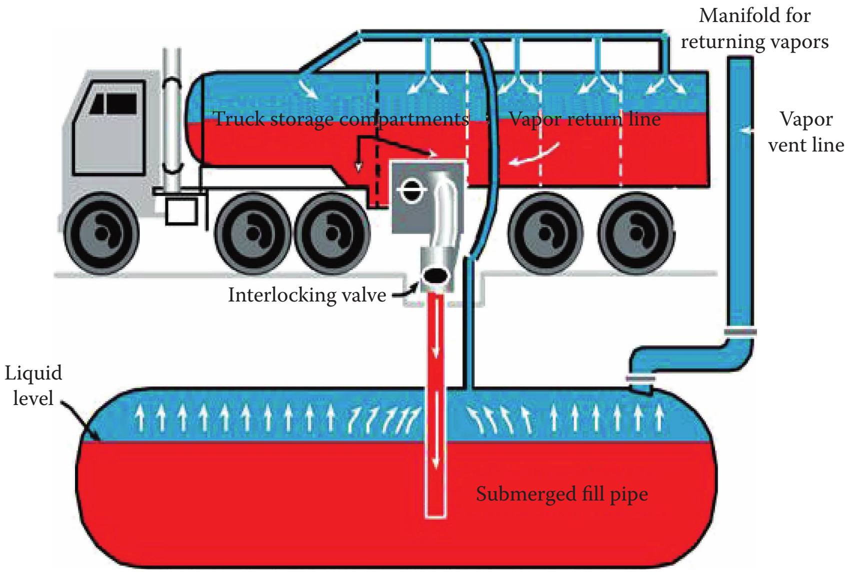


FIGURE 15.3
Bottom loading.

connections on the truck. As liquid is loaded into the trucks from the bottom, air–vapor is displaced from the top through a common vapor manifold, which is connected to a vapor collection hose allowing the vapors to be conveniently collected for processing by the vapor recovery unit (VRU). With this method, as with submerged loading, liquid–air contact is minimized; therefore, product evaporation and vapor growth are minimized. The major advantages of this type of loading for truck transports are that the entire loading operation can be done from the ground without having to get on top of the trucks, and the vapors can be easily collected for recovery. The vapor growth factor utilizing both bottom and submerged loading techniques tends to be close to 1.0.

Sometimes the vented vapor concentration is influenced by the method of unloading the liquid product. This again can be illustrated by using the truck example. One way of unloading a gasoline transport truck at a service station is to simply connect a liquid hose from the truck to the underground gasoline storage tank and then, with the truck hatches open, allow the gasoline to flow by gravity from the truck to the tank. In this type of operation, atmospheric air simply displaces the liquid product in the truck as it is being unloaded. Air/liquid contact is minimized and evaporation of liquid product is relatively small. As a result, the hydrocarbon vapor content of the residual air brought back to the terminal in the truck is much less than the theoretical concentration. One negative aspect of this method of unloading product, however, is that the displaced air–hydrocarbon vapor mixture from the underground storage tank at the service station is vented directly to the atmosphere.

A much better method of unloading gasoline, illustrated by Figure 15.4, at the service station is to practice a method of vapor balance called stage 1. With this method, two connections are made to the truck. A liquid hose is connected from the truck to the underground storage tank and a vapor hose connects the vapor space of the underground tank to the vapor space in the truck. As liquid flows from the truck to the tank, a hydrocarbon vapor–air mixture is displaced from the tank back to the truck. In this manner, minimal outside air is drawn into the truck and vapor emissions to the atmosphere are minimized. Practicing vapor balance at the service station increases the vapor concentration in the air space in



Tank truck unloading into an underground
service station storage tank. Tank truck is
practicing "vapor balance" form of vapor control.

FIGURE 15.4
Vapor control stage 1.

the returning transports to the truck loading terminal. This not only reduces air pollution, but also brings more vapors back to the terminal improving the economic justification for a VRU at the loading terminal.

15.3.1 Air Quality Emission Standards

Because of the negative effects on air quality caused by evaporative hydrocarbon vapor emissions, air quality regulatory agencies worldwide promulgate various regulations limiting the amount of these types of chemicals emitted from petroleum product distribution terminals. In the United States, the U.S. Environmental Protection Agency (EPA), in response to the Clean Air Act, first began to pass regulations affecting petroleum product terminals in the late 1970s.

The regulations were first directed at bulk distribution terminals storing and transporting gasoline by tank truck. These early regulations were generally stated in terms of requiring a specified control efficiency. For example, early regulations required the emission control device to remove 90% of the uncontrolled vented emissions from these terminals. Emission standards based on control efficiency proved to be unfair because they penalize companies that have a lower concentration of hydrocarbon vapor in the inlet feed stream to the emission control device and reward those companies that have a high vapor concentration. An example of how regulations based on a specified vapor control efficiency can be unfair is illustrated by Table 15.3. In this illustration, Terminals A and B both load the same volume of gasoline. However, with a 90% control efficiency requirement, Terminal A is allowed to emit twice as much pollution as Terminal B simply because Terminal A's inlet vapor to the emission control device averages 40 vol% hydrocarbon vapor concentration instead of the 20 vol% concentration at Terminal B. Obviously, this is not fair to Terminal B.

Because regulations based on control efficiency were not fair, and some even promoted practices that were not in the best interest of improving air quality, the U.S. EPA in 1977 promulgated a new regulation affecting gasoline bulk terminals based on limiting the mass of allowable vapor emissions per unit volume of loading activity. The U.S. EPA guideline was set at allowing 80 mg of VOC vapor emissions per liter of gasoline product loaded (mg/L). This basic practice of stating regulations in terms of mass per unit volume of loading activity has continued in the United States with the allowable emission level reduced to requirements that are ever more stringent. For example, in 1983, the U.S. EPA, citing carbon adsorption–absorption vapor recovery technology as being the "best demonstrated vapor recovery technology," reduced the Federal emission guideline at gasoline bulk terminals to 35 mg of VOC vapor per liter of gasoline loaded. After 1983, various state and local air regulatory agencies either complied with the Federal guideline or passed regulations, which are more stringent. For example, several regulatory agencies have set 10 mg/L as the emission regulation. A few regulatory agencies have set the maximum allowable emissions at 5 mg/L or less. In 1994, the U.S. EPA, based on proven performance with John Zink's ADAB™ technology, made the 10 mg/L emission limit the U.S. standard for major source gasoline bulk terminals.

Table 15.4 illustrates the relationship of various hydrocarbon vapor emission control regulations to vapor recovery efficiency.

In 2008, the U.S. EPA promulgated the gasoline distribution generally achievable control technology (GD GACT) regulation 40 CFR 63, subpart BBBBBB, for area sources of HAPs that were not subject to the major source national emission standards for HAPs National Emission Standards for Hazardous Air Pollutants (NESHAP) promulgated in 1994 (maximum achievable control technology [MACT] rule). The GD GACT rule sets national emission standards for HAPs at GD area source facilities that emit less than 10 ton/year (9.1 metric-ton/year) of a single toxic air pollutant or less than 25 ton/year (23 metric-ton/year) of any combination of toxic air pollutants. This rule specifically excludes these areas from meeting the MACT requirements. However, it does require a standard for GACT. Provisions limit air toxic emissions from bulk terminals, bulk plants, pipeline facilities, and dispensing facilities. The GD GACT regulation set an emission limit of 80 mg/L for all bulk

TABLE 15.3

Comparison of Allowable Vapor Emissions at 90% Control Efficiency

Terminal	Gasoline Loaded per Day (Liters)	Inlet Vapor Concentration (Vol%)	Uncontrolled Emissions (kg)	Allowable Emissions at 90% EFF. (kg)
A	2,000,000	40	2202	220
B	2,000,000	20	1101	110

TABLE 15.4

Comparison of U.S. Emission Regulations versus Recovery Efficiency

Inlet Vapor Hydrocarbon Concentration, Vol%	Vapor Recovery Efficiency, Vol% at VOC Emission Regulation			
	80 mg/L	35 mg/L	10 mo/L	5 mg/L
50	94.2	97.4	99.3	99.6
40	92.7	96.8	99.1	99.5
30	90.3	95.8	98.8	99.4
20	85.4	93.6	98.2	99.1
10	70.9	87.3	96.3	98.1

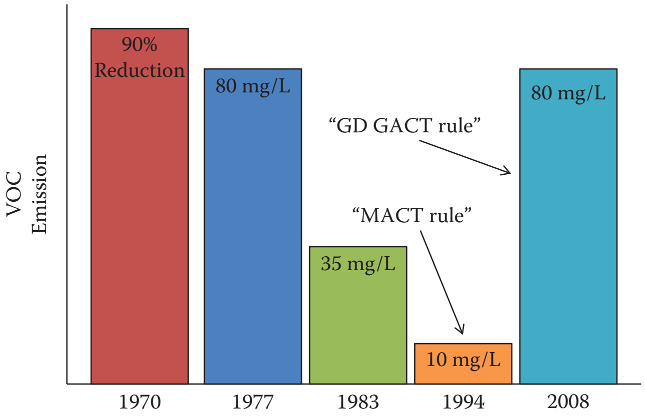


FIGURE 15.5
Summary of U.S. VOC emissions.

TABLE 15.5
Current European VOC Emission Limits

Country	Regulation	Emission Limit
Netherlands	NeR	<500 g/h or <50 mg/N-m^3 (excluding methane)
Austria, Switzerland, Lithuania, Luxembourg	TA Luft I	<150 mg/N-m^3 (excluding methane)
Germany	TA Luft II	<50 mg/N-m^3 (including methane)

loading operations with rates higher than 250,000 gal (950,000 L) within a 24 h period. Area source terminals must comply with the GD GACT rule at minimum, but most area source GD terminals are still obligated to comply with stricter regulations determined by local environmental agencies. Figure 15.5 shows a summary of the evolution of U.S. VOC emission limits.

In general, as regulations have spread throughout the world, U.S. EPA guidelines have been adopted with some slight deviations. The most stringent regulations that currently exist for gasoline bulk terminals are in Europe. For example, the European Commission (EC) regulation states the VOC vapor emission to be limited to 35 mg/L of air vented from the vapor recovery device. A summary of the current European emission limits is shown in Table 15.5.

15.4 Technologies*

15.4.1 Adsorption versus Absorption

Adsorption and absorption are two completely different and distinct physical phenomena. Their only commonality is that both involve the physical transfer of a volume of mass or energy. Adsorption is the process where the outside surface of a piece of material attracts an outside contaminant (atoms), while absorption involves the uptake of the contaminant into the literal structure of the material. A piece of dry sponge soaking up liquid best exemplifies absorption.

Adsorption is different from absorption in the sense that it focuses not on the volume, but on the surface. Figure 15.6 illustrates adsorption and absorption.

The following are technical definitions of each term:

Adsorption (AD): The adhesion of gaseous molecules, in extremely thin layers, to the surface of a solid. This adhesion is brought about by the imbalance in forces existing between the solid and the gaseous molecules. These attractive forces are known as van der Waals forces.

Absorption (AB): To take in and make a part of the whole: to soak up

15.4.2 Activated Carbon

Activated carbon is the "heart" of a carbon adsorption vapor recovery system. A thorough understanding of this remarkable material is important.

Activated carbon is a powdered, granular, or pelleted form of amorphous carbon characterized by very large surface area per unit volume because of an enormous number of fine pores.[2] It is a crude form (type F1) of graphite, with a highly porous, random structure over a broad range of pore sizes, from visible cracks and crevices to gaps and voids of molecular dimensions between the plates of graphite. Activated carbon is among the few pure forms of carbon, containing almost no nitrogen, hydrogen, halogens, sulfur, or oxygen as produced.[3] Almost any carbonaceous raw material (e.g., wood, coal, peat, coconut shells, olive pits, bone, petroleum coke, vinyl copolymers, or recycled tires[4]) can be used for the manufacture of activated carbon. Granular activated carbon is most commonly produced by grinding the raw material, adding a suitable binder to give it hardness, and then recompacting and crushing the material to the desired size.[5] Most modern vapor recovery systems are now using an extruded (pelletized) form of activated carbon.

15.4.2.1 Activation

Activation is the process of treating the carbon to open a very large number of pores from less than 5 to over 1000 Å (1 Å = 10^{-8} cm). Activation consists of removing carbon atoms by gasification reactions, which occur under controlled atmospheric conditions at high temperatures. After activation, the activated carbon has the broad pore energy distribution (or pore size) responsible for the adsorption phenomena.[6] Activation can

---

* Sections 15.4.1 and 15.4.2 were adapted and updated from Blanton, R.E., Technical Paper, presented at the *Independent Liquid Terminals Association's Supplier Patron's Trade Show*, June 12, 1996.

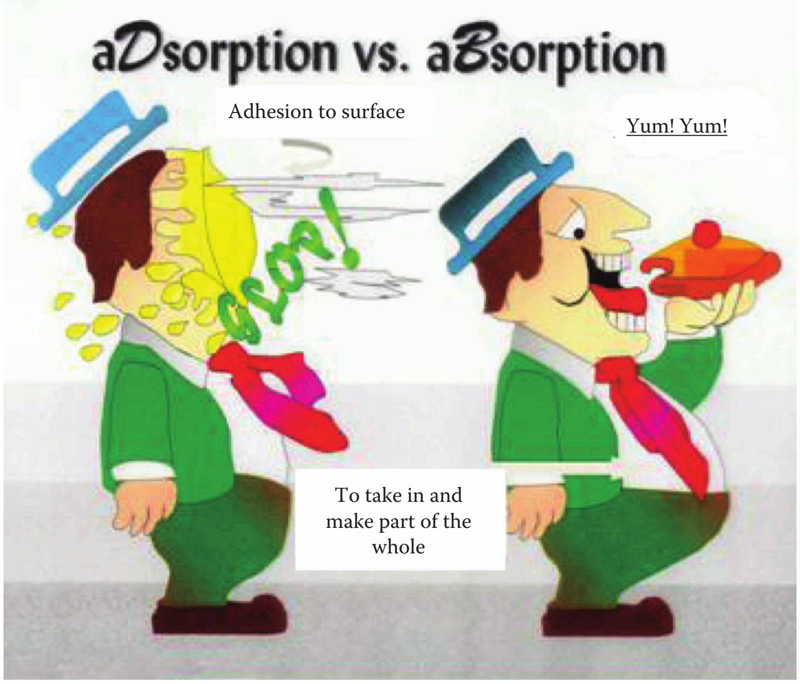


FIGURE 15.6
Adsorption versus absorption.

only "develop" an activated carbon skeleton, using the building blocks supplied by the original starting material. The resultant activated carbon skeleton defines the adsorption pore structure and, thus, all the physical adsorption characteristics of the ensuing activated carbon products and these in turn dictate the performance of the activated carbon product in any application.[3]

15.4.2.2 Adsorption

Adsorption is a separation process in which certain components (adsorbates) of a process stream are selectively transferred to the surface of a solid (adsorbent).[4] All solid surfaces tend to attract components of gases surrounding them. Nearly all vapors tend to adsorb onto inorganic solids at temperatures not too much above their boiling point.[7]

The adsorbates are adsorbed onto solid surfaces by attractive forces similar to van der Waals' forces. The intermolecular attractive forces, which cause vapors to adsorb (or condense), are generally dominated by the London dispersion force, or simply the London force, an attraction caused by the perturbation of electron orbits by adjacent atoms.[7] It is named after Fritz London, who derived the dispersion force from the first principles of quantum mechanics in 1931. London forces are the most common of the six types of van der Waals' forces that are responsible for most nonideal behavior of gases. The London force is an intermolecular interaction that exists between all molecules (both polar and nonpolar), but it is extremely short ranged. It can be considered negligible with a separation greater than about two molecular layers (much like a small magnet on a refrigerator door). Therefore, the adsorption forces will be significant only if the gaps or voids within the activated carbon structure (pore widths) are less than four or five molecular layers.[3] These attractions, generally proportional to the square of the polarizability per unit volume, are much stronger for most inorganic solids than for water or organic materials, and that is why inorganic solids are the stronger adsorbents.[7] The attraction to the surfaces of the activated carbon is so strong that the organic vapors eventually condense in the pores.[8] A condensation process accurately describes the thermodynamics of the adsorption process. The adsorbed material within the molecular scale pores is not a true condensed phase, but its properties can be approximated as such for modeling purposes.[3] Adsorption is also improved with increased pressure and reduced temperature.[2]

In simple terms, physical adsorption occurs because all molecules exert attractive forces, especially molecules at the surface of a solid (pore walls of activated carbon), and these surface molecules seek other molecules to adhere to. The large internal surface area of activated carbon has many attractive forces, which work to attract other molecules.[5]

An adsorbent must have three important features to be cost effective in a separation process. First, it must selectively concentrate the adsorbates of interest from the process stream. Second, the adsorbent must bind the adsorbates reversibly so that the adsorbent can be used repeatedly. Third, the adsorbent should possess, as high as possible, a delta loading or working capacity (the change of weight of adsorbate per volume of adsorbent between adsorbing and desorbing steps) over a reasonable range of pressure.[9] The amount of adsorbate adsorbed per unit mass of adsorbent is determined as a function of vapor pressure (or adsorbate concentration) at a given temperature: this is the adsorption isotherm.[7]

A strength of adsorption is that it can selectively remove adsorbates present in small concentrations, to obtain a high degree of removal. Additionally, the process can operate at low temperatures, important in applications where thermally sensitive products are involved.[10] Adsorption on activated carbon is a dynamic equilibrium process in the sense that the "wave front" is moving through the carbon bed. It does however reach a steady state velocity and all systems eventually reach a "pseudo" steady state.[6] In addition, poisons or solids present in the process stream can significantly affect the performance of an adsorbent.[10]

15.4.2.3 Regeneration

Desorption is the reverse of adsorption whereby the adsorbate is removed from the surface of the adsorbent.[10] The reversal of the physical adsorptive conditions (temperature, pressure, or concentration) more or less completely regenerates the activated carbon's activity and frequently allows recovery of the adsorbate.[2] The regeneration step normally represents a major component of total operating cost. In pressure swing adsorption (PSA), the bed is regenerated by reducing the pressure; in temperature swing adsorption (TSA), by increasing temperature.[10]

15.4.2.4 Granular Activated Carbon Structure

Since adsorption takes place at the molecular level, inside relatively small particles of activated carbon, a model is frequently used to visualize the process. Figures 15.7 and 15.8 illustrate the commonly envisioned model. Figure 15.7 shows a single particle of granular activated carbon with its characteristic surface cracks. Figure 15.8 shows the commonly envisioned pore structure with adsorbate molecules in the pores of the activated carbon. Suppose the large molecules shown in Figure 15.8 are adsorbed so tightly onto the surface of the activated carbon that they are not removed in the mild PSA regeneration method as is commonly used in gasoline vapor recovery applications. It appears then that over time the working capacity of the activated carbon would be critically reduced. This question presents a challenging dilemma to scientists involved in the phenomena of adsorption because the operating history of gasoline VRUs does not reflect this critical reduction in working capacity.

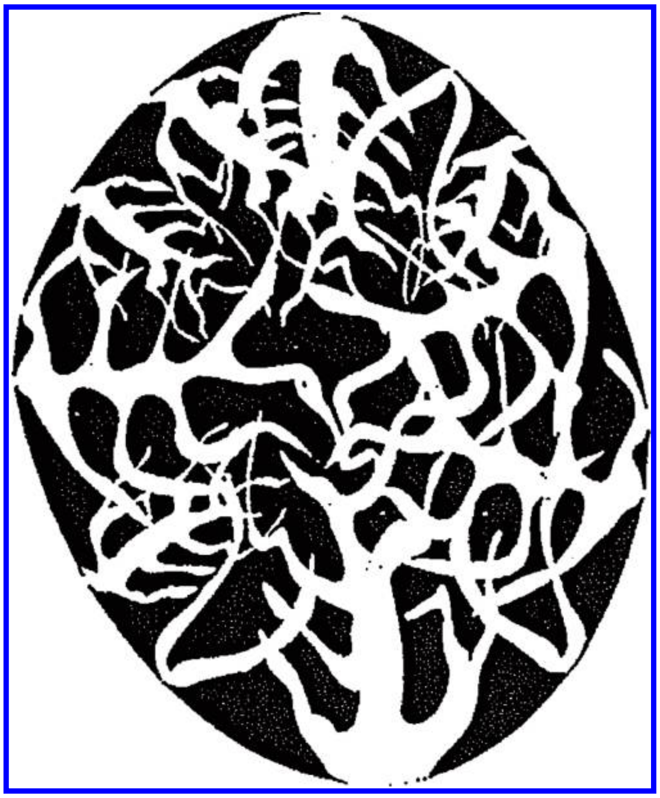


FIGURE 15.7
Single particle of granular activated carbon.

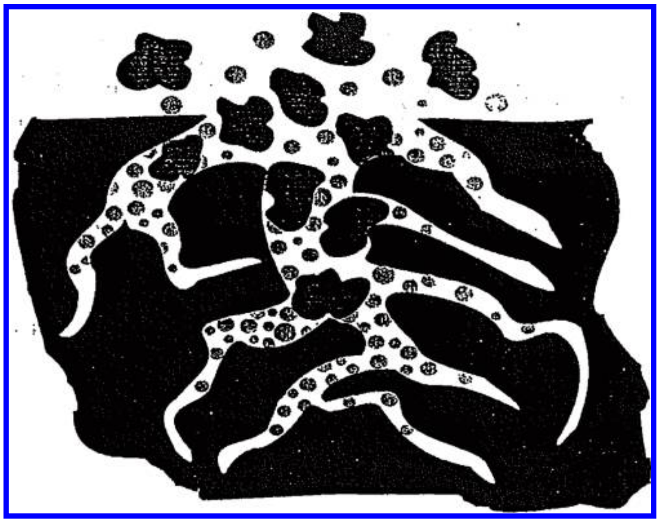


FIGURE 15.8
Pore structure with adsorbate molecules in the pores of the activated carbon.

Scientists set out to explain this problem. They started by abandoning the commonly envisioned model and worked with basic facts. As previously stated, activated carbon is a crude, type F1, form of graphite, with a random or amorphous structure, which is highly porous, over a broad range of pore sizes, from visible cracks and crevices to gaps and voids of molecular dimensions. Intermolecular attractions in the smallest pores result in adsorption forces, which cause condensation of adsorbate gases into these molecular scale pores.[11]

This work resulted in a new view of the structure of activated carbon. This view is based on previously suggested graphite plate designs, but modified to be consistent with all data available. Additional information was obtained from manufacturing process studies, microscopy, and adsorbent characterization databases. The resulting structural model is radical new thinking in the field of activated carbon and is at the forefront of activated carbon technology.[11]

There are two types of activated carbon pore structures, adsorption pores and transport pores. Adsorption pores are the finest pores in the carbon structure and are the only regions within the activated carbon particle with significant adsorption forces and any adsorption properties.[3] Transport pores are pores larger than the largest adsorption pores. They generally fall within the range from 100 to 1000 Å and act only as diffusion paths to transport adsorbates. As a granule of activated carbon is examined in increasing powers of magnification from 10X, visible scale, to 10,000,000X, molecular scale, the pore structure of the activated carbon is revealed. This is analogous to traveling across the United States along interstate freeways, to individual state highways, to major arterial avenues in a city, and to the smallest streets of a neighborhood, where the adsorbate molecules are cars, transport pores are carbon highways, and adsorption pores are carbon parking lots. The cars are only stored in the parking lots, and the highways are used to move cars to the parking lots.[3] At the molecular scale (100 Å), the graphite plate structure of the activated carbon can be observed. Because the activated carbon is a crude form of graphite, there are aliphatic dislocations (imperfections) in the plates, and there is interbonding of the plates. That is why activated carbon is not a good lubricator like graphite (it is actually abrasive in nature) even though both are composed of carbon molecules.[11]

Once this new picture of activated carbon emerged, there again was a need for a model to visually represent the structure. In thinking about the structure, flat plates with imperfections and interbonding of the plates, the similarity with peanut brittle candy was obvious. Therefore, a better model of activated carbon structure, than the commonly envisioned pore structure, is a jar of peanut brittle with flat plates either parallel or connected and the imperfections of the plates of graphite represented by the peanuts.[11] This explains the dilemma previously presented because now the "pore" is viewed as a relatively broad region compared to a more or less cylindrical pore in the commonly envisioned model.

A working definition of activated carbon is then a disperse mixture of a wide variety of graphite plates, interconnected with nongraphitic carbon bonding, organized in a random or amorphous fashion, exhibiting only a slight located order, and only on a molecular scale. This structure enables activated carbon to exhibit the strongest physical adsorption forces or the highest volume of adsorbing porosity of any material known to man.[11]

15.4.2.5 Granular Activated Carbon Properties

Activated carbon can be characterized by physical properties, chemical properties, adsorptive (thermodynamic) properties, and reactivation. All become important factors in the proper selection of activated carbon.

15.4.2.5.1 Physical Properties

Important physical properties are density, mesh size, and abrasion resistance. Density can be a major consideration in the selection of an activated carbon for a specific application. The densities of activated carbons vary with the raw material. The bed density or apparent density (AD) is the weight of carbon in a standard size cylinder, filled in a manner to provide the densest possible bed of particles, and is usually stated in units of grams per cubic centimeter (g/cm^3, also known as g/cc). Since the carbon skeleton is the only mass within the cylinder, the AD relates pore and void volume to the mass of the carbon skeleton.[3]

Mesh size (8 × 30, 12 × 40, 20 × 50, etc.) establishes the range of particle sizes and thus the effective particle size, which will be used in a carbon bed.[5] A mesh size of 6 × 16, for example, means that, in general, the particles will be larger than will pass through a 16 mesh (16 wires/in.) screen and small enough to pass through a 6 mesh screen. Particle size is an important parameter in specifying activated carbon for a specific application. The particle size directly affects the pressure drop through a carbon bed. While a smaller particle causes a higher pressure drop across a carbon bed, the diffusion path length and therefore the effective/apparent mass transfer rate of the adsorbate into the pore are shortened.[6] Subsequently, its adsorption onto the surface of the activated carbon is significantly increased.[5]

Abrasion resistance refers to an activated carbon's ability to withstand degradation during handling and system operation. It is expressed in terms of an abrasion number. The higher the abrasion number, the more resistant the carbon is to abrasion.[5]

15.4.2.5.2 Chemical Properties

Important chemical properties are ash content and reactivity. Ash level reflects the purity of the activated carbon. Ash is the inorganic residue left after activating the raw material. Common ash constituents of coal-based activated carbon are silica, alumina, iron, calcium, and magnesium.[6] Activated carbon made from a high ash content material, like wood, generally has some sort of chemical acid treatment step to reduce the inorganic constituents, which cannot be activated.

The reactivity potential of an activated carbon refers to its potential to cause exothermic (heat generating) chemical reactions within the bed of activated carbon, which, in turn, can lead to excessively high and dangerous bed temperatures. It is extremely important to select a carbon that has an acceptable reactivity level consistent with other desirable properties.[12] A test that is used to present the reactivity of activated carbon is to load it with the solvent of interest and increase the temperature of the loaded carbon in air to observe the temperature at which the solvent combusts. This test is done in a differential scanning calorimeter (DSC).

15.4.2.5.3 *Adsorptive (Thermodynamic) Properties*

Important thermodynamic properties are working capacity, butane working capacity (BWC), and surface area. The working capacity of activated carbon is the maximum quantity of adsorbate that can be adsorbed under actual, long-term operating conditions of the specific application without exceeding the allowed emission criteria.[12] The concept of volume activity then becomes important when evaluating activated carbons. It is the working capacity on a volumetric basis that is of real importance, not the working capacity on a weight basis, because the vessel volume is specified.

Design working capacities for specific applications are not to be confused with BWCs published by activated carbon manufacturers. BWC is a standardized ASTM test, which measures the maximum mass of butane that can be adsorbed onto the surface of virgin activated carbon minus the mass of butane that remains on the activated carbon after a specified air strip procedure. BWC is generally reported as grams of butane adsorbed per 100 mL of activated carbon.[12] In the related application of gasoline vapor emission controls for automobiles, the industry has recognized that butane is a poor substitute for gasoline vapor.[13]

Activity characterizations are key indicators of an activated carbon's potential performance for removing adsorbates. An important characterization tool used in determining the ability of an activated carbon to adsorb a particular adsorbate is the pore size distribution. The pore size distribution of the activated carbon is generally determined using gas-phase adsorption and liquid intrusion techniques. Gas-phase adsorption techniques (carbon dioxide, nitrogen, light alkanes) are used to measure the pore volume distribution in the adsorption region (5–100 Å) (micropore region).

Since many of the properties of activated carbon, including pore structure, can be varied by the selection of the raw material and the parameters of the activation process itself, different types of carbons are more suitable for specific applications. For example, when used in gas mask applications, a carbon with a high proportion of micropores is desirable since relatively small amounts of throwaway carbon are used, the gas molecules to be adsorbed are relatively small, and maximum adsorption capacity per unit of carbon is needed. On the other hand, regenerable gasoline vapor recovery systems used in bulk terminals require a carbon that has a higher proportion of mesopores because the carbon used in these applications is required not only to adsorb hydrocarbons, but also to release them during a carbon regeneration cycle so that the carbon can be used repeatedly.

For bulk terminal vapor recovery applications, it is important to select a carbon with properties that are appropriate for the application. While a carbon with high adsorption capacity is desirable, it is equally important to select a carbon with low retentivity. Low retentivity means that previously adsorbed molecules can be easily removed from the carbon during regeneration so that the carbon can be reused. It is also important that the carbon be of a uniform and correct particle size to allow accurate prediction and minimization of carbon bed pressure drop. Furthermore, the carbon must have thermal and chemical stability to be safely used with hazardous and flammable air–hydrocarbon vapor mixtures. The molecules encountered in gasoline vapor recovery applications are relatively small. Therefore, the carbon selected for this application will have its pore size concentrated in the micropore region. Finally, the carbon should have sufficient mechanical strength to allow a long service life.

15.4.3 Carbon Adsorption/Liquid Absorption: ADAB™ Vapor Recovery System

The John Zink Company and another company (McGill Inc.) simultaneously developed the carbon adsorption–absorption vapor recovery technology in the late 1970s and early 1980s. The John Zink Company consolidated the experience and knowledge, which was gained by both companies, with the acquisition of McGill Inc. in 1991. Since 1980, the John Zink Company has supplied over 1100 carbon adsorption–absorption-type VRUs. These installations represent a large majority of the total number of VRUs, which are currently operating in gasoline bulk distribution terminals around the world.

The John Zink hydrocarbon VRU is based on the technology of ADsorption–ABsorption (ADAB™). This technology has gained worldwide preference by both users and environmental quality control agencies as the most efficient, cost-effective, and reliable technology for evaporative hydrocarbon vapor emission control.

15.4.3.1 S3-AAW Series 3 Adsorption Absorption Wet (S3-AAW) VRU System

The VRU is designed to process an inlet feed stream consisting of hydrocarbon vapor mixed with air, or, in some cases, the hydrocarbon vapor may be mixed with an inert gas such as nitrogen. For purposes of this description, it will be assumed that the hydrocarbon vapor is mixed with air.

The ADAB™ process, when compared to other vapor control technologies, is not only very efficient, but is relatively simple. It can be summarized as a two-step process. The first processing step (adsorption) consists of passing the feed stream through a bed of activated carbon, which serves to capture the hydrocarbon vapor by adsorption onto its surface while allowing the hydrocarbon free air to pass through and vent to the atmosphere. The adsorbed hydrocarbon vapor is then desorbed (removed) from the activated carbon using a vacuum system. This desorbed hydrocarbon vapor discharging the vacuum system is then subjected to a second processing step (absorption) in which it is recovered as a liquid product by absorption into a stream of circulating liquid absorbent.

