

PROCESS PIPING: THE COMPLETE GUIDE TO ASME B31.3

by

Charles Becht IV

ASME Press . . . New York . . . 2002

© 2002 by The American Society of Mechanical Engineers
Three Park Avenue, New York, NY 10016

All rights reserved. Printed in the United States of America. Except as permitted under the United States Copyright Act of 1976, no part of this publication may be reproduced or distributed in any form or by any means, or stored in a database or retrieval system, without the prior written permission of the publisher.

INFORMATION CONTAINED IN THIS WORK HAS BEEN OBTAINED BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS FROM SOURCES BELIEVED TO BE RELIABLE. HOWEVER, NEITHER ASME NOR ITS AUTHORS OR EDITORS GUARANTEE THE ACCURACY OR COMPLETENESS OF ANY INFORMATION PUBLISHED IN THIS WORK. NEITHER ASME NOR ITS AUTHORS AND EDITORS SHALL BE RESPONSIBLE FOR ANY ERRORS, OMISSIONS, OR DAMAGES ARISING OUT OF THE USE OF THIS INFORMATION. THE WORK IS PUBLISHED WITH THE UNDERSTANDING THAT ASME AND ITS AUTHORS AND EDITORS ARE SUPPLYING INFORMATION BUT ARE NOT ATTEMPTING TO RENDER ENGINEERING OR OTHER PROFESSIONAL SERVICES. IF SUCH ENGINEERING OR PROFESSIONAL SERVICES ARE REQUIRED, THE ASSISTANCE OF AN APPROPRIATE PROFESSIONAL SHOULD BE SOUGHT.

ASME shall not be responsible for statements or opinions advanced in papers or ... printed in its publications (B7.1.3). Statement from the Bylaws.

For authorization to photocopy material for internal or personal use under circumstances not falling within the fair use provisions of the Copyright Act, contact the Copyright Clearance Center (CCC), 222 Rosewood Drive, Danvers, MA 01923, tel: 978-750-8400, www.copyright.com.

Library of Congress Cataloging-in-Publication Data

Becht, C.

Process piping : the complete guide to ASME B31.3/1

p. cm.

ISBN 0-7918-0177-2

1. Piping—Standards. I. Title

TJ930 .B347 2002

621.8'672—dc21

Cover photo: Hovensa
Refinery, St. Croix, USVI
Courtesy of Hovensa

To My Wife
Mary
And Children
Charles, Derek, John, Kristen

ABOUT THE AUTHOR

Charles Becht is a recognized authority in pressure equipment, piping, expansion joints, and elevated temperature design. Dr. Becht is President of Becht Engineering Company, Inc., a consulting engineering company that provides both process and equipment engineering services for the process and power industries. He has performed numerous expert troubleshooting and failure investigations, design reviews and construction inspections for capital projects into the billion dollar range, and consulting to manufacturers on design, development and code compliance for new and existing equipment. He was previously with Energy Systems Group, Rockwell International, and Exxon Research and Engineering as a pressure equipment specialist.

Dr. Becht is a member of the ASME Board on Pressure Technology Codes and Standards, Chairman of the ASME Post Construction Main Committee (PCC), Chairman of the Subcommittee on Repair and Testing (PCC) and Chairman of the ASME Boiler and Pressure Vessel Code Subgroup on Elevated Temperature Design. He is also a member of a number of other ASME Code Committees including B31.3, Process Piping Section Committee (incoming Vice-Chairman); B31 Code for Pressure Piping, Mechanical Design Committee; Subcommittee on Flaw Evaluation (PCC); ASME Boiler and Pressure Vessel Code Subcommittee on Design and Subcommittee on Transport Tanks (SCXII) and is Vice-Chairman of the Post Construction Executive Committee. He is also a member of ASTM Committee F-17, Plastic Piping Systems Main Committee; and the ASME, PVP Division, Design and Analysis Committee.

He received a PhD from Memorial University in Mechanical Engineering, a MS from Stanford University in Structural Engineering and a BSCE from Union College, New York. Dr. Becht is a licensed professional engineer in 14 states/provinces, an ASME Fellow since 1996 and recipient of the ASME Dedicated Service Award in 2002, and has more than 50 publications and three patents.

ACKNOWLEDGEMENTS

Thanks to my assistant, Mickey Smajda, who helped in the preparation of this book, to the ASME publications staff who were helpful, and Fred Tatar and Bob Sims who reviewed Chapter 17.

Thanks also to Bill Short who encouraged me to join the ASME B31.3 Code Committee in 1986, Sam Zamrik who encouraged me to teach a half-day tutorial on ASME B31.3 at the ASME Pressure Vessel and Piping Conference in 1992, and Chris Ziu who sponsored a series of three-day courses I taught on ASME B31.3, which, in turn, led to this book.

And to my colleagues on the ASME B31.3 Code Committee, with whom I have had many interesting discussions over the years, and the students in the courses I have taught that have, at times, asked difficult questions, that have greatly enriched this book.



Committee members, former members and visitors at the 100th meeting of the ASME B31.3 Section Committee.

Front Row (Left to Right): Ron Haupt, Robert Meyerand, Dick Hinckley, Dave Moody, Dick Hudson, Pete Gardner, John Wier, Dick Straiton, Helmut Thielsch, Tom Estilow, Don Frikken, Warren Thomas, Ken Kluge, Bill McLean, Bob McKee, Ed Bane, Mervyn Pfeiffer

Center Row (Left to Right): Bruce Bassett, Al Simmons, Roy Grichuk, Phil Ellenberger, Art Post, Glynn Woods, Jon Labrador, Roger Bradshaw, Jerry Byers, Bob Nichols, Don Glover, Chris Ziu, Rick Fairlamb, Bill Koves, Jack Gott, Zoltan Romoda, Lalgudy Balasundaram, Quy Truong, Gary Nariani, Chuck Becht, Bijan Khamanian, Larry Smith, Jim Lang, Dave Fetzner, Walt Canham

Back Row (Left to Right): Bill Short, Jim Meyer, Bryan Harris, Steve Costa, Will Parrish, Barry Agee, Ken Oswald, Chandan Nath, Fred Tatar, Bob Hookway, Rex Engle, Carlos Davila, David Coym, Dan Christian, Ralph Rapp, Jerry D'Avanzo, Mike Jeglic, Joel Andreani, Don Edwards, Robert Silvia

Also (present but missing from photo); Art Palmer, Larry Nuesslein

CONTENTS

About the Author	v
Acknowledgements	vii
List of Figures	xv
List of Tables	xvii
 Chapter 1	
Background and General Information (ASME B31.3, Foreward, Introduction and Chapter I) ...	1
1.1 History of ASME B31.3	1
1.2 Scope of ASME B31.3	2
1.3 Intent	3
1.4 Responsibilities	4
1.4.1 Owner	4
1.4.2 Designer	4
1.4.3 Manufacturer, Fabricator, and Erector	5
1.4.4 Owner's Inspector	5
1.5 How is ASME B31.3 Developed and Maintained?	5
1.6 Code Editions and Addenda	6
1.7 How Do I Get Answers to Questions About the Code?	7
1.8 How Can I Change the Code?	7
 Chapter 2	
Organization of ASME B31.3	9
2.1 Fluid Services	9
2.2 Code Organization	10
2.3 Listed Components	11
2.4 Listed Materials	11
2.5 Safeguarding	11
 Chapter 3	
Design Conditions and Criteria (ASME B31.3, Chapter II, Parts 1 and 6)	15
3.1 Design Conditions	15
3.1.1 Design Pressure	15
3.1.2 Design Temperature	15
Insert 3.1 Calculation of Wall Temperature for Refractory-Lined Pipe	16
3.1.3 Design Minimum Temperature	18
3.2 Allowable Stress	18
3.3 Allowances for Pressure and Temperature Variations	19
3.4 Overpressure Protection	20

Chapter 4

Pressure Design (ASME B31.3, Chapter II, Part 2)	23
4.1 Methods for Internal Pressure Design	23
4.2 Pressure Design of Straight Pipe for Internal Pressure	24
Insert 4.1 Sample Wall Thickness Calculation	28
Insert 4.2 Basic Stress Calculations for Cylinders Under Pressure	28
4.3 Pressure Design of Straight Pipe Under External Pressure	30
4.4 Pressure Design of Welded Branch Connections	33
4.5 Pressure Design of Extruded Outlet Header	36
4.6 Additional Considerations for Branch Connections Under External Pressure	36
4.7 Branch Connections That are Presumed to be Acceptable	36
4.8 Pressure Design of Bends and Elbows	37
4.9 Pressure Design of Miter Bends	39
4.10 Pressure Design of Closures	40
4.11 Pressure Design of Flanges	41
4.12 Pressure Design of Blind Flanges	41
4.13 Pressure Design of Blanks	41
4.14 Pressure Design of Reducers	42
4.15 Pressure Design of Unlisted Components	42

Chapter 5

Limitations on Components and Joints (ASME B31.3, Chapter II, Parts 3 and 4)	45
5.1 Overview	45
5.2 Valves	45
5.3 Flanges	46
5.4 Fittings, Bends, Miters, and Branch Connections	47
5.5 Bolting	48
5.6 Welded Joints	49
5.7 Flanged Joints	49
5.8 Threaded Joints	50
5.9 Tubing Joints	50
5.10 Miscellaneous Joints	51

Chapter 6

Design for Sustained and Occasional Loads (ASME B31.3, Chapter II, Part 1)	53
6.1 Primary Longitudinal Stress	53
6.2 Sustained Longitudinal Stress	53
6.3 Limits of Calculated Stresses Due to Occasional Loads	54

Chapter 7

Design Criteria for Thermal Stress (ASME B31.3, Chapter II, Part 1)	57
7.1 Allowable Stress for Thermal Expansion	57
7.2 How to Combine Different Displacement Cycle Conditions	61
Insert 7.1 What About Vibration?	62

Chapter 8

Flexibility Analysis (ASME B31.3, Chapter II, Part 5)	63
8.1 Flexibility Analysis	63
8.2 When is Formal Flexibility Analysis Required?	63

8.3	When is Computer Stress Analysis Typically Used?	65
8.4	Stress Intensification Factors	65
8.5	Flexibility Analysis Equations	68
8.6	Cold Spring	69
	Insert 8.1 How to Increase Piping Flexibility.....	70
8.7	Elastic Follow-up/Strain Concentration	71
8.8	Effect of Elastic Modulus Variations Due to Temperature	73
 Chapter 9		
	Supports (ASME B31.3, Chapter II, Part 5)	75
9.1	Overview of Supports	75
9.2	Materials and Allowable Stress	75
9.3	Design of Supports	76
	Insert 9.1 Spring Design	77
	Insert 9.2 Stress Classification	79
9.4	Fabrication of Supports	84
 Chapter 10		
	Load Limits for Attached Equipment	85
10.1	Overview of Equipment Load Limits	85
10.2	Pressure Vessels	85
10.3	Other Equipment Load Limits	86
10.4	Means of Reducing Loads on Equipment	86
 Chapter 11		
	Requirements for Materials (ASME B31.3, Chapter III)	87
11.1	Overview of Material Requirements	87
11.2	Qualification of Unlisted Materials	87
11.3	Use of Materials Above the Highest Temperature for Which Allowable Stresses Are Provided.....	88
11.4	Low-Temperature Service	88
11.5	When is Impact Testing Required?.....	89
11.6	Impact Test Requirements	93
11.7	Fluid Service Limitations	95
11.8	How to Use the Allowable Stress Tables in Appendix A of the Code	95
 Chapter 12		
	Fabrication, Assembly, and Erection (ASME B31.3, Chapter V)	101
12.1	Overview of Chapter V of ASME B31.3	101
12.2	General Welding Requirements	101
	Insert 12.1 Arc Welding Processes	101
12.3	Welding Procedure Specification	106
12.4	Welding Procedure Qualification Record	106
12.5	Welder Performance Qualification	107
12.6	Preheating	108
12.7	Heat Treatment	108
12.8	Governing Thickness for Heat Treatment.....	112
12.9	Pipe Bends.....	113

12.10	Brazing and Soldering	113
	Insert 12.2 Brazing	114
12.11	Bolted Joints	114
12.12	Welded Joint Details	115
12.13	Miscellaneous Assembly Requirements	117

Chapter 13

Examination (ASME B31.3, Chapter VI)		119
13.1	Overview of Examination Requirements	119
13.2	Progressive Examination	120
13.3	Types of Examination	121
13.4	Required Examination	122
13.5	Acceptance Criteria	122
13.6	Examination of Nonmetallics	122

Chapter 14

Pressure Testing (ASME B31.3, Chapter VI)		125
14.1	Overview of Pressure Test Requirements	125
14.2	Hydrostatic Test	126
14.3	Pneumatic Test	126
14.4	Hydropneumatic Test	127
14.5	Alternative Leak Test	127
14.6	Sensitive Leak Test	127
14.7	Jacketed and Vacuum Piping	127
14.8	Initial Service Leak Test	128
14.9	Requirements for Nonmetallic Piping	128

Chapter 15

Nonmetallic Piping (ASME B31.3, Chapter VII)		129
15.1	Organization	129
15.2	Design Conditions	129
15.3	Allowable Stress	129
15.4	Pressure Design	130
15.5	Limitations on Components and Joints	131
15.6	Flexibility and Support	131
15.7	Materials	132
15.8	Bonding of Plastics	133
	Insert 15.1 Bonding Processes	134
15.9	Other Fabrication, Assembly, and Erection Requirements	140
15.10	Examination and Testing	140

Chapter 16

Category M Piping (ASME B31.3, Chapter VIII)		141
16.1	When to Use the Rules for Category M Fluid Service	141
16.2	Organization of Chapter VIII of ASME B31.3	142
16.3	Overview of Metallic Rules	142
16.4	Overview of Nonmetallic Rules	143
16.5	General Comments	143

Chapter 17

High-Pressure Piping (ASME B31.3, Chapter IX)	145
17.1 Scope of Chapter IX of ASME B31.3	145
17.2 Organization of Chapter IX of ASME B31.3	145
17.3 Pressure Design of High-Pressure Piping	146
17.4 External Pressure	147
17.5 Design for Sustained and Occasional Loads	148
17.6 Design for Thermal Expansion and Fatigue	148
17.7 Materials	149
17.8 Fabrication	150
17.9 Examination	150
17.10 Testing	152
17.11 Records	152

Chapter 18

Bellows Expansion Joints (ASME B31.3, Appendix X)	153
18.1 Overview of Bellows Requirements	153
18.2 Rules for Pressure Design	153
18.3 Fatigue Design	155
18.4 Unreinforced-Bellows Fatigue Curve Development	155
18.5 Reinforced-Bellows Fatigue Curve Development	157
18.6 Fatigue Design for Other Alloys	157
18.7 Correlation Testing	158
18.8 Elevated-Temperature Design	160
18.9 Shop Test of Expansion Joints	160
18.10 Field Test of Piping Systems with Expansion Joints	161
18.11 Future Developments	162
Insert 18.1 Pressure Thrust Considerations	162