The hydrocarbon vapor feed stream, consisting of both hydrocarbon vapor and air, is displaced from the product loading operation where it is captured and piped to the VRU. Generally, this vapor feed stream is caused to flow to the VRU by a buildup of a relatively small back pressure in the transport vessel being loaded. In some cases, a vapor-assist blower is included as a motive force to cause the vapors to flow to the VRU. In other cases, a vapor holder is used to collect the vapors and a vapor blower is used to move the vapors from the vapor holder to the VRU. The vapor feed stream, once it gets to the VRU, is caused to flow through an adsorber filled with a bed of activated carbon. In the adsorber, the activated carbon adsorbs (captures) the hydrocarbon vapor from the feed stream onto its surface while allowing the residual hydrocarbon free air to vent from the adsorber to the atmosphere. Figure 15.9 illustrates this process.

Activated carbon has the ability to selectively attract and capture (adsorb) onto its surface the hydrocarbon vapor fraction from the hydrocarbon vapor–air feed stream. However, because the carbon has a finite ability to adsorb hydrocarbon molecules, it must be regenerated; otherwise, it would become saturated and would not adsorb further vapor. Factors that favor adsorption include higher hydrocarbon concentrations in the inlet vapor stream and higher pressures. The John Zink vapor recovery technology has the ability to regenerate the carbon for reuse by reversing the factors that are favorable for adsorption. During the carbon bed regeneration cycle, desorption (removal) of hydrocarbon vapor from the carbon bed is accomplished by creating a high vacuum (low absolute pressure) in the adsorber. This along with the addition of a small amount of purge (stripping) air into the adsorber at the highest vacuum level creates a condition favorable for desorption and provides a very effective means of regenerating the activated carbon for reuse in the adsorption cycle after the regeneration cycle.

In order to allow continuous and uninterrupted vapor processing capability, two identical adsorbers, filled with activated carbon, are provided. While the carbon bed in one adsorber is on-line adsorbing hydrocarbon vapors, the carbon bed in the other adsorber is off-line being regenerated. Switching valves automatically alternate the two carbon beds from the adsorption to the regeneration mode of operation. This switching typically occurs on a timed cycle lasting from 10 to 20 min. At the end of the regeneration cycle, the adsorber vessel is repressurized back to atmospheric pressure and then is placed onstream.

The vacuum system provided with each John Zink S3-AAW VRU includes a liquid ring vacuum pump (LRVP) to regenerate the carbon beds. Some larger systems may have two or more LRVPs operating in parallel. Each LRVP requires a sealing fluid to operate. This seal fluid is a specially blended ethylene glycol-based coolant.

For some applications, the VRU may be provided with an enhanced vacuum system used for carbon bed regeneration. With these designs, in addition to the LRVP, a vacuum booster blower (VBB) is provided to operate in series with the LRVP. The combination of the LRVP and VBB allows the carbon beds to be regenerated under a higher vacuum (lower absolute pressure) and provides significantly higher pumping capacity at high vacuums than is possible through the use of only the LRVP. This enhanced vacuum system more thoroughly regenerates the activated carbon and can, as a result, provide several potential benefits including use of less carbon, the achievement of lower emission levels, reduced system power requirement, and less overall capital cost. In some cases, multiple VBB and LRVP combinations may be used.

During the carbon bed regeneration cycle, the vacuum system creates a high vacuum level in the adsorber being regenerated reducing the adsorber pressure from atmospheric pressure to either 3 in.HgA (100 mbara) or 1 in.HgA (35 mbara) depending upon which type of vacuum system is supplied. The LRVP-VBB combination vacuum system is designed to achieve the latter. At these high vacuum levels, a small amount of purge (stripping) air is introduced into the adsorber to assist in the regeneration process. Regeneration of the carbon bed results in the previously adsorbed hydrocarbon vapor being extracted from the carbon bed and the reestablishment of its adsorption capacity. This extracted rich hydrocarbon

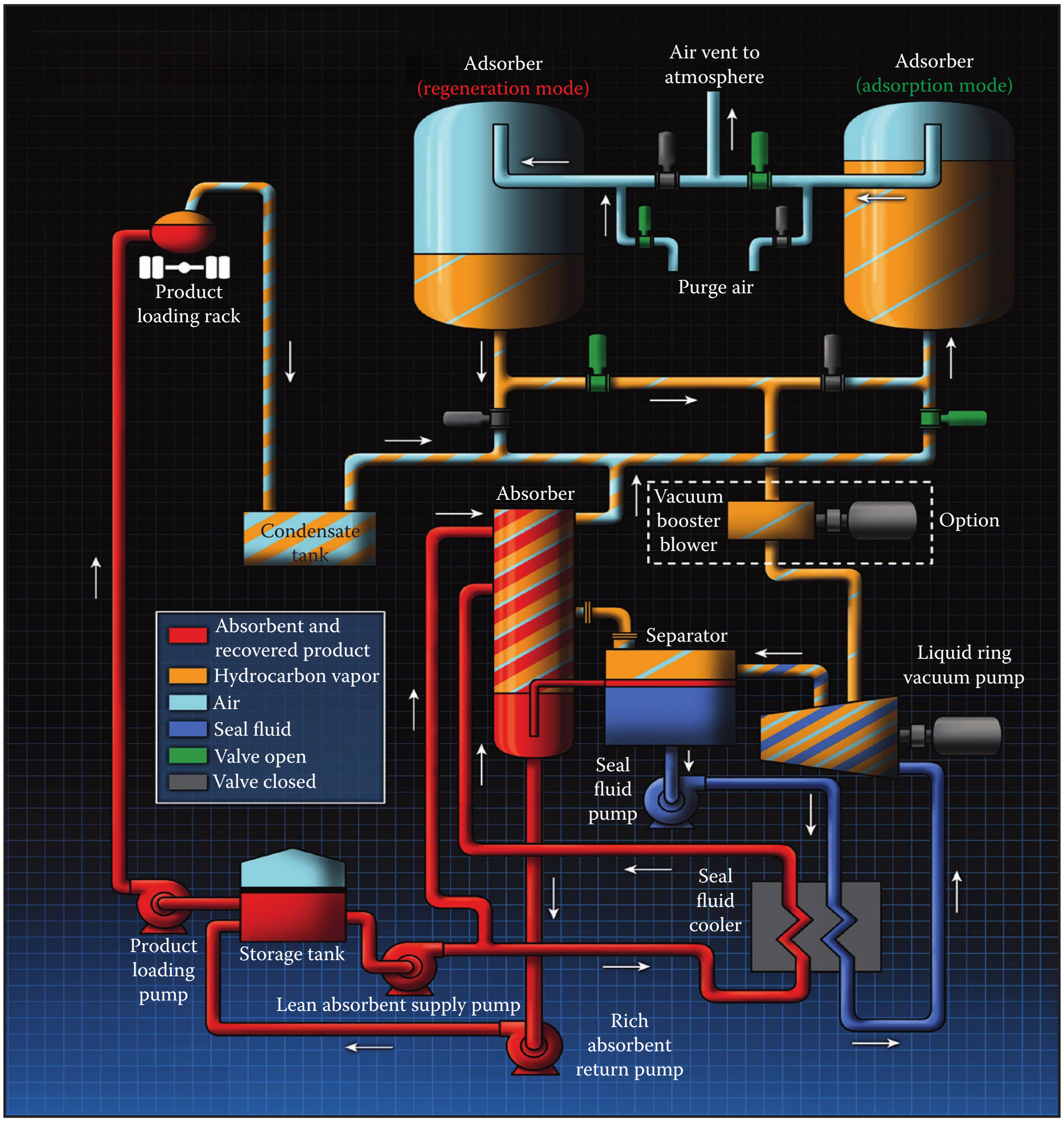


FIGURE 15.9
Carbon adsorption–absorption (ADAB™) process.

vapor stream, containing only a minor amount of air, flows from the adsorber to the vacuum system where it mixes with seal fluid in the vacuum pump and then is discharged from the vacuum pump into a three-phase separator.

The three-phase separator is provided to allow efficient separation between the vacuum pump's ethylene glycol-based seal fluid, any hydrocarbon liquid that may have condensed in the vacuum pump, and the noncondensed hydrocarbon vapor. The seal fluid is heavier than, and is immiscible with, the hydrocarbon condensed liquid. Therefore, it settles to the bottom of the separator where it is removed and circulated back to the vacuum pump using a seal fluid pump. A seal fluid cooler removes the heat of compression from the seal fluid before the seal fluid returns to the LRVP. Noncondensed hydrocarbon vapor, with a small residual air component, disengages from the liquid in the separator and flows from the separator into the base of a downstream vertical absorber column. In addition, if there is any condensed hydrocarbon liquid, it will float on top of the seal fluid in the separator as a second liquid phase. It will eventually build to a high enough level so that it will also flow, via skim piping, to the absorber column.

In the absorber, the noncondensed rich hydrocarbon vapor stream from the separator flows up through packing where it is liquefied and subsequently recovered by absorption into a down flowing liquid absorbent. The circulating absorbent, containing the recovered hydrocarbon, collects in the bottom of the absorber where it returns to the absorbent storage tank as rich absorbent. A small stream of air containing a saturated amount of nonabsorbed hydrocarbon vapor exits the top of the absorber column and is recycled to the onstream carbon bed where the hydrocarbon vapor is readsorbed.

A continuous stream of lean liquid absorbent circulates to the VRU. This liquid absorbent stream is normally the hydrocarbon liquid that was the original source of the vapor generation. In gasoline vapor control applications, for example, the gasoline product from an external storage tank is normally used as the absorbent fluid. A lean absorbent supply pump circulates the lean absorbent to the VRU. This lean absorbent serves two purposes. A portion of it flows to the top of the absorber column where it is used as the primary absorbent to recover the hydrocarbon vapor. The remainder of the lean absorbent is first used as the cooling fluid in the seal fluid cooler and then flows to the absorber where it is used as a secondary hydrocarbon vapor absorbent. The rich absorbent containing the recovered hydrocarbon vapor returns to the absorbent storage tank. A return absorbent pump is provided for this purpose.

The VRU is designed for automatic unattended operation. It will start and run automatically when transports are being loaded with product. Likewise, when the loading operation is completed, the VRU will automatically shut down in a standby mode.

In the carbon bed, there are three distinct zones, as illustrated by Figure 15.10. The first zone is the saturation zone, which is in the lower section of the bed. In this zone, the carbon pores have been filled with hydrocarbon vapor molecules such that no additional adsorption can take place. Vapors exiting the saturation zone have the same chemical composition as the inlet vapors to the carbon bed. Above the saturation zone is the mass transfer zone (MTZ) where adsorption is taking place. With multicomponent vapors such as gasoline, the MTZ can be relatively deep. In the bottom portions of the MTZ, heavier, less volatile, components are adsorbed displacing lighter, more volatile, components to the upper portions of the MTZ. The exiting air stream at each higher level of the MTZ becomes lower in hydrocarbon concentration and the hydrocarbon constituent of the stream becomes lighter. Finally, only a very small residual hydrocarbon vapor concentration is in the air stream exiting the top of the MTZ. The third zone of carbon in the carbon bed lies above the MTZ. This zone contains freshly regenerated carbon where no adsorption is taking place. During an adsorption cycle, the MTZ gradually rises in the carbon bed using more and more of the regenerated carbon. The system must be designed such that the MTZ does not rise to a point that will cause excessive hydrocarbon vapor emissions to the atmosphere before the bed is regenerated.

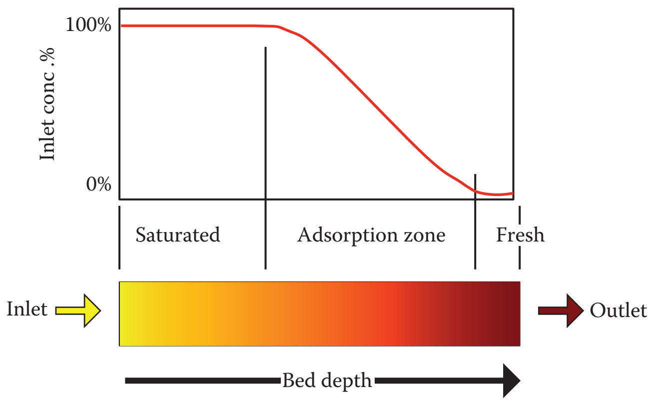


FIGURE 15.10
Three carbon bed zones.

15.4.3.2 Series 3 Adsorption Absorption Dry (S3-AAD) VRU System

For several years, the latest vapor control solutions have utilized dry vacuum pump (DVP) technology with scores of systems in operation.[14] Systems manufactured in recent years have been designed on the heels of a comprehensive research and development effort intended to refine the operation of the system and to maximize the reliability and life cycle of an essential system component: the rotary screw DVP (see Figure 15.11).

Tight clearances between two rotating screws and the walls of the pump's process cavity allow the machine to pull deep vacuum (low pressure) on the regenerating activated carbon bed. The vapor recovery process necessitates that the selected vacuum system pull down the carbon bed from atmospheric pressure down to about one-tenth of atmospheric pressure or less. At times, it is necessary for the vacuum system to generate a 10:1 or greater compression ratio, which in turn generates a great deal of heat. Figure 15.12 shows the process schematic for an S3-AAD VRU.

Traditional vapor recovery designs based upon LRVP technology (see Figure 15.13) had a built-in heat sink in the form of the circulating seal fluid stream, which would absorb the majority of the LRVP's heat of compression. The DVP, on the other hand, does not require a circulating seal fluid to generate deep vacuum. The DVP's heat of compression must still be removed, however, to prevent thermal expansion of the DVP's materials of construction. This heat is removed via the circulation of absorbent stream through an external cooling jacket and a small amount of absorbent liquid injection.

The compression performed by the DVP is variable throughout an individual regeneration cycle as the

FIGURE 15.11
John Zink DVP VRU.

pressure in the carbon bed steadily drops; therefore, the heat generated by the DVP also varies within each regeneration cycle. As a result, John Zink engineers have developed a proprietary method to optimize the absorbent liquid injection rate maximizing the life of the DVP's seals and bearings. This is one of several advancements made in recent years to improve the operability of DVP vapor recovery technology.

15.4.4 Vapor Combustors

15.4.4.1 Process Description

In order to discuss design philosophy, the design of the vapor combustor system must be broken down by application into the following three categories:

1. Truck and railcar vapors
2. Marine vapors
3. Storage tank vent vapors

Another component to the design is whether the vapor combustor will comply with 40 CFR 60.18. By meeting 40 CFR 60.18 for an open-flame combustor, the vapor combustor does not have to be tested to verify it has met the destruction level of VOC (typically 98% destruction efficiency). It does need to be tested by the customer to measure the heating value of the waste vapor, measure the flow rate to ensure maximum velocity is not exceeded, and verify the stack effluent opacity. This test ensures the vapor combustor operates with smokeless combustion. The disadvantage of this design is that the assist gas usage is higher than that of a temperature-controlled unit. In order to comply with 40 CFR 60.18, the incoming vapors must have a lower heating value of 300 Btu/scf (11 MJ/m^3). Therefore, if an application has a very lean hydrocarbon concentration or can be air at the beginning of operation, the incoming vapor must be supplemented with fuel gas to increase its lower heating value to 300 Btu/scf (11 MJ/m^3). Without some type of gas analyzer, there is no way to determine if the incoming stream has enough heating value to warrant reducing the assist gas. Therefore, in order to comply, the vapor piping must be sized for the vapor flow as well as the additional assist gas, and the vapor combustor must be big enough to handle the heat release from the vapor (at its highest heating value) plus the additional heat release from the assist gas. If complying with 40 CFR 60.18 is undesirable due to the high consumption of assist gas or the additional cost due to the larger unit, a temperature-controlled unit may be provided instead.

A temperature-controlled vapor combustor is designed to maintain a temperature setpoint. Instead of adding a predetermined amount of assist gas, the temperature-controlled unit monitors the stack exit temperature, and if the hydrocarbon level in the vapor stream is low, the unit adds assist gas until the temperature is met. As the vapor stream's heating value increases, so does the heat release yielded in the stack. As the vapor stream adds more heat to the system, the temperature-controlled unit will reduce the added assist gas to reduce the temperature. If the assist

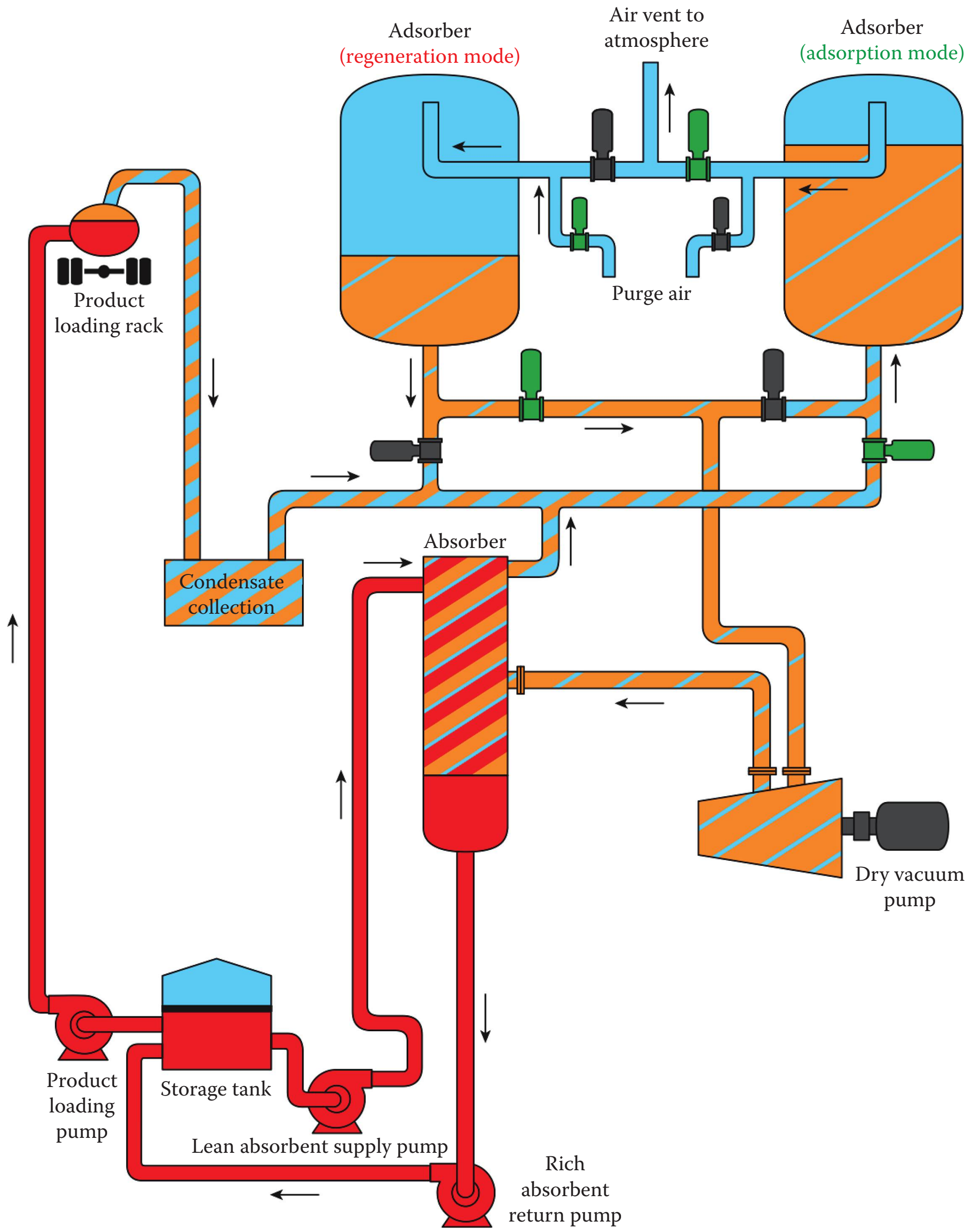


FIGURE 15.12
Process schematic for an S3-AAD VRU.

gas completely shuts off, the temperature-controlled stack will then open the quench air dampers to cool the temperature of the stack exhaust if it continues to increase.

The system normally consists of a combustion chamber, special antiflashback burners, automatic ignition pilot with continuous monitor, motor-operated vapor block valve, detonation arrestor, air-assist blower, piping, instrumentation, and master control panel packaged as an assembled unit ready for convenient field installation. Figure 15.14 illustrates these items. Figure 15.15 shows several photos of vapor combustors in the field.

15.4.4.2 Combustor Applications

15.4.4.2.1 Truck Loading or Railcar Loading

Figure 15.14 is typical of a truck or railcar loading application. This is specifically a two-stage system and is temperature controlled. However, the combustor may also be single stage. If the vapor combustor is used in an application where there is a wide variance in loading rate, a multistage combustor helps keep the flow rate across the antiflashback burners high. As shown in Figure 15.14, assist gas is only introduced on the first vapor line or stage. This is because during initial loading the vapors are very lean and will require assist

FIGURE 15.13
John Zink LRVP VRU.

gas to maintain temperature in the stack. As the flow rate increases, the heating value of the vapor typically increases and assist gas is not required on the second stage. Figure 15.16 illustrates the stratification of vapors in a tank truck.

Important questions regarding the design of a truck or railcar combustor are the maximum loading rate, the minimum loading rate, the nominal loading rate, and the products to be loaded. For example, if the maximum loading rate is four truck lanes with four arms per lane at 600 gpm (2300 Lpm) per arm, the maximum loading rate is 9600 gpm (36,000 Lpm). This equates to 1284 SCFM (36 SCMM). Figure 15.17 illustrates a truck rack vapor control system block diagram.

15.4.4.2.2 *Marine Loading*

In order to provide safe operation of the vapor combustor and prevent flashback from occurring, the vapor stream that comes from a noninerted vessel (e.g., barge) needs to be maintained at a condition above the upper

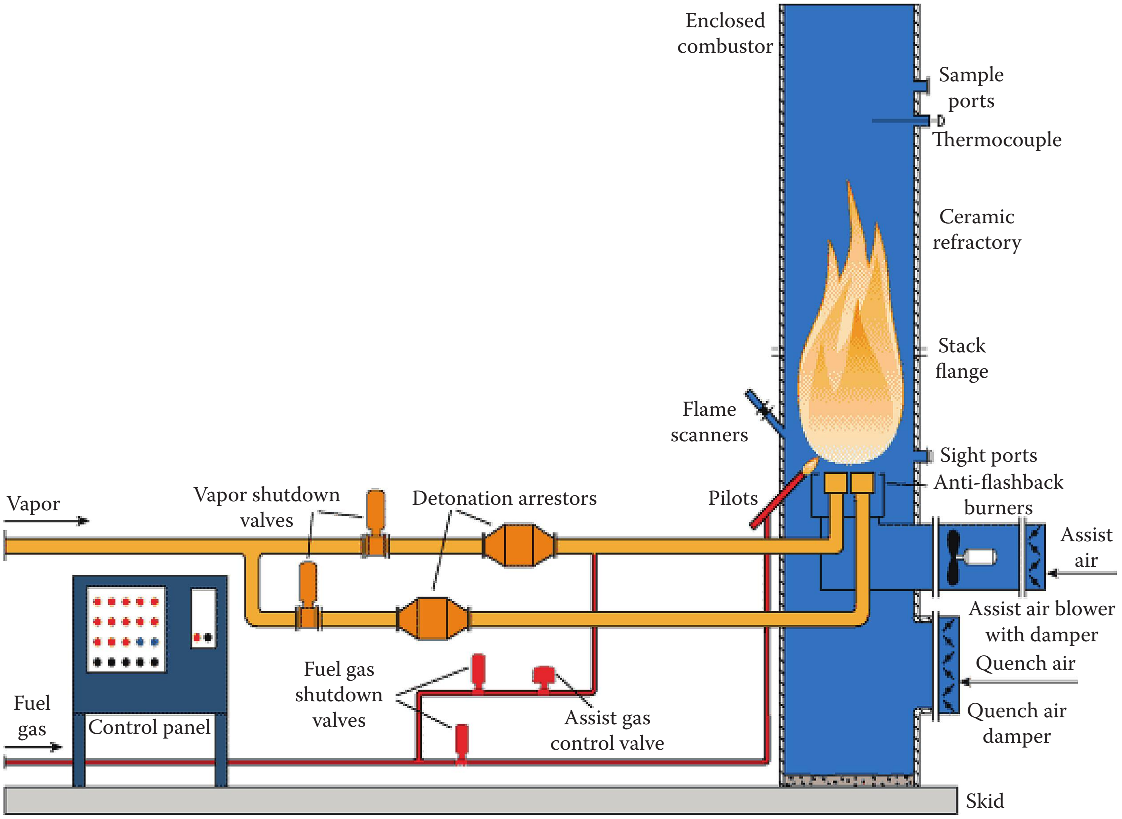


FIGURE 15.14
Diagram of a vapor combustor.

explosive limit (UEL) (see Volume 1, Chapters 3 and 4). Alternatively, for loads in which inerted vapors are displaced from the cargo hold of the vessel (e.g., ships), vapors may be confirmed to be outside of the flammable range by ensuring that the oxygen content of the vapor stream is less than the minimum oxygen content for combustion (MOCC). For the sake of this discussion, the focus of the topic will be limited to enrichment of a noninerted vapor stream to above the UEL. UEL is defined as the highest concentration (volume percentage) of a gas or vapor in air capable of producing a flash of fire in the presence of an ignition source (spark, flame, heat). Concentrations higher than UEL are too "fuel" rich to burn. UEL is also commonly called the upper flammable limit (UFL). Figure 15.18 illustrates the vapor stratification in a vessel, similar to that seen in tank trucks.

If the UEL of the vapors from the product being loaded is lower than methane, then the UEL of methane (CH_4) is 15% and controls the design. This will determine the oxygen setpoint for the control system. Typical setpoints are 150% of UEL = SHUTDOWN and 170% of UEL = ALARM; the normal operating point is substantially above 170% of the UEL to ensure that there will be no nuisance trips based upon oxygen control.

In order to determine the condition of the vapor stream, the oxygen level in the vapor mixture is measured. The oxygen level is then reduced by the addition of fuel gas (natural gas or propane) to the vapor stream to "enrich" the stream with fuel and lower the oxygen level. This gas is called enrichment gas or assist gas as shown in Figure 15.14. For marine terminal operations, the coast guard requires shutdown when the oxygen content increases above 150% of the UEL and an alarm at 170% of the UEL. Figure 15.19 illustrates a marine vapor control block diagram.

15.4.4.2.3 Storage Tanks

John Zink vapor combustion systems are designed to destroy the hydrocarbon vapors displaced when liquid hydrocarbons are filling a fixed vapor space such as is found in the space between the bottom of a tank and an elevated roof structure. The burner technology used in the John Zink combustors has evolved from flare burners that utilize the forced injection of air into the waste stream at the point of ignition. This burner technology allows the efficient combustion of waste streams of widely varying hydrocarbon compositions and concentrations.

The displaced vapors consist of hydrocarbon and air in the "noninert" atmosphere that is found below the floating deck of an internal floating roof tank. It is possible that from time to time, the noninerted vapors leaving the vapor space will be in the explosive range and precautions must be taken to prevent flashbacks from the combustor to the tank.

FIGURE 15.15
Photos of vapor combustors.

The John Zink design incorporates a proprietary antiflashback burner design as the first line of defense against flashback. In addition, the design utilizes a staging concept to achieve flow turndown conditions while maintaining the vapor flow above flame propagation velocities and a U.S. Coast Guard (USCG)-certified detonation arrester. Although the vapor blower(s) uses antisparking construction, there does remain a slight chance that a spark might be generated by the blower (or “trash” that finds its way into the blower housing) and a

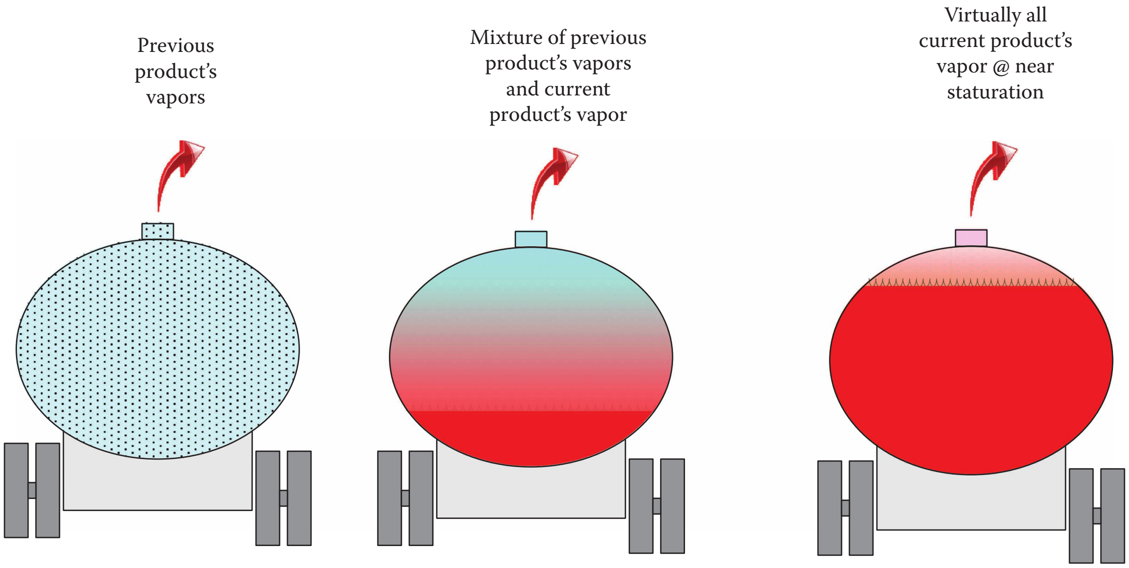


FIGURE 15.16
Stratification of vapors in a tank truck.

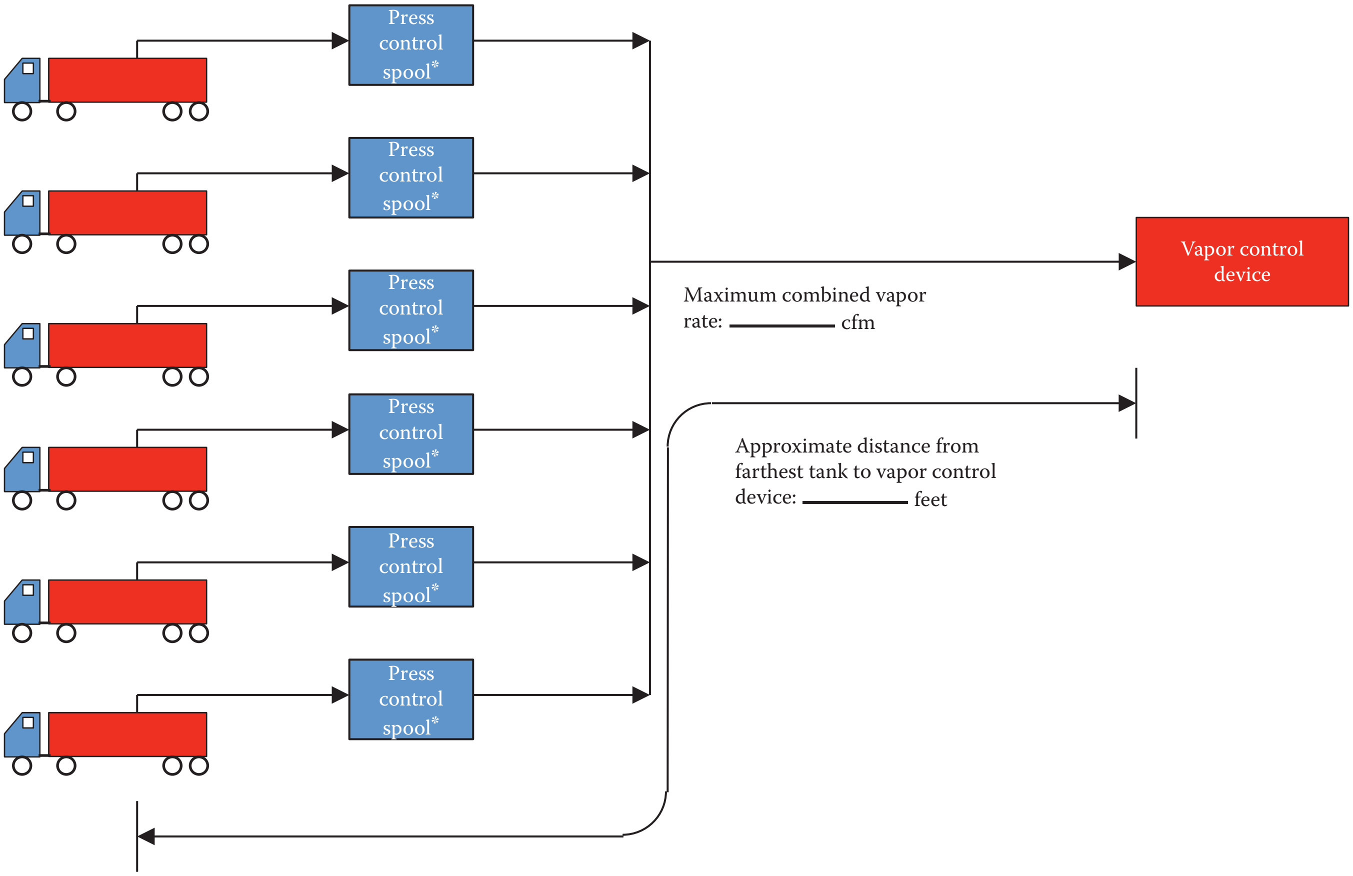


FIGURE 15.17
Truck rack vapor control system block diagram. * Press control spools are typically only incorporated in conjunction with a vacuum assist blower package.

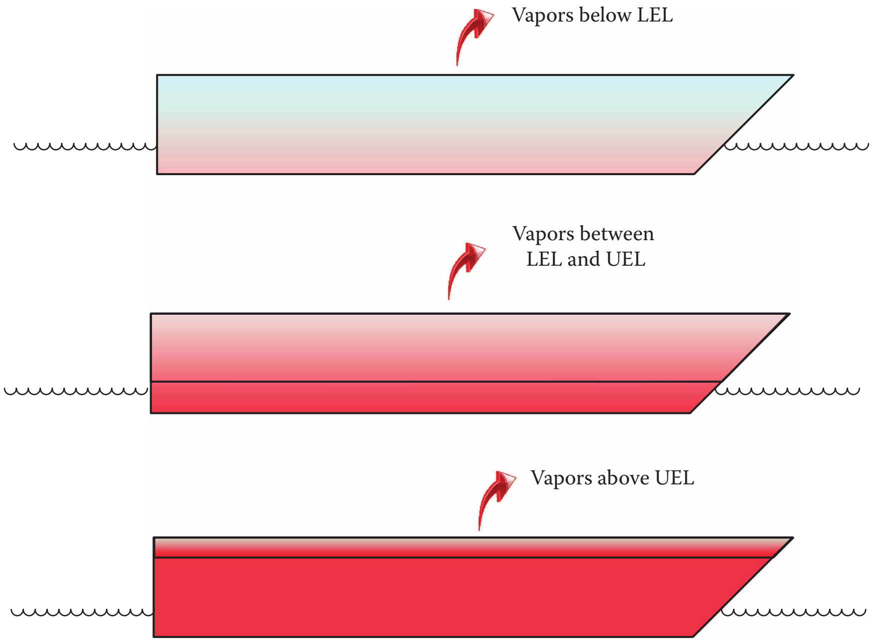


FIGURE 15.18
Vapor stratification in a vessel.

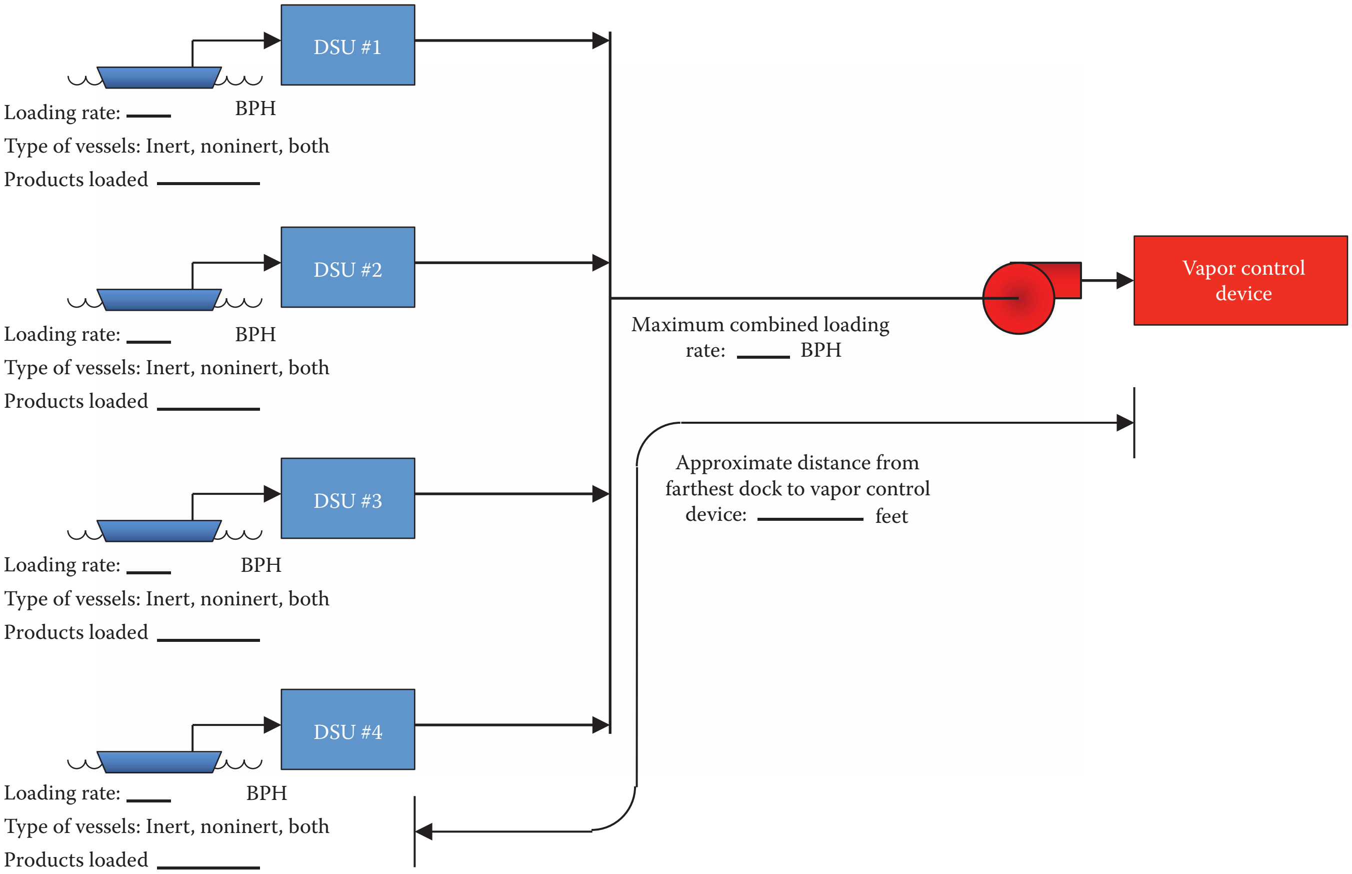


FIGURE 15.19
Marine vapor control block diagram.

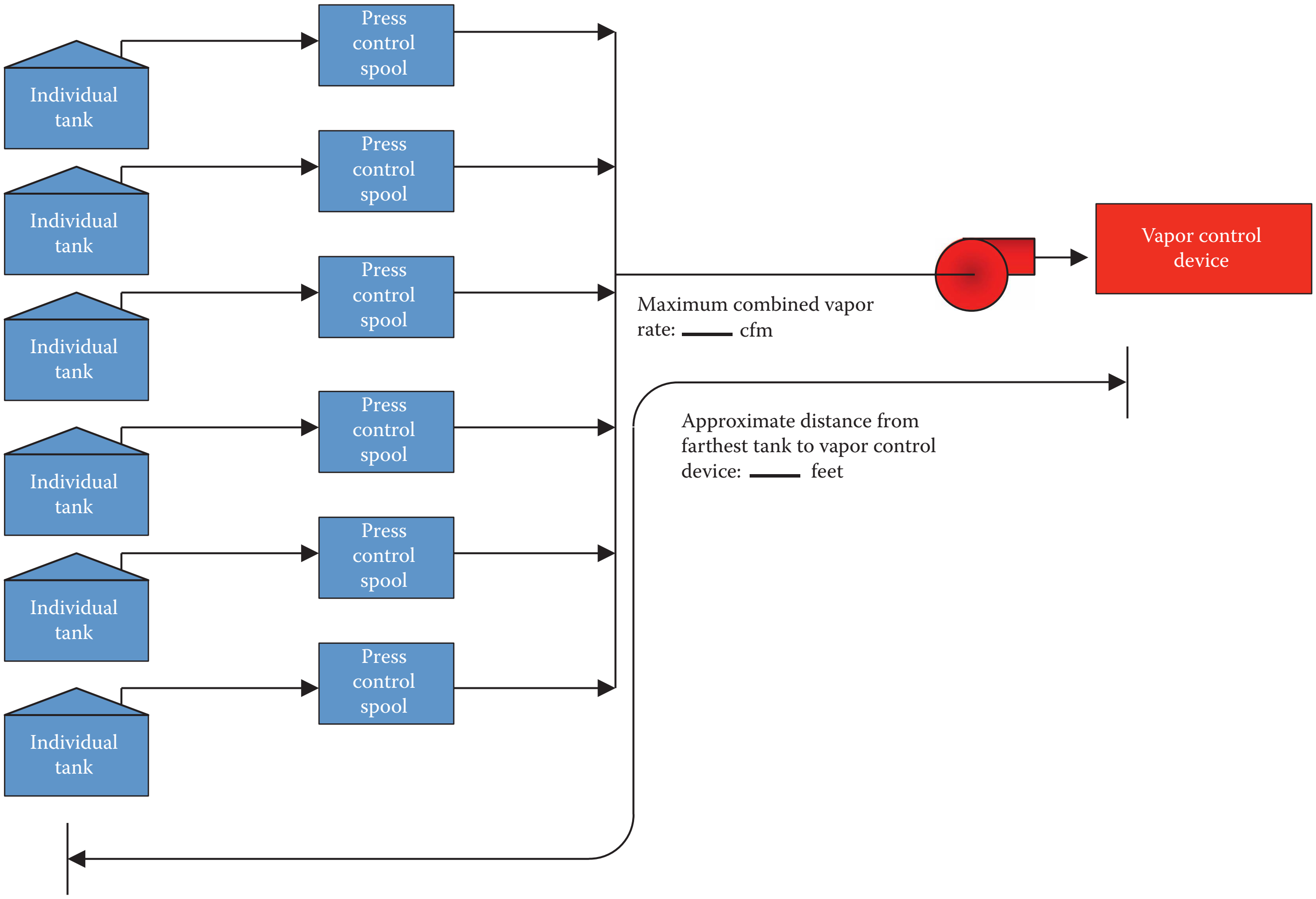


FIGURE 15.20
Internal floating roof tank vent control system block diagram.

detonation arrester is also included at the suction of each blower to protect the tanks from such an occurrence.

Lean gasoline vapors (vapors above the internal floating roof) will need to be treated at the combustor to ensure destruction. Rich gasoline vapors require air to be energetically mixed with the vapor at the point of combustion to prevent the formation of free carbon and smoking. An assist air blower is used to direct the required portion of stoichiometric air around every vapor burner. The balance of the stoichiometric air and quenching air (required to limit the combustion chamber temperature such that the refractory lining is not degraded) is directed by automatically modulating natural draft dampers located at the base of the combustion chamber.

Under most tank loading conditions, the vapors are expected to have a more than adequate concentration of hydrocarbons to maintain the necessary operating temperature. However, when venting vapors from an empty tank, it is expected that the vapors will be very lean. In this case, assist gas will be automatically introduced to the combustor as necessary to hold the minimum temperature required to achieve the necessary VOC destruction.

The tank vent manifolds will be maintained at a slight negative pressure and the vapor blower system furnished will have enough vacuum boost to overcome the tank piping to the vapor combustion unit (VCU) to assure the tanks are not over pressured. To protect the blowers (and combustor) from condensation that may form in the pipe works, a common knockout vessel is usually installed upstream of the blowers. Figure 15.20 illustrates an internal floating roof tank vent control system block diagram.

Vapor combustion systems are generally insensitive to the type of hydrocarbon vapors they receive and maximize terminal flexibility. Products other than butane and gasoline can be treated so long as the design heat release is not exceeded. Typically this is a mainly a function of total vapor pressure.

15.4.5 Flashback Protection Devices

Detonation arrestors are supplied for secondary flashback protection. The detonation arrestor is a passive in-line device that quenches a flame by absorbing its heat. Figure 15.21 shows the cross section of a detonation arrestor.

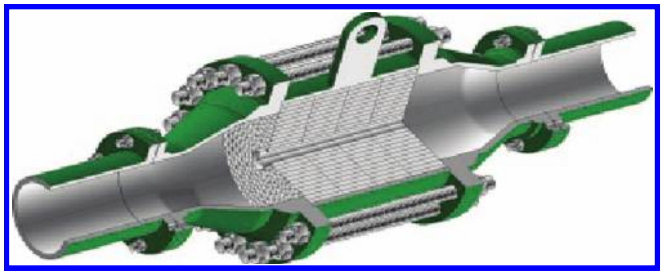


FIGURE 15.21
Cross section of a detonation arrestor.

TABLE 15.6
Groups of Various Hydrocarbons

Group A	Group B	Group C	Group D
Acetylene	Butadiene	Cyclopropane	Benzene
	Ethylene oxide	Diethyl ether	Butane
	Hydrogen	Ethylene	Ethanol
	Propylene oxide	Hydrogen sulfide	Gasoline
		Methyl mercaptan	Methanol[a]
			Pentanes

[a] One exception is methanol; experience indicates this should be treated as a Group C gas to prevent flashback.

Care should be taken to select a *detonation* and not a *deflagration* flame arrestor. Most hydrocarbons in vapor control applications are classified as a Group D gas. There are four groups per National Electric Code (NEC). Table 15.6 gives examples of each type. No detonation arrestor manufacturer makes a Group A arrestor.

Detonation is a process of supersonic combustion in which a shock wave is propagated forward due to energy release in a reaction zone behind it. It is the more powerful of the two general classes of combustion, the other one being deflagration. Figure 15.22 illustrates the pressure waves generated by a flame front in a pipe.

In a detonation, the shock compresses the material thus increasing the temperature to the point of autoignition. The ignited material burns behind the shock and releases energy that supports the shock propagation. Because detonations generate high pressures, they are usually much more destructive than deflagrations. In a pipe length of 200 ft, an overdriven detonation can achieve velocities of 5100 mph (2300 m/s) and reach pressures of 3100 psig (210 barg). Deflagration is a technical term describing subsonic combustion that usually propagates through thermal conductivity (hot burning material heats the next layer of cold material and ignites it).

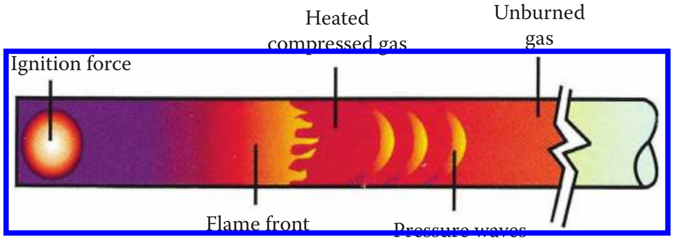


FIGURE 15.22
Pressure waves generated by a flame front in a pipe.

Definitions

Absorption (AB): To take in and make a part of the whole: to soak up.

Activated carbon: Is a form of carbon, which is extremely porous with little, if any, volatile material remaining in its structure. Activated carbon is different from activated charcoal. Activated charcoal has a relatively large amount of volatile components remaining in its structure.

Adsorption (AD): The adhesion of gaseous molecules, in extremely thin layers, to the surface of a solid. This adhesion is brought about by the imbalance in forces existing between the solid and the gaseous molecules. These attractive forces are known as van der Waals forces.

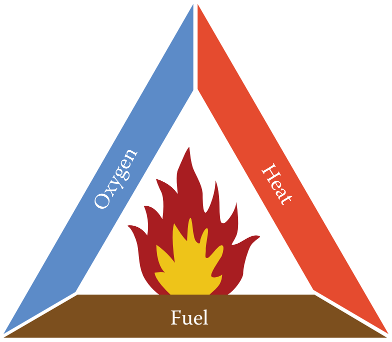
Autoignition: Sufficient temperature can provide the activation energy required for combustion reaction to occur.

Bleedthrough: Low concentrations of unadsorbed hydrocarbons detected at the outlet of the carbon bed during operation.

Breakthrough: The event when the outlet hydrocarbon concentration exceeds the upper concentration limit during operation.

British thermal unit (BTU): Energy required to heat 1 lb of water 1°F.

Combustion triangle: Fuel, air, and ignition/heat.



Detonation: Supersonic combustion in which a shockwave is propagated due to the energy released behind it.

Deflagration: Subsonic combustion that propagates through thermal conductivity.

Dry vacuum pump (DVP): A vacuum pump that does not require a liquid seal to seal the gap between the impeller/screw and the pump walls. These pumps have very small tolerances between the wall and the screw.

Emission limits: In North America, the emission limits are usually defined in terms of mass of hydrocarbon emitted per volume of liquid loaded at the rack, for instance, 80, 35, or 10 mg/L (milligram of hydrocarbon emitted per liter of product loaded). This is a measurement that is averaged over a period of time. The U.S. EPA specifies a 6 h test as a minimum, though some states require a shorter averaging period. In Europe (E.U.), the majority of the countries have adopted a mass emitted per volume emitted. Typically this is 35 g of hydrocarbon emitted per normal cubic meter vented (35 g/N-m^3). The older U.S. standard of 35 mg/L is roughly equivalent to the E.U. standard of 35 g/N-m^3. Germany has adopted the most stringent standard of all at 50 mg/N-m^3, which is approximately 0.05 mg/L.

Heat of adsorption: The latent heat given off by a material as it is adsorbed onto the surface of another material.

Liquid ring vacuum pump (LRVP): A vacuum pump that utilizes a liquid seal to seal the gap between the wall and the pump impeller.

Purge gas strip: Desorption of hydrocarbon from the carbon's surface by passing an inert gas through the carbon bed.

Regeneration: The process that allows the carbon to be reused numerous times.

Stack: Self-supported vertical carbon steel cylinder shell lined with ceramic blanket refractory.

Vacuum level: The term vacuum is used to denote a pressure below atmospheric pressure. In referring to a "vacuum," it must be noted that it is the opposite of pressure; a high vacuum means a low pressure. A high or deep vacuum is farther from atmospheric pressure than a low vacuum, for example, a low vacuum is near atmospheric pressure. Absolute vacuum, zero absolute pressure, as measured at sea level, is 0 in. of mercury absolute or 29.92 in. of mercury vacuum (29.92 in. HgV). A standard efficiency activated carbon system is regenerated to 3 in. of mercury absolute (3 in. HgA), which is equal to 27 in. of mercury vacuum (27 in. HgV) at sea level or 90% absolute vacuum. Vacuum is also referred to in mm instead of inches. (mmHgA is also referred to as torr.)

Vacuum pump capacity: The volume of vapor/air removed by the vacuum pump at a specified vacuum level. The vacuum pump capacity is usually measured in actual cubic feet per minute at a specific vacuum level (acfm).

Vacuum strip: Desorption of hydrocarbon from the carbon's surface by pulling a vacuum on the carbon bed.

Volatile organic compound: Any nonmethane ROC that participates in atmospheric photochemical reactions.

Nomenclature

AAD	Adsorption–absorption dry
AAW	Adsorption–absorption wet
AC	Alternating current
ACFM	Actual cubic feet per minute
AD	Apparent density
ADAB™	Carbon adsorption–absorption
ADCON™	Carbon adsorption–condensation
AIChE	American Institute of Chemical Engineers
API	American Petroleum Institute
ASME	American Society of Mechanical Engineers
BACT	Best available control technology
BTU	British Thermal Unit
CE	Combustion efficiency
CEM	Continuous emission monitor
CFM	Cubic feet per minute
CFR	Code of federal regulations
CIM	Continuous inlet monitor
CPI	Chemical processing industry
CS	Carbon steel
CSA	Canadian Standards Association
DA	Detonation arrestor
DC	Direct current
DDT	Deflagration to detonation transformation
DE	Destruction efficiency
DRE	Destruction removal efficiency
DSC	Differential scanning calorimeter
DVP	Dry vacuum pump
EG	Ethylene glycol
EOL	End of load
EPA	U.S. Environmental Protection Agency
ESD	Emergency shutdown
FBN	Fuel bound NOx
FPS	Feet per second
GD GACT	Gasoline distribution generally achievable control technology

HAP	Hazardous air pollutant
HC	Hydrocarbon
HCV	Hand control valve
HE-ADAB™	High efficiency adsorption–absorption (HEAA)
HMI	Human–machine interface
HOA	Hand, off, auto
HP	Horsepower
HPI	Hydrocarbon processing industry
ILTA	Independent liquid terminal association
I/O	Input/output
KO	Knockout
LEL/LFL	Lower explosive limit/lower flammable limit
LRV	Lower range value
LRVP	Liquid ring vacuum pump
LVDT	Linear variable displacement transducer
MACT	Maximum achievable control technology
MAPT	Master absolute pressure transmitter
MESG	Maximum experimental safe gap
MMBTU/h	Million BTU/h
MOCC	Minimum oxygen concentration for combustion
MOV	Motor-operated valves
MTZ	Mass transfer zone
NEC	National Electrical Code
NEMA	National Electrical Manufacturers Association
NESHAP	National Emission Standards for Hazardous Air Pollutants
OIP	Operator interface panel
OSHA	Occupational safety and health administration
P&ID	Process and instrumentation diagram
PECS	Portable emission control system
PID	Proportional, integral, differential
PLC	Programmable logic controller
PPM	Parts per million
PSA	Pressure swing adsorption
PSI	Pounds per square inch
psia	Pounds per square inch absolute
psig	Pounds per square inch gauge
PSH	Pressure switch high
PSL	Pressure switch low
PSM	Process safety management
ROCs	Reactive organic compounds
RPM	Revolutions per minute
RTD	Resistance temperature detector
RVP	Reid vapor pressure
SCF	Standard cubic foot
SCFM	Standard cubic feet per minute
SOL	Start of load
SOV	Shutoff valve
SS	Stainless steel
SV	Solenoid valve
TCV	Temperature control valve
TFE	Tetrafluoroethylene (teflon)
TSA	Temperature swing adsorption
TVP	Total vapor pressure
UEL/UFL	Upper explosive limit/upper flammable limit
UL	Underwriters laboratories
URV	Upper range value
VAVACS™	Vapor vacuum control system
VBB	Vacuum booster blower
VCU	Vapor combustion unit
VFD	Variable frequency drive
Vmax	Maximum velocity
VOCs	Volatile organic compounds
VP	Vacuum pump
VRU	Vapor recovery unit
ZTOF	Zink thermal oxidizing flare

Useful Conversions

Barrel per hour to gallons per minute

$$\text{bph} \times \frac{1\text{ h}}{60\text{ min}} \times \frac{42\text{ gal}}{1\text{ barrel}} = \text{gpm}$$

Gallons per minute to cubic feet per minute (cfm)

$$\text{gpm} \times \frac{1\text{ ft}^3}{7.48\text{ gal}} = \frac{\text{ft}^3}{\text{min}}$$

lb mol/h to cubic feet per minute (cfm)

$$\frac{\text{lbmol}}{\text{h}} \times \frac{379.5\text{ ft}^3}{1\text{ lbmol}} \times \frac{1\text{ h}}{60\text{ min}} = \frac{\text{ft}^3}{\text{min}}$$

Useful Industry References

AP 42	*AP 42*, Fifth edn., *Compilation of Air Pollutant Emission Factors, Volume 1: Stationary Point and Area Sources*	
	Section 5.2	Transportation and marketing of petroleum liquids
GPSA	(Gas processors suppliers association), engineering data book	
	Volume 1:	Section 6—Storage
	Volume 2:	Section 23—Physical properties
CFR title 33:	Navigation and navigable waters	
	Part 154	Facilities transferring oil or hazardous material in bulk
	Part 155	Oil or hazardous material pollution prevention regulations for vessels
	Part 156	Oil and hazardous material transfer operations
CFR title 46:	Subchapter on d-tank vessels	
	Part 30	General provisions
	Part 32	Special equipment, machinery, and hull requirements
	Part 35	Operations
	Part 39	Vapor control systems

References

1. Dinsmore, H. and de Silva, R., *High Efficiency Hydrocarbon Vapour Recovery Systems for Petroleum Products Distribution Terminals*, presented at the ASCOPE Refining Workshop, Bangkok, Thailand, November 10–11, 1995.
2. Allport, H. B., Activated carbon, in *McGraw-Hill Encyclopedia of Science & Technology*, 5th ed., Vol. 1, McGraw-Hill, New York, p. 94, 1982.
3. Greenbank, M., Ph.D., and Steve spots, Carbon filters—Effects of starting material on activated carbon characteristics and performance, *Industrial Water Treatment*, January/February 1995, 19–27.
4. Knaebel, K. S., For your next separation consider adsorption, *Chemical Engineering*, November 1995, 92–102.
5. Deithorn, R. T. and Anthony, F. M., Activated carbon—What it is, how it works, *Water Technology*, November 1986, 26–29.
6. Activated carbon product brochure, Calgon Carbon Corporation, Pittsburgh, PA.
7. Fowkes, F. M., Adsorption, in *McGraw-Hill Encyclopedia of Science & Technology*, 5th Ed., Vol. 1, McGraw-Hill, New York, pp. 124–127, 1982.
8. Boppart, S., Get the most from activated-carbon systems, *Environmental Engineering World*, May–June 1995, 12–14.
9. Keller, G. E. II, Adsorption: Building upon a solid foundation, *Chemical Engineering Progress*, October 1995, 56–67.
10. Humphrey, J. L., Separation processes: Playing a critical role, *Chemical Engineering Progress*, October 1995, 31–41.
11. Notes from personal lecture given by Dr. M. Greenbank, PhD, Surface Chemist, Calgon Carbon Corporation, 1995.
12. Cooper, H. and Dinsmore, H., *Service Bulletin 12—Activated Carbon Replacement*, *John Zink and McGill Vapor Recovery Systems*, John Zink Company, Tulsa, OK, August 4, 1994.
13. Linger, G. G. and Toan, P. Vo., White Paper Response to Zink Service Bulletin 12, Calgon Carbon Corporation, Internal Memo, November 8, 1994.
14. McElroy, T. It's a dry heat, *Hazardous Cargo Bulletin*, 32(5), 70–71, 2011.

Appendix A: Units and Conversions

TABLE A.1

Prefixes

Multiplier	Prefix	Symbol
10^{18}	Exa	E
10^{15}	Peta	P
10^{12}	Tera	T
10^{9}	Giga	G
10^{6}	Mega	M
10^{3}	Kilo	k
10^{2}	Hecta	h
10	Deca	da
10^{-1}	Deci	d
10^{-2}	Centi	c
10^{-3}	Milli	m
10^{-6}	Micro	m
10^{-9}	Nano	n
10^{-12}	Pico	p
10^{-15}	Femto	f
10^{-18}	Atto	a

Source: Annamalai, K. and Puri, I.K., *Combustion Science and Engineering*, CRC Press, Boca Raton, FL, 2007, p. 981, Table A.1A.