Appendix I

Useful Information for Pressure Design	165
---	-----

Appendix II

Guidelines for Computer Flexibility Analysis	185
---	-----

Appendix III

Useful Information for Flexibility Analysis	189
--	-----

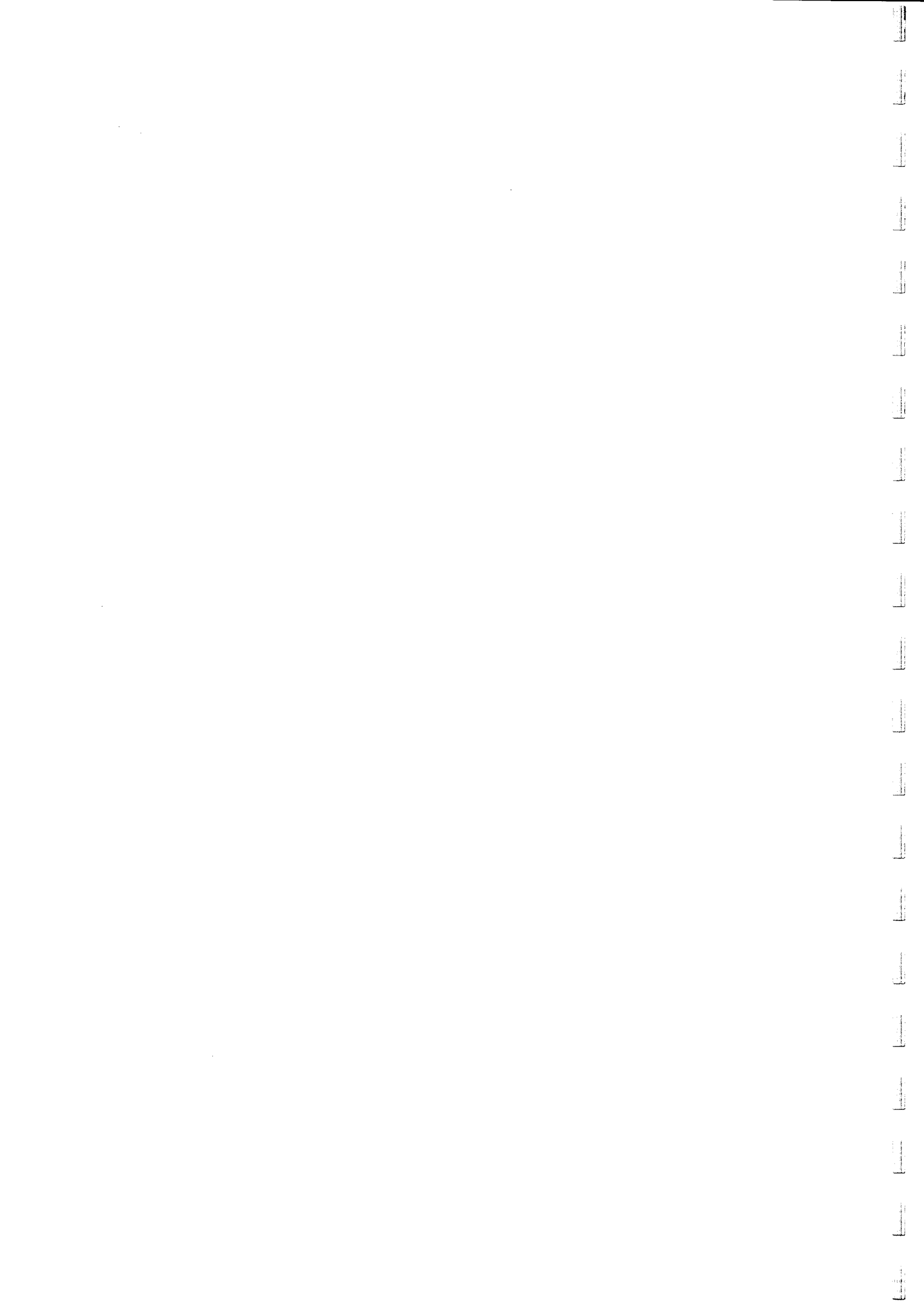
Appendix IV

Expansion Joint Guide	224
------------------------------	-----

Appendix V

Conversion Factors	250
---------------------------	-----

References	257
Index	265



LIST OF FIGURES

Figure Number

3.1	Effective Outside Film Coefficient	17
4.1	Stress Distribution Through Pipe Wall Thickness Due to Internal Pressure	25
4.2	Comparison of Lamé and Boardman Equations	27
4.3	Equilibrium at a Circumferential Cut	28
4.4	Equilibrium at a Longitudinal Cut	29
4.5	Chart for Determining <i>A</i> (ASME BPVC, Section II, Part D, Subpart 3, Fig. G)	31
4.6	Typical Chart for Determining <i>B</i> (ASME BPVC, Section II, Part D, Subpart 3, Fig. CS-2)	33
4.7	Illustration of Fabricated Branch Connection Showing Nomenclature (ASME B31.3, Fig. 304.3.3)	35
4.8	Illustration of Extruded Outlet Fitting Showing Nomenclature (ASME B31.3, Fig. 304.3.4)	37
4.9	Illustration of Miter Bend Showing Nomenclature (ASME B31.3, Fig. 304.2.3)	39
5.1	Taper Thread	50
7.1	Load-Controlled Versus Deformation-Controlled Behavior	58
7.2	Stress-Strain Behavior Illustrating Shakedown	59
7.3	Markl Fatigue Curve for Butt-Welded Steel Pipe	60
8.1	Markl-Type Fatigue Test Machine (Courtesy of WFI)	66
8.2	In-Plane and Out-Plane Bending Moments in Bends and Branch Connections (ASME B31.3, Figs. 319.4.4A and 319.4.4B)	67
8.3	Piping Layout 1	70
8.4	Piping Layout 2	71
8.5	Strain Concentration Two-Bar Model	71
9.1	Variable-Spring Hanger Table (Courtesy of Anvil International)	78
9.2	Constant-Effort-Spring Hanger Table (Courtesy of Anvil International)	80
11.1	Minimum Temperature Without Impact Testing for Carbon Steel Materials (ASME B31.3, Fig. 323.2.2A)	91
11.2	Reduction in Minimum Design Metal Temperature Without Impact Testing (ASME B31.3, Fig. 323.2.2B)	92
11.3	Schematic of Charpy Impact Test and Test Specimens Before and After Fracture	94
12.1	Shielded Metal Arc Welding (Courtesy of The James F. Lincoln Foundation)	102
12.2	Gas Tungsten Arc Welding (Courtesy of The James F. Lincoln Foundation)	103
12.3	Gas Metal Arc Welding (Courtesy of The James F. Lincoln Foundation)	104
12.4	Gas-Shielded Fluxed Cored Arc Welding (Courtesy of The James F. Lincoln Foundation)	104
12.5	Submerged Arc Welding (Courtesy of The James F. Lincoln Foundation)	105
12.6	Typical Details for Double-Welded Slip-on and Socket Welding Flange Attachment Welds (ASME B31.3, Fig. 328.5.2B)	116

12.7	Minimum Welding Dimensions for Socket Welding Components Other Than Flanges (ASME B31.3, Fig. 328.5.2C)	116
12.8	Acceptable Details for Branch Attachment Welds (ASME B31.3, Fig. 328.5.4D)	117
15.1	Fully Tapered Thermosetting Adhesive Joint (ASME B31.3, Fig. A328.5.6)	135
15.2	Thermosetting Wrapped Joints (ASME B31.3, Fig. A328.5.7)	136
15.3	Thermoplastic Solvent-Cemented Joint (ASME B31.3, Fig. A328.5.3)	137
15.4	Hot Gas Welding	141
15.5	Steps for Heat-Element Butt Fusion (Courtesy of Chris Ziu)	139
15.6	Thermoplastic Heat Fusion Joints (ASME B31.3, Fig. A328.5.4)	139
15.7	Thermoplastic Electrofusion Joints (ASME B31.3, Fig. A328.5.5)	140
17.1	Comparison of Wall Thickness Required by Base Code Pressure Design Rules [ASME B31.3, Eq. (3a)] versus those of ASME B31.3, Chapter IX [ASME B31.3, Eqs. (34a), (34b)]	147
18.1	Bellows Pressure Test Data	154
18.2	Fatigue Data for Unreinforced As-Formed Stainless Steel Bellows	156
18.3	Fatigue Data for Reinforced As-Formed Stainless Steel Bellows	157
18.4	Procedure for Constructing Fatigue Curves for Other Materials	159

LIST OF TABLES

Table Number

4.1	Values of Coefficient Y	25
5.1	Minimum Thickness for Threaded Pipe	51
7.1	Combination of Different Displacement Cycles	61
11.1	Requirements for Low-Temperature Toughness Tests for Metals	90
11.2	Typical Appendix A Allowable Stress Table	96
12.1	Preheat Temperatures	109
12.2	Requirements for Heat Treatment	110
13.1	Required Examination for Metallic Piping	123
17.1	Acceptance Criteria for Welds	151

BACKGROUND AND GENERAL INFORMATION

This book is based on the 1999 Edition of ASME B31.3, *Process Piping Code*, including the 2000 Addenda. Because changes, some very significant, are made to the Code every year, the reader should refer to the Code for any specific requirements. This book should be considered as providing background information and not specific current Code rules.

The equations in this book are numbered sequentially in each chapter. When equations from ASME B31.3 are reproduced herein the latter equation numbers are given as well.

1.1 HISTORY OF ASME B31.3

In 1926 the American Standards Association initiated Project B31 to develop a piping code. ASME was the sole administrative sponsor. The first publication of this document, *American Tentative Standard Code for Pressure Piping*, was in 1935. From 1942 through 1955 the Code was published as the *American Standard Code for Pressure Piping*, ASA B31.1. It was composed of separate Sections for different industries.

These Sections were split off, starting in 1955 with *Gas Transmission and Distribution Piping Systems*, ASA B31.8. ASA B31.3, *Petroleum Refinery Piping Code*, was first published in 1959. A number of separate Sections have been prepared, most of which have been published. The various Section designations are as follows:

- B31.1 *Power Piping*
- B31.2 *Fuel Gas Piping* [withdrawn in 1988]
- B31.3 *Process Piping*
- B31.4 *Liquid Transportation Systems for Hydrocarbons, Liquid Petroleum Gas, Anhydrous Ammonia, and Alcohols*
- B31.5 *Refrigeration Piping*
- B31.6 *Chemical Plant Piping* [never published]
- B31.7 *Nuclear Piping* [moved to *ASME Boiler and Pressure Vessel Code*, Section III]
- B31.8 *Gas Transmission and Distribution Piping Systems*
- B31.9 *Building Services Piping*
- B31.10 *Cryogenic Piping* [never published]
- B31.11 *Slurry Piping*

A draft of the Section for Chemical Plant Piping, B31.6, was completed in 1974. However, it was decided to merge this Section into ASME B31.3, because the two Code Sections were closely related. A joint Code Section, Chemical Plant and Petroleum Refinery Piping, was published in 1976. At this

time items such as fluid service categories (e.g., Category M), Nonmetallic Piping, and Safeguarding were introduced into ASME B31.3. In 1980 the Nonmetal portions of the B31.3 Code were gathered and combined into one Chapter, Chapter VIII.

A draft Code for Cryogenic Piping was prepared by Section Committee B31.10 and was ready for approval in 1981. Again, since the coverage overlapped with ASME B31.3, it was decided to merge the Section Committees and develop a single, inclusive Code. This Code was issued in 1984. In the same year another potentially separate Code was added as a new chapter to ASME B31.3, High-Pressure Piping, Chapter IX.

The resulting document is a Code that is very broad in scope. It covers fluids as benign as water and as hazardous as mustard gas. It covers temperatures from cryogenic conditions to 815°C (1500°F) and beyond, and pressures from vacuum and atmospheric to 340,000 kPa (50,000 psi) and higher. Part of the philosophy of the Code stems from this broad coverage. There is a great deal of responsibility placed with the owner and latitude to use good engineering.

The acronym that appears in front of B31.3 in the title of the Code has changed from ASA to ANSI to ASME. The initial designation, ASA, referred to the American Standards Association. Between 1967 and 1969 ASA became first the United States of America Standards Institute and then the American National Standards Institute (ANSI). In 1978 the Standards Committee was reorganized as a committee operating under the American Society of Mechanical Engineers (ASME) with ANSI approval. Thus, it is currently correct to refer to the Code as ASME B31.3. These changes have not changed the Committee structure or the Code.

1.2 SCOPE OF ASME B31.3

ASME B31.3 is the generally accepted standard for process piping, worldwide. It is the de facto standard for piping for the oil, petro-chemical and chemical industries, worldwide. To facilitate its international use, ISO/WD 15649, *Petroleum and natural gas industries—piping*, has been approved. This ISO standard makes a normative reference to ASME B31.3. It is an evergreen reference, as it is not tied to a specific edition. The ISO Standard also provides additional technical requirements, such as provisions for buried piping.

To facilitate use of ASME B31.3, as referenced by ISO 15649, in Europe, a guide is being prepared by EEMUA. This guide, entitled "Guide to the Use of ISO 15649 and ANSI/ASME B31.3 for Piping in Europe in Compliance With the Pressure Equipment Directive," will provide specific guidance on how to comply with the European Pressure Equipment Directive (PED) while using ISO 15649. Following this guide does not provide a presumption of conformity with the PED, therefore, the conformity assessment requirements of the PED must be followed in full.

Within the United States, ASME B31.3 is the recognized and generally accepted good engineering practice for process piping. It is specifically referenced by legislation in the following states (Alaska, California, Maryland and Oklahoma) and provinces of Canada (British Columbia, Northwest Territories, Ontario, Prince Edward Island, Quebec and Saskatchewan). However, it is also referenced by other Codes, Standards and regulations, such as NFPA and CFR. As such, it is often required by law, although not directly specified in state regulations.

The ASME B31.3 Code was written with process piping in mind. It is very broad in scope, reflecting the scope of process piping services. Although ASME B31 allows owners to select the piping code most appropriate to their piping installation, the following are typical examples of the types of facilities that ASME B31.3 is intended to cover:

- Chemical plants
- Oil refineries
- Loading terminals
- Bulk processing plants
- Cryogenic piping

The Code was previously intended to cover all piping within the property limits of a process plant, with certain exclusions. The ASME B31 Codes now state that it is the owners' responsibility to decide which Code is most applicable to their piping system. However, these previously listed exclusions show the intent of the writers of the Code. The following items were excluded from coverage by earlier editions of ASME B31.3:

- Fire protection systems
- Sanitary and storm water systems
- Plumbing
- Piping in property set aside for transportation piping (ASME B31.4, B31.8, and B31.11)
- Piping within the jurisdiction of Section I of the *ASME Boiler and Pressure Vessel Code* (BPV) (i.e., boiler external piping)

Certain piping was considered optional:

- Steam piping from a boiler that is outside the jurisdiction of ASME BPV, Section I, which could be either ASME B31.1 or ASME B31.3
- Refrigeration piping that is part of a packaged system (factory-assembled), which could be either ASME B31.5 or ASME B31.3

Boiler external piping is the piping between a boiler casing and the first block valve outside of the boiler. This is within the jurisdiction of Section I of ASME BPVC, although Section I refers to the rules of ASME B31.1 for this piping.

Although the Code included building systems piping (e.g., heating systems) that are on the process plant property, other than items specifically excluded (e.g., sanitary), these are better covered by ASME B31.9. In practice, it is doubtful that many people have used ASME B31.3 for their building systems piping in any case.

1.3 INTENT

The ASME B31.3 Code provides minimum requirements for safety. It is not a design handbook. Furthermore, it is for new piping. Although the intent of ASME B31.3 may be considered in evaluating existing piping, piping that has been in service is not within the scope of the Code. Other standards, such as API 570, *Piping Inspection Code*, should be considered for use with piping after it has been placed in service. The ASME B31.3 Code does not address operation or maintenance of piping systems.

The following paragraphs, quoted from para. 300(c) at the beginning of the Code, provide important insights into its basic intent:

- “(3) Engineering requirements of this Code, while considered necessary and adequate for safe design, generally employ a simplified approach to the subject. A designer capable of applying a more rigorous analysis shall have the latitude to do so, however, the approach must be documented in the engineering design and its validity accepted by the owner. The approach used shall provide details of design, construction, examination, inspection, and testing for the design conditions of para. 301, with calculations consistent with the design criteria of this Code.”
- “(4) Piping elements should, insofar as practicable, conform to the specifications and standards listed in this Code. Piping elements neither specifically approved nor specifically prohibited by this Code may be used provided they are qualified for use as set forth in applicable Chapters of this Code.”
- “(5) The engineering design shall specify any unusual requirements for a particular service. Where service requirements necessitate measures beyond those required by this Code, such measures

shall be specified by the engineering design. Where so specified, the Code requires that they be accomplished.”

These statements should help in understanding the philosophy of the ASME B31.3 Code. The Code is not intended to rigorously set forth every procedure, approve every component, and approve every material. Instead, procedures are set forth for evaluating the use of unlisted components and unlisted materials. This differs, for example, from the *ASME Boiler and Pressure Vessel Code* and *Power Piping Code* requirements that Code Cases be prepared to approve any material for use that is not listed by the Code.

Paragraph 300(c)3 states that more rigorous analysis methods may be used. Finite-element analysis is an example of one such method. However, the designer must be able to demonstrate (to the owner) the validity of the more rigorous analysis method. The paragraph was revised in the 2000 Addendum of the Code. It was clarified to specifically state that the validity must be accepted by the owner and the approach documented in the engineering design. Further, the last sentence was added. This change was made to address concerns that 300(c)3 could be interpreted too liberally, and to more specifically state the intent.