TABLE A.2

Basic Units, Conversions, and Molecular Properties

Area
1 acre = 4046.9 m^2
1 m^2 = 10^{-6} km^2 = 10^4 cm^2 = 10^6 mm^2
1 m^2 = 10.764 ft^2 = 1550 in.^{2}
1 ft^2 = 144 in.^{2} = 0.0929 m^2
1 hectare = 10,000 m^2 = 2.5 acres = 108,000 ft^2

Density
1 g/cm^3 = 1 kg/L = 1000 kg/m^3 = 62.43 lb$_m$/ft^3 = 0.03613 lb$_m$/in^3
1kg/m^3 = 0.06243 lb$_m$/ft^3, 1lb$_m$/ft^3 = 16.018 kg/m^3
Specific gravity = density/reference density
For liquids, reference density of water at 15.74°C (60°F) = 999 kg/m^3, 62.4 lb/ft^3
For gases, reference density of air at 15.74°C (60°F) = 1.206 kg/m^3

Energy
1 eV ≈ 1.602 × 10^{-19} J.
1 MBtu = 1 kBtu = 1000 Btu, 1 MMBtu = 1000 kBtu = 10^6 Btu
1 TBtu = 10^9 Btu or 1 GBtu
1 quad = 10^{15} Btu or 1.05 × 10^{15} kJ or 2.93 × 10^{11} kWh = 172.4 million barrels of crude oil
1 kWh = 0.0036 GJ = 3.6 MJ = 3412 Btu, 1 hp h = 0.00268 GJ = 2.68 MJ = Btu
1 Btu = 778.14 ft lb$_f$ = 1.0551 kJ, 1 kJ = 0.94782 Btu = 25,037 lb$_m$ ft/s^2
1 cal = 4.1868 J, 1 (food) cal = 1000 cal or 1 kcal

TABLE A.2 (continued)

Basic Units, Conversions, and Molecular Properties

1 kJ/kg = 0.43 Btu/lb, 1 Btu/lb = 2.326 kJ/kg, 1 kg/GJ = 1 g/MJ = 2.326 lb$_m$/mmBtu
1 Btu/SCF = 37 kJ/m^3, 1 m^3/GJ = 37.3 ft^3/mmBtu, 1 lb$_m$/mmBtu = 0.430 kg/GJ = 0.430 g/MJ
1 Therm = 10^5 Btu = 1.055 × 10^5 kJ
1 hp = 0.7064 Btu/s = 0.7457 kW = 745.7 W = 550 lb$_f$·ft/s = 42.41 Btu/min
1 boiler HP = 33,475 Btu/h, 1 Btu/h = 1.0551 kJ/h
1 barrel (42 gallons) of crude oil = 5,800,000 Btu = 6120 MJ
1 gallon of gasoline = 124,000 Btu = 131 MJ
1 gallon of heating oil = 139,000 Btu = 146.7 MJ
1 gallon of diesel fuel = 139,000 Btu = 146.7 MJ
1 barrel of residual fuel oil = 6,287,000 Btu = 6633 MJ
1 cubic foot of natural gas = 1,026 Btu = 1.082 MJ
1 gallon of propane = 91,000 Btu = 96 MJ
1 short ton of coal = 20,681,000 Btu = 21,821 MJ

Force
1 lb$_f$ = 4.4482 N = 32.174 lb$_m$ · ft/s^2 or g$_c$ = 32.174 lb$_m$ ft/s^2 lb$_f$

Ideal Gas Law

$$Pv = RT;\ PV = mRT;\ PV = n\bar{R}T,\ P\bar{v} = \bar{R}T,$$

$$\begin{aligned}\bar{R} &= 8.314\,\text{kPa m}^3/\text{k mol k} = 0.08314\,\text{bar m}^3/\text{k mol K}\\ &= 1.986\,\text{Btu/lb mol °R} = 1545\text{ft lb}_f/\text{lb mol °R}\\ &= 0.7299\,\text{atm ft}^3/\text{lb mol °R}\end{aligned}$$

Length/Velocity
1 in. = 0.0254 m
1 ft = 12 in. = 0.3048 m
1 mile = 5280 ft = 1609.3 m
1 statute mile = 1 smi = 0.87 nmi = 1.609 km
1 nautical mile = 1.15 smi = 1.85 km
1 mi/h = 1.46667 ft/s = 0.447 m/s = 1.609 km/h
1 m/s = 3.2808 ft/s = 2.237 mi/h = 1.96 kt = 1.15 smi/h = 3.63 km/h
Speed of light in vacuum, *c* = 2.998 × 10^8 m/s
Speed of sound = $\sqrt{\gamma RT}\,1$

Mass
1 teragram (Tg) = 1 million metric-ton
Mass of an electron = 0.5 MeV (1 Mev = 10^6 eV; for mass, use $E = me^2$) = 9.109 × 10^{31} kg
Mass of proton = 940 MeV = 1.67 × 10^{-27} kg, Mass of neutron = 1.675 × 10^{-27} kg
1 lb$_m$ = 0.45359 kg = 7000 grains
1 short ton = 2000 lb = 907.2 kg
1 long ton = 2240 lb or 1016.1 kg
1 metric-ton = 1000 kg
1 ounce = 28.3495 g
1 kg = 2.2046 lb

(*continued*)

TABLE A.2 (continued)

Basic Units, Conversions, and Molecular Properties

Molecular Properties

1 Angstrom = 1.0×10^{-10} m

$N_{Avog} = 6.023 \times 10^{26}$ molecules/kmol for a molecular substance (e.g., oxygen)

= 6.023×10^{26} atoms/atom mole for an atomic substance (e.g., He)

Boltzmann constant, $k_B = 1.38 \times 10^{-26}$ kJ/molecule K

Planck's constant, $h_P = 6.626 \times 10^{-37}$ kJ s/molecule

Stefan–Boltzmann constant, $\sigma = 5.66961 \times 10^{-11}$ kW/m^2 K^4

Charge of an electron = 1.602×10^{-19} coulombs, orbit radius (nm) = $0.0529n^2$, n: orbit number

Energy level of an orbit (eV) = $13.56/n^2$

Numbers

$\ln x = 2.303 \log_{10} x$

$\log_{10} x = 0.4343 \ln x$

e = 2.718

$\pi = 3.142$

1 deg = 0.0175 radians

Pressure

1 bar = 10^5 Pa, 1 mm Hg = 133.3 Pa

1 in Hg = 3.387 kPa = 0.491 psi

1 in water (4°C) = 0.03613 psi

1 atm = 14.696°lb$_f$/in.^{2} = 1.0133 bar = 10.3323 mm of H_2O (4°C) = 760 mm of Hg(0°C)

1 psi = 1 lb$_f$/in.^{2} = 144 lb$_f$/ft^2 = 6.894 kPa = 6894 Pa = 27.653 in water (4°C)

Specific Heat

1 Btu/lb °F = 4.1868 kJ/kg °C

1 kJ/kg °C = 0.23885 Btu/lb °F

Temperature

T(°C) = (T(°F) − 32) * (5/9)

T(°F) = T(°C) * 1.8 + 32

T(K) = T(°C) + 273.15

T(°R) = T(°F) + 459.67

1°R = 0.556 K, 1 K = 1.8°R

To convert electron volts into the corresponding temperature in Kelvin, multiply by 11,604.

Volume

1 m^3= 1000 L

1 fluid ounce = 29.5735 cm^3 = 0.0295735 L

1 m^3/kg = 1000 L/kg = 16.02 ft^3/lb, 1 m^3/GJ = 37.26 ft^3/mmBtu

1 ft^3/lb$_m$ = 0.062428 m^3/kg

1 U.S. gallon = 128 fluid ounce = 3.786 L

1 barrel = 42 U.S. gallons = 35 imperial gallons = 158.98 L = 5.615 ft^3 = 231 in.^{3} = 0.1337 ft^3

TABLE A.2 (continued)

Basic Units, Conversions, and Molecular Properties

Volume of 1 kmol (SI) and 1 lb mol (English) of an ideal gas at STP conditions as defined below:

Scientific or SATP	U.S. Standard (1976) or ISA	Chemists' Standard or CSA	NTP (Gas Industry)
25°C (77°F), 101.3 kPa (14.7 psi, 29.92 in. of Hg)	15°C (60°F), 101.33 kPa (1 atm, 14.696 psi, 29.92 in. of Hg)	0°C (32°F), 101.33 kPa (1 atm, 14.7 psi, 29.92 in. of Hg)	20°C (65°F), 101.33 kPa (1 atm)
24.5 m^3/kmol (392 ft^3/lb mol)	23.7 m^3/kmol (375.6 ft^3/lb mol)	22.4 m^3/kmol (359.2ft^3/lb mol)	23.89 m^3/kmol (382.7 ft^3/lb mol)

SATP, standard ambient temperature and pressure; ISA, International Standard Atmosphere; NTP, normal temperature and pressure.

Air Composition

Species	Mole %	Mass %	Molecular Weight
Ar	0.934	1.288287	39.948
CO_2	0.0314	0.047715	44.01
N_2	78.084	75.51721	28.01
O_2	20.9476	23.14489	32
Ne	0.001818	0.001267	20.18
He	0.000524	7.24E−05	4.0026
Krypton	0.000114	0.00033	83.8
Xe	8.70E−06	3.94E−05	131.3
H_2	0.00005	3.48E−06	2.016
CH_4	0.0002	0.000111	16.043
N_2O	0.00005	7.6E−05	44.013
SO_2, NO_2, CO, I_2	0.000235	—	—

Source: Annamalai, K. and Puri, I.K., *Combustion Science and Engineering*, CRC Press, Boca Raton, FL, 2007, p. 981, Table A.1A.

Note: Molecular weight (mass) of air = 28.96 kg/kmol.

TABLE A.3

Atomic Weights for Common Elements

Name	Symbol	Atomic Number	Atomic Weight
Aluminum	Al	13	26.98
Antimony	Sb	51	121.76
Argon	Ar	18	39.95
Arsenic	As	33	74.92
Barium	Ba	56	137.32
Beryllium	Be	4	9.01
Bismuth	Bi	83	208.98
Boron	B	5	10.811
Bromine	Br	35	79.90
Cadmium	Cd	48	112.41
Calcium	Ca	20	40.08
Carbon	C	6	12.01
Cesium	Cs	55	132.91
Chlorine	Cl	17	35.45
Chromium	Cr	24	52.00
Cobalt	Co	27	58.93
Copper	Cu	29	63.55
Fluorine	F	9	19.00
Germanium	Ge	32	72.61
Gold	Au	79	196.97
Helium	He	2	4.00
Hydrogen	H	1	1.01
Indium	In	49	114.82
Iodine	I	53	126.90
Iridium	If	77	192.22
Iron	Fe	26	55.85
Krypton	Kr	36	83.80
Lead	Pb	82	207.20
Lithium	Li	3	6.94
Magnesium	Mg	12	24.31
Manganese	Mn	25	54.94
Mercury	Hg	80	200.59
Molybdenum	Mo	42	95.94
Neon	Ne	10	20.18
Nickel	Ni	28	58.69
Nitrogen	N	7	14.01
Oxygen	O	8	16.00
Palladium	Pd	46	106.42
Phosphorus	P	15	30.97
Platinum	Pt	78	195.08
Plutonium	Pu	94	244.00
Potassium	K	19	39.10
Radium	Ra	88	226.00
Radon	Rn	86	222.00
Rhodium	Rh	45	102.91
Selenium	Se	34	78.96
Silicon	Si	14	28.09
Silver	Ag	47	107.87
Sodium	Na	11	22.99
Strontium	Sr	38	87.62
Sulfur	S	16	32.07
Tantalum	Ta	73	180.95
Thallium	Tl	81	204.38
Tin	Sn	50	118.71
Titanium	Ti	22	47.87
Tungsten	W	74	183.84
Uranium	U	92	238.03
Vanadium	V	23	50.94
Xenon	Xe	54	131.29
Zinc	Zn	30	65.39
Zirconium	Zr	40	92.22

Source: Annamalai, K. and Puri, I.K., *Combustion Science and Engineering*, CRC Press, Boca Raton, FL 2007, p. 985, Table A.1B.

Reference

1. Annamalai, K. and Puri, I.K., *Combustion Science and Engineering*, CRC Press, Boca Raton, FL

Appendix B: Physical Properties of Materials

TABLE B.1

Physical Properties of Pipe

Nominal Pipe Size, OD (in.)	Schedule Number			Wall Thickness (in.)	ID (in.)	Inside Area (in.^{2})	Metal Area (in.^{2})	Sq. Ft. Outside Surface (per ft)	Sq. Ft. Inside Surface (per ft)	Weight per ft (lb)	Weight of Water per ft (lb)	Moment of Inertia (in.^{4})	Section Modulus (in.^{3})	Radius Gyration (in.)
	a	b	c											
	—	—	10S	0.049	0.307	0.0740	0.0548	0.106	0.0804	0.186	0.0321	0.00088	0.00437	0.1271
1/8	40	Std	40S	0.068	0.269	0.0568	0.0720	0.106	0.0705	0.245	0.0246	0.00106	0.00525	0.1215
0.405	80	XS	80S	0.095	0.215	0.0364	0.0925	0.106	0.0563	0.315	0.0157	0.00122	0.00600	0.1146
	—	—	10S	0.065	0.410	0.1320	0.0970	0.141	0.1073	0.330	0.0572	0.00279	0.01032	0.1694
1/4	40	Std	40S	0.088	0.364	0.1041	0.1250	0.141	0.0955	0.425	0.0451	0.00331	0.01230	0.1628
0.540	80	XS	80S	0.119	0.302	0.0716	0.1574	0.141	0.0794	0.535	0.0310	0.00378	0.01395	0.1547
	—	—	10S	0.065	0.545	0.2333	0.1246	0.177	0.1427	0.423	0.1011	0.00586	0.01737	0.2169
3/8	40	Std	40S	0.091	0.493	0.1910	0.1670	0.177	0.1295	0.568	0.0827	0.00730	0.02160	0.2090
0.675	80	XS	80S	0.126	0.423	0.1405	0.2173	0.177	0.1106	0.739	0.0609	0.00862	0.02554	0.1991
	—	—	10S	0.083	0.674	0.3570	0.1974	0.220	0.1765	0.671	0.1547	0.01431	0.0341	0.2692
	40	Std	40S	0.109	0.622	0.3040	0.2503	0.220	0.1628	0.851	0.1316	0.01710	0.0407	0.2613
1/2	80	XS	80S	0.147	0.546	0.2340	0.3200	0.220	0.1433	1.088	0.1013	0.02010	0.0478	0.2505
0.840	160	—	—	0.187	0.466	0.1706	0.3830	0.220	0.1220	1.304	0.0740	0.02213	0.0527	0.2402
	—	XXS	—	0.294	0.252	0.0499	0.5040	0.220	0.0660	1.714	0.0216	0.02425	0.0577	0.2192
	—	—	5S	0.065	0.920	0.6650	0.2011	0.275	0.2409	0.684	0.2882	0.02451	0.0467	0.349
	—	—	10S	0.083	0.884	0.6140	0.2521	0.275	0.2314	0.857	0.2661	0.02970	0.0566	0.343
3/4	40	Std	40S	0.113	0.824	0.5330	0.3330	0.275	0.2157	1.131	0.2301	0.0370	0.0706	0.334
1.050	80	XS	80S	0.154	0.742	0.4320	0.4350	0.275	0.1943	1.474	0.1875	0.0448	0.0853	0.321
	160	—	—	0.218	0.614	0.2961	0.5700	0.275	0.1607	1.937	0.1284	0.0527	0.1004	0.304
	—	XXS	—	0.308	0.434	0.1479	0.7180	0.275	0.1137	2.441	0.0641	0.0579	0.1104	0.284
	—	—	5S	0.065	1.185	1.1030	0.2553	0.344	0.3100	0.868	0.478	0.0500	0.0760	0.443
	—	—	10S	0.109	1.097	0.9450	0.4130	0.344	0.2872	1.404	0.409	0.0757	0.1151	0.428
1	40	Std	40S	0.133	1.049	0.8640	0.4940	0.344	0.2746	1.679	0.374	0.0874	0.1329	0.421
1.315	80	XS	80S	0.179	0.957	0.7190	0.6390	0.344	0.2520	2.172	0.311	0.1056	0.1606	0.407
	160	—	—	0.250	0.815	0.5220	0.8360	0.344	0.2134	2.844	0.2261	0.1252	0.1903	0.387
	—	XXS	—	0.358	0.599	0.2818	1.0760	0.344	0.1570	3.659	0.1221	0.1405	0.2137	0.361
	—	—	5S	0.065	1.530	1.839	0.326	0.434	0.401	1.107	0.797	0.1038	0.1250	0.564
	—	—	10S	0.109	1.442	1.633	0.531	0.434	0.378	1.805	0.707	0.1605	0.1934	0.550
1 1/4	40	Std	40S	0.140	1.380	1.496	0.669	0.434	0.361	2.273	0.648	0.1948	0.2346	0.540
1.660	80	XS	80S	0.191	1.278	1.283	0.881	0.434	0.335	2.997	0.555	0.2418	0.2913	0.524
	160	—	—	0.250	1.160	1.057	1.107	0.434	0.304	3.765	0.458	0.2839	0.342	0.506
	—	XXS	—	0.382	0.896	0.631	1.534	0.434	0.2346	5.214	0.2732	0.341	0.411	0.472
	—	—	5S	0.065	1.770	2.461	0.375	0.497	0.463	1.274	1.067	0.1580	0.1663	0.649
	—	—	10S	0.109	1.682	2.222	0.613	0.497	0.440	2.085	0.962	0.2469	0.2599	0.634
1 1/2	40	Std	40S	0.145	1.610	2.036	0.799	0.497	0.421	2.718	0.882	0.310	0.326	0.623
1.900	80	XS	80S	0.200	1.500	1.767	1.068	0.497	0.393	3.631	0.765	0.391	0.412	0.605
	160	—	—	0.281	1.338	1.406	1.429	0.497	0.350	4.859	0.608	0.483	0.508	0.581
	—	XXS	—	0.400	1.100	0.950	1.885	0.497	0.288	6.408	0.412	0.568	0.598	0.549
	—	—	5S	0.065	2.245	3.960	0.472	0.622	0.588	1.604	1.716	0.315	0.2652	0.817
	—	—	10S	0.109	2.157	3.650	0.776	0.622	0.565	2.638	1.582	0.499	0.420	0.802
2	40	Std	40S	0.154	2.067	3.360	1.075	0.622	0.541	3.653	1.455	0.666	0.561	0.787
2.375	80	XS	80S	0.218	1.939	2.953	1.477	0.622	0.508	5.022	1.280	0.868	0.731	0.766
	160	—	—	0.343	1.689	2.240	2.190	0.622	0.442	7.444	0.971	1.163	0.979	0.729
	—	XXS	—	0.436	1.503	1.774	2.656	0.622	0.393	9.029	0.769	1.312	1.104	0.703
	—	—	5S	0.083	2.709	5.76	0.728	0.753	0.709	2.475	2.499	0.710	0.494	0.988
	—	—	10S	0.120	2.635	5.45	1.039	0.753	0.690	3.531	2.361	0.988	0.687	0.975

(continued)

TABLE B.1 (continued)

Physical Properties of Pipe

Nominal Pipe Size, OD (in.)	Schedule Number			Wall Thickness (in.)	ID (in.)	Inside Area (in.^{2})	Metal Area (in.^{2})	Sq. Ft. Outside Surface (per ft)	Sq. Ft. Inside Surface (per ft)	Weight per ft (lb)	Weight of Water per ft (lb)	Moment of Inertia (in.^{4})	Section Modulus (in.^{3})	Radius Gyration (in.)
	a	b	c											
2 1/2	40	Std	40S	0.203	2.469	4.79	1.704	0.753	0.646	5.793	2.076	1.530	1.064	0.947
2.875	80	XS	80S	0.276	2.323	4.24	2.254	0.753	0.608	7.661	1.837	0.193	1.339	0.924
	160	—	—	0.375	2.125	3.55	2.945	0.753	0.556	10.01	1.535	2.353	1.637	0.894
	—	XXS	—	0.552	1.771	2.46	4.030	0.753	0.464	13.70	1.067	2.872	1.998	0.844
	—	—	5S	0.083	3.334	8.73	0.891	0.916	0.873	3.03	3.78	1.301	0.744	1.208
	—	—	10S	0.120	3.260	8.35	1.274	0.916	0.853	4.33	3.61	1.822	1.041	1.196
3	40	Std	40S	0.216	3.068	7.39	2.228	0.916	0.803	7.58	3.20	3.02	1.724	1.164
3.500	80	XS	80S	0.300	2.900	6.61	3.020	0.916	0.759	10.25	2.864	3.90	2.226	1.136
	160	—	—	0.437	2.626	5.42	4.210	0.916	0.687	14.32	2.348	5.03	2.876	1.094
	—	XXS	—	0.600	2.300	4.15	5.470	0.916	0.602	18.58	1.801	5.99	3.43	1.047
	—	—	5S	0.083	3.834	11.55	1.021	1.047	1.004	3.47	5.01	1.960	0.980	1.385
3 1/2	—	—	10S	0.120	3.760	11.10	1.463	1.047	0.984	4.97	4.81	2.756	1.378	1.372
4.000	40	Std	40S	0.226	3.548	9.89	2.68	1.047	0.929	9.11	4.28	4.79	2.394	1.337
	80	XS	80S	0.318	3.364	8.89	3.68	1.047	0.881	12.51	3.85	6.28	3.14	1.307
	—	—	5S	0.083	4.334	14.75	1.152	1.178	1.135	3.92	6.40	2.811	1.249	1.562
	—	—	10S	0.120	4.260	14.25	1.651	1.178	1.115	5.61	6.17	3.96	1.762	1.549
4	40	Std	40S	0.237	4.026	12.73	3.17	1.178	1.054	10.79	5.51	7.23	3.21	1.510
4.500	80	XS	80S	0.337	3.826	11.50	4.41	1.178	1.002	14.98	4.98	9.61	4.27	1.477
	120	—	—	0.437	3.626	10.33	5.58	1.178	0.949	18.96	4.48	11.65	5.18	1.445
	160	—	—	0.531	3.438	9.28	6.62	1.178	0.900	22.51	4.02	13.27	5.90	1.416
	—	XXS	—	0.674	3.152	7.80	8.10	1.178	0.825	27.54	3.38	15.29	6.79	1.374
	—	—	5S	0.109	5.345	22.44	1.868	1.456	1.399	6.35	9.73	6.95	2.498	1.929
	—	—	10S	0.134	5.295	22.02	2.285	1.456	1.386	7.77	9.53	8.43	3.03	1.920
5	40	Std	40S	0.258	5.047	20.01	4.30	1.456	1.321	14.62	8.66	15.17	5.45	1.878
5.563	80	XS	80S	0.375	4.813	18.19	6.11	1.456	1.260	20.78	7.89	20.68	7.43	1.839
	120	—	—	0.500	4.563	16.35	7.95	1.456	1.195	27.04	7.09	25.74	9.25	1.799
	160	—	—	0.625	4.313	14.61	9.70	1.456	1.129	32.96	6.33	30	10.8	1.760
	—	XXS	—	0.750	4.063	12.97	11.34	1.456	1.064	38.55	5.62	33.6	12.1	1.722
	—	—	5S	0.109	6.407	32.20	2.231	1.734	1.677	5.37	13.98	11.85	3.58	2.304
	—	—	10S	0.134	6.357	31.70	2.733	1.734	1.664	9.29	13.74	14.4	4.35	2.295
6	40	Std	40S	0.280	6.065	28.89	5.58	1.734	1.588	18.97	12.51	28.14	8.5	2.245
6.625	80	XS	80S	0.432	5.761	26.07	8.40	1.734	1.508	28.57	11.29	40.5	12.23	2.195
	120	—	—	0.562	5.501	23.77	10.70	1.734	1.440	36.39	10.30	49.6	14.98	2.153
	160	—	—	0.718	5.189	21.15	13.33	1.734	1.358	45.30	9.16	59	17.81	2.104
	—	XXS	—	0.864	4.897	18.83	15.64	1.734	1.282	53.16	8.17	66.3	20.03	2.060
	—	—	5S	0.109	8.407	55.5	2.916	2.258	2.201	9.91	24.07	26.45	6.13	3.01
	—	—	10S	0.148	8.329	54.5	3.94	2.258	2.180	13.40	23.59	35.4	8.21	3.00
	20	—	—	0.250	8.125	51.8	6.58	2.258	2.127	22.36	22.48	57.7	13.39	2.962
	30	—	—	0.277	8.071	51.2	7.26	2.258	2.113	24.70	22.18	63.4	14.69	2.953
	40	Std	40S	0.322	7.981	50.0	8.40	2.258	2.089	28.55	21.69	72.5	16.81	2.938
8	60	—	—	0.406	7.813	47.9	10.48	2.258	2.045	35.64	20.79	88.8	20.58	2.909
8.625	80	XS	80S	0.500	7.625	45.7	12.76	2.258	1.996	43.39	19.80	105.7	24.52	2.878
	100	—	—	0.593	7.439	43.5	14.96	2.258	1.948	50.87	18.84	121.4	28.14	2.847
	120	—	—	0.718	7.189	40.6	17.84	2.258	1.882	60.63	17.60	140.6	32.6	2.807
	140	—	—	0.812	7.001	38.5	19.93	2.258	1.833	67.76	16.69	153.8	35.7	2.777
	—	XXS	—	0.875	6.875	37.1	21.30	2.258	1.800	72.42	16.09	162	37.6	2.757
	160	—	—	0.906	6.813	36.5	21.97	2.258	1.784	74.69	15.80	165.9	38.5	2.748
	—	—	5S	0.134	10.482	86.3	4.52	2.815	2.744	15.15	37.4	63.7	11.85	3.75
	—	—	10S	0.165	10.420	85.3	5.49	2.815	2.728	18.70	36.9	76.9	14.3	3.74
	20	—	—	0.250	10.250	82.5	8.26	2.815	2.683	28.04	35.8	113.7	21.16	3.71
	—	—	—	0.279	10.192	81.6	9.18	2.815	2.668	31.20	35.3	125.9	23.42	3.70
	30	—	—	0.307	10.136	80.7	10.07	2.815	2.654	34.24	35.0	137.5	25.57	3.69

TABLE B.1 (continued)

Physical Properties of Pipe

Nominal Pipe Size, OD (in.)	Schedule Number			Wall Thickness (in.)	ID (in.)	Inside Area (in.^{2})	Metal Area (in.^{2})	Sq. Ft. Outside Surface (per ft)	Sq. Ft. Inside Surface (per ft)	Weight per ft (lb)	Weight of Water per ft (lb)	Moment of Inertia (in.^{4})	Section Modulus (in.^{3})	Radius Gyration (in.)
	a	b	c											
10	40	Std	40S	0.365	10.020	78.9	11.91	2.815	2.623	40.48	34.1	160.8	29.9	3.67
10.750	60	XS	80S	0.500	9.750	74.7	16.10	2.815	2.553	54.74	32.3	212	39.4	3.63
	80	—	—	0.593	9.564	71.8	18.92	2.815	2.504	64.33	31.1	244.9	45.6	3.60
	100	—	—	0.718	9.314	68.1	22.63	2.815	2.438	76.93	29.5	286.2	53.2	3.56
	120	—	—	0.843	9.064	64.5	26.24	2.815	2.373	89.20	28.0	324	60.3	3.52
	140	—	—	1.000	8.750	60.1	30.6	2.815	2.291	104.13	26.1	368	68.4	3.47
	160	—	—	1.125	8.500	56.7	34.0	2.815	2.225	115.65	24.6	399	74.3	3.43
	—	—	5S	0.165	12.420	121.2	6.52	3.34	3.25	19.56	52.5	129.2	20.27	4.45
	—	—	10S	0.180	12.390	120.6	7.11	3.34	3.24	24.20	52.2	140.5	22.03	4.44
	20	—	—	0.250	12.250	117.9	9.84	3.34	3.21	33.38	51.1	191.9	30.1	4.42
	30	—	—	0.330	12.090	114.8	12.88	3.34	3.17	43.77	49.7	248.5	39.0	4.39
	—	Std	40S	0.375	12.000	113.1	14.58	3.34	3.14	49.56	49.0	279.3	43.8	4.38
12	40	—	—	0.406	11.938	111.9	15.74	3.34	3.13	53.53	48.5	300	47.1	4.37
12.750	—	XS	80S	0.500	11.750	108.4	19.24	3.34	3.08	65.42	47.0	362	56.7	4.33
	60	—	—	0.562	11.626	106.2	21.52	3.34	3.04	73.16	46.0	401	62.8	4.31
	80	—	—	0.687	11.376	101.6	26.04	3.34	2.978	88.51	44.0	475	74.5	4.27
	100	—	—	0.843	11.064	96.1	31.5	3.34	2.897	107.20	41.6	562	88.1	4.22
	120	—	—	1.000	10.750	90.8	36.9	3.34	2.814	125.49	39.3	642	100.7	4.17
	140	—	—	1.125	10.500	86.6	41.1	3.34	2.749	139.68	37.5	701	109.9	4.13
	160	—	—	1.312	10.126	80.5	47.1	3.34	2.651	160.27	34.9	781	122.6	4.07
	10	—	—	0.250	13.500	143.1	10.80	3.67	3.53	36.71	62.1	255.4	36.5	4.86
	20	—	—	0.312	13.376	140.5	13.42	3.67	3.5	45.68	60.9	314	44.9	4.84
	30	Std	—	0.375	13.250	137.9	16.05	3.67	3.47	54.57	59.7	373	53.3	4.82
	40	—	—	0.437	13.126	135.3	18.62	3.67	3.44	63.37	58.7	429	61.2	4.80
	—	XS	—	0.500	13.000	132.7	21.21	3.67	3.4	72.09	57.5	484	69.1	4.78
	—	—	—	0.562	12.876	130.2	23.73	3.67	3.37	80.66	56.5	537	76.7	4.76
14	60	—	—	0.593	12.814	129.0	24.98	3.67	3.35	84.91	55.9	562	80.3	4.74
14.000	—	—	—	0.625	12.750	127.7	26.26	3.67	3.34	89.28	55.3	589	84.1	4.73
	—	—	—	0.687	12.626	125.2	28.73	3.67	3.31	97.68	54.3	638	91.2	4.71
	80	—	—	0.750	12.500	122.7	31.2	3.67	3.27	106.13	53.2	687	98.2	4.69
	—	—	—	0.875	12.250	117.9	36.1	3.67	3.21	122.66	51.1	781	111.5	4.65
	100	—	—	0.937	12.126	115.5	38.5	3.67	3.17	130.73	50.0	825	117.8	4.63
	120	—	—	1.093	11.814	109.6	44.3	3.67	3.09	150.67	47.5	930	132.8	4.58
	140	—	—	1.250	11.500	103.9	50.1	3.67	3.01	170.22	45.0	1127	146.8	4.53
	160	—	—	1.406	11.188	98.3	55.6	3.67	2.929	189.12	42.6	1017	159.6	4.48
	10	—	—	0.250	15.500	188.7	12.37	4.19	4.06	42.05	81.8	384	48	5.57
	20	—	—	0.312	15.376	185.7	15.38	4.19	4.03	52.36	80.5	473	59.2	5.55
	30	Std	—	0.375	15.250	182.6	18.41	4.19	3.99	62.58	79.1	562	70.3	5.53
	—	—	—	0.437	15.126	179.7	21.37	4.19	3.96	72.64	77.9	648	80.9	5.50
	40	XS	—	0.500	15.000	176.7	24.35	4.19	3.93	82.77	76.5	732	91.5	5.48
	—	—	—	0.562	14.876	173.8	27.26	4.19	3.89	92.66	75.4	813	106.6	5.46
	—	—	—	0.625	14.750	170.9	30.2	4.19	3.86	102.63	74.1	894	112.2	5.44
16	60	—	—	0.656	14.688	169.4	31.6	4.19	3.85	107.50	73.4	933	116.6	5.43
16.000	—	—	—	0.687	14.626	168.0	33.0	4.19	3.83	112.36	72.7	971	121.4	5.42
	—		—	0.750	14.500	165.1	35.9	4.19	3.8	122.15	71.5	1047	130.9	5.40
	80	—	—	0.842	14.314	160.9	40.1	4.19	3.75	136.46	69.7	1157	144.6	5.37
	—	—	—	0.875	14.250	159.5	41.6	4.19	3.73	141.35	69.1	1193	154.1	5.36
	100	—	—	1.031	13.938	152.6	48.5	4.19	3.65	164.83	66.1	1365	170.6	5.30
	120	—	—	1.218	13.564	144.5	56.6	4.19	3.55	192.29	62.6	1556	194.5	5.24
	140	—	—	1.437	13.126	135.3	65.7	4.19	3.44	223.50	58.6	1760	220.0	5.17

(continued)

TABLE B.1 (continued)

Physical Properties of Pipe

Nominal Pipe Size, OD (in.)	Schedule Number			Wall Thickness (in.)	ID (in.)	Inside Area (in.^{2})	Metal Area (in.^{2})	Sq. Ft. Outside Surface (per ft)	Sq. Ft. Inside Surface (per ft)	Weight per ft (lb)	Weight of Water per ft (lb)	Moment of Inertia (in.^{4})	Section Modulus (in.^{3})	Radius Gyration (in.)
	a	b	c											
	160	—	—	1.593	12.814	129.0	72.1	4.19	3.35	245.11	55.9	1894	236.7	5.12
	10	—	—	0.250	17.500	240.5	13.94	4.71	4.58	47.39	104.3	549	61.0	6.28
	20	—	—	0.312	17.376	237.1	17.34	4.71	4.55	59.03	102.8	678	75.5	6.25
	—	Std	—	0.375	17.250	233.7	20.76	4.71	4.52	70.59	101.2	807	89.6	6.23
	30	—	—	0.437	17.126	230.4	24.11	4.71	4.48	82.06	99.9	931	103.4	6.21
	—	XS	—	0.500	17.000	227.0	27.49	4.71	4.45	93.45	98.4	1053	117.0	6.19
	40	—	—	0.562	16.876	223.7	30.8	4.71	4.42	104.75	97.0	1172	130.2	6.17
	—	—	—	0.625	16.750	220.5	34.1	4.71	4.39	115.98	95.5	1289	143.3	6.15
18	—	—	—	0.687	16.626	217.1	37.4	4.71	4.35	127.03	94.1	1403	156.3	6.13
18.000	60	—	—	0.750	16.500	213.8	40.6	4.71	4.32	138.17	92.7	1515	168.3	6.10
	—	—	—	0.875	16.250	207.4	47.1	4.71	4.25	160.04	89.9	1731	192.8	6.06
	80	—	—	0.937	16.126	204.2	50.2	4.71	4.22	170.75	88.5	1834	203.8	6.04
	100	—	—	1.156	15.688	193.3	61.2	4.71	4.11	207.96	83.7	2180	242.2	5.97
	120	—	—	1.375	15.250	182.6	71.8	4.71	3.99	244.14	79.2	2499	277.6	5.90
	140	—	—	1.562	14.876	173.8	80.7	4.71	3.89	274.23	75.3	2750	306	5.84
	160	—	—	1.781	14.438	163.7	90.7	4.71	3.78	308.51	71.0	3020	336	5.77
	10	—	—	0.250	19.500	298.6	15.51	5.24	5.11	52.73	129.5	757	75.7	6.98
	—	—	—	0.312	19.376	294.9	19.30	5.24	5.07	65.40	128.1	935	93.5	6.96
	20	Std	—	0.375	19.250	291.0	23.12	5.24	5.04	78.60	126.0	1114	111.4	6.94
	—	—	—	0.437	19.126	287.3	26.86	5.24	5.01	91.31	124.6	1286	128.6	6.92
	30	XS	—	0.500	19.000	283.5	30.6	5.24	4.97	104.13	122.8	1457	145.7	6.90
	—	—	—	0.562	18.876	279.8	34.3	5.24	4.94	116.67	121.3	1624	162.4	6.88
20	40	—	—	0.593	18.814	278.0	36.2	5.24	4.93	122.91	120.4	1704	170.4	6.86
20.000	—	—	—	0.625	18.750	276.1	38.0	5.24	4.91	129.33	119.7	1787	178.7	6.85
	—	—	—	0.687	18.626	272.5	41.7	5.24	4.88	141.71	118.1	1946	194.6	6.83
	—	—	—	0.750	18.500	268.8	45.4	5.24	4.84	154.20	116.5	2105	210.5	6.81
	60	—	—	0.812	18.376	265.2	48.9	5.24	4.81	166.40	115.0	2257	225.7	6.79
	—	—	—	0.875	18.250	261.6	52.6	5.24	4.78	178.73	113.4	2409	240.9	6.77
	80	—	—	1.031	17.938	252.7	61.4	5.24	4.70	208.87	109.4	2772	277.2	6.72
	100	—	—	1.281	17.438	238.8	75.3	5.24	4.57	256.10	103.4	3320	332	6.63
	120	—	—	1.500	17.000	227.0	87.2	5.24	4.45	296.37	98.3	3760	376	6.56
	140	—	—	1.750	16.500	213.8	100.3	5.24	4.32	341.10	92.6	4220	422	6.48
	160	—	—	1.968	16.064	202.7	111.5	5.24	4.21	379.01	87.9	4590	459	6.41
	10	—	—	0.250	23.500	434	18.65	6.28	6.15	63.41	188.0	1316	109.6	8.40
	—	—	—	0.312	23.376	430	23.20	6.28	6.12	78.93	186.1	1629	135.8	8.38
	20	Std	—	0.375	23.250	425	27.83	6.28	6.09	94.62	183.8	1943	161.9	8.35
	—	—	—	0.437	23.126	420	32.4	6.28	6.05	109.97	182.1	2246	187.4	8.33
	—	XS	—	0.500	23.000	415	36.9	6.28	6.02	125.49	180.1	2550	212.5	8.31
24	30	—	—	0.562	22.876	411	41.4	6.28	5.99	140.80	178.1	2840	237.0	8.29
24.000	—	—	—	0.625	22.750	406	45.9	6.28	5.96	156.03	176.2	3140	261.4	8.27
	40	—	—	0.687	22.626	402	50.3	6.28	5.92	171.17	174.3	3420	285.2	8.25
	—	—	—	0.750	22.500	398	54.8	6.28	5.89	186.24	172.4	3710	309	8.22
	60	—	—	0.968	22.064	382	70.0	6.28	5.78	238.11	165.8	4650	388	8.15
	80	—	—	1.218	21.564	365	87.2	6.28	5.65	296.36	158.3	5670	473	8.07
	100	—	—	1.531	20.938	344	108.1	6.28	5.48	367.40	149.3	6850	571	7.96
	120	—	—	1.812	20.376	326	126.3	6.28	5.33	429.39	141.4	7830	652	7.87
	140	—	—	2.062	19.876	310	142.1	6.28	5.20	483.13	134.5	8630	719	7.79

TABLE B.1 (continued)

Physical Properties of Pipe

Nominal Pipe Size, OD (in.)	Schedule Number a	Schedule Number b	Schedule Number c	Wall Thickness (in.)	ID (in.)	Inside Area (in.²)	Metal Area (in.²)	Sq. Ft. Outside Surface (per ft)	Sq. Ft. Inside Surface (per ft)	Weight per ft (lb)	Weight of Water per ft (lb)	Moment of Inertia (in.⁴)	Section Modulus (in.³)	Radius Gyration (in.)
	160	—	—	2.343	19.314	293	159.4	6.28	5.06	541.94	127.0	9460	788	7.70
	10	—	—	0.312	29.376	678	29.1	7.85	7.69	98.93	293.8	3210	214	10.50
30	20	—	—	0.500	29.000	661	46.3	7.85	7.59	157.53	286.3	5040	336	10.43
30.000	30	—	—	0.625	28.750	649	57.6	7.85	7.53	196.08	281.5	6220	415	10.39

[a] ASA B36.10 Steel-pipe schedule numbers.

[b] ASA B36.10 Steel-pipe nominal wall-thickness designations.

[c] ASA B36.19 Stainless-steel-pipe schedule numbers.

TABLE B.2

Commercial Copper Tubing[a]

Size, OD		Wall Thickness			Flow Area			Surface Area		
in.	mm	in.	mm	gauge	in.^{2}	mm^2	Metal Area (in.^{2})	Inside (ft^2/ft)	Outside (ft^2/ft)	Weight (lb/ft)
1/8	3.2	0.030	0.76	A	0.003	1.9	0.012	0.017	0.033	0.035
3/16	4.76	0.030	0.76	A	0.013	8.4	0.017	0.034	0.049	0.058
1/4	6.4	0.030	0.76	A	0.028	18.1	0.021	0.050	0.066	0.080
1/4	6.4	0.049	1.24	18	0.018	11.6	0.031	0.038	0.066	0.120
5/16	7.94	0.032	0.81	21A	0.048	31.0	0.028	0.065	0.082	0.109
3/8	9.53	0.032	0.81	21A	0.076	49.0	0.033	0.081	0.098	0.134
3/8	9.53	0.049	1.24	18	0.060	38.7	0.050	0.072	0.098	0.195
1/2	12.7	0.032	0.81	21A	0.149	96.1	0.047	0.114	0.131	0.182
1/2	12.7	0.035	0.89	20L	0.145	93.6	0.051	0.113	0.131	0.198
1/2	12.7	0.049	1.24	18K	0.127	81.9	0.069	0.105	0.131	0.269
1/2	12.7	0.065	1.65	16	0.108	69.7	0.089	0.97	0.131	0.344
5/8	15.9	0.035	0.89	20A	0.242	156	0.065	0.145	0.164	0.251
5/8	15.9	0.040	1.02	L	0.233	150	0.074	0.143	0.164	0.285
5/8	15.9	0.049	1.24	18K	0.215	139	0.089	0.138	0.164	0.344
3/4	19.1	0.035	0.89	20A	0.363	234	0.079	0.178	0.196	0.305
3/4	19.1	0.042	1.07	L	0.348	224	0.103	0.174	0.196	0.362
3/4	19.1	0.049	1.24	18K	0.334	215	0.108	0.171	0.196	0.418
3/4	19.1	0.065	1.65	16	0.302	195	0.140	0.162	0.196	0.542
3/4	19.1	0.083	2.11	14	0.268	173	0.174	0.151	0.196	0.674
7/8	22.2	0.045	1.14	L	0.484	312	0.117	0.206	0.229	0.455
7/8	22.2	0.065	1.65	16K	0.436	281	0.165	0.195	0.229	0.641
7/8	22.2	0.083	2.11	14	0.395	255	0.206	0.186	0.229	0.800
1	25.4	0.065	1.65	16	0.594	383	0.181	0.228	0.262	0.740
1	25.4	0.083	2.11	14	0.546	352	0.239	0.218	0.262	0.927
1 1/8	28.6	0.050	1.27	L	0.825	532	0.176	0.268	0.294	0.655
1 1/8	28.6	0.065	1.65	16K	0.778	502	0.216	0.261	0.294	0.839
1 1/4	31.8	0.065	1.65	16	0.985	636	0.242	0.293	0.327	0.938
1 1/4	31.8	0.083	2.11	14	0.923	596	0.304	0.284	0.327	1.18
1 3/8	34.9	0.055	1.40	L	1.257	811	0.228	0.331	0.360	0.884
1 3/8	34.9	0.065	1.65	16K	1.217	785	0.267	0.326	0.360	1.04
1 1/2	38.1	0.065	1.65	16	1.474	951	0.294	0.359	0.393	1.14
1 1/2	38.7	0.083	2.11	14	1.398	902	0.370	0.349	0.393	1.43
1 5/8	41.3	0.060	1.52	L	1.779	1148	0.295	0.394	0.425	1.14
1 5/8	41.3	0.072	1.83	K	1.722	1111	0.351	0.388	0.425	1.36
2	50.8	0.083	2.11	14	2.642	1705	0.500	0.480	0.628	1.94
2	50.8	0.109	2.76	12	2.494	1609	0.620	0.466	0.628	2.51
2 1/8	54.0	0.070	1.78	L	3.095	1997	0.449	0.520	0.556	1.75
2 1/8	54.0	0.083	2.11	14K	3.016	1946	0.529	0.513	0.556	2.06
2 5/8	66.7	0.080	2.03	L	4.77	3078	0.645	0.645	0.687	2.48
2 5/8	66.7	0.095	2.41	13K	4.66	3007	0.760	0.637	0.687	2.93
3 1/8	79.4	0.090	2.29	L	6.81	4394	0.950	0.771	0.818	3.33
3 1/8	79.4	0.109	2.77	12K	6.64	4284	1.034	0.761	0.818	4.00

TABLE B.2 (continued)

Commercial Copper Tubing[a]

Size, OD		Wall Thickness			Flow Area			Surface Area		
in.	mm	in.	mm	gauge	in.^{2}	mm^2	Metal Area (in.^{2})	Inside (ft^2/ft)	Outside (ft^2/ft)	Weight (lb/ft)
3 5/8	92.1	0.100	2.54	L	9.21	5942	1.154	0.897	0.949	4.29
3 5/8	92.1	0.120	3.05	11K	9.00	5807	1.341	0.886	0.949	5.12
4 1/8	104.8	0.110	2.79	L	11.92	7691	1.387	1.022	1.080	5.38
4 1/8	104.8	0.134	3.40	10K	11.61	7491	1.682	1.009	1.080	6.51

Source: *The CRC Handbook of Mechanical Engineering*, CRC Press, Boca Raton, FL, 1998.

Notes: The table above gives dimensional data and weights of copper tubing used for automotive, plumbing, refrigeration, and heat exchanger services. For additional data see the standards handbooks of the Copper Development Association, Inc., the ASTM standards, and the "SAE Handbook."

Dimensions in this table are actual specified measurements, subject to accepted tolerances. Trade size designations are usually by actual OD, except for water and drainage tube (plumbing), which measures 1/8 in. larger OD. A 1/2 in. plumbing tube, for example, measures 5/8 in. OD, and a 2 in. plumbing tube measures 2 1/8 in. OD.

Key to Gauge Sizes

Standard-gauge wall thicknesses are listed by numerical designation (14–21), BWG or Stubs gauge. These gauge sizes are standard for tubular heat exchangers. The letter A designates SAE tubing sizes for automotive service. Letter designations *K* and *L* are the common sizes for plumbing services, soft or hard temper.

Other Materials

These same dimensional sizes are also common for much of the commercial tubing available in aluminum, mild steel, brass, bronze, and other alloys. Tube weights in this table are based on copper at 0.323 lb/in^3. For other materials the weights should be multiplied by the following approximate factors:

Aluminum	0.30
Monel	0.96
Mild steel	0.87
Stainless steel	0.89
Brass	0.95

[a] Compiled and computed.

Appendix C: Properties of Gases and Liquids

TABLE C.1

Properties of Gases at Atmospheric Pressure (101.3 kPa = 14.7 psia): Air (Gas Constant = 286.8 J/(kg K) = 53.3 ft lbf/lbm °R; $\gamma = c_p/c_v = 1.4$)

Temp, T		Density, p		Specific Heat, c_p		Kinematic Viscosity, v		Thermal Conductivity, k		Thermal Diffusivity, α		Prandtl Number, P_r
K	°R	kg/m^3	lbm/ft^3	J/kg K	BTU/ lbm °R	m^2/s	ft^2/s	W/m K	BTU/h ft °R	m^2/s	ft^2/h	
100	180	3.601	0.225	1026.6	0.245	1.923×10^{-6}	2.070×10^{-5}	0.009246	0.005342	0.02501×10^{-6}	0.0869	0.770
150	270	2.368	0.148	1009.9	0.241	4.343	4.674	0.013735	0.007936	0.05745	0.223	0.753
200	360	1.768	0.110	1006.1	0.240	7.490	8.062	0.01809	0.01045	0.10165	0.394	0.739
250	450	1.413	0.0882	1005.3	0.240	9.49	10.2	0.02227	0.02287	0.13161	0.510	0.722
300	540	1.177	0.0735	1005.7	0.240	15.68	16.88	0.02624	0.01516	0.22160	0.859	0.708
350	630	0.998	0.0623	1009.0	0.241	20.76	22.35	0.03003	0.01735	0.2983	1.156	0.697
400	720	0.883	0.0551	1014.0	0.242	25.90	27.88	0.03365	0.01944	0.3760	1.457	0.689
450	810	0.783	0.489	1020.7	0.244	28.86	31.06	0.037.7	0.02142	0.4222	1.636	0.683
500	900	0.705	0.0440	1029.5	0.245	37.90	40.80	0.04038	0.02333	0.5564	2.356	0.680
550	990	0.642	0.0401	1039.2	0.248	44.34	47.73	0.04360	0.02519	0.6532	2.531	0.680
600	1000	0.589	0.0367	1055.1	0.252	51.34	55.26	0.04659	0.02682	0.7512	2.911	0.680
650	1170	0.543	0.0339	1063.5	0.254	58.51	62.98	0.00953	0.02862	0.8578	3.324	0.682
700	1260	0.503	0.0314	1075.2	0.257	66.25	7131	0.05230	0.030023	0.9672	3.748	0.684
750	1350	0.471	0.0594	1085.6	0.259	73.91	79.56	0.05509	0.03183	1.0774	4.175	0.686
800	1440	0.441	0.0275	1097.8	0.262	8229	88.58	0.05779	0.03339	1.1951	4.631	0.689
850	1530	0.415	0.0259	1109.5	0.265	90.75	97.68	0.06028	0.03483	1.3097	5.075	0.692
900	1620	0.393	0.0245	1121.2	0.268	99.3	107	0.06279	0.03628	1.4278	5.530	0.696
950	1710	0.372	0.0232	1132.1	0.270	108.2	116.5	0.06525	0.03770	1.5510	6.010	0.699
1000	1800	0.352	0.0220	1141.7	0.273	117.8	126.8	0.06752	0.03901	1.6779	6502	0.702
1100	1980	0.320	0.0120	1160	0.277	138.6	149.2	0.0732	0.0423	1.969	7.630	0.704
1200	2160	0.295	0.0184	1179	0.282	159.1	171.3	0.0782	0.0423	1.969	7.630	0.707
1300	2340	0.271	0.0189	1197	0.286	182.1	196.0	0.0837	0.0434	2.583	10.01	0.705
1400	2520	0.252	0.0157	1214	0.290	205.5	221.2	0.0891	0.0515	2.920	11.32	0.705
1500	2700	0.236	0.0147	1230	0.294	229.1	246.6	0.0946	0.0547	3.262	1264	0.705
1600	2880	0.221	0.0138	1248	0.298	254.5	273.9	0.100	0.0578	3.609	13.98	0.705
1700	36060	0.208	0.0130	1267	0.303	280.5	301.9	0.105	0.0607	3.977	15.41	0.705
1800	3240	0.197	0.0123	1287	0.307	308.1	331.6	0.111	0.0641	4.379	16.97	0.704
1900	3420	0.186	0.0115	1309	0.383	338.5	364.4	0.117	0.0676	4.811	18.64	0.704
2000	3600	0.176	0.0110	1338	0.320	369.0	397.2	0.124	0.0716	5.260	20.38	0.702
2100	3780	0.168	0.0105	1372	0.328	399.6	430.1	0.131	0.0757	5.715	22.15	0.700
2200	3960	0.160	0.0100	1419	0.339	432.6	465.6	0.139	0.0803	6120	2372	0.707
2300	4140	0.154	0.00955	1482	0.354	464.0	499.4	0.149	0.0861	6.540	25.34	0.710
2400	4320	0.146	0.00905	1574	0376	504.0	542.5	0.161	0.0930	7.020	27.20	0.718
2500	4500	0.139	0.00868	1688	0.403	543.5	585.0	0.175	0.101	7.441	28.83	0.730

Source: Janna, W.S., *Engineering Heat Transfer*, 2nd edn., CRC Press, Boca Raton, FL, 2000, p. 654, Table D.1.

TABLE C.2

Properties of Gases at Atmospheric Pressure (101.3 kPa = 14.7 psia): Nitrogen (Gas Constant = 296.8 J/(kg K) = 55.16 ft lbf/lbm °R; $\gamma = c_p/c_v = 1.40$)

Temp, T		Density, p		Specific Heat, c_p		Kinematic Viscosity, v		Thermal Conductivity, k		Thermal Diffusivity, α		
K	°R	kg/m^3	lbm/ft^3	J/kg K	BTU/lbm °R	m^2/s	ft^2/s	W/m K	BTU/h ft °R	m^2/s	ft^2/h	Prandtl Number, P_r
100	180	3.4808	0.2173	1072.2	0.2561	1.971×10^{-6}	2.122×10^{-5}	0.009450	0.005460	0.025319×10^{-4}	0.09811	0.786
200	360	1.7108	0.1068	1042.9	0.2491	7.568	8.146	0.01824	0.01054	0.10224	0.3962	0.747
300	540	1.1421	0.0713	1040.8	0.2486	15.63	16.82	0.02620	0.01514	0.22044	0.8542	0.713
400	720	0.8538	0.0533	1045.9	0.2498	25.74	27.71	0.03335	0.01927	0.3734	1.447	0.691
500	900	0.6824	0.0426	1055.5	0.2521	37.66	40.54	0.03984	0.02302	0.5530	2.143	0.684
600	1080	0.5687	0.0355	1075.6	0.2569	51.19	55.10	0.4580	0.02646	0.7486	2.901	0.686
700	1260	0.4934	0.0308	1096.9	0.2620	65.13	7010	0.05123	0.02960	0.9466	3.668	0.691
800	1440	04277	0.0267	1122.5	0.2681	81.46	87.68	0.05609	0.03241	1.1685	4.528	0.700
900	1620	0.3796	0.0237	1146.4	0.2738	91.06	98.02	0.06070	0.03507	1.3946	5.404	0.711
1000	1800	0.3412	0.0213	1167.7	0.2789	117.2	126.2	0.06475	0.03741	1.6250	6.297	0.724
1100	1980	0.3108	0.0194	1185.7	0.2382	136.0	146.4	0.06850	0.03958	1.8591	7.204	0.736
1200	2160	0.2851	0.0178	1203.7	0.2875	156.1	168.0	0.07184	0.04151	2.0932	8.111	0.748

Source: Janna, W.S., *Engineering Heat Transfer*, 2nd edn., CRC Press, Boca Raton, FL, 2000, p. 657, Table D.5.

TABLE C.3

Properties of Gases at Atmospheric Pressure (101.3 kPa = 14.7 psia): Oxygen (Gas Constant = 260 J/(kg K) = 48.3 ft lbf/lbm °R; $\gamma = c_p/c_v = 1.40$)

Temp, T		Density, p		Specific Heat, c_p		Kinematic Viscosity, v		Thermal Conductivity, k		Thermal Diffusivity, α		
K	°R	kg/m^3	lbm/ft^3	J/kg K	BTU/lbm °R	m^2/s	ft^2/s	W/m K	BTU/h ft °R	m^2/s	ft^2/h	Prandtl Number, P_r
100	180	3.9118	0.2492	947.9	0.2264	1.946×10^{-6}	2.095×10^{-5}	0.00903	0.00522	0.023876×10^{-4}	0.09252	0.815
150	270	26190	0.1635	917.8	0.2192	4.387	4.722	0.01367	0.00790	0.05688	0.2204	0.773
200	360	1.9559	0.1221	913.1	0.2181	7.593	8.173	0.01824	0.01054	0.10214	0.3958	0.745
250	450	1.5618	0.0975	915.7	0.2187	11.45	12.32	0.02259	0.01305	0.15794	0.6120	0.725
300	540	1.3007	0.0812	920.3	0.2198	15.86	17.07	0.02676	0.01546	0.22353	0.8662	0.709
350	630	1.1133	0.0695	929.1	0.2219	20.80	22.39	0.03070	0.01774	0.2968	1.150	0.702
400	720	0.9755	0.0609	942.0	0.2250	26.18	2818	0.03461	0.02000	0.3768	1.460	0.695
450	810	0.8682	0.0542	956.7	0.2285	31.99	34.43	0.03828	0.02212	0.4609	1.786	0.694
500	900	0.7801	0.0487	972.2	0.2322	38.37	41.27	0.04173	0.02411	0.5502	2.132	0.697
550	990	0.7096	0.0443	988.1	0.2360	45.05	48.49	0.04517	0.02610	06441	2.496	0.700
600	1080	0.6508	0.0406	1004.4	0.2399	52.15	56.13	0.04882	0.02792	0.7399	2.867	0.704

Source: Janna, W.S., *Engineering Heat Transfer*, 2nd edn., CRC Press, Boca Raton, FL, 2000, p. 658, Table D.6.

TABLE C.4

Properties of Gases at Atmospheric Pressure (101.3 kPa = 14.7 psia): Carbon Dioxide (Gas Constant = 188.9 J/(kg K) = 35.11 ft lbf/lbm °R; $\gamma = c_p/c_v = 1.30$)

Temp, *T*		Density, *p*		Specific Heat, c_p		Kinematic Viscosity, *v*		Thermal Conductivity, *k*		Thermal Diffusivity, α		
K	°R	kg/m³	lbm/ft³	J/kg K	BTU/lbm °R	m²/s	ft²/s	W/m K	BTU/h ft °R	m²/s	ft²/h	Prandtl Number, P_r
220	396	2.4733	0.1544	783	0.187	4.490×10^{-6}	4.833×10^{-5}	0.010805	0.006243	0.05920×10^{-4}	0.2294	0.818
250	450	2.1657	0.1352	804	0.192	5.813	6.257	0.012884	0.007444	0.07401	0.2868	0.793
300	540	1.7973	0.1122	871	0.208	8.321	8.957	0.016572	0.009575	0.10588	0.4103	0.770
350	630	1.5362	0.0959	900	0.215	11.19	12.05	0.02047	0.01183	0.14808	0.5738	0.755
400	720	1.3424	0.0838	942	0.225	14.39	15.49	0.02461	0.01422	0.19463	0.7542	0.738
450	810	1.1918	0.0744	980	0.234	17.90	19.27	0.02897	0.01674	0.24813	0.9615	0.721
500	900	1.0732	0.0670	1013	0.242	21.67	23.33	0.03352	0.01937	0.3084	1.195	0.702
550	990	0.9739	0.0608	1047	0.250	25.74	27.71	0.03821	0.02208	0.3750	1.453	0.685
600	1080	0.8938	0.0558	1076	0.257	30.02	32.31	0.04313	0.02491	0.4483	1.737	0.668

Source: Janna, W.S., *Engineering Heat Transfer*, 2nd edn., CRC Press, Boca Raton, FL, 2000, p. 655, Table D.2.

TABLE C.5

Properties of Gases at Atmospheric Pressure (101.3 kPa = 14.7 psia): Water Vapor or Steam (Gas Constant = 461.5 J/(kg K) = 85.78 ft lbf/lbm °R; $\gamma = c_p/c_v = 1.33$)

Temp, *T*		Density, *p*		Specific Heat, c_p		Kinematic Viscosity, *v*		Thermal Conductivity, *k*		Thermal Diffusivity, α		
K	°R	kg/m³	lbm/ft³	J/kg K	BTU/lbm °R	m²/s	ft²/s	W/m K	BTU/h ft °R	m²/s	ft²/h	Prandtl Number, P_r
380	684	0.5863	0.0366	2060	0.492	2.16×10^{-6}	2.33×10^{-5}	0.0246	0.0142	0.2036×10^{-4}	0.789	1.060
400	720	0.5542	0.0346	2014	0.481	2.42	2.61	0.0261	0.0151	0.2338	0.906	1.040
450	810	0.4902	0.0306	1980	0.473	3.11	3.35	0.0299	0.0173	0.307	1.19	1.010
500	900	0.4005	0.0275	1985	0.474	3.86	4.16	0.0339	0.0196	0.387	1.50	0.996
550	990	0.4005	0.0250	1997	0.477	4.70	5.06	0.0379	0.0219	0.475	1.84	0.991
600	1080	0.3652	0.0228	2026	0.484	5.66	6.09	0.0422	0.0244	0.573	2.22	0.986
650	1170	0.3380	0.0211	2056	0.491	6.64	7.15	0.0464	0.0268	0.666	2.58	0.995
700	1260	13140	0.0196	2085	0.498	7.75	8.31	0.0505	0.0292	0.772	2.99	1.000
750	1350	0.2931	0.0183	2119	0.506	8.88	9.56	0.0549	0.0317	0.883	3.42	0.005
800	1440	0.2739	0.0171	2152	0.514	10.20	10.98	0.0592	0.0342	1.001	3.88	1.010
850	1530	0.2579	0.0161	2186	0.522	11.52	12.40	0.0637	0.0368	1.130	4.38	1.019

Source: Janna, W.S., *Engineering Heat Transfer*, 2nd edn., CRC Press, Boca Raton, FL, 2000, p. 659, Table D.7.

TABLE C.6

Thermodynamic Properties of Steam: Temperature Table (SI Units)

T_{sat} (°C)	P_{sat} (kPa)	Specific Volume (m³/kg)			Internal Energy (kJ/kg)			Enthalpy (kJ/kg)			Entropy (kJ/kg K)		
		v_f	v_{fg}	v_g	u_f	u_{fg}	u_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
0	0.61	0.001000	206.13	206.13	0.00	2373.9	2373.9	0.0	2500.0	2500.0	−0.0012	9.1590	9.1578
5	0.87	0.001000	147.20	147.20	21.04	2361.1	2382.1	21.0	2489.6	2510.6	0.0757	8.9510	9.0267
10	1.23	0.001000	106.36	106.36	42.02	2347.8	2389.8	42.0	2478.4	2520.4	0.1509	8.7511	8.9020
15	1.71	0.001001	78.036	78.037	62.95	2333.7	2396.7	63.0	2466.8	2529.7	0.2244	8.5582	8.7827
20	2.34	0.001002	57.801	57.802	83.86	2319.9	2403.7	83.9	2455.0	2538.9	0.2965	8.3718	8.6684
25	3.17	0.001003	43.446	43.447	104.75	2305.5	2410.3	104.8	2443.1	2547.9	0.3672	8.1919	8.5591
30	4.24	0.001004	32.907	32.908	125.63	2291.6	2417.2	125.6	2431.2	2556.8	0.4367	8.0180	8.4546
35	5.62	0.001006	25.250	25.251	146.50	2277.3	2423.8	146.5	2419.2	2565.7	0.5049	7.8496	8.3545
40	7.37	0.001008	19.536	19.537	167.37	2263.2	2430.6	167.4	2407.3	2574.6	0.5720	7.6864	8.2584
45	9.58	0.001010	15.262	15.263	188.24	2249.1	2437.3	188.3	2395.3	2583.5	0.6381	7.5281	8.1662
50	12.33	0.001012	12.046	12.047	209.12	2234.7	2443.8	209.1	2383.2	2592.3	0.7031	7.3745	8.0776
55	15.74	0.001014	9.5771	9.5781	230.01	2220.4	2450.4	230.0	2371.1	2601.1	0.7672	7.2253	7.9925
60	19.92	0.001017	7.6776	7.6786	250.91	2206.0	2456.9	250.9	2358.9	2609.8	0.8303	7.0804	7.9107
65	25.00	0.001020	6.1996	6.2006	271.83	2191.6	2463.4	271.9	2346.6	2618.4	0.8926	6.9394	7.8320
70	31.15	0.001023	5.0452	5.0462	292.76	2177.0	2469.7	292.8	2334.2	2626.9	0.9540	6.8023	7.7563
75	38.54	0.001026	4.1328	4.1338	313.70	2162.3	2476.0	313.7	2321.6	2635.4	1.0146	6.6687	7.6834
80	47.35	0.001029	3.4074	3.4085	334.67	2147.6	2482.3	334.7	2309.8	2643.7	1.0744	6.5387	7.6131
85	57.80	0.001032	2.8276	2.8286	355.65	2132.8	2488.4	355.7	2296.2	2651.9	1.1335	6.4118	7.5453
90	70.10	0.001036	2.3604	2.3614	376.66	2117.8	2494.5	376.7	2283.3	2660.0	1.1917	6.2881	7.4798
95	84.52	0.001039	1.9806	1.9817	397.69	2102.8	2500.5	397.8	2270.2	2668.0	1.2493	6.1673	7.4166
100	101.32	0.001043	1.6689	1.6699	418.75	2087.9	2506.6	418.9	2257.0	2675.8	1.3062	6.0492	7.3554
105	120.80	0.001047	1.4142	1.4152	439.83	2072.8	2512.6	440.0	2243.6	2683.6	1.3624	5.9338	7.2962
110	143.27	0.001051	1.2063	1.2074	460.95	2057.2	2518.2	461.1	2230.0	2691.1	1.4179	5.8209	7.2388
115	169.07	0.001056	1.0350	1.0361	482.10	2041.3	2523.4	482.3	2216.3	2698.6	1.4728	5.7105	7.1833
120	198.55	0.001060	0.89100	0.8921	503.28	2025.4	2528.7	503.5	2202.3	2705.8	1.5271	5.6023	7.1293
125	232.11	0.001065	0.76938	0.7704	524.51	2009.6	2534.1	524.8	2188.2	2712.9	1.5807	5.4962	7.0770
130	270.15	0.001070	0.66702	0.6681	545.78	1993.6	2539.4	546.1	2173.8	2719.9	1.6338	5.3922	7.0261
135	313.09	0.001075	0.58074	0.5818	567.09	1977.3	2544.4	567.4	2159.2	2726.6	1.6864	5.2902	6.9766
140	361.39	0.001080	0.50739	0.5085	588.46	1960.9	2549.3	588.8	2144.3	2733.1	1.7384	5.1900	6.9284
145	415.53	0.001085	0.44462	0.4457	609.88	1944.3	2554.2	610.3	2129.1	2739.4	1.7899	5.0916	6.8815
150	475.99	0.001091	0.39100	0.3921	631.35	1927.5	2558.8	631.9	2113.6	2745.5	1.8409	4.9948	6.8358
155	543.30	0.001096	0.34514	0.3462	652.89	1910.3	2563.2	653.5	2097.8	2751.3	1.8915	4.8996	6.7911
160	618.00	0.001102	0.30566	0.3068	674.50	1892.8	2567.3	675.2	2081.7	2756.9	1.9416	4.8059	6.7475
165	700.68	0.001108	0.27131	0.2724	696.18	1875.1	2571.3	697.0	2065.2	2762.2	1.9912	4.7135	6.7048
170	791.86	0.001114	0.24141	0.2425	717.93	1857.2	2575.2	718.8	2048.4	2767.2	2.0405	4.6224	6.6630
175	892.20	0.001121	0.21538	0.2165	739.77	1839.0	2578.8	740.8	2031.2	2772.0	2.0894	4.5325	6.6220
180	1002.3	0.001127	0.19266	0.1938	761.69	1820.5	2582.1	762.8	2013.6	2776.4	2.1380	4.4437	6.5817
185	1122.9	0.001134	0.17272	0.1739	783.70	1803.6	2585.3	785.0	1995.5	2780.5	2.1862	4.3559	6.5421
190	1254.5	0.001141	0.15513	0.1563	805.80	1782.4	2588.3	807.2	1977.1	2784.3	2.2341	4.2691	6.5032
195	1398.0	0.001148	0.13964	0.1408	828.01	1762.9	2590.9	829.6	1958.1	2787.8	2.2817	4.1834	6.4651
200	1553.9	0.001156	0.12597	0.1271	850.32	1743.0	2593.3	852.1	1938.8	2790.9	2.3290	4.0986	6.4276
205	1723.1	0.001164	0.11386	0.1150	872.74	1722.7	2595.4	874.7	1918.9	2793.6	2.3761	4.0147	6.3908
210	1906.3	0.001172	0.10307	0.1042	895.28	1702.0	2597.3	897.5	1898.5	2796.0	2.4230	3.9314	6.3544
215	2104.3	0.001180	0.09345	0.0946	917.94	1681.0	2598.9	920.4	1877.6	2798.0	2.4696	3.8485	6.3181
220	2317.8	0.001189	0.08486	0.0860	940.73	1659.5	2600.2	943.5	1856.2	2799.7	2.5161	3.7661	6.2821
225	2547.8	0.001198	0.07716	0.0784	963.66	1637.6	2601.3	966.7	1834.2	2800.9	2.5623	3.6841	6.2464
230	2795.0	0.001208	0.07022	0.0714	986.73	1615.4	2602.2	990.1	1811.7	2801.8	2.6084	3.6025	6.2109
235	3060.3	0.001218	0.06400	0.0652	1010.0	1592.7	2602.7	1033.7	1788.6	2802.3	2.6544	3.5213	6.1757
240	3344.7	0.001228	0.05851	0.0597	1033.6	1569.1	2602.5	1037.5	1764.8	2802.3	2.7002	3.4404	6.1406
245	3649.0	0.001239	0.05353	0.0548	1056.9	1545.2	2602.1	1061.4	1740.5	2801.9	2.7460	3.3597	6.1057