The same approach can be found throughout the Code. For example, heat treatments other than those specified may be used, components in listed standards may be rerated, and the temperatures for which allowable stresses are provided may be exceeded. There is a great deal of freedom for good engineering practice and much responsibility for owners.

1.4 RESPONSIBILITIES

1.4.1 Owner

The owner's first responsibility is to determine which Code Section should be used. The owner has overall responsibility within the ASME B31.3 Code for compliance with the Code and for establishing the requirements for design, fabrication, examination, inspection, and testing. Owners are also responsible for designating the fluid service if they desire Category D or M and selecting Chapter IX if they wish it to be used for high-pressure piping.

The owner is the individual or organization that will own and operate the facility after it is constructed. For example, in turnkey construction, the owner is not the contractor, but the organization to which the facility will be turned over.

1.4.2 Designer

The Designer is responsible to the owner for assurance that the engineering design of piping complies with the requirements of the Code and with any additional requirements established by the owner. Qualifications for the designer were added in the 2000 Addendum, as para. 301.1. Note that the Designer is the person in charge of the engineering design of the piping system. This is not the job classification typically called “piping designer” nor the design firm.

The qualifications are stated as minimum experience, including some combination of education or professional registration, and experience in the design of related pressure piping. Experience that includes design calculations for pressure, sustained and occasional loads, and piping flexibility is considered to satisfy this experience requirement.

Different systems may require more or less experience, depending on their complexity and criticality. A Designer with less experience may be acceptable in some circumstances; this is permissible with the approval of the owner.

The number of years of experience is not specified for engineers with Professional Engineering registration. Reliance is instead placed on jurisdictional laws that prohibit Professional Engineers from taking responsible charge of work for which they do not have the necessary competence.

1.4.3 Manufacturer, Fabricator, and Erector

The manufacturer, fabricator, and erector of piping are responsible for providing materials, components, and workmanship in compliance with the requirements of the Code and of the engineering design. Remember that additional requirements specified in the engineering design become requirements of the Code for that piping installation.

1.4.4 Owner's Inspector

The owner's Inspector is responsible to the owner for ensuring that the requirements of the Code for inspection, examination, and testing are met. The Inspector is the owner's representative and oversees the examination and testing work performed by the contractor. The Inspector, therefore, cannot be an employee of the contractor, but could be a third party retained by the owner. The Inspector is a representative of the owner, not an independent third party, such as an authorized inspector. Required qualifications of the Inspector are provided in Chapter V of the Code and are discussed below in Chapter 13. Note that the Inspector is not required to perform examinations or check design.

1.5 HOW IS ASME B31.3 DEVELOPED AND MAINTAINED?

ASME B31.3 is a consensus document. It is written by a Committee intended to be made up of a balanced representation from a variety of interests. These include members with the following perspectives:

- Manufacturer (AK)
- User (AW)
- Designer/Constructor (AC)
- Regulatory (AT)
- Insurance Inspection (AH)
- General Interest (AF)

The members of the Committee are not intended to be representatives of specific organizations; their membership is considered based on the qualifications of the individuals and the desire for a balanced representation of various interest groups.

ASME B31.3 is written as a consensus code and is intended to reflect industry practice. This differs from a regulatory approach, in which rules may be written by a government body.

Changes to the Code are prepared by the B31.3 Section Committee. Within the Section Committee, responsibility for specific portions of the Code are split among the following Task Groups:

- Task Group A: Chapters I, II (Parts 3 and 4), IV, and VIII
Appendices G, M, and Q
Coordination of Appendices E and F
Technical Oversight of Code
- Task Group B: Chapter II (Parts 1, 2, 5, and 6)
Appendices D, H, V, and X
- Task Group D: Chapters II (302.3.1 through 302.3.4 only) and III
Appendices A and C (metals only)
- Task Group E: Chapters V and VI
- Task Group F: Chapter VII
Appendices B and C (nonmetals only)
- Task Group G: Chapter IX
Appendix K

These task groups can be roughly categorized as (A) general rules, (B) design/analysis, (D) materials, (E) fabrication, examination, testing, and erection, (F) nonmetallics, and (G) high pressure.

To make a change to the Code, the responsible Task Group prepares the change and sends out a letter ballot for the Section Committee as a whole to vote on the change. Anyone who votes against the change, i.e., votes negatively, must state their reason for doing so, which is sent to the entire Section Committee. The responsible Task Group usually makes an effort to resolve any negatives. In fact, it is rare for any change to go forward with more than a couple of negative votes out of a Section Committee that consists of 50 or more individuals. Only two-thirds approval is required for the change to move forward to the next level of approval, the B31 Standards Committee.

Any changes to the Code are forwarded to the B31 Standards Committee along with the written reasons for any negative votes and the Section Committee responses to those negatives. In this fashion, the Standards Committee is given the opportunity to see any opposing viewpoints. If anyone on the B31 Standards Committee votes negatively on a change on first consideration, the item is returned to the Section Committee with written reasons for the negative vote. The Section Committee must consider and respond to any negatives, either by withdrawing or modifying the proposed change or by providing explanations that respond to the negative. If the item is returned to the Standards Committee for second consideration, it requires a two-thirds approval to pass.

Once an item is passed by the Standards Committee, it is forwarded to the Board on Pressure Technology Codes and Standards. The Board is the final level at which the item is voted on within ASME. Again, any negative vote at this level returns the item to the Section Committee and a second consideration requires two-thirds approval to pass.

Although the Board on Pressure Technology Codes and Standards reports to the Council on Codes and Standards, the Council does not vote on changes to the Code.

The final step is a public review process. The availability of drafts is announced in two publications, *ANSI Standards Action* and *ASME Mechanical Engineering*. Copies of the proposed changes are also forwarded to the B31 Conference Group and B31 National Interest Review Group for review. Any comments from the public or the Groups are considered by the Section Committee. These procedures provide for careful consideration and public review of any proposed change to the Code.

1.6 CODE EDITIONS AND ADDENDA

A new edition of the Code is issued every three years. Addenda are issued in each of the two years between editions. These addenda are designated "a" and "b," respectively. The Addenda and the subsequent new edition include the following:

- Technical changes that have been approved by letter ballot
- Editorial changes that clarify the Code, but do not change technical requirements
- Errata to correct typographical errors and the like

This publication schedule was a change from prior policy as of the 1999 edition. Prior policy was to issue three Addenda, with one Addendum issued the year the new edition was published. All technical changes were made in Addenda, and only editorial changes and Errata were included in the new Edition.

This book is based on the 1999 Edition, including Addenda a. Addenda "a" and "b" were issued in 2000 and 2001, respectively. A new Edition will appear in 2002.

Significant changes can occur with each Addendum, as well as each Edition. An engineer whose practice includes process piping should keep current Codes. New Editions of the Code, which include delivery of the Addenda, Errata, and interpretations associated with them, may be obtained from ASME.

It should be noted that a change to the above procedure is under discussion. It has been proposed to issue new Editions every 2 years starting with the 2002 Edition and eliminate the Addenda. Any changes that are sufficiently urgent would be issued as Code Cases. The reason for such a change in publication practice is that the Addenda service is incompatible with electronic publishing/sales of the document.

1.7 HOW DO I GET ANSWERS TO QUESTIONS ABOUT THE CODE?

The B31.3 Section Committee responds to all questions on the Code via an inquiry process. Instructions for writing a request for an interpretation are provided in ASME B31.3, Appendix Z. The Committee will provide a strict interpretation of the existing rules. However, as a matter of policy, the Committee will not approve, certify, rate, or endorse anything, nor will it act as a consultant on specific engineering problems or the general understanding or application of Code rules. Further, it will not provide explanations as to background or reasons for Code rules. If you need one of these to be done, you should take a course, read this book, or hire a consultant, as appropriate.

The Section Committee will answer any request for interpretation with a literal interpretation of the Code. It will not create rules that do not exist in the Code, and will state that the Code does not address an item if it is not specifically covered by rules written into the Code. Even if the Section Committee disagrees with how the Code is written, it will answer in accordance with how it is written, and then possibly consider changes in the Code.

This strict procedure is considered to be necessary, because interpretations are often asked due to disagreements that may be subject to contractual terms or litigation. If the Code is the binding document, it must be taken as it was written at the time, not how one may prefer it to have been written.

Answering requests for interpretations is one of the highest priorities in the Section Committee meetings. The response is almost always prepared at the first Section Committee meeting following receipt of the request, and is then forwarded to the inquirer by ASME staff. Interpretations are published with Addenda for the benefit of all Code users.

1.8 HOW CAN I CHANGE THE CODE?

The simplest means of trying to change the Code is to write a letter suggesting a change. Any requests for revision to the Code are considered by the Task Group responsible for the respective Section of the Code. Note that the response of the Task Group must be approved by the full Section Committee. To increase the likelihood of a suggested change being adopted, the individual should come to the meeting at which the item will be discussed. ASME B31.3 Section Committee meetings are open to the public, and participation of interested parties is generally welcomed. Having a person explain the proposed change and the need for it is even more effective than sending a letter. If you become an active participant and have appropriate professional and technical qualifications, you could be invited to become a B31.3 Section Committee member.

Your request for a Code change may be passed to one of two Technical Committees under ASME B31. These are the Fabrication and Examination Technical Committee and the Mechanical Design Technical Committee. These Technical Committees exist to provide technical advice and ensure consistency among the various Code Sections.

ORGANIZATION OF ASME B31.3

2.1 FLUID SERVICES

The Code includes three Categories of fluid service, which provides a means of discriminating among possible degrees of hazard. Less stringent design, examination, and testing are permitted for fluid service of lower hazard (Category D) and more stringent requirements are applied for more hazardous fluid service (Category M). All fluid services are considered Normal unless the owner designates them as Category D or Category M.

It is the owner's responsibility to select the fluid service category. Selections of Category D or Category M cannot be made without the owner's permission. The fluid service is assumed to be Normal Fluid Service unless a different selection is made. Because some owners may not be familiar with this responsibility, the piping designer should inform the owner of the responsibility and may offer advice with respect to the selection process.

Category D Fluid Service is the less hazardous service. It includes fluids that are nontoxic, non-flammable, and not dangerous to human tissue and are at pressure less than 1035 kPa (150 psi) and temperature from -29°C through 186°C (-20°F through 366°F). These criteria can be found in the definitions in Chapter I of the Code, under "*fluid service*." Water piping is an obvious candidate for Category D Fluid Service. For example, 150# steam can be classified as Category D Fluid Service even though an individual in a container filled with 150# steam would obviously suffer tissue damage. The important additional distinction is that it is the condition of the fluid on leakage, not in the pipe, that must be considered. Rules are provided in the Code that basically permit less expensive construction for these less hazardous services.

Category M Fluid Service is reserved for extremely hazardous fluid services. Examples of fluid services that would usually be designated as Category M include systems containing methyl isocyanate, phosgene, and nerve gas. Systems containing fluids such as H_2S and hydrogen cyanide are not typically designated as being in Category M Fluid Service. However, it is not possible to create a list of Category M fluids, because the conditions of the installation must be considered in making the classification. For a fluid service to be Category M, the potential for personnel exposure must be judged to be significant. If a piping is double-contained, for example, it could be judged that even highly toxic fluids such as phosgene do not make the system Category M, because the potential for personnel exposure is not significant. For Category M fluid service, the rules for Normal Fluid Service are not applicable. Instead, additional rules that lead to more costly construction, with provisions designed to enhance piping system tightness, are provided in Chapter VIII.

The definition of Category M fluid service is as follows, based on the definition of fluid service in Chapter I of ASME B31.3. Note that, for purposes of emphasis, it has been broken into subparts, all of which must be satisfied for the service to meet the definition of Category M:

“a fluid service
in which the potential for personnel exposure is judged to be significant
and in which a single exposure of a very small quantity
of a toxic fluid,
caused by leakage,
can produce serious irreversible harm to persons on breathing or bodily contact,
even when prompt restorative measures are taken.”

Note that the Code considers many very hazardous fluid services to be normal fluid service. The design and construction rules for normal fluid service are suitable for hazardous services. Category M provides a higher level. If higher integrity piping is desired by the owner, even though the fluid service does not meet the definition of Category M, the owner can still specify the additional design, construction, examination, and testing requirements that are provided in Chapter VIII. Hydrofluoric acid is one example of a fluid for which many owners specify more stringent requirements than are provided in the Code for Normal Fluid Service, although it actually would be considered Normal Fluid Service.

In addition, it may be desirable to use more stringent rules than those provided in Chapter VIII for Category M Fluid Service. For example, it may be considered appropriate to perform 100% radiographic examination rather than 20% (the requirement of Chapter VIII) or to double-contain a fluid that would otherwise be considered Category M.

A flowchart is provided in ASME B31.3, Appendix M, Guide to Classifying Fluid Services. This Appendix is considered by the Code to provide guidance, not Code requirements. The actual Code requirements for fluid service classification are the definitions in Chapter I.

2.2 CODE ORGANIZATION

The ASME B31.3 Code has certain organizational features that, when understood, make it easier to follow. The Code consists of nine chapters. The first six chapters are what is called the base Code. It provides the basic piping Code requirements for Normal and Category D fluid service metallic piping. Chapter VII contains rules for nonmetallic piping and additional requirements for metallic piping lined with nonmetals. Chapter VIII contains rules for piping in Category M Fluid Service.

Chapter IX contains rules for high-pressure piping. This Chapter is never a required selection. It is always the option of the owner to select the use of Chapter IX. Although the Code provides a guideline stating that high pressure can be considered to refer to over Class 2500 flanges, this is not a requirement, and the base Code may be satisfactorily used to pressures higher than this.

The paragraphs in the Code follow a specific numbering scheme: Paragraphs are designated in the 300 range. The 300 series paragraphs are exclusive to the ASME B31.3 Code Section of the ASME B31 *Code for Pressure Piping*. The paragraph numbers in the base Code are repeated in Chapters VII through IX. Those paragraphs are differentiated by a letter placed in front of the paragraph numbers in each case. Chapter VII uses the letter A, Chapter VIII uses the letter M for metallic piping and MA for nonmetallic piping, and Chapter IX uses the letter K.

Using the same paragraph numbers permits easy cross-referencing between each of the last three chapters and the base Code requirements. The last three chapters refer back to base Code requirements whenever they are applicable. In this arrangement, the same requirements need not be repeated multiple times. Also, these three chapters tell users whenever they need to refer back to base Code requirements.

2.3 LISTED COMPONENTS

Chapter IV of the Code, Standards for Piping Components, tabulates the listed standards. Listed piping components are those that are in accordance with the standards listed in Table 326.1. Components that do not comply with those standards are considered to be unlisted components. Note that listed components for Chapter VII (nonmetallic) are listed in Table A326.1, metallic components for Chapter VIII (Category M) are listed in Table 326.1, nonmetallic components for Chapter VIII are listed in Table A326.1 and components for Chapter IX (high pressure) are listed in Table K326.1.

2.4 LISTED MATERIALS

Listed materials are those that are listed in the allowable stress tables of the Code and those permitted in component standards listed in the Code, as in Table 326.1, Table A326.1, or Table K326.1, as appropriate.

2.5 SAFEGUARDING

ASME B31.3 incorporates the concept of safeguarding. Safeguarding involves consideration of factors beyond the simple design of the pipe in the overall safety of the piping installation. It brings in the concepts of consequences of failure and probable sources of damage, which essentially considers risk.

Safeguarding is a concept that works well in the context of ASME B31.3, because the owner has overall responsibility for all aspects of the piping system. This differs from the much more limited scope of responsibilities in ASME B31.1 and ASME BPVC, Section VIII, Division 1. Because the owner has complete responsibility for both design and operation, the owner also has the ability to effectively specify and implement safeguarding provisions.

ASME B31.3 permits the use of certain components, joining methods, and other procedures when appropriate safeguards are provided. For example, brazed joints are prohibited from use in piping systems containing flammable or toxic fluids, unless safeguarded. Because the concern with brazed joints is failure of the joint on fire exposure due to melting of the brazing material, appropriate safeguards could involve protecting the joints from fire exposure.

A partial list of conditions where safeguarding is required includes the following:

- Paragraph 305.2.2, use of ASTM A 134 pipe made from ASTM A 285 plate and A 139 pipe for other than Category D Fluid Service
- Paragraph 308.2.4, use of flanges other than weld neck flanges meeting certain criteria for severe cyclic conditions
- Paragraph 313, use of expanded joints when the fluid is toxic or damaging to human tissue
- Table 314.2.1, use of threaded joints in sizes larger than DN 50 (NPS 2) when the fluid is flammable, toxic, or damaging to human tissue
- Paragraph 314.2.2, use of straight-threaded joints under severe cyclic conditions when the joint is subject to external moment loadings
- Paragraph 315.2, use of flared, flareless, or compression-type tubing fittings covered by listed standards, in normal fluid service, subject to severe cyclic conditions
- Paragraph 317.2, use of brazed and braze-welded joints in fluid services that are flammable, toxic, or damaging to human tissue
- Paragraph 318.2.3, use of bell and gland-type joints (other than caulked joints, which are subject to further limitations in para. 316) under severe cyclic conditions

- Paragraph 323.2.2(d), exemption of safeguarded piping from impact testing based on the stress state from external loads
- Paragraph 323.4.2, use of cast iron other than ductile iron permitted for specified conditions only when safeguarded against excessive heat and thermal shock and mechanical shock and abuse
- Paragraph A323.4.1, safeguarding against excessive temperature, shock, vibration, pulsation, and mechanical abuse when nonmetallic materials are used in any fluid service
- Paragraph A323.4.2(a), use of thermoplastics in other than Category D fluid service
- Paragraph A323.4.2(b), use of reinforced plastic mortar (RPM) in other than Category D Fluid Service
- Paragraph A323.4.2(c), use of reinforced thermosetting resin (RTR) piping in toxic or flammable fluid service
- Paragraph A323.4.2(d), use of safeguarding against large, rapid temperature changes when borosilicate glass and porcelain are used in any fluid service; additionally, use of general safeguarding when they are used in toxic or flammable fluid services

The following safeguards are also mentioned in the Code:

- Paragraph M300(d) requires consideration be given to additional engineered safeguards for Category M fluid service.
- Paragraph FA323.4(b) (Appendix F) recommends safeguards for thermoplastic piping in aboveground compressed gas (including air) service.

Appendix G of ASME B31.3 discusses the concept of safeguarding and provides examples of safeguarding by plant layout and operation and engineered safeguards.

Safeguarding requires addressing the potential consequences of failure. Therefore, the hazardous properties of the fluid, the quantity of the fluid that could be released by a piping failure, the consequences of such a release with respect to personnel exposure and equipment damage (with potential additional consequences), and the conditions of the environment and their effect on the hazards caused by a possible piping failure are all considerations of safeguarding. These address the consequence aspect of risk. The safety inherent in the piping by virtue of materials of construction, methods of joining, and history of service reliability, also mentioned, addresses the probability-of-failure aspect of risk.

Safeguarding by plant layout and operation includes the following examples from Appendix G of ASME B31.3:

“(a) Plant layout features, such as open-air process equipment structures; spacing and isolation of hazardous areas; slope and drainage; buffer areas between plant operations and populated communities; and control over plant access.

(b) Protective installations, such as fire protection systems; barricades or shields; ventilation to remove corrosive or flammable vapors; instruments for remote monitoring and control; and containment and/or recovery facilities or facilities (e.g., incinerators) for emergency disposal of hazardous materials.

(c) Operating practices, such as restricted access to processing areas; work permit system for hazardous work; and special training for operating, maintenance, and emergency crews.

(d) Means for safe discharge of fluids released during pressure relief device operation, blowdown, cleanout, etc.

(e) Procedures for startup, shutdown, and management of operating conditions, such as gradual pressurization or depressurization and gradual warmup or cooldown, to minimize the possibility of piping failure, e.g., brittle fracture.”

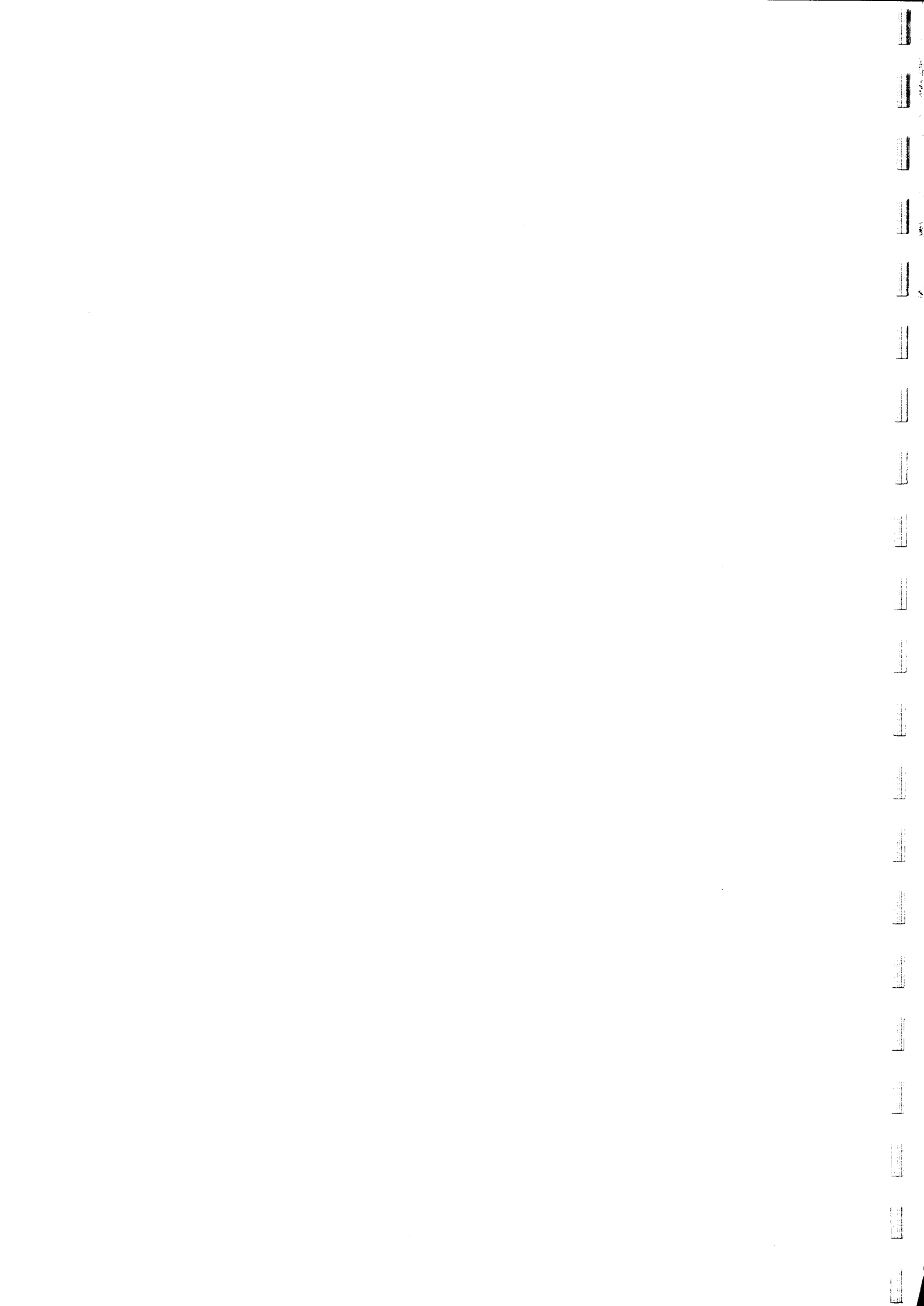
Examples of engineered safeguards include the following:

“(a) Means to protect piping against possible failures, such as (1) thermal insulation, shields, or process controls to protect from excessively high or low temperature and thermal shock; (2) armor,

guards, barricades, or other protection from mechanical abuse; (3) damping or stabilization of process or fluid flow dynamics to eliminate or to minimize or protect against destructive loads (e.g., severe vibration pulsations, cyclic operating conditions).

(b) Means to protect people and property against harmful consequences of possible piping failure, such as confining and safely disposing of escaped fluid by shields for flanged joints, valve bonnets, gages, or sight glasses; or for the entire piping system if of frangible material; limiting the quantity or rate of fluid escaping by automatic shut-off or excess flow valves, additional block valves, flow-limiting orifices, or automatic shutdown of pressure source; limiting the quantity of fluid in process at any time, where feasible."

The above serve as examples of what is meant by safeguarding. They do not, however, represent all the possibilities of how piping systems can be safeguarded.



DESIGN CONDITIONS AND CRITERIA

3.1 DESIGN CONDITIONS

Design conditions in ASME B31.3 are specifically intended for pressure design. The design pressure and temperature are the most severe *coincident* conditions, defined as the conditions that result in the greatest pipe wall thickness or highest required pressure class or other component rating. Design conditions are not intended to be a combination of the highest potential pressure and the highest potential temperature, unless such conditions occur at the same time.

Although it is possible for one operating condition to govern the design of one component in a piping system (and be the design condition for that component) and another to govern the design of another component, this is a relatively rare event. If it is encountered, the two different components in the piping system should have different design conditions.

3.1.1 Design Pressure

In determining the design pressure, all conditions of internal pressure must be considered. These include thermal expansion of trapped fluids, surge, and failure of control devices. The determination of design pressure can be significantly affected by the means used to protect the pipe from overpressure. Process piping systems are permitted to be used without protection of safety relief valves. However, in the event that none are provided on the pipe (or on attached equipment that would also protect the pipe), the piping system must be designed to safely contain the maximum pressure that can occur in the piping system, including consideration of failure of any and all control devices. Such a pressure could be, for example, the dead-head pressure of a pump. Unlike some industry standards, there is no consideration given to whether the source of failure depends on single- versus double-contingency events.

Although all these pressures must be considered, they do not necessarily become the design pressure. The Code permits pressure and temperature variations per para. 302.2.4. If the event being considered complies with the Code requirements of 302.2.4, the allowable stress and/or component pressure rating may be exceeded for a short time, as discussed below in Section 3.3. Although this is often considered to be an allowable variation above the design condition, the variation limitations are related to the maximum allowable working pressure of the piping, not the design conditions, which could be lower than the maximum allowable pressure at temperature.

3.1.2 Design Temperature

It is the metal temperature that is of interest in establishing the design temperature. Thus, the design temperature does not necessarily coincide with the temperature of the process fluid. In addition to

fluid temperature, other considerations include ambient cooling, ambient heating, solar radiation, and maximum heat tracing temperature.

For an insulated pipe, the metal temperature is generally considered to be the fluid temperature, unless some other means of heating is a consideration, such as the presence of steam tracing, which could potentially heat the pipe to the steam temperature.

For uninsulated pipe, the metal temperature is taken as the fluid temperature if the temperature is below 65°C (150°F), unless some other effect, such as solar heating of the pipe, could result in a greater temperature. A commonly assumed potential temperature due to solar heating for piping exposed to the sun is 50°C (120°F); local conditions may dictate a higher temperature.

For an uninsulated pipe, the Code permits consideration of ambient cooling effects if the fluid temperature is 65°C (150°F) or greater. Of course, if solar radiation can bring the metal temperature to higher than this temperature, as in Saudi Arabia, ambient cooling effects could only be considered at some higher temperature. The metal temperature of the pipe can be determined via heat transfer calculations or testing, such as measurement of the temperature on a similar system, or based on presumptive values given in the following table, which provides conservatively high estimates of wall temperature:

Assumed Metal Temperature	Component
95% of fluid temperature	Components having a wall thickness comparable to pipe, such as valves, pipe, lapped ends, and welding fittings
90% of fluid temperature	Flanges (except lap joint), including those that are on fittings and valves
85% of fluid temperature	Lap joint flanges
80% of fluid temperature	Bolting

For an internally insulated pipe, such as a refractory-lined pipe, the metal temperature must be determined by heat transfer calculations or test. Note that the heat transfer calculations must consider the appropriate range of conditions of wind and ambient temperatures to which the pipe will be subjected.

Insert 3.1 Calculation of Wall Temperature for Refractory-Lined Pipe

In lines for high-temperature fluid solids (e.g., in fluid catalytic cracking units), it is sometimes necessary to calculate the metal temperature with an internal, insulating refractory lining. In this case, the metal temperature may be significantly less than the fluid temperature. The refractory lining resists erosion while at the same time permitting use of carbon steel with a fluid that would otherwise be prohibitively hot for its use. Note that the metal temperature is often deliberately kept warm, often above 150°C (300°F), to prevent condensation against the inside of the steel pipe (of gasses passing through refractory pores and cracks) and resultant corrosion (e.g., due to sulfuric acid and sulfurous acid in flue gas) of the metal under the refractory. The following equations and a chart of film coefficient are provided as means for calculating the metal temperature. Note that, for design temperature, conditions that maximize the metal temperature are assumed (zero wind and hottest ambient temperature).

The heat flow through the pipe, per square foot per hour, is given by

$$Q = \frac{2\pi \Delta T}{\frac{1}{r_1 h_i} + \frac{\ln(r_2/r_1)}{K_r} + \frac{\ln(r_3/r_2)}{K_s} + \frac{1}{r_3 h_o}} \quad (3.1)$$

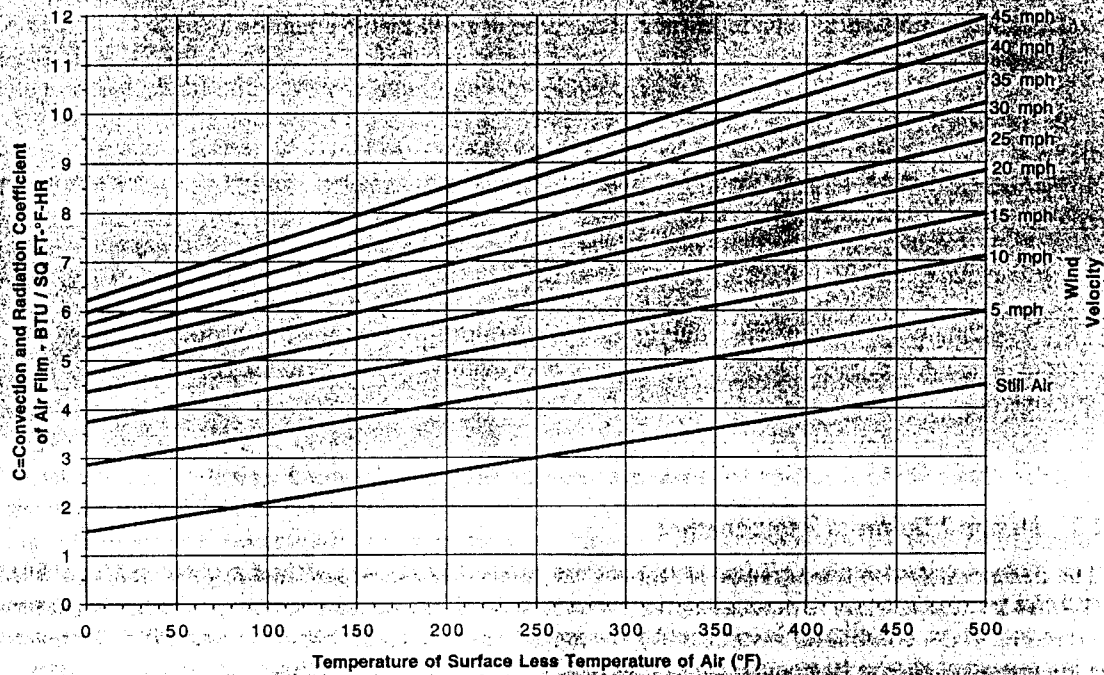


FIG. 3.1
EFFECTIVE OUTSIDE FILM COEFFICIENT

where

Q = heat flow, Btu/ft-h

ΔT = temperature difference between fluid inside the pipe and ambient temperature, °F

h_i = film coefficient on the inside of the refractory (often assumed to be very high, so the term including this in the denominator may be neglected), Btu/ft² · °F · h

h_o = film coefficient on the outside of the pipe (see Fig. 3.1), Btu/ft² · °F · h

r_1 = radius inside the refractory, in.

r_2 = radius outside of the refractory, in.

r_3 = outside radius of the pipe, in.