TABLE C.6 (continued)

Thermodynamic Properties of Steam: Temperature Table (SI Units)

T_{sat} (°C)	P_{sat} (kPa)	Specific Volume (m³/kg)			Internal Energy (kJ/kg)			Enthalpy (kJ/kg)			Entropy (kJ/kg K)		
		ν_f	ν_{fg}	ν_g	u_f	u_{fg}	u_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
250	3974.2	0.001250	0.04893	0.0502	1080.7	1521.0	2601.7	1085.6	1715.5	2801.2	2.7917	3.2792	6.0708
255	4321.3	0.001262	0.04471	0.0460	1104.6	1496.7	2601.3	1110.1	1689.9	2800.0	2.8373	3.1986	6.0359
260	4691.2	0.001275	0.04086	0.0421	1128.8	1471.9	2600.7	1134.8	1663.5	2798.3	2.8829	3.1180	6.0009
265	5085.0	0.001288	0.03738	0.0387	1153.2	1446.4	2599.6	1159.8	1636.5	2796.3	2.9286	3.0372	5.9657
270	5503.8	0.001302	0.03424	0.0355	1177.9	1420.3	2598.1	1185.1	1608.7	2793.7	2.9743	2.9560	5.9303
275	5948.6	0.001317	0.03139	0.0327	1202.8	1393.4	2596.3	1210.7	1580.1	2790.8	3.0200	2.8745	5.8945
280	6420.5	0.001333	0.02878	0.0301	1228.1	1366.0	2594.0	1236.6	1550.8	2787.4	3.0660	2.7924	5.8584
285	6920.8	0.001349	0.02639	0.0277	1253.7	1337.9	2591.6	1263.0	1520.6	2783.6	3.1121	2.7097	5.8218
290	7450.6	0.001366	0.02418	0.0255	1279.6	1297.7	2577.3	1289.8	1477.9	2767.7	3.1585	2.6262	5.7847
295	8011.1	0.001385	0.02214	0.0235	1306.0	1265.5	2571.5	1317.1	1442.8	2759.9	3.2052	2.5417	5.7469
300	8603.7	0.001404	0.02025	0.0217	1332.8	1232.0	2564.8	1344.9	1406.2	2751.1	3.2523	2.4560	5.7083
305	9214.4	0.001425	0.01850	0.0199	1360.2	1197.5	2557.6	1373.3	1367.9	2741.2	3.3000	2.3688	5.6687
310	9869.4	0.001447	0.01688	0.0183	1388.0	1161.1	2549.2	1402.3	1327.8	2730.1	3.3483	2.2797	5.6279
315	10,561.0	0.001470	0.01538	0.0169	1416.5	1123.0	2539.6	1432.1	1285.5	2717.6	3.3973	2.1884	5.5858
320	11,289.0	0.001499	0.01398	0.0155	1445.7	1083.0	2528.7	1462.6	1240.9	2703.5	3.4473	2.0947	5.5420
325	12,056.0	0.001528	0.01267	0.0142	1475.5	1040.9	2516.4	1494.0	1193.6	2687.3	3.4984	1.9979	5.4962
330	12,862.0	0.001561	0.01143	0.0130	1506.2	996.3	2502.5	1526.3	1143.3	2669.6	3.5507	1.8973	5.4480
335	13,712.0	0.001598	0.01026	0.0119	1537.8	949.0	2486.8	1559.7	1089.6	2649.3	3.6045	1.7922	5.3967
340	14,605.0	0.001639	0.00914	0.0108	1570.4	898.7	2469.0	1594.3	1032.2	2626.5	3.6601	1.6820	5.3420
345	15,545.0	0.001686	0.00808	0.0098	1606.3	842.1	2448.4	1632.5	967.7	2600.2	3.7176	1.5658	5.2834
350	16,535.0	0.001741	0.00706	0.0088	1643.0	780.5	2423.5	1671.8	897.2	2569.0	3.7775	1.4416	5.2191
355	17,577.0	0.001808	0.00605	0.0079	1682.1	710.9	2393.0	1713.9	817.3	2531.2	3.8400	1.3054	5.1454
360	18,675.0	0.001896	0.00504	0.0069	1726.2	629.5	2355.7	1761.6	723.7	2485.3	3.9056	1.1531	5.0587
365	19,833.0	0.002016	0.00400	0.0060	1777.9	531.0	2308.9	1817.8	610.3	2428.1	3.9746	0.9822	4.9569
370	21,054.0	0.002225	0.00274	0.0050	1843.3	394.1	2237.3	1890.1	451.9	2342.0	4.0476	0.7555	4.8030
374.4	22,090.0	0.00315	0.00000	0.00315	2029.6	0.0	2029.6	2099.3	0.0	2099.3	4.4298	0.0	4.4298

Source: Properties obtained from software, *STEAMCALC*, John Wiley & Sons, New York, 1983; *Introduction to Thermal and Fluid Engineering*, CRC Press, Boca Raton, FL, 2012, p. 901, Table A.3.

TABLE C.7

Thermodynamic Properties of Steam: Pressure Table (SI Units)

P_{sat}	T_{sat}	Specific Volume (m³/kg)			Internal Energy (kJ/kg)			Enthalpy (kJ/kg)			Entropy (kJ/kg K)		
(kPa)	(°C)	v_f	v_{fg}	v_g	u_f	u_{fg}	u_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
1.00	7.0	0.001000	129.08	129.08	29.40	2356.1	2385.5	29.4	2485.2	2514.6	0.1058	8.8704	8.9763
1.50	13.0	0.001001	88.067	88.068	54.68	2339.3	2394.0	54.7	2471.4	2526.3	0.1956	8.6337	8.8292
2.00	17.5	0.001001	67.073	67.074	73.41	2326.8	2400.2	73.4	2460.9	2534.3	0.2607	8.4642	8.7249
2.50	21.1	0.001002	54.290	54.291	88.41	2316.7	2405.1	88.4	2452.4	2540.8	0.3120	8.3322	8.6442
3.00	24.1	0.001003	45.751	45.752	100.96	2308.0	2409.0	101.0	2445.3	2546.2	0.3545	8.2240	8.5786
3.50	26.7	0.001003	39.483	39.484	111.81	2300.9	2412.7	111.8	2439.1	2550.9	0.3908	8.1324	8.5233
4.00	29.0	0.001004	34.779	34.780	121.37	2294.5	2415.9	121.4	2433.6	2555.0	0.4226	8.0529	8.4756
4.50	31.0	0.001005	31.128	31.129	129.95	2288.6	2418.6	130.0	2428.7	2558.7	0.4509	7.9827	8.4336
5.00	32.9	0.001005	28.194	28.195	137.73	2283.3	2421.0	137.7	2424.3	2562.0	0.4764	7.9197	8.3961
5.50	34.6	0.001006	25.773	25.774	144.86	2278.4	2423.3	144.9	2420.2	2565.0	0.4996	7.8626	8.3622
6.00	36.2	0.001006	23.742	23.743	151.45	2274.0	2425.4	151.5	2416.4	2567.9	0.5209	7.8104	8.3313
6.50	37.7	0.001007	22.013	22.014	157.58	2269.8	2427.4	157.6	2412.9	2570.5	0.5407	7.7623	8.3030
7.00	39.0	0.001007	20.522	20.523	163.31	2266.0	2429.3	163.3	2409.6	2572.9	0.5590	7.7177	8.2768
7.50	40.3	0.001008	19.225	19.226	168.70	2262.3	2431.0	168.7	2406.5	2575.2	0.5763	7.6762	8.2524
8.00	41.5	0.001008	18.086	18.087	173.79	2258.9	2432.7	173.8	2403.6	2577.4	0.5924	7.6372	8.2296
8.50	42.7	0.001009	17.080	17.081	178.61	2255.6	2434.2	178.6	2400.8	2579.4	0.6077	7.6006	8.2083
9.00	43.8	0.001009	16.185	16.186	183.19	2252.5	2435.7	183.2	2398.2	2581.4	0.6222	7.5660	8.1881
9.50	44.8	0.001010	15.383	15.384	187.56	2249.5	2437.1	187.6	2395.7	2583.2	0.6359	7.5332	8.1691
10.00	45.8	0.001010	14.660	14.661	191.74	2246.7	2438.4	191.7	2393.3	2585.0	0.6490	7.5021	8.1511
15.00	54.0	0.001014	10.020	10.021	225.83	2223.2	2449.0	225.8	2373.5	2599.4	0.7544	7.2548	8.0092
20.00	60.1	0.001017	7.6483	7.6493	251.28	2205.7	2457.0	251.3	2358.7	2610.0	0.8314	7.0779	7.9093
25.00	65.0	0.001020	6.2015	6.2025	271.80	2191.6	2463.3	271.8	2346.6	2618.4	0.8925	6.9396	7.8321
30.00	69.1	0.001022	5.2277	5.2287	289.09	2379.5	2468.6	289.1	2336.3	2625.5	0.9433	6.8260	7.7693
35.00	72.7	0.001024	4.5249	4.5259	304.11	2169.0	2473.1	304.1	2327.4	2631.5	0.9869	6.7295	7.7164
40.00	75.9	0.001026	3.9918	3.9929	317.42	2159.7	2477.1	317.5	2319.4	2636.8	1.0253	6.6455	7.6707
45.00	78.7	0.001028	3.5744	3.5755	329.40	2151.3	2480.7	329.4	2312.2	2641.6	1.0595	6.5710	7.6305
50.00	81.3	0.001030	3.2389	3.2398	340.31	2143.6	2483.9	340.4	2305.5	2645.9	1.0904	6.5042	7.5946
60.00	86.0	0.001033	2.7305	2.7316	359.66	2129.9	2489.6	359.7	2293.7	2653.5	1.1446	6.3880	7.5326
70.00	90.0	0.001036	2.3638	2.3648	376.49	2117.9	2494.4	376.6	2283.4	2659.9	1.1913	6.2891	7.4804
80.00	93.5	0.001038	2.0859	2.0869	391.43	2107.2	2498.7	391.5	2274.1	2665.6	1.2323	6.2029	7.4352
90.00	96.7	0.001041	1.8667	1.8678	404.90	2097.7	2502.6	405.0	2265.7	2670.7	1.2689	6.1265	7.3954
100.00	99.6	0.001043	1.6898	1.6908	417.20	2089.0	2506.2	417.3	2258.0	2675.3	1.3020	6.0578	7.3598
101.32	100.0	0.001043	1.66895	1.6700	418.74	2087.9	2506.6	418.8	2257.0	2675.8	1.3062	6.0493	7.3554
125.00	106.0	0.001048	1.36965	1.3707	444.01	2069.7	2513.7	444.1	2240.9	2685.1	1.3734	5.9113	7.2847
150.00	111.4	0.001053	1.15612	1.1572	466.74	2052.9	2519.6	466.9	2226.3	2693.2	1.4330	5.7904	7.2234
175.00	116.1	0.001057	1.00248	1.0035	486.58	2037.9	2524.5	486.8	2213.4	2700.1	1.4844	5.6873	7.1717
200.00	120.2	0.001060	0.88498	0.8860	504.25	2024.7	2529.0	504.5	2201.7	2706.2	1.5295	5.5974	7.1269

225.00	124.0	0.001064	0.79229	0.7934	520.22	2012.8	2533.0	520.5	2191.1	2711.5	1.5700	5.5175	7.0874
250.00	127.4	0.001067	0.71751	0.7186	534.82	2001.9	2536.7	535.1	2181.2	2716.3	1.6066	5.4455	7.0521
275.00	130.6	0.001070	0.65602	0.6571	548.30	1991.7	2540.0	548.6	2172.1	2720.7	1.6401	5.3800	7.0201
300.00	133.5	0.001073	0.60457	0.6056	560.83	1982.1	2542.9	561.2	2163.5	2724.6	1.6710	5.3199	6.9910
325.00	136.3	0.001076	0.56082	0.5619	572.57	1973.1	2545.7	572.9	2155.4	2728.3	1.6998	5.2643	6.9641
350.00	138.9	0.001079	0.52305	0.5241	583.60	1964.6	2548.2	584.0	2147.7	2731.6	1.7266	5.2126	6.9392
375.00	141.3	0.001081	0.49007	0.4911	594.03	1956.6	2550.6	594.4	2140.3	2734.8	1.7519	5.1642	6.9161
400.00	143.6	0.001084	0.46105	0.4621	603.93	1948.9	2552.8	604.4	2133.3	2737.7	1.7757	5.1187	6.8944
425.00	145.8	0.001086	0.43534	0.4364	613.36	1941.6	2554.9	613.8	2126.6	2740.4	1.7982	5.0758	6.8740
450.00	147.9	0.001088	0.41242	0.4135	622.35	1934.5	2556.9	622.8	2120.1	2743.0	1.8196	5.0352	6.8548
475.00	149.9	0.001091	0.39188	0.3930	630.97	1927.7	2558.7	631.5	2113.9	2745.4	1.8400	4.9966	6.8366
500.00	151.8	0.001093	0.37336	0.3745	639.24	1921.2	2560.4	639.8	2107.9	2747.6	1.8595	4.9598	6.8193
550.00	155.5	0.001097	0.34129	0.3424	654.86	1908.7	2563.5	655.5	2096.4	2751.8	1.8960	4.8910	6.7871
600.00	158.8	0.001101	0.31443	0.3155	669.42	1896.9	2566.3	670.1	2085.5	2755.6	1.9298	4.8278	6.7576
650.00	162.0	0.001104	0.29151	0.2926	683.06	1885.8	2568.8	683.8	2075.2	2759.0	1.9613	4.7692	6.7305
700.00	164.9	0.001108	0.27168	0.2728	695.93	1875.3	2571.2	696.7	2065.4	2762.1	1.9907	4.7146	6.7053
750.00	167.7	0.001111	0.25439	0.2555	708.11	1865.3	2573.4	708.9	2056.0	2765.0	2.0183	4.6634	6.6817
800.00	170.4	0.001115	0.23919	0.2403	719.68	1855.7	2575.4	720.6	2047.0	2767.6	2.0445	4.6152	6.6597
850.00	172.9	0.001118	0.22572	0.2268	730.71	1846.5	2577.2	731.7	2038.4	2770.0	2.0692	4.5696	6.6388
900.00	175.3	0.001121	0.21372	0.2148	741.27	1837.7	2578.9	742.3	20300	2.772.3	2.0928	4.5264	6.6192
950.00	177.7	0.001124	0.20295	0.2041	751.39	1829.1	2580.5	752.5	2021.9	2774.3	2.1152	4.4853	6.6005
1000	179.9	0.001127	0.19322	0.1943	761.11	1820.8	2581.9	762.2	2014.0	2776.3	2.1367	4.4460	6.5827
1100	184.1	0.001133	0.17631	0.1774	779.52	1805.1	2584.6	780.8	1999.0	2779.8	2.1771	4.3725	6.5495
1200	187.9	0.001138	0.16209	0.1632	796.71	1790.2	2586.9	798.1	1984.7	2782.8	2.2145	4.3047	6.5191
1300	191.6	0.001143	0.14998	0.1511	812.87	1776.1	2589.0	814.4	1971.1	2785.4	2.2493	4.2417	6.4910
1400	195.0	0.001148	0.13956	0.1407	828.13	1762.7	2590.8	829.7	1958.0	2787.8	2.2820	4.1829	6.4649
1500	198.3	0.001153	0.13050	0.1317	842.61	1749.8	2592.4	844.3	1945.5	2789.8	2.3127	4.1277	6.4405
1600	201.4	0.001158	0.12254	0.1237	856.39	1737.3	2593.7	858.2	1933.4	2791.7	2.3418	4.0757	6.4176
1700	204.3	0.001163	0.11549	0.1167	869.56	1725.4	2595.0	871.5	1921.7	2793.3	2.3695	4.0265	6.3960
1800	207.1	0.001167	0.10918	0.1104	882.18	1713.9	2596.1	884.3	1910.4	2794.7	2.3958	3.9797	6.3755
1900	209.8	0.001172	0.10351	0.1047	894.30	1702.7	2597.0	896.5	1899.4	2795.9	2.4210	3.9350	6.3559
2000	212.4	0.001176	0.09839	0.0996	905.97	1691.9	2597.9	908.3	1888.7	2797.0	2.4450	3.8922	6.3372
2250	218.4	0.001186	0.08751	0.0887	933.41	1666.2	2599.6	936.1	1863.1	2799.2	2.5012	3.7925	6.2936
2500	223.9	0.001196	0.07873	0.0799	958.75	1642.1	2600.9	961.7	1839.0	2800.7	2.5525	3.7015	6.2540
2750	229.1	0.001206	0.07147	0.0727	982.37	1619.4	2601.8	985.7	1816.0	28017	2.5997	3.6178	6.2176
3000	233.8	0.001215	0.06538	0.0666	1004.5	1597.9	2602.4	1008.2	1794.0	2802.2	2.6437	3.5402	6.1839
3250	238.3	0.001225	0.06028	0.0615	1025.4	1577.0	2602.5	1029.4	1772.9	2802.3	2.6848	3.4676	6.1524
3500	242.5	0.001234	0.05594	0.0572	1045.3	1556.8	2602.1	1049.6	1752.6	2802.2	2.7234	3.3995	6.1229
3750	246.5	0.001242	0.05208	0.0533	1064.2	1537.6	2601.8	1068.8	1732.9	2801.7	2.7600	3.3350	6.0950
4000	250.3	0.001251	0.04864	0.0499	1082.2	1519.3	2601.5	1087.2	1713.9	2801.1	2.7947	3.2739	6.0686
5000	263.9	0.001285	0.03811	0.0394	1147.9	1451.9	2599.8	1154.3	1642.4	2796.7	2.9186	3.0548	5.9734

(continued)

TABLE C.7 (continued)

Thermodynamic Properties of Steam: Pressure Table (SI Units)

P_{sat}	T_{sat}	Specific Volume (m³/kg)			Internal Energy (kJ/kg)			Enthalpy (kJ/kg)			Entropy (kJ/kg K)		
(kPa)	**(°C)**	v_f	v_{fg}	v_g	u_f	u_{fg}	u_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
6000	275.5	0.001319	0.03109	0.0324	1205.6	1390.4	2596.0	1213.5	1576.9	2790.5	3.0251	2.8655	5.8906
7000	285.8	0.001352	0.02603	0.0274	1257.8	1333.5	2591.3	1267.2	1515.7	2782.9	3.1194	2.6965	5.8159
8000	295.0	0.001385	0.02214	0.0235	1305.9	1265.8	2571.7	1317.0	1442.9	2759.9	3.2050	2.5421	5.7471
9000	303.3	0.001418	0.01907	0.0205	1351.0	1209.3	2560.3	1363.7	1380.9	2744.7	3.2840	2.3981	5.6821
10000	311.0	0.001452	0.01658	0.0180	1393.7	1153.8	2547.5	1408.2	1319.5	2727.7	3.3580	2.2616	5.6196
11000	318.1	0.001489	0.01450	0.0160	1434.6	1098.6	2533.1	1451.0	1258.0	2709.0	3.4283	2.1305	5.5588
12000	324.7	0.001527	0.01273	0.0143	1474.0	1043.3	2517.3	1492.4	1196.0	2688.4	3.4958	2.0028	5.4986
13000	331.0	0.001568	0.01119	0.0128	1512.4	987.6	2499.9	1532.8	1133.1	2665.8	3.5611	1.8771	5.4382
14000	336.9	0.001612	0.00984	0.0114	1549.8	931.1	2480.9	1572.4	1068.8	2641.2	3.6249	1.7520	5.3769
15000	342.4	0.001661	0.00862	0.0103	1586.6	873.4	2460.0	1611.5	1002.8	2614.3	3.6877	1.6265	5.3142
16000	347.7	0.001715	0.00752	0.0092	1626.1	810.1	2436.2	1653.6	930.4	2584.0	3.7498	1.4996	5.2494
17000	352.3	0.001769	0.00660	0.0084	1660.2	750.3	2410.5	1690.3	862.5	2552.8	3.8054	1.3819	5.1872
18000	357.0	0.001839	0.00566	0.0075	1698.6	680.8	2379.4	1731.7	782.6	2514.3	3.8652	1.2481	5.1134
19000	361.4	0.001926	0.00475	0.0067	1740.1	603.6	2343.6	1776.7	693.9	2470.5	3.9249	1.1063	5.0312
20000	365.7	0.002037	0.00384	0.0059	1785.8	515.1	2300.9	1826.6	591.9	2418.5	3.9846	0.9568	4.9414
21000	369.8	0.002208	0.00281	0.0050	1839.7	401.7	2241.4	1886.0	460.8	2346.8	4.0443	0.7681	4.8124
22000	373.7	0.002623	0.00114	0.0038	1944.6	174.0	2118.6	2002.3	199.0	2201.3	4.1042	0.4563	4.5605
22090	374.4	0.00315	0.00000	0.00315	2029.6	0.0	2029.6	2099.3	0.0	2099.3	4.4298	0.0	4.4298

Source: Properties obtained from software, *STEAMCALC*, John Wiley & Sons, New York, 1983; *Introduction to Thermal and Fluid Engineering*, CRC Press, Boca Raton, FL, 2012, p. 904, Table A.4.

TABLE C.8

Thermodynamic Properties of Steam: Superheated Vapor Table (SI Units)

P (kPa)	*T* (°C)	*v* (m³/kg)	*u* (kJ/kg)	*h* (kJ/kg)	*s* (kJ/kg k)
10 (T_{sat} = 45.8°C)					
	100	17.196	2516.2	2688.1	8.4498
	150	19.513	2588.2	2783.3	8.6893
	200	21.826	2661.2	2879.5	8.9040
	250	24.136	2735.5	2976.8	9.0996
	300	26.446	2811.2	3075.6	9.2799
	350	28.754	2888.3	3175.9	9.4476
	400	31.063	2967.1	3277.7	9.6048
	450	33.371	3047.4	3381.1	9.7530
	500	35.679	3129.4	3486.2	9.8935
	550	37.987	3213.1	3593.0	10.027
	600	40.295	3298.5	3701.5	10.155
	650	42.603	3385.7	3811.7	10.278
	700	44.911	3474.5	3923.7	10.396
	750	47.219	3565.2	4037.4	10.510
	800	49.526	3657.6	4152.9	10.620
	850	51.834	3751.8	4270.2	10.727
50 (T_{sat} = 81.3°C)					
	100	3.4182	2512.0	2682.9	7.6959
	150	3.8894	2586.0	2780.5	7.9413
	200	4.3561	2659.8	2877.6	8.1583
	250	4.8206	2734.5	2975.6	8.3551
	300	5.2840	2810.5	3074.7	8.5360
	350	5.7468	2887.8	3175.1	8.7040
	400	6.2092	2966.6	3277.1	8.8614
	450	6.6715	3047.1	3380.6	9.0098
	500	7.1336	3129.1	3485.8	9.1504
	550	7.5956	3212.9	3592.6	9.2843
	600	8.0575	3298.3	3701.2	9.4123
	650	8.5193	3385.5	3811.4	9.5351
	700	8.9811	3474.4	3923.4	9.6532
	750	9.4428	3565.0	4037.2	9.7672
	800	9.9045	3657.5	4152.7	9.8775
	850	10.366	3751.7	4270.0	9.9843
100 (T_{sat} = 99.6°C)					
	100	1.6956	2506.4	2676.0	7.3610
	150	1.9363	2583.1	2776.8	7.6146
	200	2.1724	2658.1	2875.3	7.8347
	250	2.4062	2733.3	2974.0	8.0329
	300	2.6388	2809.6	3073.5	8.2146
	350	2.8708	2887.1	3174.2	8.3831
	400	3.1025	2966.1	3276.4	8.5407
	450	3.3340	3046.6	3380.0	8.6893
	500	3.5654	3128.8	3485.3	8.8300
	550	3.7966	3212.5	3592.2	8.9640
	600	4.0277	3298.0	3700.8	9.0921
	650	4.2588	3385.2	3811.1	9.2149
	700	4.4898	3474.1	3923.1	9.3331
	750	4.7208	3564.8	4036.9	9.4471
	800	4.9518	3657.3	4152.4	9.5574
	850	5.1827	3751.5	4269.8	9.6643

TABLE C.8 (continued)

Thermodynamic Properties of Steam: Superheated Vapor Table (SI Units)

P (kPa)	*T* (°C)	*v* (m³/kg)	*u* (kJ/kg)	*h* (kJ/kg)	*s* (kJ/kg k)
101.32 (T_{sat} = 100.0°C)					
	150	1.9109	2583.1	2776.7	7.6084
	200	2.1439	2658.0	2875.3	7.8286
	250	2.3747	2733.3	2973.9	8.0268
	300	2.6043	2809.6	3073.4	8.2085
	350	2.8334	2887.1	3174.2	8.3770
	400	3.0621	2966.1	3276.3	8.5347
	450	3.2905	3046.6	3380.0	8.6832
	500	3.5189	3128.7	3485.3	8.8240
	550	3.7471	3212.5	3592.2	8.9580
	600	3.9752	3298.0	3700.8	9.0860
	650	4.2033	3385.2	3811.1	9.2089
	700	4.4313	3474.1	3923.1	9.3271
	750	4.6593	3564.8	4036.9	9.4411
	800	4.8872	3657.3	4152.4	9.5513
	850	5.1152	3751.5	4269.8	9.6582
200 (T_{sat} = 120.2°C)					
	150	0.9596	2577.2	2769.1	7.2804
	200	1.0804	2654.5	2870.6	7.5072
	250	1.1989	2730.9	2970.7	7.7084
	300	1.3162	2807.8	3071.1	7.8916
	350	1.4329	2885.8	3172.4	8.0610
	400	1.5492	2965.0	3274.9	8.2192
	450	1.6653	3045.7	3378.8	8.3682
	500	1.7813	3128.0	3484.3	8.5092
	550	1.8971	3211.9	3591.3	8.6434
	600	2.0129	3297.4	3700.0	8.7716
	650	2.1286	3384.7	3810.4	8.8945
	700	2.2442	3473.7	3922.5	9.0128
	750	2.3598	3564.4	4036.4	9.1268
	800	2.4754	3656.9	4152.0	9.2372
	850	2.5909	3751.1	4269.3	9.3441
300 (T_{sat} = 133.5°C)					
	150	0.6338	2570.8	2760.9	7.0779
	200	0.7164	2650.8	2865.7	7.3122
	250	0.7965	2728.5	2967.4	7.5165
	300	0.8753	2806.1	3068.7	7.7014
	350	0.9535	2884.4	3170.5	7.8717
	400	1.0314	2963.9	3273.4	8.0305
	450	1.1091	3044.8	3377.6	8.1798
	500	1.1866	3127.2	3483.2	8.3211
	550	1.2639	3211.2	3590.4	8.4554
	600	1.3413	3296.9	3699.2	8.5838
	650	1.4185	3384.2	3809.7	8.7068
	700	1.4957	3473.2	3921.9	8.8252
	750	1.5728	3564.0	4035.8	8.9393
	800	1.6499	3656.5	4151.5	9.0497
	850	1.7270	3750.8	4268.9	9.1566

(continued)

TABLE C.8 (continued)

Thermodynamic Properties of Steam: Superheated Vapor Table (SI Units)

P (kPa)	T (°C)	v (m^3/kg)	u (kJ/kg)	h (kJ/kg)	s (kJ/kg k)
400 (T_{sat} = 143.6°C)					
	150	0.4707	2563.9	2752.2	6.9287
	200	0.5343	2647.0	2860.7	7.1712
	250	0.5952	2726.0	2964.1	7.3789
	300	0.6549	2804.3	3066.2	7.5655
	350	0.7139	2883.1	3168.6	7.7367
	400	0.7725	2962.9	3271.9	7.8961
	450	0.8309	3044.0	3376.3	8.0458
	500	0.8892	3126.5	3482.2	8.1873
	550	0.9474	3210.6	3589.5	8.3219
	600	1.0054	3296.3	3698.5	8.4504
	650	1.0635	3383.7	3809.0	8.5735
	700	1.1214	3472.7	3921.3	8.6919
	750	1.1793	3563.5	4035.3	8.8062
	800	1.2372	3656.1	4151.0	8.9166
	850	1.2951	3750.4	4268.4	9.0236
600 (T_{sat} = 158.8°C)					
	200	0.3521	2639.0	2850.2	6.9669
	250	0.3939	2720.8	2957.2	7.1819
	300	0.4344	2800.6	3061.3	7.3719
	350	0.4742	2880.3	3164.8	7.5451
	400	0.5136	2960.7	3268.9	7.7057
	450	0.5528	3042.2	3373.8	7.8562
	500	0.5919	3125.0	3480.1	7.9982
	550	0.6308	3209.3	3587.7	8.1332
	600	0.6696	3295.1	3696.9	8.2619
	650	0.7084	3382.6	3807.7	8.3853
	700	0.7471	3471.8	3920.1	8.5039
	750	0.7858	3562.7	4034.2	8.6182
	800	0.8245	3655.3	4150.0	8.7287
	850	0.8631	3749.7	4267.6	8.8358
800 (T_{sat} = 170.4°C)					
	200	0.2608	2630.4	2839.1	6.8156
	250	0.2932	2715.5	2950.1	7.0388
	300	0.3241	2796.9	3056.2	7.2326
	350	0.3544	2877.5	3161.0	7.4079
	400	0.3842	2958.5	3265.8	7.5697
	450	0.4137	3040.4	3371.4	7.7209
	500	0.4432	3123.5	3478.0	7.8635
	550	0.4725	3208.0	3586.0	7.9988
	600	0.5017	3294.0	3695.4	8.1278
	650	0.5309	3381.6	3806.3	8.2514
	700	0.5600	3470.9	3918.9	8.3702
	750	0.5891	3561.9	4033.1	8.4846
	800	0.6181	3654.6	4149.1	8.5953
	850	0.6471	3749.0	4266.7	8.7024