K_r = conductivity of the refractory material, Btu-in./ft² · °F

K_s = conductivity of the pipe material, Btu-in./ft² · °F

The change in temperature through any layer is equal to the heat flow times the resistance. Thus, the temperature on the outside surface of the pipe is given by the equation

$$T_o = \frac{Q}{2\pi r_3 h_o} + T_a \quad (3.2)$$

where

T_o = temperature on the outside surface of the pipe, °F

T_a = ambient temperature, °F

The temperature on the inside surface of the pipe is given by the equation

$$T_i = Q \frac{\ln(r_3/r_2)}{2\pi K_r} + T_o \quad (3.3)$$

where

T_i = temperature on the inside surface of the metal pipe, °F

This is normally essentially the same as the temperature on the outside surface, because the conductivity of the pipe material is generally high relative to the refractory and outside film coefficient.

3.1.3 Design Minimum Temperature

The design minimum temperature is required to determine the impact test requirements of ASME B31.3. As the temperature is decreased, the notch toughness of the material drops. Thus, impact tests are required under conditions of sufficiently low temperature and high stress to ensure that the material has adequate toughness to avoid a brittle fracture.

In establishing the minimum temperature, one must consider the lowest component temperature that can be expected in service. This can include effects such as autorefrigeration (the cooling that occurs due to depressurization) and low ambient temperatures.

With the inclusion of stress as a consideration in the need to perform impact testing, it is possible that more than one minimum temperature, each with different stress conditions, will require consideration. This is because the loads may be different at different low-temperature conditions, and lower temperatures without impact testing may be permitted at lower stress conditions.

3.2 ALLOWABLE STRESS

The Code provides allowable stresses for metallic piping covered by the base Code and Chapter VIII (Category M) based on criteria listed in para. 302.3. These are, with certain exceptions, the lowest of the following:

- One-third of the specified minimum tensile strength (which is at room temperature)
- One-third of the tensile strength at temperature
- Two-thirds of the specified minimum yield strength (which is at room temperature)
- Two-thirds of the "minimum" yield strength at temperature
- Average stress for a minimum creep rate of 0.01%/1000 hours
- Two-thirds of the average stress for creep rupture in 100,000 hours
- 80% of the minimum stress for a creep rupture in 100,000 hours

Specified values are the minimum required in the material specifications. For unlisted material, the "minimum" at temperature is required in para. 302.3.2(f) to be determined by multiplying the specified (room temperature) values by the ratio of the average strength at temperature to that at room temperature. The allowable stresses listed in the Code are determined by the ASME Boiler and Pressure Vessel Code Subcommittee II, and are based on trend curves that show the effect of temperature on yield and tensile strengths. (The trend curve provides the aforementioned ratio.) An additional factor of 1.1 is used with the tensile strength at temperature.

These paragraphs were updated in the 2000 Addendum to provide wording that is more consistent with Section II of the BPVC. However, there was no change in the allowable stress basis, even though the factor of 1.1 for tensile strength at temperature was added to the wording [in para. 301.7.3(8)], whereas it had not previously been mentioned.

An exception to the above criteria is made for austenitic stainless steel and nickel alloys with similar stress-strain behavior, for which one can use as high as 90% of the yield strength at temperature. This is due not to a desire to be less conservative, but to a recognition of the differences in behavior of these alloys. The quoted yield strength is determined by drawing a line parallel to the elastic loading curve, but with a 0.2% offset in strain. The yield strength is the intercept of this line with the stress-strain curve. Such an evaluation provides a good yield strength value of carbon steel and alloys with similar behavior, but it does not represent the strength of austenitic stainless steel, which has considerable hardening and additional strength beyond this strain value. However, the additional strength is achieved at the cost of additional deformation. Thus, the higher allowable stresses relative to yield are only applicable to components that are not deformation-sensitive. Thus, although one might use the higher allowable stress for pipe, it should not be used for flange design. The allowable stress tables use a style that lets the Code user know when the allowable stress exceeds two-thirds of the yield strength at temperature, as will be described in Section 10.8.

Use of the higher allowable stress can create some complications, because the Code has limitations in the form of stress in the piping relative to yield strength. For example, variations above what would otherwise be considered to be the maximum allowable working pressure of the piping are permitted, to as high as 33% above the allowable stress. However, the Code requires that the stress not exceed yield stress. Thus, if the pipe was designed using 90% of yield stress, the allowable variation would be much less than 33%. Similar circumstances occur for occasional loads (see para. 302.3.6; stress is limited to yield).

The allowable stress for bolting is similar, except that it is based on one-fourth of the tensile strength rather than one-third, and special consideration is given to bolting materials for which the strength has been enhanced by heat treatment or strain hardening. They are limited to one-fifth of the specified minimum tensile strength and one-fourth of the specified minimum yield strength, unless these are lower than the values for the annealed material, in which case the annealed material values of allowable stress are used.

For cast and ductile iron materials, the behavior is brittle and the allowable stress differs accordingly. For cast iron, the basic allowable stress is the lower of one-tenth of the specified minimum tensile strength (at room temperature) and one-tenth of the "minimum" strength at temperature, also based on the trend of average material strength with temperature. For ductile iron, a factor of one-fifth is used rather than a factor of one-tenth. For malleable iron the same criteria are used as for other metallic materials (e.g., carbon steel).

3.3 ALLOWANCES FOR PRESSURE AND TEMPERATURE VARIATIONS

Although the Code does not use the term "maximum allowable working pressure," the concept is useful in a discussion of the allowances for variations. Pressure design of piping systems is based on the design conditions. However, because piping systems are assemblies of standardized parts, there is quite often significant pressure capacity in the piping beyond the design conditions of the system. The allowances for variations are relative to the maximum permissible design pressure for the system. The allowances for variations are not used in sustained (longitudinal), occasional (wind, earthquake), or displacement (thermal expansion) stress evaluations. They are only used in pressure design.

Increases in pressure and temperature above the design conditions are permitted for short-term events, as long as several conditions are satisfied, one of which is that this maximum allowable working

pressure is not exceeded by more than some percentage. Thus, the variation can be much higher than the design conditions, yet remain permissible.

The following conditions are requirements for use of the variations:

- The piping system shall not have pressure-containing components of cast iron or other nonductile metal.
- The nominal pressure stress (hoop stress for straight pipe or, for rated components, the pressure divided by the allowable pressure times two-thirds the yield strength) must be less than the yield strength of the material.
- The longitudinal stresses must be within the normally permitted limits.
- The total number of pressure-temperature variations above the design conditions must be less than 1000 over the life of the system (note that this is the number anticipated in the design of the system, not some count taken during operation of the system; the ASME B31.3 Code is for design of new piping systems).
- The maximum pressure must be less than the test pressure; this can be a limitation if pneumatic or alternative leak testing was used.

If the above conditions are satisfied, and if the owner approves, the pressure rating or allowable stress (essentially the maximum allowable working pressure) may be exceeded by 33% for events that are not more than 10 hours at any one time nor more than 100 hours per year, and by 20% for events that are not more than 50 hours at any one time nor more than 500 hours per year. It is clear how a variation in pressure is handled. There is sometimes confusion relative to variations in temperature. The variation in temperature decreases the allowable stress or pressure rating. Thus, the stress or pressure may exceed the allowable value during a variation in temperature, without a change in pressure.

If the above variations are used, the designer must determine that the piping system, including the effects of the variations, is safe over the service life of the piping, using methods that are acceptable to the owner. Note that the pressure test provides such assurance for piping operating below the creep regime. For piping at elevated temperatures, within the creep regime for the material of construction, the pressure test does not ensure long-term pressure integrity. Therefore, Appendix V of ASME B31.3 was provided to evaluate the effect of short-term variations at elevated temperature on the life of the piping system.

As an alternative to the prior two paragraphs, if the variation is self-limiting, such as by accumulation during a pressure-relieving event, and lasts no more than 50 hours at any one time nor more than 500 hours per year, an allowable variation of 20% is permitted without the owner's approval and without requiring that the designer determine that the piping system is safe with the variations.

3.4 OVERPRESSURE PROTECTION

As discussed above in the section on design pressure, the piping system must either be designed to safely contain the maximum possible pressure, considering such factors as failure of control devices and dynamic events such as surge, or be provided with overpressure protection, such as a safety relief valve. This basic requirement is provided in para. 301.2.2, whereas requirements for relief valves are provided in para. 322.6, Pressure Relieving Systems.

For example, if a 600-psi system goes through a pressure-letdown valve (irrespective of fail-closed features or other safeguards) to a 300-psi system and then through another letdown valve to a 150-psi system, if no safety relief devices are provided, the 150-psi system would have to be designed to safely contain 600 psi. This is because possible failure of both valves must be considered. There is no provision for double contingency or other considerations relative to probability of occurrence in the Code requirements.

If a pressure-relieving device is used, ASME B31.3 refers to the requirements of ASME BPVC, Section VIII, Division 1. However, there are some exceptions. For example, design pressure is substituted

for maximum allowable working pressure (MAWP), because the latter term is not used in the Piping Code.

More significantly, the set pressure requirements of ASME BPVC Section VIII, Division 1, are not followed. If they were, then the variations could never realistically be permitted, because any variation in pressure above the design pressure would result in opening of the pressure relief device. Rather than limit the set pressure to the design pressure, the Piping Code allows the set pressure to be any value, as long as the maximum pressure during the relieving event, including consideration of potential accumulation (additional pressure buildup beyond the pressure at which the valve opens), does not exceed one of the following two alternatives a) the maximum relieving pressure permitted by BPVC Section VIII, Division 1, or b) the maximum pressure permitted in the allowances for variations provisions of ASME B31.3. However, the owner's approval is required for the set pressure to exceed the design pressure.

For liquid thermal expansion relief devices, the set pressure may simply be established at 120% of the design pressure, without any requirement for owner's approval nor any need to comply with the additional requirements necessary to use the allowances for variations. The set pressure can be set at higher than 120%; however, then the other requirements of justifications and approvals explained above must be satisfied.

Block valves are permitted in the inlet and discharge lines to the relief valve, with certain limitations. These requirements are outlined in para. 322.6.1.

PRESSURE DESIGN

4.1 METHODS FOR INTERNAL PRESSURE DESIGN

The ASME B31.3 Code provides four basic methods for the design of components for internal pressure, as described in para. 302.2:

(a) Components in accordance with listed standards for which pressure ratings are provided in the standard, such as ASME B16.5 for flanges, are considered suitable by ASME B31.3 for the pressure rating specified in the standard. Note that the other methods of pressure design provided in ASME B31.3 can be used to rerate such listed components and/or extend their temperature range.

(b) Some listed standards, such as ASME B16.9 for pipe fittings, state that the fitting has the same pressure rating as matching seamless pipe. ASME B31.3 modifies this slightly by stating that the fittings are accepted to have the same rating as the matching seamless pipe, considering only 87.5% of the wall thickness (removing the 87.5% allowance is presently under consideration). This takes into consideration the typical mill tolerance for pipe. Note that design calculations are not usually performed for these components; design calculations are performed for the straight pipe, and matching fittings are simply selected.

(c) Design equations for some components, such as straight pipe and branch connections, are provided in para. 304 of ASME B31.3. These can be used to determine the required wall thickness with respect to internal pressure of components. Furthermore, some specific branch connection designs are assumed to be acceptable.

(d) Components that are not in accordance with a listed standard and for which design equations are not provided in the Code are treated in para. 304.7.2. This paragraph provides accepted methods, such as burst testing, for determining the pressure capacity of unlisted components.

The equations in the Code provide the minimum thickness required to limit the membrane, and in some cases bending stresses in the piping component, to the appropriate allowable stress. Mechanical and corrosion/erosion allowances must be added to this thickness. Finally, the nominal thickness selected must be such that the minimum thickness that may be provided, per specifications and considering mill tolerance, is at least equal to the required minimum thickness.

Mechanical tolerances include physical reductions in wall thickness, such as due to threading and grooving the pipe. Corrosion and erosion allowances are based on the anticipated corrosion and/or erosion over the life of the pipe. This is based on estimates, experience, or literature, such as National Association of Corrosion Engineers (NACE) publications. These allowances are added to the pressure design thickness to determine the minimum required thickness of the pipe or component when it is new.

For threaded components, the nominal thread depth (dimension h of ASME B1.20.1; see Appendix I or equivalent) is used for the mechanical allowance. For machined surfaces or grooves where the tolerance is not specified, the Code requires that a tolerance of 0.5 mm (0.02 in.) on the depth of the cut be assumed.

Mill tolerances are provided in specifications. The most common tolerance on the wall thickness of straight pipe is 12.5%. This means that the wall thickness at any given location around the circumference of the pipe must not be less than 87.5% of the nominal wall thickness. Note that the tolerance on pipe weight is typically tighter, so that the volume of metal and its weight may be present although a thin region would control design for hoop stress due to internal pressure.

The appropriate specification for the pipe must be referred to in order to determine the specified mill tolerance. For example, plate typically has an undertolerance of 0.25 mm (0.01 in.). However, pipe formed from plate does not have this undertolerance; it can be much greater. The pipe specification, which can permit a greater undertolerance, governs for the pipe. The manufacturer of pipe can order plate that is thinner than the nominal wall thickness for manufacturing the pipe, as long as the pipe specification mill tolerances are satisfied. However, the weight tolerance could then govern. For example, the thickness tolerance for A53 pipe is 12.5%, but the weight tolerance is 10%. As a result, the minimum thickness for welded pipe made from plate material would be 10% under thickness because of the weight tolerance.

4.2 PRESSURE DESIGN OF STRAIGHT PIPE FOR INTERNAL PRESSURE

Equations for pressure design of straight pipe are provided in para. 304.1. The minimum thickness of the pipe selected, considering manufacturer's minus tolerance, must be at least equal to t_m , defined as¹

$$t_m = t + c \quad (4.1)$$

where

c = sum of the mechanical allowances plus corrosion and erosion allowances

t = pressure design thickness

t_m = minimum required thickness including allowances

For pipe with $t < D/6$, the basic equation for determining pressure design thickness is provided in the Code,²

$$t = \frac{PD}{2(SE + PY)} \quad (4.2)$$

where

D = pipe outside diameter (not nominal diameter)

E = quality factor

P = internal design gage pressure

S = allowable stress value

Y = coefficient provided in Table 304.1.1 of the Code and Table 4.1 here

Note that Eq. (4.2) is based on the outside diameter, rather than the inside diameter, which is used in Pressure Vessel Codes. This is for a very good reason: The outside diameter of pipe is independent of wall thickness. That is, an NPS 6 pipe will have an outside diameter of 6.625 in., regardless of the

¹ASME B31.3, Eq. (2).

²ASME B31.3, Eq. (3a).

TABLE 4.1
VALUES OF COEFFICIENT Y FOR $t < D/6$

TABLES OF COEFFICIENT γ FOR $t \leq D/6$						
Material	Temperature, °C (°F)					
	≤482 (900 & lower)	510 (950)	538 (1000)	566 (1050)	593 (1100)	≥621 (1150 & up)
Ferritic steels	0.4	0.5	0.7	0.7	0.7	0.7
Austenitic steels	0.4	0.4	0.4	0.4	0.5	0.7
Other ductile metals	0.4	0.4	0.4	0.4	0.4	0.4
Cast iron	0.0	...	...	...	...	...

wall thickness. Therefore, the wall thickness can be directly calculated when the outside diameter is used in the equation.

Equation (4.2) is an empirical approximation of the more accurate and complex Lamé equation (ca. 1833). The hoop or circumferential stress is higher toward the inside of the pipe than toward the outside. This stress distribution is illustrated in Fig. 4.1. The Lamé equation, provided below, can be used to calculate the stress as a function of location through the wall thickness. Equation (4.2) is the Boardman equation. Although it has no theoretical basis, it provides a good match to the more accurate and complex Lamé equation for a wide range of diameter-to-thickness ratios. It becomes increasingly conservative for lower D/t ratios (thicker pipe) if Y is held constant.

The Lamé equation for hoop stress on the inside surface of pipe follows. Note that for internal pressure, the stress is higher on the inside than the outside. This is because strain in the longitudinal direction of the pipe must be constant through the thickness, so that any longitudinal strain caused by the compressive radial stress (due to Poisson effects and considering that the radial stress on the inside surface is equal to the surface traction of internal pressure) must be offset by a corresponding increase in hoop tensile stress to cause an offsetting Poisson effect on longitudinal strain. The Lamé equation is

$$\sigma_h = P \frac{0.5(D/t)^2 - (D/t) + 1}{(D/t) - 1} \quad (4.3)$$

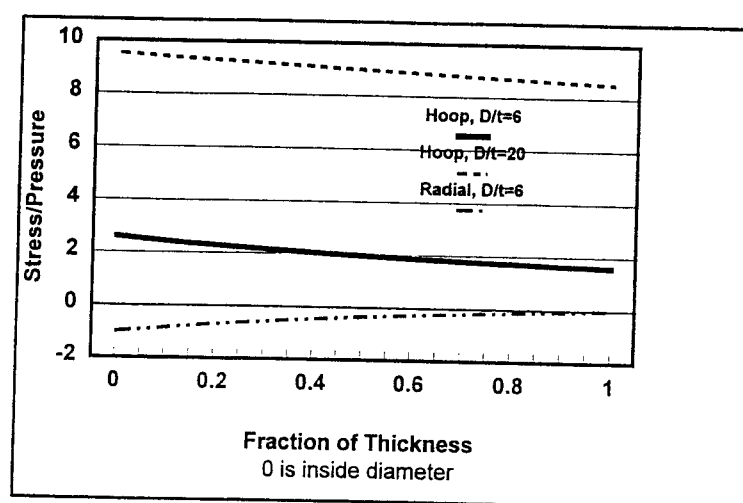


FIG. 4.1
STRESS DISTRIBUTION THROUGH PIPE WALL THICKNESS DUE TO
INTERNAL PRESSURE

where

σ_h = hoop stress

The empirical Boardman representation of this simply bases the calculation of pressure stress on some intermediate diameter, between the inside and outside diameters of the pipe, as

$$\sigma_h = P \frac{D - 2Yt}{2t} \quad SE = PD \cdot 2PYt \quad (4.4)$$

where

$$Y = 0.4 \quad (4.5)$$

Simple rearrangement of the above equation and substitution of SE for σ_h leads to Eq. (3a) of the code [Eq. (4.2) here]. Further, inside-diameter-based formulas add 0.6 times the thickness to the inside radius of the pipe rather than subtract 0.4 times the thickness from the outside radius. Thus, the inside-diameter-based formula in the Pressure Vessels Codes and Eqs. (3a) and (3b) of ASME B31.3, the Piping Code, are consistent. The additional consideration in Eq. (3b) of ASME B31.3 is the addition of the allowances (internal corrosion increases the inside diameter in the corroded condition). With this additional consideration, Eq. (3b) of ASME B31.3 based on inside diameter provides the same required thickness as Eq. (3a) based on outside diameter.

A comparison of hoop stress calculated using the Lamé equation versus the Boardman equation (4.2) is provided in Fig. 4.2. Remarkably, the deviation of the Boardman equation from the Lamé equation is less than 1% for D/t ratios greater than 5.1. Thus, the Boardman equation can be directly substituted for the more complex Lamé equation.

For thicker wall pipe, ASME B31.3 provides the following equation for the calculation of the Y factor in the definition of Y in para. 304.1.1. Use of this equation to calculate Y results in Eq. (4.2), matching the Lamé equation for heavy wall pipe as well:

$$Y = \frac{d + 2c}{D + d + 2c} \quad (4.6)$$

The factor Y depends on temperature. At elevated temperatures, when creep effects become significant, creep leads to a more even distribution of stress across the pipe wall thickness. Thus, the factor Y increases, leading to a decrease in the calculated required wall thickness (for a constant allowable stress).

Three additional equations were formerly provided by the Code, but two were removed to be consistent with ASME B31.1 and simplify the Code. They may continue to be used. The first of the removed equations is

$$t = \frac{PD}{2SE} \quad (4.7)$$

This equation is the simple Barlow equation, which is based on the outside diameter and is always conservative. It may be used, because it is always more conservative than the Boardman equation, which is based on a smaller diameter (except when $Y = 0$). The second removed equation is

$$t = \frac{D}{2} \left(1 - \sqrt{\frac{SE - P}{SE + P}} \right) \quad (4.8)$$

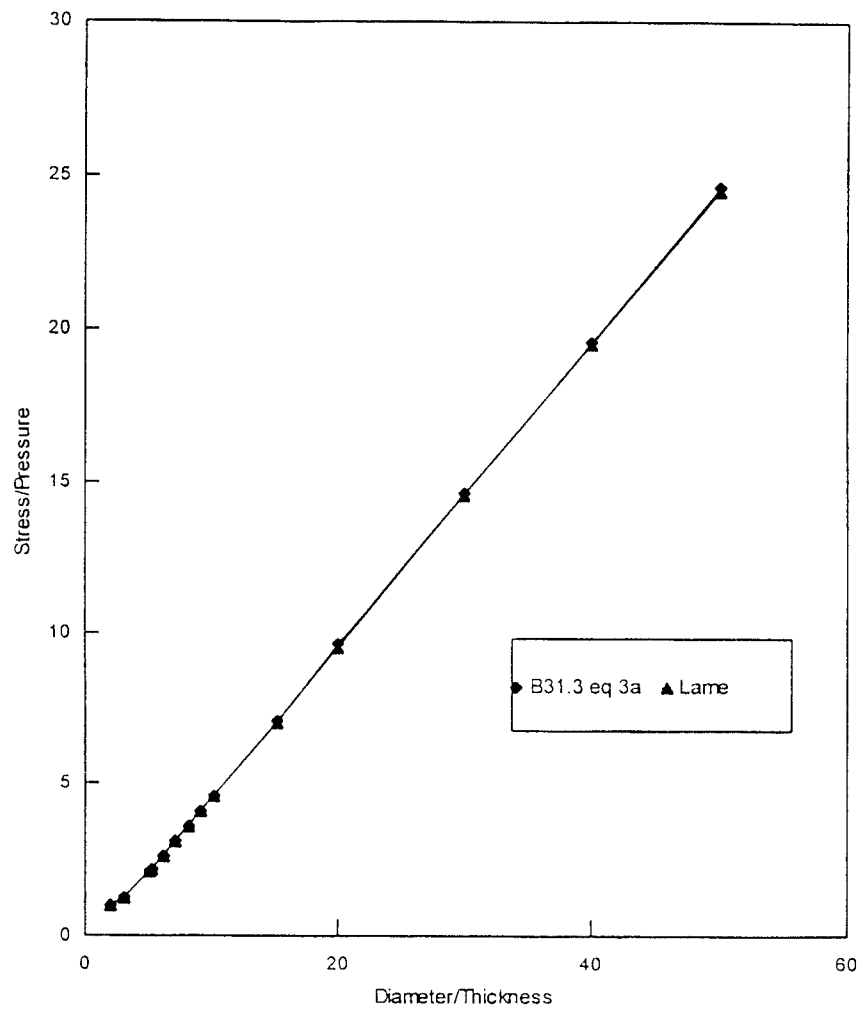


FIG. 4.2
COMPARISON OF LAMÉ AND BOARDMAN EQUATIONS

This equation is the Lamé equation rearranged to calculate thickness. Although it is not specifically included, it could be used, in accord with para. 300(c)3. However, it should not make a significant difference in the calculated wall thickness.

The following optional equation remains in ASME B31.3³:

$$t = \frac{P(d + 2c)}{2[SE - P(1 - Y)]} \quad (4.9)$$

where

d = inside diameter

³ASME B31.3, Eq. (3b).

Equation (4.9) is the same as (4.2), but with $(d + 2c + 2t)$ substituted for D and rearranged to keep thickness on the left side.

Insert 4.1 Sample Wall Thickness Calculation

What is the required thickness of NPS 2 threaded A53 Grade B seamless pipe for the following conditions?:

- Design pressure = 150 psi
- Design temperature = 500°F
- Corrosion allowance (CA) = 1/16 in.
- $SE = 18,900$ psi
- $D = 2.375$ in.
- $t = 150(2.375)/[2(18,900 + 0.4 \times 150)] = 0.0094$ in.
- $c = CA + \text{thread depth} = 0.0625 + 0.07$ in.
- $t_m = 0.0094 + 0.0625 + 0.07 = 0.14$ in.

$$t = \frac{PD}{2(SE + Pt)} = \frac{150 \times 2.375}{2(18,900 + 150 \times 0.4)} = 0.0094 \text{ in.}$$

The minimum nominal pipe thickness, considering mill tolerance, is

$$0.875 \bar{T}_{min} = t_m$$

$$\bar{T}_{min} = \frac{t_m}{0.875} = 0.16 \text{ in.} \quad (4.10)$$

Schedule 80, XS pipe, with a nominal wall thickness of 0.218 in. is acceptable.

Insert 4.2 Basic Stress Calculations for Cylinders Under Pressure

The average (through-thickness) circumferential and longitudinal (axial) stresses in a cylinder due to internal pressure can be calculated from equilibrium considerations. The circumferential stresses can be calculated from a longitudinal section, as shown in Fig. 4.3. The forces acting on that section must equilibrate, or, per Newton's law, the parts on either side of the section will start accelerating away from each other.

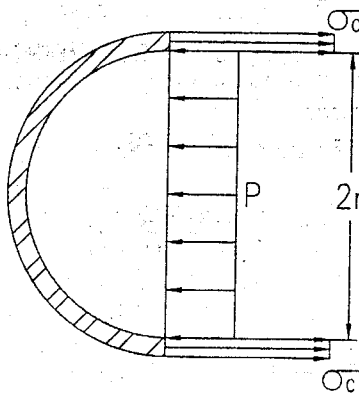


FIG. 4.3
EQUILIBRIUM AT A CIRCUMFERENTIAL CUT

The pressure force acting on the section is $2rP$, where r is the inside radius and P is the internal pressure. The circumferential force in the pipe wall resisting the pressure force is $2t\sigma_c$, where t is the thickness (times two, because there are two sides), and σ_c is the average circumferential stress. These must be equal, and solving for σ_c , one arrives at the following equation:

$$\sigma_c = \frac{Pr}{t} \quad (4.11)$$

The longitudinal stress can be determined by making a girth cut (as a guillotine cut) on the pipe, as shown in Fig. 4.4. The pressure force acting on the section is $\pi r^2 P$ and the longitudinal force in the pipe wall resisting this pressure force is $2\pi r t \sigma_\ell$. Note that, to be more precise, the mean radius of the pipe should be used to calculate the area of the pipe wall, but using the inside radius is generally close enough and conservative. The longitudinal stress can be calculated by equating these two forces and solving for σ_ℓ :

$$\sigma_\ell = \frac{Pr}{2t} \quad (4.12)$$

Thus, the longitudinal stress in a cylinder due to internal pressure is about one-half of the circumferential stress. This is quite convenient in the design of piping, because the wall thickness is determined based on pressure design. This leaves at least one-half of the strength in the longitudinal direction available for supporting the pipe weight.

A common example of the fact that the stress in the circumferential direction is twice that in the longitudinal direction can be found when cooking a hot dog. A hot dog has a pressure-containing skin. When the internal temperature reaches the point where the fluids contained inside begin to vaporize, the hot dog skin is pressurized. When the skin is overpressurized and fails, the split is always longitudinal, transverse to the direction of highest stress, the circumferential direction. Hot dogs, at least in the experience of this author, never experience guillotine failures during cooking.

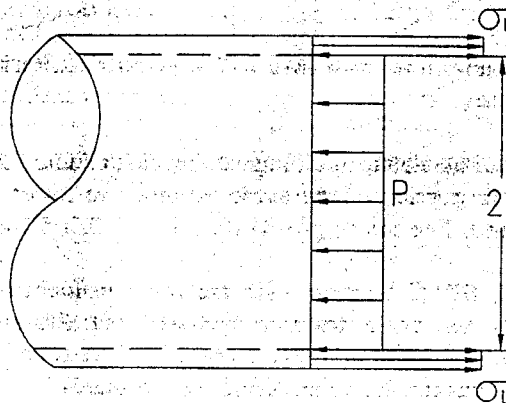


FIG. 4.4
EQUILIBRIUM AT A LONGITUDINAL CUT

4.3 PRESSURE DESIGN OF STRAIGHT PIPE UNDER EXTERNAL PRESSURE

For straight pipe under external pressure, there is a membrane stress check in accordance with Eq. (3a) [or (3b)] of ASME B31.3 [the equation for internal pressure; Eq. (4.2) or Eq. (4.9) here], as well as a buckling check in accordance with the external pressure design rules of ASME BPVC, Section VIII, Division 1.

Flanges, heads, and stiffeners that comply with ASME BPVC, Section VIII, Division 1, para. UG-29 are considered stiffeners. The length between stiffeners is the length between such components. The buckling pressure is a function of geometrical parameters and material properties.

Buckling pressure calculations in ASME BPVC, Section VIII, Division 1 require first the calculation of a parameter A , which is a function of geometry, and then a parameter B , which depends on A and a material property curve. The charts that provide the parameter B account for plasticity between the proportional limit of the stress-strain curve and the 0.2% offset yield stress. The chart for determination of A is provided in Fig. 4.5. A typical chart for B is provided in Fig. 4.6.

Two equations are provided for calculating the maximum permissible external pressure. The first uses the parameter B ,

$$p = \frac{4B}{3D/t} \quad (4.13)$$

where

- B = parameter from material curves in ASME BPVC, Section II, Part D, Subpart 3
- D = inside diameter (note that this differs from the ASME B31.3 definition of D , which is outside diameter; this is the Pressure Vessel Code definition of D)
- p = allowable external pressure
- t = pressure design thickness

The second equation is for elastic buckling, and must be used when the value of A falls to the left of the material property curves that provide B . This equation is

$$p = \frac{4AE}{3} \quad (4.14)$$

where

- A = parameter from geometry curves in ASME BPVC Section II, Part D, Subpart 3, Fig. G (see Fig. 4.5 here)

The second equation is based on elastic buckling, so the elastic modulus is used. A chart of B could be used, with the linear elastic portion of the curve extended to lower values of B , but this would unnecessarily enlarge the charts. The charts provided in ASME B31.5 have this form, with the elastic lines extended.

The procedures of ASME BPVC Section VIII include consideration of the allowable out-of-roundness in pressure vessels, and use a design margin of three. Although pipe is not generally required to comply with the same out-of-roundness tolerance as is required for pressure vessels, this has historically been ignored, and has not led to any apparent problems.

The basis for the approach of ASME BPVC Section VIII is provided in Bergman (1960), Holt (1960), Saunders and Windenburg (1960), Windenburg and Trilling (1960), and Windenburg (1960).