TABLE C.8 (continued)

Thermodynamic Properties of Steam: Superheated Vapor Table (SI Units)

P (kPa)	T (°C)	v (m^3/kg)	u (kJ/kg)	h (kJ/kg)	s (kJ/kg k)
1000 (T_{sat} = 179.9°C)					
	200	0.2059	2621.4	2827.3	6.6930
	250	0.2328	2710.0	2942.8	6.9251
	300	0.2580	2793.1	3051.1	7.1229
	350	0.2824	2874.7	3157.1	7.3003
	400	0.3065	2956.3	3262.7	7.4633
	450	0.3303	3038.5	3368.8	7.6154
	500	0.3540	3121.9	3475.9	7.7585
	550	0.3775	3206.6	3584.2	7.8942
	600	0.4010	3292.8	3693.8	8.0235
	650	0.4244	3380.6	3805.0	8.1473
	700	0.4477	3470.0	3917.7	8.2662
	750	0.4710	3561.0	4032.0	8.3808
	800	0.4943	3653.8	4148.1	8.4916
	850	0.5175	3748.3	4265.8	8.5988
1500 (T_{sat} = 198.3°C)					
	250	0.15199	2695.4	2923.4	6.7093
	300	0.16971	2783.3	3037.8	6.9183
	350	0.18654	2867.4	3147.2	7.1014
	400	0.20292	2950.6	3255.0	7.2677
	450	0.21906	3034.0	3362.5	7.4219
	500	0.23503	3118.1	3470.6	7.5664
	550	0.25089	3203.3	3579.7	7.7030
	600	0.26666	3289.9	3689.9	7.8331
	650	0.28237	3378.0	3801.5	7.9574
	700	0.29803	3467.6	3914.7	8.0767
	750	0.31364	3558.9	4029.4	8.1917
	800	0.32921	3651.8	4145.7	8.3027
	850	0.34475	3746.5	4263.6	8.4101
2000 (T_{sat} = 212.4°C)					
	250	0.11145	2679.5	2902.4	6.5451
	300	0.12550	2772.9	3023.9	6.7671
	350	0.13856	2860.0	3137.1	6.9565
	400	0.15113	2944.8	3247.1	7.1263
	450	0.16343	3029.3	3356.1	7.2826
	500	0.17556	3114.2	3465.3	7.4286
	550	0.18757	3200.0	3575.1	7.5662
	600	0.19950	3287.0	3686.0	7.6970
	650	0.21137	3375.4	3798.1	7.8218
	700	0.22318	3465.3	3911.6	7.9416
	750	0.23494	3556.8	4026.7	8.0569
	800	0.24667	3649.9	4143.2	8.1681
	850	0.25836	3744.7	4261.5	8.2758

TABLE C.8 (continued)

Thermodynamic Properties of Steam: Superheated Vapor Table (SI Units)

P (kPa)	**T (°C)**	**v (m^3/kg)**	**u (kJ/kg)**	**h (kJ/kg)**	**s (kJ/kg k)**
2500 (T_{sat} = 223.9°C)					
	250	0.08699	2662.2	2879.7	6.4076
	300	0.09893	2762.0	3009.3	6.6446
	350	0.10975	2852.2	3126.6	6.8409
	400	0.12004	2938.9	3239.1	7.0145
	450	0.13005	3024.6	3349.7	7.1730
	500	0.13987	3110.3	3459.9	7.3205
	550	0.14958	3196.6	3570.6	7.4592
	600	0.15921	3284.1	3682.1	7.5906
	650	0.16876	3372.8	3794.7	7.7161
	700	0.17827	3462.9	3908.6	7.8362
	750	0.18772	3554.6	4024.0	7.9518
	800	0.19714	3648.0	4140.8	8.0633
	850	0.20653	3742.9	4259.3	8.1712
3000 (T_{sat} = 233.8°C)					
	250	0.07055	2643.2	2854.9	6.2855
	300	0.08116	2750.6	2994.1	6.5399
	350	0.09053	2844.3	3115.9	6.7437
	400	0.09931	2932.9	3230.9	6.9213
	450	0.10779	3019.8	3343.1	7.0822
	500	0.11608	3106.3	3454.5	7.2312
	550	0.12426	3193.2	3566.0	7.3709
	600	0.13234	3281.1	3678.1	7.5031
	650	0.14036	3370.2	3791.3	7.6291
	700	0.14832	3460.6	3905.6	7.7497
	750	0.15624	3552.5	4021.2	7.8656
	800	0.16412	3646.0	4138.4	7.9774
	850	0.17197	3741.1	4257.1	8.0855
4000 (T_{sat} = 250.3°C)					
	300	0.058835	2725.8	2961.2	6.3622
	350	0.066448	2827.6	3093.4	6.5835
	400	0.073377	2920.6	3214.1	6.7699
	450	0.079959	3010.0	3329.8	6.9358
	500	0.086343	3098.2	3443.6	7.0879
	550	0.092599	3186.4	3556.8	7.2298
	600	0.098764	3275.2	3670.2	7.3636
	650	0.10486	3364.9	3784.3	7.4907
	700	0.11090	3455.9	3899.5	7.6121
	750	0.11690	3548.2	4015.8	7.7287
	800	0.12285	3642.1	4133.5	7.8411
	850	0.12878	3737.6	4252.7	7.9496

TABLE C.8 (continued)

Thermodynamic Properties of Steam: Superheated Vapor Table (SI Units)

P (kPa)	**T (°C)**	**v (m^3/kg)**	**u (kJ/kg)**	**h (kJ/kg)**	**s (kJ/kg k)**
5000 (T_{sat} = 263.9°C)					
	300	0.045302	2698.2	2924.7	6.2085
	350	0.051943	2809.9	3069.6	6.4512
	400	0.057792	2907.7	3196.6	6.6474
	450	0.063252	3000.0	3316.2	6.8188
	500	0.068495	3090.0	3432.5	6.9743
	550	0.073603	3179.5	3547.5	7.1184
	600	0.078617	3269.2	3662.3	7.2538
	650	0.083560	3359.6	3777.4	7.3820
	700	0.088447	3451.1	3893.4	7.5044
	750	0.093289	3543.9	4010.4	7.6216
	800	0.098094	3638.2	4128.7	7.7345
	850	0.102867	3734.0	4248.3	7.8435
6000 (T_{sat} = 275.6°C)					
	300	0.036146	2667.1	2884.0	6.0669
	350	0.042223	2790.9	3044.2	6.3354
	400	0.047380	2894.3	3178.6	6.5429
	450	0.052104	2989.7	3302.3	6.7202
	500	0.056592	3081.7	3421.3	6.8793
	550	0.060937	3172.5	3538.1	7.0258
	600	0.065185	3263.1	3654.2	7.1627
	650	0.069360	3354.3	3770.4	7.2922
	700	0.073479	3446.4	3887.2	7.4154
	750	0.077552	3539.6	4004.9	7.5333
	800	0.081588	3634.3	4123.8	7.6467
	850	0.085592	3730.4	4243.9	7.7562
7000 (T_{sat} = 285.8°C)					
	300	0.029459	2631.9	2838.1	5.9299
	350	0.035234	2770.6	3017.2	6.2301
	400	0.039922	2880.4	3159.8	6.4504
	450	0.044132	2979.1	3288.1	6.6343
	500	0.048087	3073.2	3409.8	6.7971
	550	0.051890	3165.4	3528.6	6.9460
	600	0.055591	3257.0	3646.1	7.0846
	650	0.059218	3348.9	3763.4	7.2153
	700	0.062788	3441.6	3881.1	7.3394
	750	0.066312	3535.3	3999.5	7.4580
	800	0.069798	3630.3	4118.9	7.5720
	850	0.073253	3726.7	4239.5	7.6818

Source: *Introduction to Thermal and Fluid Engineering*, CRC Press, Boca Raton, FL, 2012, p. 907, Table A.5.

TABLE C.9

Combustion Data for Hydrocarbons (Metric and English Units)

Hydrocarbon	Formula	Higher Heating Value (Vapor) kJ/kg	Higher Heating Value (Vapor) Btu/lbm	Theor. Air/ Fuel Ratio (by mass)	Max Flame Speed m/s	Max Flame Speed ft/s	Adiabatic Flame Temp (in Air) °C	Adiabatic Flame Temp (in Air) °F	Ignition Temp (in Air) °C	Ignition Temp (in Air) °F	Flash Point °C	Flash Point °F	Flammability Limits (in Air) (% by volume) LFL	Flammability Limits (in Air) (% by volume) UFL
Paraffins or alkanes														
Methane	CH_4	55,533	23,875	17.195	0.34	1.1	1918	3484	705	1301	Gas	Gas	5.0	15.0
Ethane	C_2H_6	51,923	22,323	15.899	0.40	1.3	1949	3540	520–630	968–1166	Gas	Gas	3.0	12.5
Propane	C_3H_8	50,402	21,669	15.246	0.40	1.3	1967	3573	466	871	Gas	Gas	2.1	10.1
n-Butane	C_4H_{10}	49,593	21,321	14.984	0.37	1.2	1973	3583	405	761	–60	–76	1.86	8.41
iso-Butane	C_4H_{10}	49,476	21,271	14.984	0.37	1.2	1973	3583	462	864	–83	–117	1.80	8.44
n-Pentane	C_5H_{12}	49,067	21,095	15.323	0.40	1.3	2232	4050	309	588	< –40	< –40	1.40	7.80
iso-Pentane	C_5H_{12}	48,955	21,047	15.323	0.37	1.2	2235	4055	420	788	< –51	< –60	1.32	9.16
Neopentane	C_5H_{12}	48,795	20,978	15.323	0.34	1.1	2238	4060	450	842	Gas	Gas	1.38	7.22
n-Hexane	C_6H_{14}	48,767	20,966	15.238	0.40	1.3	2221	4030	248	478	–22	–7	1.25	7.00
Neohexane	C_6H_{14}	48,686	20,931	15.238	0.37	1.2	2235	4055	425	797	–48	–54	1.19	7.58
n-Heptane	C_7H_{16}	48,506	20,854	15.141	0.40	1.3	2196	3985	223	433	–4	25	1.00	6.00
Triptane	C_7H_{16}	48,437	20,824	15.141	0.37	1.2	2224	4035	454	849	—	—	1.08	6.69
n-Octane	C_8H_{18}	48,371	20,796	15.093	—	—	—	—	220	428	13	56	0.95	3.20
iso-Octane	C_8H_{18}	48,311	20,770	15.093	0.34	1.1	—	—	447	837	–12	10	0.76	5.94
Olefins or alkenes														
Ethylene	C_2H_4	50,325	21,636	14.807	0.67	2.2	2343	4250	490	914	Gas	Gas	2.75	28.6
Propylene	C_3H_6	48,958	21,048	14.807	0.43	1.4	2254	4090	458	856	gas	gas	2.00	11.1
Butylene	C_4H_8	48,506	20,854	14.807	0.43	1.4	2221	4030	443	829	Gas	Gas	1.98	9.65
iso-Butene	C_4H_8	48,234	20,737	14.807	0.37	1.2	—	—	465	869	Gas	Gas	1.80	9.00
n-Pentene	C_5H_{10}	48,195	20,720	14.807	0.43	1.4	2296	4165	298	569	—	—	1.65	7.70
Aromatics														
Benzene	C_6H_6	42,296	18,184	13.297	0.40	1.3	2266	4110	562	1044	–11	12	1.35	6.65
Toluene	C_7H_8	43,033	18,501	13.503	0.37	1.2	2232	4050	536	997	4	40	1.27	6.75
p-Xylene	C_8H_{10}	43,410	18,663	13.663	—	—	2210	4010	464	867	17	63	1.00	6.00
Other hydrocarbons														
Acetylene	C_2H_2	50,014	21,502	13.297	1.40	4.6	2632	4770	406–440	763–824	Gas	Gas	2.50	81.0
Naphthalene	$C_{10}H_8$	40,247	17,303	12.932	—	—	2260	4100	515	959	79	174	0.90	5.9

Source: Adapted from Baukal, C.E. (ed.), *The John Zink Combustion Handbook*, CRC Press, Boca Raton, FL, 2001, p. 715, Table B.1.

TABLE C.10

Chemical, Physical, and Thermal Properties of Gases: Gases and Vapors, Including Fuels and Refrigerants, English and Metric Units

Common Name(s)	**Acetylene (Ethyne)**	**Butadiene**	***n*-Butane**	**Isobutane (2-Methyl Propane)**
Chemical Formula	C_2H_2	C_4H_6	C_4H_{10}	C_4H_{10}
Refrigerant Number	—	—	600	600a
Chemical and physical properties				
Molecular weight	26.04	54.09	58.12	58.12
Specific gravity, air = 1	0.90	1.87	2.07	2.07
Specific volume, ft^3/lb	14.9	7.1	6.5	6.5
Specific volume, m^3/kg	0.93	0.44	0.405	0.418
Density of liquid (at atm bp), lb/ft^3	43.0		37.5	37.2
Density of liquid (at atm bp), kg/m^3	693.0		604.0	599.0
Vapor pressure at 25°C, psia			35.4	50.4
Vapor pressure at 25°C, MN/m^2			0.0244	0.347
Viscosity (abs), lbm/ft s	6.72 × 10^{-6}		4.8 × 10^{-6}	
Viscosity (abs), centipoises[a]	0.01		0.007	
Sound velocity in gas, m/s	343	226	216	216
Thermal and thermodynamic properties				
Specific heat, c_p, Btu/lb °F or cal/g °C	0.40	0.341	0.39	0.39
Specific heat, c_p, J/kg K	1 674.0	1 427.0	1 675.0	1 630.0
Specific heat ratio, c_p/c_v	1.25	1.12	1.096	1.10
Gas constant *R*, ft lb/lb °R	59.3	28.55	26.56	26.56
Gas constant *R*, J/kg °C	319	154.0	143.0	143.0
Thermal conductivity, Btu/h ft °F	0.014		0.01	0.01
Thermal conductivity, W/m °C	0.024		0.017	0.017
Boiling point (sat 14.7 psia), °F	−103	24.1	31.2	10.8
Boiling point (sat 760 mm), °C	−75	−4.5	−0.4	−11.8
Latent heat of evap. (at bp), Btu/lb	264		165.6	157.5
Latent heat of evap. (at bp), J/kg	614,000		386,000	366,000
Freezing (melting) point, °F (1 atm)	−116	−164.0	−217.0	−229
Freezing (melting) point, °C (1 atm)	−82.2	−109.0	−138	−145
Latent heat of fusion, Btu/lb	23.		19.2	
Latent heat of fusion, J/kg	53,500		44 700	
Critical temperature, °F	97.1		306	273.0
Critical temperature, °C	36.2	171.0	152.0	134.0
Critical pressure, psia	907.0	652.0	550.0	537.0
Critical pressure, MN/m^2	6.25		3.8	3.7
Critical volume, ft^3/lb			0.070	
Critical volume, m^3/kg			0.0043	
Flammable (yes or no)	Yes	Yes	Yes	Yes
Heat of combustion, Btu/ft^3	1450	2950	3300	3300
Heat of combustion, Btu/lb	21,600	20,900	21,400	21,400
Heat of combustion, kJ/kg	50,200	48,600	49,700	49,700

(*continued*)

TABLE C.10 (continued)

Chemical, Physical, and Thermal Properties of Gases: Gases and Vapors, Including Fuels and Refrigerants, English and Metric Units

Common Name(s)	**1-Butene (Butylene)**	***cis*-2-Butene**	***trans*-2-Butene**	**Isobutene**
Chemical Formula	C_4H_8	C_4H_8	C_4H_8	C_4H_8
Refrigerant Number	—	—	—	—
Chemical and physical properties				
Molecular weight	56.108	56.108	56.108	56.108
Specific gravity, air = 1	1.94	1.94	1.94	1.94
Specific volume, ft^3/lb	6.7	6.7	6.7	6.7
Specific volume, m^3/kg	0.42	0.42	0.42	0.42
Density of liquid (at atm bp), lb/ft^3				
Density of liquid (at atm bp), kg/m^3				
Vapor pressure at 25°C, psia				
Vapor pressure at 25°C, MN/m^2				
Viscosity (abs), lbm/ft s				
Viscosity (abs), centipoises[a]				
Sound velocity in gas, m/s	222	223.0	221.0	221.0
Thermal and thermodynamic properties				
Specific heat, c_p, Btu/lb °F or cal/g °C	0.36	0.327	0.365	0.37
Specific heat, c_p, J/kg K	1 505.0	1 368.0	1 527.0	1 548.0
Specific heat ratio, c_p/c_v	1.112	1.121	1.107	1.10
Gas constant R, ft lb/lb °F	27.52			
Gas constant R, J/kg °C	148.0			
Thermal conductivity, Btu/h ft °F				
Thermal conductivity, W/m °C				
Boiling point (sat 14.7 psia), °F	20.6	38.6	33.6	19.2
Boiling point (sat 760 mm), °C	−6.3	3.7	0.9	−7.1
Latent heat of evap. (at bp), Btu/lb	167.9	178.9	174.4	169.
Latent heat of evap. (at bp), J/kg	391,000	416,000.0	406,000.0	393,000.0
Freezing (melting) point, °F (1 atm)	−301.6	−218.0	−158.0	
Freezing (melting) point, °C (1 atm)	−185.3	−138.9	−105.5	
Latent heat of fusion, Btu/lb	16.4	31.2	41.6	25.3
Latent heat of fusion, J/kg	38,100	72,600.0	96,800.0	58,800.0
Critical temperature, °F	291.0			
Critical temperature, °C	144.0	160.0	155.0	
Critical pressure, psia	621.0	595.0	610.0	
Critical pressure, MN/m^2	4.28	4.10	4.20	
Critical volume, ft^3/lb	0.068			
Critical volume, m^3/kg	0.0042			
Flammable (yes or no)	Yes	Yes	Yes	Yes
Heat of combustion, Btu/ft^3	3150	3150.0	3150.0	3150.0
Heat of combustion, Btu/lb	21,000	21,000.0	21,000.0	21,000.0
Heat of combustion, kJ/kg	48,800	48,800.0	48,800.0	48,800.0

TABLE C.10 (continued)

Chemical, Physical, and Thermal Properties of Gases: Gases and Vapors, Including Fuels and Refrigerants, English and Metric Units

Common Name(s)	**Carbon Dioxide**	**Carbon Monoxide**	**Ethane**	**Ethylene (Ethene)**
Chemical Formula	CO_2	CO	C_2H_6	C_2H_4
Refrigerant Number	**744**	**—**	**170**	**1150**
Chemical and physical properties				
Molecular weight	44.01	28.011	30.070	28.054
Specific gravity, air = 1	1.52	0.967	1.04	0.969
Specific volume, ft^3/lb	8.8	14.0	13.025	13.9
Specific volume, m^3/kg	0.55	0.874	0.815	0.87
Density of liquid (at atm bp), lb/ft^3	—		28.0	35.5
Density of liquid (at atm bp), kg/m^3	—		449.0	569.0
Vapor pressure at 25°C, psia	931.0			
Vapor pressure at 25°C, MN/m^2	6.42			
Viscosity (abs), lbm/ft s	9.4×10^{-6}	12.1×10^{-6}	$64. \times 10^{-6}$	6.72×10^{-6}
Viscosity (abs), centipoises[a]	0.014	0.018	0.095	0.010
Sound velocity in gas, m/s	270.0	352.0	316.0	331.0
Thermal and thermodynamic properties				
Specific heat, c_p, Btu/lb °F or cal/g °C	0.205	0.25	0.41	0.37
Specific heat, c_p, J/kg K	876.0	1 046.0	1 715.0	1 548.0
Specific heat ratio, c_p/c_v	1.30	1.40	1.20	1.24
Gas constant R, ft lb/lb °F	35.1	55.2	51.4	55.1
Gas constant R, J/kg °C	189.0	297.0	276.0	296.0
Thermal conductivity, Btu/h ft °F	0.01	0.014	0.010	0.010
Thermal conductivity, W/m °C	0.017	0.024	0.017	0.017
Boiling point (sat 14.7 psia), °F	−109.4[b]	−312.7	−127.0	−155.0
Boiling point (sat 760 mm), °C	−78.5	−191.5	−88.3	−103.8
Latent heat of evap. (at bp), Btu/lb	246.0	92.8	210.0	208.0
Latent heat of evap. (at bp), J/kg	572,000.0	216,000.0	488,000.0	484,000.0
Freezing (melting) point, °F (1 atm)		−337.0	−278.0	−272.0
Freezing (melting) point, °C (1 atm)		−205.0	−172.2	−169.0
Latent heat of fusion, Btu/lb	—	12.8	41.0	51.5
Latent heat of fusion, J/kg	—		95,300.0	120,000.0
Critical temperature, °F	88.0	−220.0	90.1	49.0
Critical temperature, °C	31.0	−140.0	32.2	9.5
Critical pressure, psia	1072.0	507.0	709.0	741.0
Critical pressure, MN/m^2	7.4	3.49	4.89	5.11
Critical volume, ft^3/lb		0.053	0.076	0.073
Critical volume, m^3/kg		0.0033	0.0047	0.0046
Flammable (yes or no)	No	Yes	Yes	Yes
Heat of combustion, Btu/ft^3	—	310.0		1 480.
Heat of combustion, Btu/lb	—	4340.0	22,300.0	20,600.0
Heat of combustion, kJ/kg	—	10,100.0	51,800.0	47,800.0

(*continued*)

TABLE C.10 (continued)

Chemical, Physical, and Thermal Properties of Gases: Gases and Vapors, Including Fuels and Refrigerants, English and Metric Units

Common Name(s)	**Hydrogen**	**Methane**	**Nitric Oxide**	**Nitrogen**
Chemical Formula	H_2	CH_4	NO	N_2
Refrigerant Number	**702**	**50**	**—**	**728**
Chemical and physical properties				
Molecular weight	2.016	16.044	30.006	28.013 4
Specific gravity, air = 1	0.070	0.554	1.04	0.967
Specific volume, ft^3/lb	194.0	24.2	13.05	13.98
Specific volume, m^3/kg	12.1	1.51	0.814	0.872
Density of liquid (at atm bp), lb/ft^3	4.43	26.3		50.46
Density of liquid (at atm bp), kg/m^3	71.0	421.0		808.4
Vapor pressure at 25°C, psia				
Vapor pressure at 25°C, MN/m^2				
Viscosity (abs), lbm/ft s	6.05×10^{-6}	7.39×10^{-6}	12.8×10^{-6}	12.1×10^{-6}
Viscosity (abs), centipoises[a]	0.009	0.011	0.019	0.018
Sound velocity in gas, m/s	1315.0	446.0	341.0	353.0
Thermal and thermodynamic properties				
Specific heat, c_p, Btu/lb °F or cal/g °C	3.42	0.54	0.235	0.249
Specific heat, c_p, J/kg K	14,310.0	2260.0	983.0	1040.0
Specific heat ratio, c_p/c_v	1.405	1.31	1.40	1.40
Gas constant R, ft lb/lb °F	767.0	96.0	51.5	55.2
Gas constant R, J/kg °C	4,126.0	518.0	277.0	297.0
Thermal conductivity, Btu/h ft °F	0.105	0.02	0.015	0.015
Thermal conductivity, W/m °C	0.0182	0.035	0.026	0.026
Boiling point (sat 14.7 psia), °F	−423.0	−259.0	−240.0	−320.4
Boiling point (sat 760 mm), °C	20.4 K	−434.2	−151.5	−195.8
Latent heat of evap. (at bp), Btu/lb	192.0	219.2		85.5
Latent heat of evap. (at bp), J/kg	447,000.0	510,000.0		199,000.0
Freezing (melting) point, °F (1 atm)	−434.6	−296.6	−258.0	−346.0
Freezing (melting) point, °C (1 atm)	−259.1	−182.6	−161.0	−210.0
Latent heat of fusion, Btu/lb	25.0	14.0	32.9	11.1
Latent heat of fusion, J/kg	58,000.0	32,600.0	76,500.0	25,800.0
Critical temperature, °F	−399.8	−116.0	−136.0	−232.6
Critical temperature, °C	−240.0	−82.3	−93.3	−147.0
Critical pressure, psia	189.0	673.0	945.0	493.0
Critical pressure, MN/m^2	1.30	4.64	6.52	3.40
Critical volume, ft^3/lb	0.53	0.099	0.0332	0.051
Critical volume, m^3/kg	0.033	0.0062	0.00207	0.00318
Flammable (yes or no)	Yes	Yes	No	No
Heat of combustion, Btu/ft^3	320.0	985.0	—	—
Heat of combustion, Btu/lb	62,050.0	22,900.0	—	—
Heat of combustion, kJ/kg	144,000.0		—	—

TABLE C.10 (continued)

Chemical, Physical, and Thermal Properties of Gases: Gases and Vapors, Including Fuels and Refrigerants, English and Metric Units

Common Name(s)	**Nitrous Oxide**	**Oxygen**	**Propane**	**Propylene (Propene)**
Chemical Formula	**N_2O**	**O_2**		**C_3H_6**
Refrigerant Number	**744A**	**732**	**290**	**1270**
Chemical and physical properties				
Molecular weight	44.012	31.998 8	44.097	42.08
Specific gravity, air = 1	1.52	1.105	1.52	1.45
Specific volume, ft^3/lb	8.90	12.24	8.84	9.3
Specific volume, m^3/kg	0.555	0.764	0.552	0.58
Density of liquid (at atm bp), lb/ft^3	76.6	71.27	36.2	37.5
Density of liquid (at atm bp), kg/m^3	1227.0	1142.0	580.0	601.0
Vapor pressure at 25°C, psia			135.7	166.4
Vapor pressure at 25°C, MN/m^2			0.936	1.147
Viscosity (abs), lbm/ft s	10.1×10^{-6}	13.4×10^{-6}	53.8×10^{-6}	57.1×10^{-6}
Viscosity (abs), centipoises[a]	0.015	0.020	0.080	0.085
Sound velocity in gas, m/s	268.0	329.0	253.0	261.0
Thermal and thermodynamic properties				
Specific heat, c_p, Btu/lb °F or cal/g °C	0.21	0.220	0.39	0.36
Specific heat, c_p, J/kg K	879.0	920.0	1630.0	1506.0
Specific heat ratio, c_p/c_v	1.31	1.40	1.2	1.16
Gas constant R, ft lb/lb °F	35.1	48.3	35.0	36.7
Gas constant R, J/kg °C	189.0	260.0	188.0	197.0
Thermal conductivity, Btu/h ft °F	0.010	0.015	0.010	0.010
Thermal conductivity, W/m °C	0.017	0.026	0.017	0.017
Boiling point (sat 14.7 psia), °F	−127.3	−297.3	−44.0	−54.0
Boiling point (sat 760 mm), °C	−88.5	−182.97	−42.2	−48.3
Latent heat of evap. (at bp), Btu/lb	161.8	91.7	184.0	188.2
Latent heat of evap. (at bp), J/kg	376,000.0	213,000.0	428,000.0	438,000.0
Freezing (melting) point, °F (1 atm)	−131.5	−361.1	−309.8	−301.0
Freezing (melting) point, °C (1 atm)	−90.8	−218.4	−189.9	−185.0
Latent heat of fusion, Btu/lb	63.9	5.9	19.1	
Latent heat of fusion, J/kg	149,000.0	13,700.0	44,400.0	
Critical temperature, °F	97.7	−181.5	205.0	197.0
Critical temperature, °C	36.5	−118.6	96.0	91.7
Critical pressure, psia	1052.0	726.0	618.0	668.0
Critical pressure, MN/m^2	7.25	5.01	4.26	4.61
Critical volume, ft^3/lb	0.036	0.040	0.073	0.069
Critical volume, m^3/kg	0.0022	0.0025	0.0045	0.0043
Flammable (yes or no)	No	No	Yes	Yes
Heat of combustion, Btu/ft^3	—	—	2450.0	2310.0
Heat of combustion, Btu/lb	—	—	21,660.0	21,500.0
Heat of combustion, kJ/kg	—	—	50,340.0	50,000.0

Source: *The CRC Press Handbook of Thermal Engineering*, CRC Press, Boca Raton, FL, 2000; *JZ Handbook*, 1st edn., Table B.4, p. 719.

Note: The properties of pure gases are given at 25°C (77°F, 298 K) and atmospheric pressure (except as stated).

[a] For $N s/m^2$ divide by 1000.