A new buckling evaluation procedure, provided in Code Case 2286, is more relevant to piping, because it permits consideration of combined loads, including external pressure, axial load, and

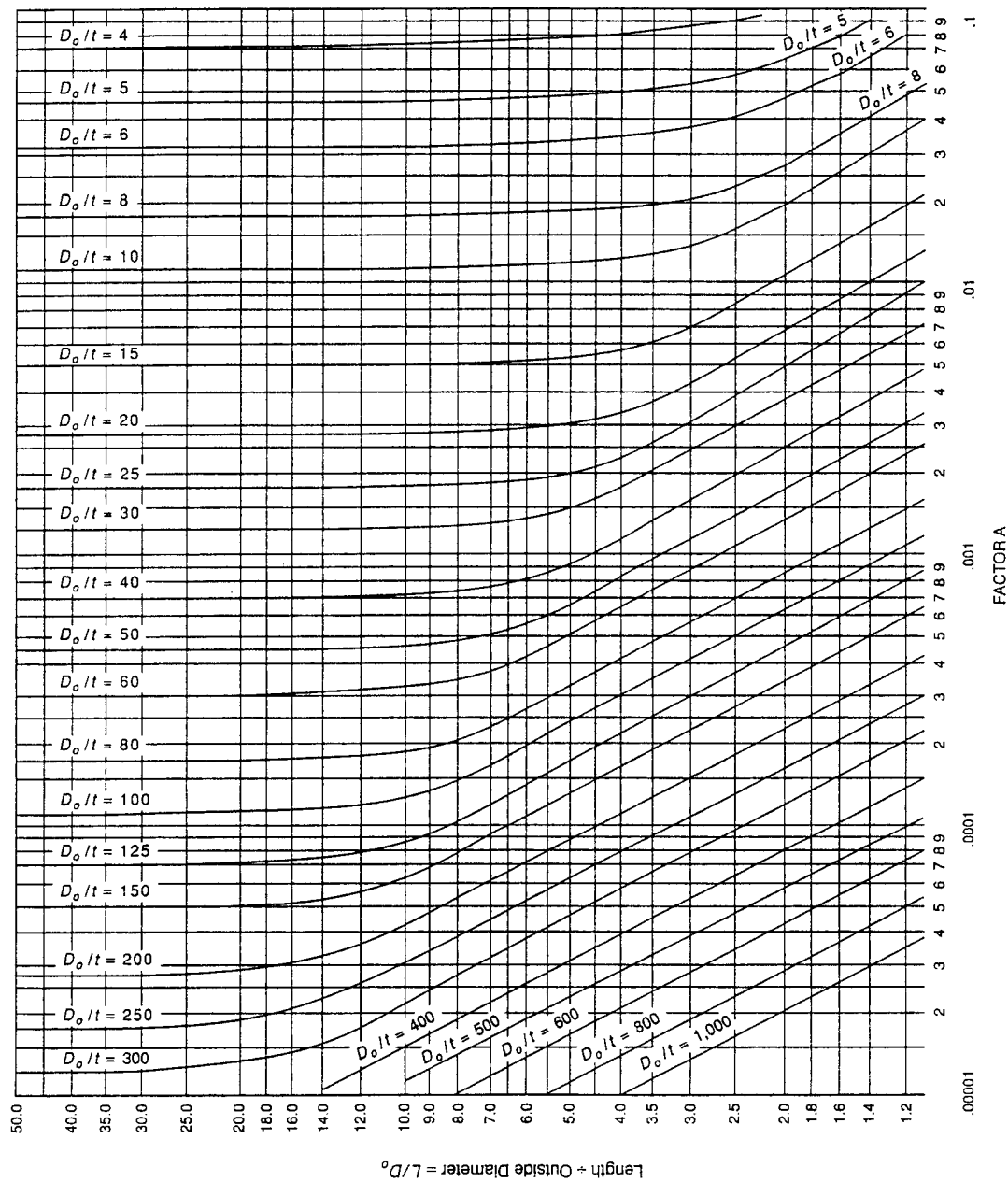


FIG. 4.5
 CHART FOR DETERMINING A (ASME BPVC, SECTION II, PART D, SUBPART 3, FIG. G). TABLE G CITED IN THE FIGURE IS GIVEN
 IN ASME BPVC, SECTION II

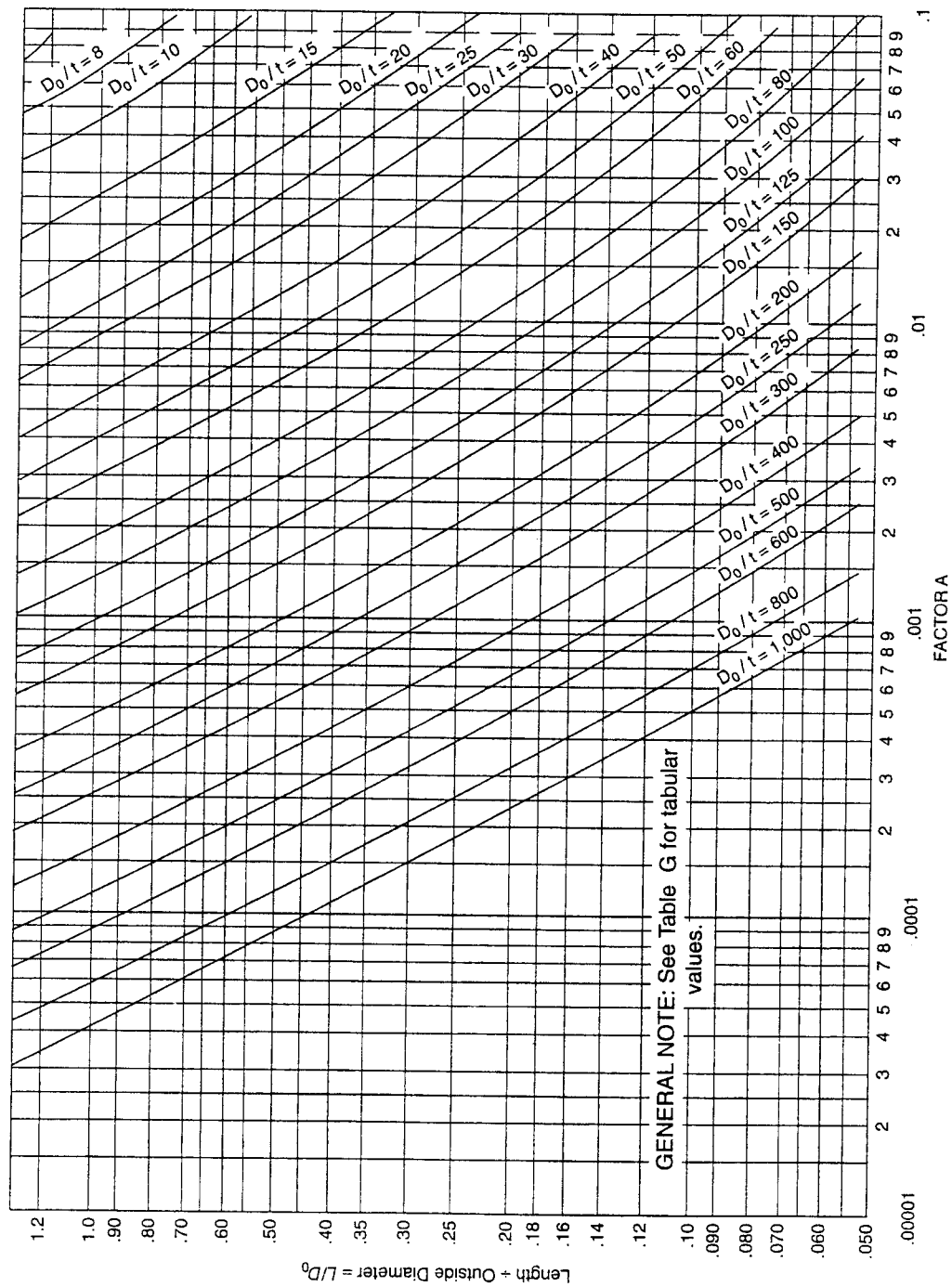


FIG. 4.5
CONTINUED

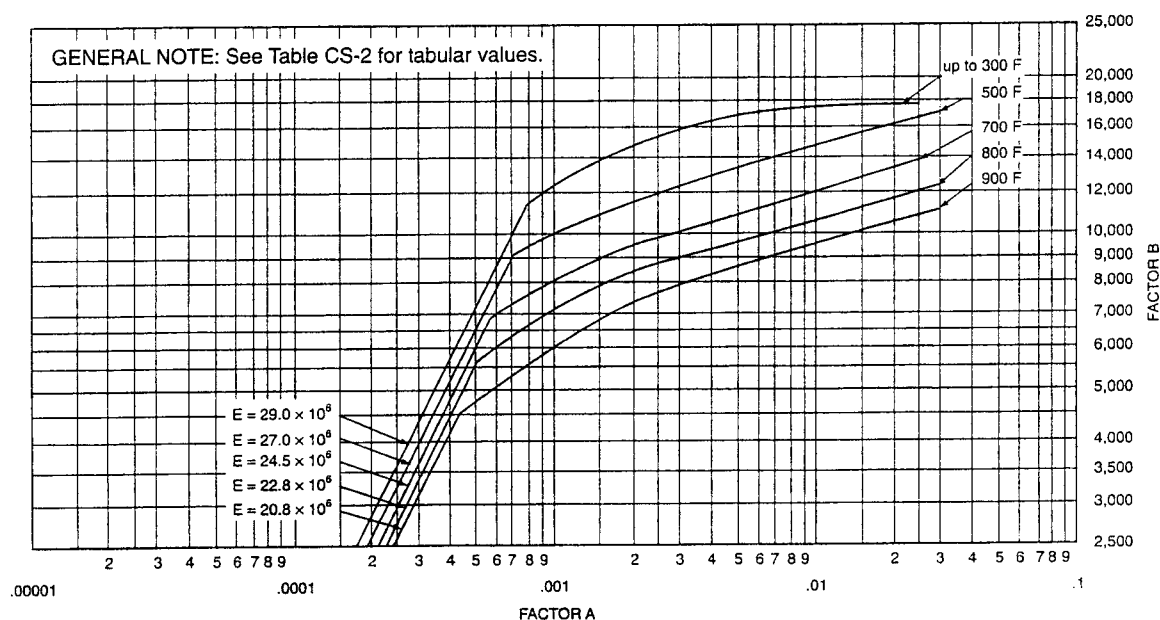


FIG. 4.6

TYPICAL CHART FOR DETERMINING B (ASME BPVC, SECTION II, PART D, SUBPART 3, FIG. CS-2). TABLE CS-2 CITED IN THE FIGURE IS GIVEN IN ASME BPVC, SECTION II

gross bending moment. Code Case 2286 also provides a more uniform margin of safety and a higher allowable external pressure. It is not at present explicitly recognized in ASME B31.3, but could be considered, given para. 300(c)3.

4.4 PRESSURE DESIGN OF WELDED BRANCH CONNECTIONS

The pressure design of branch connections is based on a rather simple approach, although the resulting design calculations are the most complex of the design-by-formula approaches for pressure design provided in the Code. A branch connection cuts a hole in the run pipe. The metal removed is no longer available to carry the forces due to internal pressure. An area replacement concept is used for those branch connections that do not either comply with listed standards or with certain designs (see Section 4.7). The area of metal removed by cutting the hole, to the extent that it was required for internal pressure, must be replaced by extra metal in a region around the branch connection. This region is within the limits of reinforcement, defined later.

The simplified design approach is limited with respect to the geometries to which it is considered applicable. These limitations are as follows:

- The run pipe diameter-to-thickness ratio is less than 100.
- The branch-to-run diameter ratio is not greater than one.
- The angle β (angle between branch and run pipe axes) is at least 45° .
- The axis of the branch intersects the axis of the run pipe.

Where the above limitations are not satisfied, the designer is referred to para. 304.7.2 (see Section 4.15). Alternatives in that paragraph include proof testing and finite-element analysis. Paragraph 304.3.5(e) suggests consideration of integral reinforcement or complete encirclement reinforcement for such branch connections.

The area A_1 of metal removed is defined as

$$A_1 = t_h d_1 (2 - \sin \beta) \quad (4.15)$$

where

- d_1 = effective length removed from the run pipe at the branch connection
- t_h = pressure design thickness of the header
- β = the smaller angle between axes of branch and run

For a 90 deg. branch connection, d_1 is effectively, the largest possible inside diameter of the branch pipe; the inside diameter of the pipe if fully corroded and with the full mill tolerance removed from the inside of the pipe.

Figure 4.7 illustrates the nomenclature and process.

The angle β is used in the evaluation because a lateral connection, a branch connection with a β other than 90°, creates a larger hole in the run pipe. This larger hole must be considered in d_1 . For a lateral, d_1 is the branch pipe inside diameter, considering mill tolerance and corrosion/erosion allowance, divided by $\sin \beta$. The $(2 - \sin \beta)$ term is used to provide additional reinforcement that is considered to be appropriate because of the geometry of the branch connection.

The pressure design thickness is the pressure design thickness of the run pipe, with one exception. If the run pipe is welded and the branch does not intersect the weld, the weld quality factor E should not be used in calculating the wall thickness. The weld quality factor only reduces the allowable stress at the location of the weld.

Only the pressure design thickness is used in calculating the required area since only the pressure design thickness is required to resist internal pressure. Corrosion allowance and mill tolerance at the hole are obviously of no consequence.

The area removed, A_1 , must be replaced by available area around the opening. This area is available from excess wall thickness that may be available in the branch and run pipes as well as added reinforcement, and the fillet welds that attach the added reinforcement. This metal must be relatively close to the opening of the run pipe to reinforce it. Thus, the metal must be within a certain limited area in order to be considered appropriate reinforcement of the opening.

The limit of reinforcement along the run pipe, taken as a dimension from the centerline of the branch pipe where it intersects the run pipe wall, is d_2 , defined by

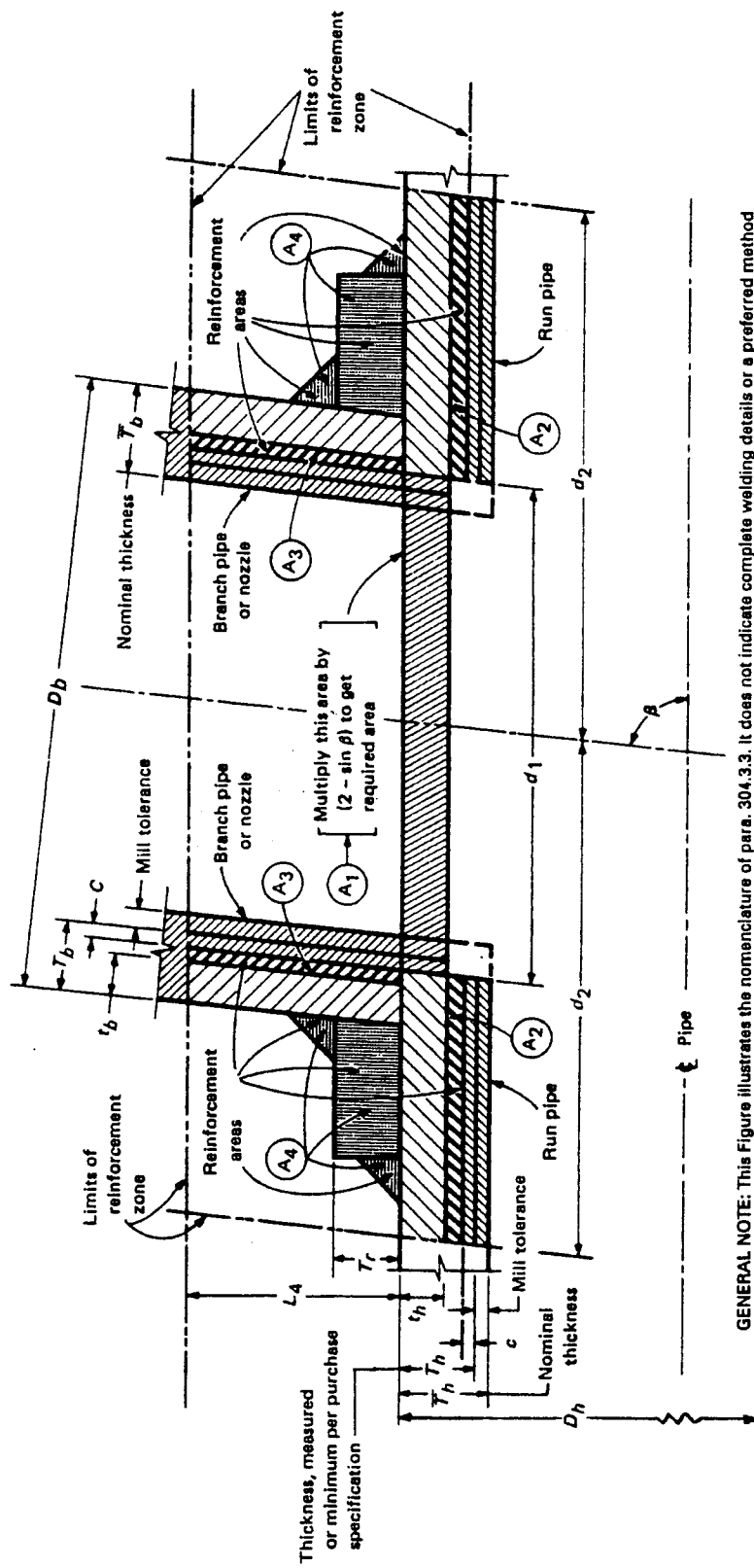
$$d_2 = \text{greater of } [d_1, (T_b - c) + (T_h - c) + d_1/2] \quad (4.16) \\ \text{but } d_2 \leq D_h$$

where

- D_h = outside diameter of header or run pipe
- T_b = minimum thickness of branch pipe
- T_h = minimum thickness of run pipe
- c = allowance (mechanical, corrosion, erosion)

The limit of reinforcement along the branch pipe measured from the outside surface of the run pipe is L_4 , defined as the lesser of $2.5 (T_h - c)$ and $2.5 (T_b - c) + T_r$, where T_r is the minimum thickness of reinforcement.

The reinforcement within this zone is required to exceed A_1 . This reinforcement consists of excess thickness available in the run pipe, A_2 , excess thickness available in the branch pipe, A_3 , and additional



GENERAL NOTE: This Figure illustrates the nomenclature of para. 304.3.3. It does not indicate complete welding details or a preferred method of construction. For typical weld details, see Fig. 328.5.4D.