TABLE C.11

Burning Velocities of Various Fuels

	$\phi = 0.7$	$\phi = 0.8$	$\phi = 0.9$	$\phi = 1.0$	$\phi = 1.1$	$\phi = 1.2$	$\phi = 1.3$	$\phi = 1.4$	S_{max}	ϕ at S_{max}
Saturated hydrocarbons										
Ethane	30.6	36.0	40.6	44.5	47.3	47.3	44.4	37.4	47.6	1.14
Propane			42.3	45.6	46.2	42.4	34.3		46.4	1.06
n-Butane		38.0	42.6	44.8	44.2	41.2	34.4	25.0	44.9	1.03
Methane		30.0	38.3	43.4	44.7	39.8	31.2		44.8	1.08
n-Pentane		35.0	40.5	42.7	42.7	39.3	33.9		43.0	1.05
n-Heptane		37.0	39.8	42.2	42.0	35.5	29.4		42.8	1.05
2,2,4-Trimethylpentane		37.5	40.2	41.0	37.2	31.0	23.5		41.0	0.98
2,2,3-Trimethylpentane		37.8	39.5	40.1	39.5	36.2			40.1	1.00
2,2-Dimethylbutane		33.5	38.3	39.9	37.0	33.5			40.0	0.98
Isopentane		33.0	37.6	39.8	38.4	33.4	24.8		39.9	1.01
2,2-Dimethylpropane			31.0	34.8	36.0	35.2	33.5	31.2	36.0	1.10
Unsaturated hydrocarbons										
Acetylene		107	130	144	151	154	154	152	155	1.25
Ethylene	37.0	50.0	60.0	68.0	73.0	72.0	66.5	60.0	73.5	1.13
Propyne		62.0	66.6	70.2	72.2	71.2	61.0		72.5	1.14
1,3-Butadiene			42.6	49.6	55.0	57.0	56.9	55.4	57.2	1.23
n-1-Heptyne		46.8	50.7	52.3	50.9	47.4	41.6		52.3	1.00
Propylene			48.4	51.2	49.9	46.4	40.8		51.2	1.00
n-2-Pentene		35.1	42.6	47.8	46.9	42.6	34.9		48.0	1.03
2,2,4-Trimethyl-3-pentene		34.6	41.3	42.2	37.4	33.0			42.5	0.98
Substituted alkyls										
Methanol		34.5	42.0	48.0	50.2	47.5	44.4	42.2	50.4	1.08
Isopropyl alcohol		34.4	39.2	41.3	40.6	38.2	36.0	34.2	41.4	1.04
Triethylamine		32.5	36.7	38.5	38.7	36.2	28.6		38.8	1 06
n-Butyl chloride	24.0	30.7	33.8	34.5	32.5	26.9	20.0		34.5	1.00
Allyl chloride	30.6	33.0	33.7	32.4	29.6				33.8	0.89
Isopropyl mercaptan		30.0	33.5	33.0	26.6				33.8	0.44
Ethylamine		28.7	31.4	32.4	31.8	29.4	25.3		32.4	1.00
Isopropylamine		27.0	29.5	30.6	29.8	27.7			30.6	1 01
n-Propyl chloride		24.7	28.3	27.5	24.1				28.5	0.93
Isopropyl chloride		24.8	27.0	27.4	25.3				27.6	0.97
n-Propyl bromide	No ignition									
Silanes										
Tetramethylsilane	39.5	49.5	57.3	58.2	57.7	54.5	47.5		58.2	1.01
Trimethylethoxysilane	34.7	41.0	47.4	50.3	46.5	41.0	35.0		50.3	1.00
Aldehydes										
Acrolein	47.0	58.0	66.6	65.9	56.5				67.2	0.95
Propionaldehyde		37.5	44.3	49.0	49.5	46.0	41.6	37.2	50.0	1.06
Acetaldehyde		26.6	35.0	41.4	41.4	36.0	30.0		42.2	1.05
Ketones										
Acetone		40.4	44.2	42.6	38.2				44.4	0.93
Methyl ethyl ketone		36.0	42.0	43.3	41.5	37.7	33.2		43.4	0.99
Esters										
Vinyl acetate	29.0	36.6	39.8	41.4	42.1	41.6	35.2		42.2	1.13
Ethyl acetate		30.7	35.2	37.0	35.6	30.0			37.0	1.00
Ethers										
Dimethyl ether		44.8	47.6	48.4	47.5	45.4	42.6		48.6	0.99
Diethyl ether	30.6	37.0	43.4	48.0	47.6	40.4	32.0		48.2	1.05
Dimethoxymethane	32.5	38.2	43.2	46.6	48.0	46.6	43.3		48.0	1.10
Diisopropyl ether		30.7	35.5	38.3	38.6	36.0	31.2		38.9	1.06

TABLE C.11 (continued)

Burning Velocities of Various Fuels

	$\phi = 0.7$	$\phi = 0.8$	$\phi = 0.9$	$\phi = 1.0$	$\phi = 1.1$	$\phi = 1.2$	$\phi = 1.3$	$\phi = 1.4$	S_{max}	ϕ at S_{max}
Thio ethers										
Dimethyl sulfide		29.9	31.9	33.0	30.1	24.8			33.0	1.00
Peroxides										
Di-*t*-butyl peroxide		41.0	46.8	50.0	49.6	46.5	42.0	35.5	50.4	1.04
Aromatic compounds										
Furan	48.0	55.0	60.0	62.5	62.4	60.0			62.9	1.05
Benzene		39.4	45.6	47.6	44.8	40.2	35.6		47.6	1.00
Thiophane	33.8	37.4	40.6	43.0	42.2	37.2	24.6		43.2	1.03
Cyclic compounds										
Ethylene oxide	57.2	70.7	83.0	88.8	89.5	87.2	81.0	73.0	89.5	1.07
Butadiene monoxide		36.6	47.4	57.8	64.0	66.9	66.8	64.5	67.1	1.24
Propylene oxide	41.6	53.3	62.6	66.5	66.4	62.5	53.8		67.0	1.05
Dihydropyran	39.0	45.7	51.0	54.5	55.6	52.6	44.3	32.0	55.7	1.08
Cyclopropane		40.6	49.0	54.2	55.6	53.5	44.0		55.6	1.10
Tetrahydropyran	44.8	51.0	53.6	51.5	42.3				53.7	0.93
Cyclic compounds										
Tetrahydrofuran			43.2	48.0	50.8	51.6	49.2	44.0	51.6	1.19
Cyclopentadiene	36.0	41.8	45.7	47.2	45.5	40.6	32.0		47.2	1.00
Ethylenimine		37.6	43.4	46.0	45.8	43.4	38.9		46.4	1.04
Cyclopentane	31.0	38.4	43.2	45.3	44.6	41.0	34.0		45.4	1.03
Cyclohexane			41.3	43.5	43.9	38.0			44.0	1.08
Inorganic compounds										
Hydrogen	102	120	145	170	204	245	213	290	325	1.80
Carbon disulfide	50.6	58.0	59.4	58.8	57.0	55.0	52.8	51.6	59.4	0.91
Carbon monoxide					28.5	32.0	34.8	38.0	52.0	2.05
Hydrogen sulfide	34.8	39.2	40.9	39.1	32.3				40.9	0.90
Propylene oxide	74.0	86.2	93.0	96.6	97.8	94.0	84.0	71.5	97.9	1.09
Hydrazine	87.3	90.5	93.2	94.3	93.0	90.7	87.4	83.7	94.4	0.98
Furfural	62.0	73.0	83.3	87.0	87.0	84.0	77.0	65.5	87.3	1.05
Ethyl nitrate	70.2	77.3	84.0	86.4	83.0	72.3			86.4	1.00
Butadiene monoxide	51.4	57.0	64.5	73.0	79.3	81.0	80.4	76.7	81.1	1.23
Carbon disulfide	64.0	72.5	76.8	78.4	75.5	71.0	66.0	62.2	78.4	1.00
n-Butyl ether		67.0	72.6	70.3	65.0				72.7	0.91
Methanol	50.0	58.5	66.9	71.2	72.0	66.4	58.0	48.8	72.2	1.08
Diethyl cellosolve	49.5	56.0	63.0	69.0	69.7	65.2			70.4	1.05
Cyclohexene										
Monoxide	54.5	59.0	63.5	67.7	70.0	64.0			70.0	1.10
Epichlorohydrin	53.0	59.5	65.0	68.6	70.0	66.0	58.2		70.0	1.10
n-Pentane		50.0	55.0	61.0	62.0	57.0	49.3	42.4	62.9	1.05
n-Propyl alcohol	49.0	56.6	62.0	64.6	63.0	50.0	37.4		64.8	1.03
n-Heptane	41.5	50.0	58.5	63.8	59.5	53.8	46.2	38.8	63.8	1.00
Ethyl nitrite	54.0	58.8	62.6	63.5	59.0	49.5	42.0	36.7	63.5	1.00
Pinene	48.5	58.3	62.5	62.1	56.6	50.0			63.0	0.95
Nitroethane	51.5	57.8	61.4	57.2	46.0	28.0			61.4	0.92
Isooctane		50.2	56.8	57.8	53.3	50.5			58.2	0.98
Pyrrole		52.0	55.6	56.6	56.1	52.8	48.0	43.1	56.7	1.00
Aniline		41.5	45.4	46.6	42.9	37.7	32.0		46.8	0.98
Dimethyl formamide		40.0	43.6	45.8	45.5	40.7	36.7		46.1	1.04

Source: The compilation of laminar flame speed data given in tables is from Gibbs and Calcote, *J. Chem. Eng. Data*, 4, 2226, 1959; *Combustion Science and Engineering*, CRC Press, Boca Raton, FL, 2012, p. 1057, Table A.39D.

Notes: $T = 25°C$ (air–fuel temperature); $P = 1$ atm (0.31 mol % H_2O in air); burning velocity S as a function of equivalence ratio ϕ in cm/s. The data are for premixed fuel–air mixtures at 100°C and 1 atm pressure; 0.31 mol % H_2O in air; burning velocity S as a function of ϕ in cm/s.

Appendix D: Properties of Solids

TABLE D.1

Thermal Properties of Selected Metallic Elements at 293 K (20°C) or 528°R (65°F)

Element	Specific Gravity	Specific Heat, c_p		Thermal Conductivity, k		Diffusivity, α		Melting Temperature	
		J/(kg K)	BTU/(lbm °R)	W/(m K)	BTU/(h ft °R)	$m^2/s \times 10^6$	$ft^2/s \times 10^3$	K	°R
Aluminum	2.702	896	0.214	236	136	97.5	1.05	933	1680
Beryllium	1.850	1750	0.418	205	118	63.3	0.681	1550	2790
Chromium	7.160	440	0.105	91.4	52.8	29.0	0.312	2118	3812
Copper	8.933	383	0.0915	399	231	116.6	1.26	1356	2441
Gold	19.300	129	0.0308	316	183	126.9	1.37	1336	2405
Iron	7.870	452	0.108	31.1	18.0	22.8	0.245	1810	3258
Lead	11.340	129	0.0308	35.3	20.4	24.1	0.259	601	1082
Magnesium	1.740	1017	0.243	156	90.1	88.2	0.949	923	1661
Manganese	7.290	486	0.116	7.78	4.50	2.2	0.0236	1517	2731
Molybdenum	10.240	251	0.0600	138	79.7	53.7	0.578	2883	5189
Nickel	8.900	446	0.107	91	52.6	22.9	0.246	1726	3107
Platinum	21.450	133	0.0318	71.4	41.2	25.0	0.269	2042	3676
Potassium	0.860	741	0.177	103	59.6	161.6	1.74	337	607
Silicon	2.330	703	0.168	153	88.4	93.4	1.01	1685	3033
Silver	10.500	234	0.0559	427	247	173.8	1.87	1234	2221
Tin	5.750	227	0.0542	67.0	38.7	51.3	0.552	505	909
Titanium	4.500	611	0.146	22.0	12.7	8.0	0.0861	1953	3515
Tungsten	19.300	134	0.0320	179	103	69.2	0.745	3653	6575
Uranium	19.070	113	0.0270	27.4	15.8	12.7	0.137	1407	2533
Vanadium	6.100	502	0.120	31.4	18.1	10.3	0.111	2192	3946
Zinc	7.140	385	0.0920	121	69.9	44.0	0.474	693	1247

Source: Janna, W.S., *Engineering Heat Transfer*, 2nd edn., CRC Press, Boca Raton, FL, 2000, p. 643, Table B.1.

Notes: Density = ρ = specific gravity × 62.4 lbm/ft^3 = specific gravity × 1000 kg/m^3.
Diffusivity = α; for aluminum, α m^2/s × 106 = 97.5; so $\alpha = 97.5 \times 10^{-6}$ m^2/s.
Also, $\alpha = k/\rho c_p$.

TABLE D.2

Thermal Properties of Selected Alloys

			Specific Heat, c_p		Thermal Conductivity, k		Diffusivity, α		Coeff. of Linear Expansion		Approximate Melting point	
Alloy	**Composition**	**Specific Gravity**	**J/(kg K)**	**BTU/ (lbm °R)**	**W/(m K)**	**BTU/(h ft °R)**	**m²/s × 10⁵**	**ft²/s × 10⁴**	**μ m/m K**	**μ in./ in. °F**	**°C**	**°F**
Aluminum												
Aluminum alloy 3003, rolled	ASTM B221	2.73			155.7	90			23.2	12.9	649	1200
Aluminum alloy 2017, annealed	ASTM B221	2.8			164.4	95			22.9	12.7	641	1185
Aluminum alloy 380	ASTM SC84B	2.7			96.9	56			20.9	11.6	566	1050
Duralumin	95 Al, 5 Cu	2.787	833	0.199	164	94.7	6.676	7.187				
Silumin	87 Al, 13 Si	2.659	871	0.208	164	94.7	7.099	7.642				
Copper	ASTM B152, B124, B133, B1, B2, B3	8.91			389.3	225			16.7	9.3	1082	1980
Red brass (cast)	ASTM B30, No. 4A	8.7			72.7	42			18.0	10.0	996	1825
Yellow brass (high brass)	ASTM B36, B134, B135	8.47			119.4	69			18.9	10.5	932	1710
Aluminum bronze	ASTM B169, Alloy A; ASTM B124, B150	7.8			70.9	41			16.6	9.2	1038	1900
Beryllium copper 25	ASTM B194	8.25			12.1	7			16.7	9.3	927	1700
A-bronze	95 Cu, 5 Al	8.666	410	0.0979	83	47.9	2.330	2.508				
Bronze	75 Cu, 25 Sn	8.666	343	0.0819	26	15.0	0.859	0.925				
Red brass	85 Cu, 9 Sn, 6 Zn	8.714	385	0.0920	61	35.2	1.804	1.942				
Brass	70 Cu, 30 Zn	8.522	385	0.0920	111	64.1	3.412	3.673				
German silver	62 Cu, 15 Ni, 22 Zn	8.618	394	0.0941	24.9	14.4	0.733	0.789				
Constantan	60 Cu, 40 Ni	8.922	410	0.0979	22.7	13.1	0.612	0.659				
Cupronickel	30%	8.95			29.4	17			15.3	8.5	1227	2240
Cupronickel	55-45 (Constantan)	8.9			22.5	13			14.6	8.1	1260	2300
Iron												
Ingot iron		7.86			72.7	42			12.2	6.8	1538	2800
Cast gray iron	ASTM A48-48, Class 25	7.2			45.0	26			12.1	6.7	1177	2150
Malleable iron	ASTM A47	7.32							11.9	6.6	1232	2250
Ductile cast iron	ASTM A339, A395	7.2			32.9	19			13.5	7.5	1149	2100
Ni-resist cast iron	type 2	7.3			39.8	23			17.3	9.6	1232	2250
Cast iron	4 C	7.272	420	0.100	52	30.0	1.702	1.832				
Wrought iron	0.5 CH	7.849	460	0.110	59	34.1	1.626	1.750				

Steel												
Plain carbon steel	AISI-SAE 1020	7.86			51.9	30			12.1	6.7	1516	2760
Carbon steel	1 C	7.801	473	0.113	43	24.8	1.172	1.262				
	1.5 C	7.753	486	0.113	36	20.8	0.970	1.040				
Chrome steel	1 Cr	7.865	460	0.110	61	35.2	1.665	1.792				
	5 Cr	7.833	460	0.110	40	23.1	1.110	1.195				
	10 Cr	7.785	460	0.110	31	17.9	0.867	0.933				
Chrome-nickel steel	15 Cr, 10 Ni	7.865	460	0.110	19	11.0	0.526	0.577				
	20 Cr, 15 Ni	7.833	460	0.110	15.1	8.72	0.415	0.447				
Nickel steel	10 Ni	7.945	460	0.110	26	15.0	0.720	0.775				
	20 Ni	7.993	460	0.110	19	11.0	0.526	0.566				
	40 Ni	8.169	460	0.110	10	5.78	0.279	0.300				
	60 Ni	8.378	460	0.110	19	11.0	0.493	0.531				
Nickel-chrome steel	80 Ni, 15 C	8.522	460	0.110	17	9.82	0.444	0.478				
	40 Ni, 15 C	8.073	460	0.110	11.6	6.70	0.305	0.328				
Manganese steel	1 Mn	7.865	460	0.110	50	28.9	1.388	1.494				
	5 Mn	7.849	460	0.110	22	12.7	0.637	0.686				
Silicon steel	1 Si	7.769	460	0.110	42	24.3	1.164	1.164				
	5 Si	7.417	460	0.110	19	11.0	0.555	0.597				
Stainless steel	Type 304	8.02	461	0.110	14.4	8.32	0.387	0.417	17.3	9.6	1427	2600
	Type 347	7.97	461	0.110	14.3	8.26	0.387	0.417				
Tungsten steel	1 W	7.913	448	0.107	66	31.1	1.858	2.000				
	5 W	8.073	435	0.104	54	31.2	1.525	1.642				
Other												
Chemical lead		11.35			34.6	20			29.5	16.4	327	621
Antimonial lead (hard lead)		10.9			29.4	17			27.2	15.1	290	554
Magnesium alloy	AZ31B	1.77			77.9	45			26.1	14.5	627	1160
Nickel	ASTM B160, B161, B162	8.89			60.6	35			11.9	6.6	1441	2625
Nickel silver 18% alloy A (wrought)	ASTM B122, No. 2	8.8			32.9	19			16.2	9.0	1110	2030
Commercial titanium		5			17.3	10			8.8	4.9	1816	3300
Zinc	ASTM B69	7.14			107.3	62			32.4	18.0	418	785
Zirconium, commercial		6.5			17.3	10			5.2	2.9	1843	3350
Cast 28-7 alloy (HD)	ASTM A297-63T	7.6			2.6	1.5			16.6	9.2	1482	2700
Hastelloy	C	3.94			8.7	5			11.3	6.3	1288	2350
Inconel	X, annealed	8.25			15.6	9			12.1	6.7	1399	2550
Haynes	alloy 25 (L605)	9.15			9.5	5.5			13.7	7.6	1371	2500
Stellite												
K Monel		8.47			19.0	11			13.3	7.4	1349	2460
Solder	50-50	8.89			45.0	26			23.6	13.1	216	420

Source: Kreith, F., *The CRC Handbook of Mechanical Engineering*, CRC Press, Boca Raton, FL, 1998, p. C-56, Table C.6; Janna, W.S., *Engineering Heat Transfer*, 2nd edn., CRC Press, Boca Raton, FL, 2000, p. 644, Table B.2.

TABLE D.3

Thermal Properties of Selected Building Materials and Insulations at 293 K (20°C) or 528°R (65°F)

Material	**Specific Gravity**	**Specific Heat, c_p**		**Thermal Conductivity, k**		**Diffusivity, α**	
		J/(kg K)	**BTU/(lbm °R)**	**w/(m K)**	**BTU/(h ft °R)**	**$m^2/s \times 10^5$**	**$ft^2/s \times 10^6$**
Asbestos	0.383	816	0.195	0.113	0.0653	0.036	3.88
Asphalt	2.120			0.698	0.403		
Bakelite	1.270			0.233	0.135		
Brick							
Carborundum (50%SiC)	2.200			5.82	3.36		
Common	1.800	840	0.201	0.38–0.52	0.22–0.30	0.028–0.034	3.0–3.66
Magnesite (50%MgO)	2.000			2.68	1.55		
Masonry	1.700	837	0.200	0.658	0.38	0.046	5.0
Silica (95%SiO_2)	1.900			1.07	0.618		
Cardboard				0.14–0.35	0.08–0.2		
Cement (hard)				1.047	0.605		
Clay (48.7%moist)	1.545	880	0.210	1.26	0.728	0.101	10.9
Coal (anthracite)	1.370	1260	0.301	0.238	0.137	0.013–0.015	1.4–1.6
Concrete(dry)	0.500	837	0.200	0.128	0.074	0.049	5.3
Cork board	0.150	1880	0.449	0.042	0.0243	0.015–0.044	1.6–4.7
Cork (expanded)	0.120			0.036	0.0208		
Earth (diatomaceous)	0.466	879	0.210	0.126	0.072	0.031	3.3
Earth (clay with 28% moist)	1.500			1.51	0.872		
Earth (sandy with 8% moist)	1.500			1.05	0.607		
Glass fiber	0.220			0.035	0.02		
Glass (window pane)	2.800	800	0.191	0.81	0.47	0.034	3.66
Glass (wool)	0.200	670	0.160	0.040	0.023	0.028	3.0
Granite	2.750			3.0	1.73		
Ice at 0°C	0.913	1830	0.437	2.22	1.28	0.124	13.3
Kapok	0.025			0.035	0.02		
Linoleum	0.535			0.081	0.047		
Mica	2.900			0.523	0.302		
Pine bark	0.342			0.080	0.046		
Plaster	1.800			0.814	0.47		

Source: Janna, W.S., *Engineering Heat Transfer*, 2nd edn., CRC Press, Boca Raton, FL, 2000, p. 645, Table B.3.

Notes: Density = ρ = specify gravity × 62.4 lbm/ft^3 = specific gravity × 1000 kg/m^3.
Diffusivity = α; for asbestos, $\alpha \times 10^3 = 0.036\,m^2/s$; so $\alpha = 0.036 \times 10^{-3}\,m^2/s$ also, $\alpha = k/\rho\, c_p$.

Appendix E: Trademark Disclaimer

At the time of first publication of this Volume 3 of the *John Zink Hamworthy Combustion Handbook*, the following trademarks, which may or may not be mentioned in this book, were owned as indicated:

A.C.® is a registered trademark of Coen Company, Inc. in the U.S.
AEROSIM® is a registered trademark of John Zink Company LLC in Canada.
AEROSWIRL® is a registered trademark of John Zink Company LLC in Canada.
AIROIL-FLAREGAS® is a registered trademark of Hamworthy Combustion GMBH in Germany, and the WIPO.
AMOXSAFE® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in China, WIPO, Japan, Korea, U.K., and the U.S.
APR® is a registered trademark of Hamworthy Peabody Combustion, Inc. in the U.S.
AXIFLO® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.K.
AZDAIR® is a registered trademark of John Zink Company LLC in the U.S.
BI-MIX® is a registered trademark of John Zink Company LLC in the U.S.
CEA® & Design is a registered trademark of John Zink Company LLC in Canada.
COEN® is a registered trademark of Coen Company, Inc. in CTM, India, Japan, Mexico, U.S.
COEN & DESIGN is a registered trademark of Coen Company, Inc. in China, CTM, France, Mexico, and the U.S.
CONDOX® is a registered trademark of John Zink KEU GmbH in Germany.
COOLstar® is a registered trademark of John Zink Company LLC in Brazil, China, CTM, India, Japan Korea, Mexico, and the U.S.
DEEPSTAR® is a registered trademark of John Zink Company LLC in the U.S.
DFL® is a registered trademark of Hamworthy Peabody Combustion, Inc. in the U.S. and Canada.
DYNASWIRL® is a registered trademark of John Zink Company LLC in Canada.
ECOJET® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.S., Canada, and the CTM.
ECOPHASE/ECOPHASE INDUSTRIAL/ECOPHASE MARINE® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.K.
ECOPHASE VOC/ECOPHASE VOC RECOVERY® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.K.
ENVIROJET® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the CTM.
ENVIROJET® is a registered trademark of Hamworthy Peabody Combustion, Inc. in the U.S.
ENVIROKING® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in International, and the U.K.
ENVIROTHERM® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.K.
ESP® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.S., CTM, and the WIPO.
FRIGOSOLVER® is a registered trademark of John Zink KEU GmbH in Germany.
FYR-COMPAK® is a registered trademark of Coen Company, Inc. in the U.S.
HALO® is a registered trademark of John Zink Company LLC in the U.S.
HAMWORTHY is a registered trademark of Hamworthy Combustion Engineering, Ltd. in Australia, Brazil, China, CTM, India, WIPO, Japan, U.K., and the U.S.
HAMWORTHY AIROIL PEABODY LOGO® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in Korea.
HAMWORTHY COMBUSTION LOGO® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in Australia, China, CTM, WIPO, Japan, U.K., and the U.S.
HAMWORTHY ELECTROTEC® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in CTM, and the U.K.
HAMWORTHY FLARESENSE® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the CTM.
HAMWORTHY PEABODY COMBUSTION® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.S.
HAMWORTHY SMARTPILOT® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.S. and the CTM.
HAMWORTHY-COMBUSTION® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in Australia, China, CTM, WIPO, Japan, Korea, Russian Federation, Singapore, U.K., and the U.S.
HERO® is a registered trademark of John Zink Company LLC in the U.S.

HiTurb® is a registered trademark of John Zink KEU GmbH in Germany.

HXGL® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.K.

INDAIR® is a registered trademark of John Zink Company LLC in Canada, Denmark, Germany, U.K., and the U.S.

InstaFire® is a registered trademark of John Zink Company LLC in the U.S.

ISCAN® is a registered trademark of Coen Company, Inc. in the U.S.

JOHN ZINK COMPANY® is a registered trademark of John Zink Company LLC in Chile.

JOHN ZINK W/DESIGN is a registered trademark of John Zink Company LLC in Benelux.

JOHN ZINK® is a registered trademark of John Zink Company LLC in Argentina, Australia, Benelux, Brazil, Canada, China, Columbia, France, Germany, India, Indonesia, Italy, Japan, Korea, Mexico, New Zealand, Philippines, Singapore, Spain, Switzerland, Taiwan, Thailand, U.K., U.S., Venezuela, and Vietnam.

JZ JOHN ZINK COMPANY is a registered trademark of John Zink Company LLC in Japan.

JZ W/DESIGN is a registered trademark of John Zink Company LLC in Benelux.

JZ® is a registered trademark of John Zink Company LLC in Argentina, Australia, Brazil, Chile, Columbia, India, Korea, Malaysia, Mexico, Philippines, Singapore, Taiwan, U.K., U.S., and Venezuela.

(Kaldair) K DESIGN® is a registered trademark of John Zink Company LLC in Austria, Benelux, Denmark, Finland, France, Germany, India, Japan, Korea, Norway, Portugal, Singapore, Spain, Sweden, Switzerland, Taiwan, Turkey, U.K., and the U.S.

K JOHN ZINK is a registered trademark of John Zink Company LLC in the U.S.

KALDAIR® is a registered trademark of John Zink Company LLC in Canada, Denmark, Germany, Indonesia, Norway, U.K., and the U.S.

KEP® is a registered trademark of John Zink Company LLC in the U.S.

KJZ® is a registered trademark of John Zink Company LLC in the U.S.

LISCOTHERM® is a registered trademark of John Zink KEU GmbH in Germany.

MARDAIR® is a registered trademark of John Zink Company LLC in Canada, Denmark, Germany, Norway, and the U.K.

NO$_x$IDIZER® is a registered trademark of John Zink Company LLC in France, Japan, U.K., and the U.S.

PEABODY® is a registered trademark of Hamworthy Peabody Combustion, Inc. in Austria, Benelux, Brazil, Canada, Denmark, Finland, France, Italy, Norway, Switzerland, U.K., U.S.

PEABODY® is a registered trademark of Hamworthy Combustion GmbH in Germany and the WIPO.

PEABODY & Design (Rising Sun) is a registered trademark of Hamworthy Peabody Combustion, Inc. in Canada, Mexico, U.K., and the U.S.

PECS® is a registered trademark of John Zink Company LLC in the U.S.

PrimEx® is a registered trademark of John Zink KEU GmbH in Germany.

QLN® is a registered trademark of Coen Company, Inc. in the U.S.

QLN Stylized® is a registered trademark of Coen Company, Inc. in the U.S.

QUANTUM LOW NOX® is a registered trademark of Coen Company, Inc. in the U.S.

RADOL® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.K.

RIMFIRE® is a registered trademark of John Zink Company LLC in the U.S.

SMARTSCAN® is a registered trademark of Hamworthy Peabody Combustion, Inc. in the U.S.

SMARTSPARK® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.S., Canada, WIPO, and the U.K.

SMARTVIEW® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.S., CTM, and the WIPO.

Sound*PROOF*® is a registered trademark of John Zink Company LLC in the U.S.

STABELITE® is a registered trademark of John Zink Company LLC in Canada.

STEAMIZER® is a registered trademark of John Zink Company LLC in the U.S.

STEDAIR® is a registered trademark of John Zink Company LLC in Canada, Denmark, Germany, Norway, and the U.K.

THERMAL OXIDIZER® is a registered trademark of John Zink Company LLC in Germany.

THERMULITE® is a registered trademark of John Zink Company LLC in Argentina, Brazil, Canada, China, CTM, India, Japan, Korea, Mexico, Saudi Arabia, Singapore, Taiwan, U.S., and Venezuela.

TODD® is a registered trademark of John Zink Company LLC in the U.S., Canada, and the CTM.

TODD W/DESIGN® is a registered trademark of John Zink Company LLC in the U.S.

TOPDOWNTOP® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the CTM.

TURBOGRID® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the CTM.

WALFIRE® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.S., CTM, WIPO, and the U.K.

WALRAD® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.S.

XCCOM® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.K.

XCVIEW® is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.K.

ZINK® is a registered trademark of John Zink Company LLC in Argentina, Australia, Brazil, Chile, China, Columbia, India, Indonesia, Japan, Korea, Malaysia, New Zealand, Philippines, Singapore, Taiwan, Thailand, U.K., U.S., Venezuela, and Vietnam.

ZINK JAPAN (Katakana Characters)® is a registered trademark of John Zink Company LLC in Japan.

ZINKCO JAPAN® is a registered trademark of John Zink Company LLC in Japan.

ZINKCO JAPAN (Katakana Characters)® is a registered trademark of John Zink Company LLC in Japan.

ZINKCO® is a registered trademark of John Zink Company LLC in Japan and Taiwan.

ZTOF® is a registered trademark of John Zink Company LLC in Benelux, CTM, France, Germany, Italy, Japan, U.K., and the U.S.

ZULE® is a registered trademark of John Zink Company LLC in the U.S.

ECOGEN® STYLIZED is a registered trademark of Hamworthy Combustion Engineering, Ltd. in the U.S.

LINFLO® is a registered trademark of John Zink Company LLC in Canada.

ADAB™, ADCON™, AG-HE™, AIRRESTOR™, CONDAIR™,COOLburn™,COOLflow™,COOLflow™, COOLfuel™, COOLfuel™, COOLmix™, COOLteam™, COOLtechnologies™, COOLtech™, DRAGONstar™, DSMG™, EEP-500™, eMISSIONS™, HAWAstar™ HYDRA™, INFURNOx™, JOHN ZINK INSTITUTE℠, KEC™, KE-2™, KEU™, KMI™, LDRW™, LHLB™, LHTS™, LH™, LPMF™, LPMW™, LRGO™, LS™, LSV™, MAC Torch™, OWB™, PilotEye2000™, PMS™, PXMR™,QMR™,QS™,RMB™,SFFG™,SFFR™,SFG™, SFG™, SKEC™, Slide-N-Lock™, SOLAR™, Steamizer XP™, TANGENT™, TEMPURGE™, VaporEqualizer™, VaporWatch™, VARIFLAME II™, VARIFLAME™, VAVACS™, WindPROOF™, XP™, ZEF™, ZEUS™, ZNIX™, ZOOM™, Z-Seal™, and Z-STAR™ are trademarks of John Zink Company LLC.

CPF™, C-RMB™, DAF™, DAZ™, Delta Power™, Delta-NOx ULN™, Delta-NOx™, DLN, D-Register™, D-RMB™, Dynaswirl-LN™, Fyr-Ball™, Fyr-Bolt™, FYR-LOGIX™, Fyr-Monitor™, Fyr-Storm™, HEEI™, iScan2™ are trademarks of Coen Company, Inc.

ProLine™, QLN ULN™, QLN II™, and QL™ are trademarks of Coen Company, Inc.

VOCONOMISER™ is a trademark of Hamworthy Combustion Engineering, Ltd.

Index

Combustion

JOHN ZINK
HAMWORTHY
COMBUSTION

The John Zink Hamworthy Combustion Handbook, Second Edition: Volume 3 — Applications

Edited by Charles E. Baukal, Jr.,
John Zink Company LLC, Tulsa, Oklahoma, USA

Despite the length of time it has been around, its importance, and vast amounts of research, combustion is still far from being completely understood. Issues regarding the environment, cost, and fuel consumption add further complexity, particularly in the process and power generation industries. Dedicated to advancing the art and science of industrial combustion, **The John Zink Hamworthy Combustion Handbook, Second Edition: Volume 3 – Applications** offers comprehensive, up-to-date coverage of equipment used in the process and power generation industries.

Under the leadership of Charles E. Baukal, Jr., top engineers and technologists from John Zink Hamworthy Combustion examine industry applications such as process burners, boiler burners, process flares, thermal oxidizers, and vapor control. This volume builds on the concepts covered in the first two volumes and shows how they are used in combustion applications. The book also features a wealth of color illustrations, photographs, and tables throughout.

What's New in This Edition

- Expanded to three volumes, with Volume 3 focusing on important industry applications
- Extensive updates and revisions throughout, reflecting new standards, energy sources, processes, and conservation concerns
- Expanded coverage of flares and new coverage of biogas flares and flare gas recovery
- Information on vapor combustors
- Discussion of pollution control equipment
- Expanded coverage of commercial and utility boiler burners
- Chapters on process and air heaters
- More material on thermal oxidizers
- A new chapter on marine and offshore applications

The third of three volumes in the new, expanded edition of the bestselling handbook, this volume helps you broaden your knowledge of industrial combustion applications to better meet the challenges of this field.

CRC Press
Taylor & Francis Group
an informa business
www.taylorandfrancisgroup.com

6000 Broken Sound Parkway, NW
Suite 300, Boca Raton, FL 33487
711 Third Avenue
New York, NY 10017
2 Park Square, Milton Park
Abingdon, Oxon OX14 4RN, UK

K11816
ISBN: 978-1-4398-3966-9
90000
9 781439 839669
www.crcpress.com