FIG. 4.7
ILLUSTRATION OF FABRICATED BRANCH CONNECTION SHOWING NOMENCLATURE (ASME B31.3, FIG. 304.3.3)

reinforcement, A_4 . These can be calculated as follows:

$$A_2 = (2d_2 - d_1)(T_h - t_h - c) \quad (4.17)$$

$$A_3 = 2L_4(T_b - t_b - c)/\sin \beta \quad (4.18)$$

The area A_4 is the area of properly attached reinforcement and the welds that are within the limits of reinforcement. The Code specifies minimum weld sizes in para. 328.5.4. The designer is directed to assume that the minimum dimensions specified by the Code are provided, unless the welder is specifically directed to make larger welds. The ASME B31.3 Code does not require the designer to specify branch connection weld size, because generally acceptable minimum sizes are specified by the Code. Additionally, the ASME B31.3 Code differs from the Pressure Vessel Code in that strength calculations for load paths through the weld joints are not required.

4.5 PRESSURE DESIGN OF EXTRUDED OUTLET HEADER

An extruded outlet header is a branch connection formed by extrusion, using a die or dies to control the radii of the extrusion. Paragraph 304.3.4 provides area replacement rules for such connections; they are applicable for 90-deg. branch connections where the branch pipe centerline intercepts the run pipe centerline, and where there is no additional reinforcement. Figure 4.8 (ASME B31.3, Fig. 304.3.4) shows the geometry of an extruded outlet header. Extruded outlet headers are subject to minimum external contour radius requirements, depending on the diameter of the branch connection.

A similar area replacement calculation is used as described in Section 4.4 for fabricated branch connections, except that the required replacement area is reduced for smaller branch-to-run pipe diameter ratios. The replacement area is from additional metal in the branch pipe, additional metal in the run pipe, and additional metal in the extruded outlet lip.

4.6 ADDITIONAL CONSIDERATIONS FOR BRANCH CONNECTIONS UNDER EXTERNAL PRESSURE

Branch connections under external pressure are covered in para. 304.3.6. The same rules described in Sections 4.4 and 4.5 are used. However, only one-half of the area described in Section 4.4, covering welded branch connections, requires replacement. In other words, only one-half of the area A_1 requires replacement. Also, the thicknesses used in the calculation are the required thicknesses for the external pressure condition.

4.7 BRANCH CONNECTIONS THAT ARE PRESUMED TO BE ACCEPTABLE

Some specific types of branch connections are presumed to be acceptable. This, of course, includes listed fittings (e.g., ASME B16.9 tees and MSS SP-97 branch connection fittings). It also includes the following:

- For branch connections DN 50 (NPS 2) or less that do not exceed one-fourth of the nominal size of the run pipe, threaded or socket welding couplings or half-couplings (Class 2000 or greater) are presumed to provide sufficient reinforcement as long as the minimum thickness of the coupling is at least as thick as the branch pipe.
- Branch connection fittings qualified per para. 304.7.2 are acceptable.

4.8 PRESSURE DESIGN OF BENDS AND ELBOWS

Bends were required to have, after bending, a wall thickness at least equal to the minimum required wall thickness for straight pipe in para. 304.2.1. However, this was changed in the 2000 Addendum. The Lorenz equation (ca. 1910) was included; it provides a means of calculating the required wall thickness. Note that the prior requirement that simply stated the thickness should be the same as required for straight pipe was deleted. The new requirement is more conservative for the intrados (inside curves) of bends and less conservative for the extrados (outside curve).

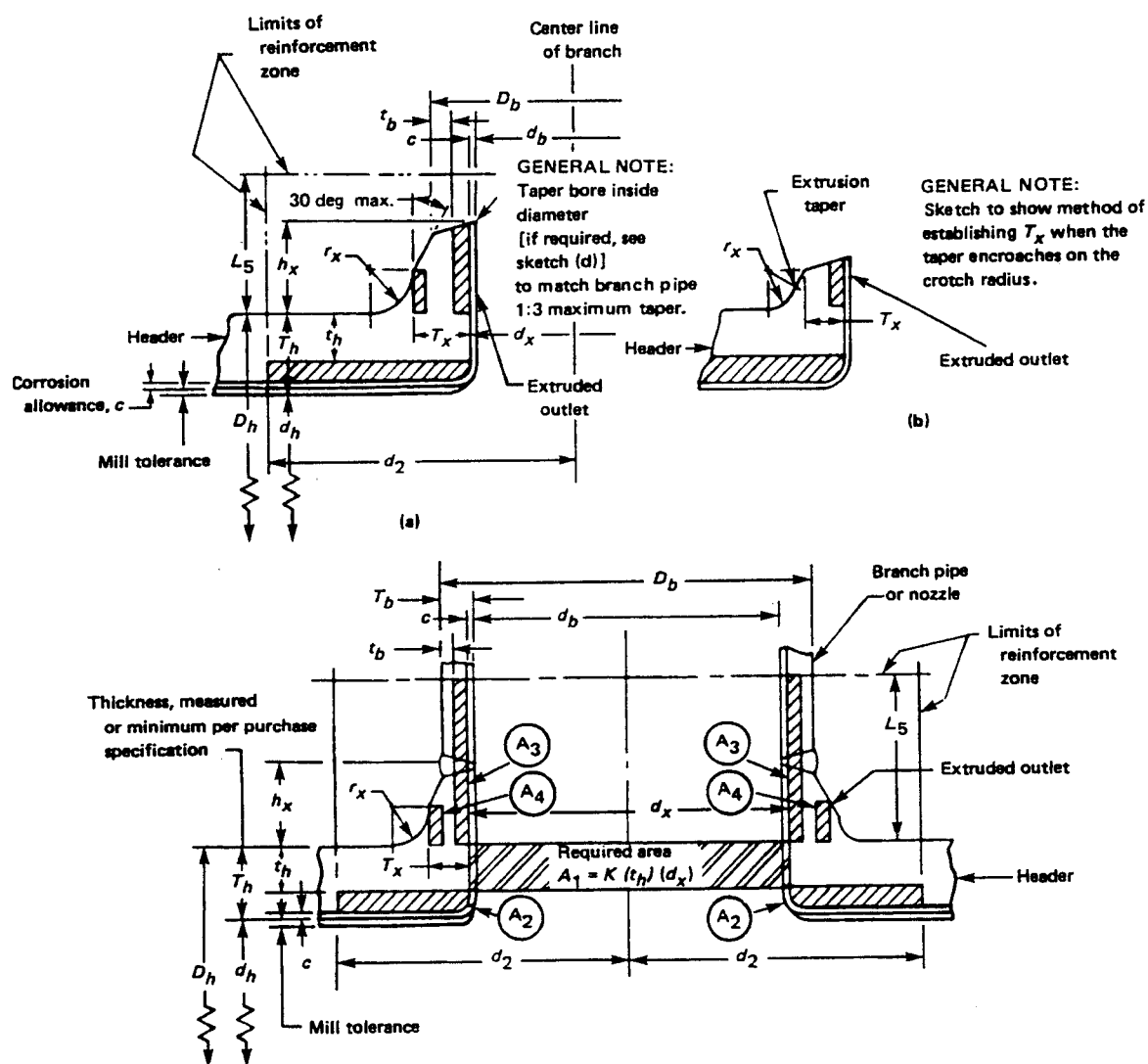


FIG. 4.8

ILLUSTRATION OF EXTRUDED OUTLET FITTING SHOWING NOMENCLATURE (ASME B31.3, FIG. 304.3.4). THIS FIGURE ILLUSTRATES THE NOMENCLATURE OF PARA. 304.3.4. IT DOES NOT INDICATE COMPLETE DETAILS OR A PREFERRED METHOD OF CONSTRUCTION

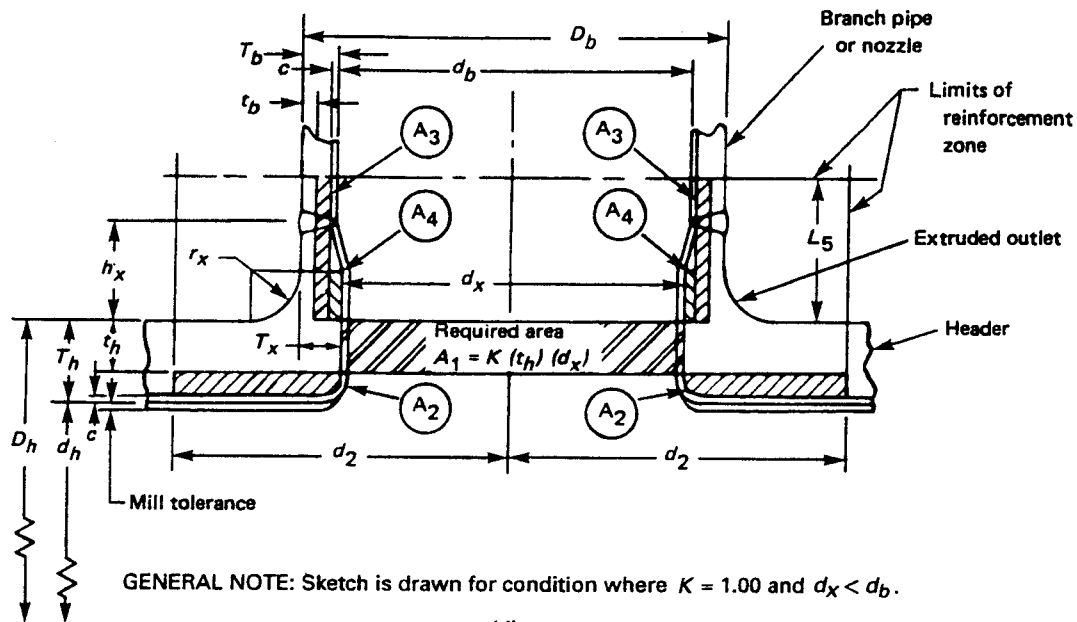


FIG. 4.8
CONTINUED

The Lorenz equation is basically the equation for a toroid. If the intrados and extrados had the same wall thickness, the inside would be subjected to higher hoop stress than straight pipe and the outside would be subjected to lower hoop stress than straight pipe. A simple way to envision this is that the inside has less metal over the curve and the outside has more metal over the curve. The Lorenz equation for an elbow or bend is given by⁴

$$t = \frac{PD}{2(SE/I + PY)} \quad (4.19)$$

where the terms are as defined in Section 4.2 for Eq. (4.2), except for I , which is a stress index that accounts for the difference in hoop stress due to internal pressure in bends versus straight pipe.

On the inside curve of the bend, the intrados, we have⁵

$$I = \frac{4R_1/D - 1}{4R_1/D - 2} \quad (4.20)$$

where

R_1 = radius of bend

⁴ ASME B31.3, Eq. (3c).

⁵ ASME B31.3, Eq. (3d).

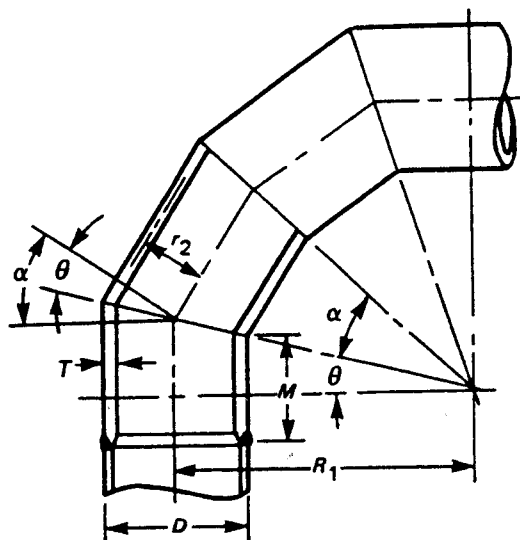


FIG. 4.9
ILLUSTRATION OF MITER BEND SHOWING NOMENCLATURE (ASME B31.3, FIG. 304.2.3)

On the outside of the bend, or the extrados, we have⁶

$$I = \frac{4R_1/D + 1}{4R_1/D + 2} \quad (4.21)$$

On the side of the elbow, or the crown, $I = 1.0$ (i.e., the hoop stress is the same as in straight pipe).

The thickness variation from the intrados to the extrados is required to be gradual, and the requirements are stated to apply at the midspan of the bend. The thickness at the ends is required to satisfy the required thickness for straight pipe per para. 304.1.

The normal process of making a bend by bending straight pipe produces this type of thickness variation. Part of the reason for providing these new rules is due to the practice of fabricating elbows by forming two “clamshells” out of plate and welding them together. This produces a bend of uniform thickness, and the thickness on the intrados would be too thin if it simply satisfied the required thickness for straight pipe.

Elbows in accordance with listed standards, or qualified by para. 304.7.2, are also permitted.

4.9 PRESSURE DESIGN OF MITER BENDS

Miter changes in direction with an angular offset of 3 deg. or less (angle α in Fig. 4.9) do not require design consideration as a miter bend. The required wall thickness for this condition is the same as straight pipe. Design equations for multiple and single miter bends follow.

The design equation for determining the required wall thickness for internal pressure for multiple miter bends is the lesser of the following two equations, 4.22 and 4.23. Note that these equations are

⁶ASME B31.3, Eq. (3e).

only valid when the angle θ does not exceed 22.5 deg. Equation 4c applies for angles θ greater than 22.5 deg.^{7,8}

$$P_m = \frac{SE(T-c)}{r_2} \left(\frac{T-c}{(T-c) + 0.643 \tan \theta \sqrt{r_2(T-c)}} \right) \quad (4.22)$$

$$P_m = \frac{SE(T-c)}{r_2} \left(\frac{R_1 - r_2}{R_1 - 0.5r_2} \right) \quad (4.23)$$

Single miters with an angle θ not greater than 22.5 deg. are calculated per equation 4.22. Otherwise, they are calculated per equation 4.24.⁹

$$P_m = \frac{SE(T-c)}{r_2} \left(\frac{T-c}{(T-c) + 1.25 \tan \theta \sqrt{r_2(T-c)}} \right) \quad (4c) \quad (4.24)$$

The miter wall thickness, T , used in Eqs. (4.22)–(4.24) is required to extend a distance at least M from the inside crotch of the miter end welds.

In the above equations, the following definitions apply:

E = quality factor

M = greater of $2.5(r_2 T)^{0.5}$ and $(\tan \theta)(R_1 - r_2)$

P_m = maximum allowable internal pressure for miter bends

R_1 = effective radius of miter bend, defined as the shortest distance from the pipe centerline to the intersection of the planes of adjacent miter joints

S = allowable stress from Table A-1 in Appendix A of ASME B31.3

T = miter pipe wall thickness (measured or minimum per purchase specification)

c = sum of mechanical allowances plus corrosion and erosion allowances

r_2 = mean radius of pipe using nominal wall

θ = angle of miter cut

α = angle of change in direction at miter joint

= 2θ

4.10 PRESSURE DESIGN OF CLOSURES

Closures are covered in para. 304.4.1. Listed components, such as ASME B16.9 pipe caps, can be used for closures. The other two options provided in ASME B31.3 are to design the closure in accordance with ASME BPVC Section VIII, Division 1, or to qualify it as an unlisted component in accordance with para. 304.7.2 (see Section 4.15). Specific references to ASME BPVC, Section VIII, Division 1 paragraphs are provided for ellipsoidal, torispherical, hemispherical, conical, toriconical, and flat heads.

Openings in closures are covered in para. 304.4.2. These requirements are summarized below:

- If the opening is greater than one-half of the inside diameter of the closure (as defined in ASME BPVC Section VIII, Division 1, para. UG-36), it should be designed as a reducer

⁷ASME B31.3, Eq. (4a).

⁸ASME B31.3, Eq. (4b).

⁹ASME B31.3, Eq. (4c).

per para. 304.6 if the closure is dished and as a flange per para. 304.5 if the closure is flat.

- Small openings and connections using branch connection fittings that comply with para. 304.3.2(b) or 304.3.2(c) are considered to be inherently adequately reinforced.
- The required area of reinforcement is determined per the relevant ASME BPVC Section VIII, Division 1 requirements [UG-37(b), UG-38, or UG-39], which depend on the type of closure. For example, only one-half of the area requires replacement for a flat head.
- The available area of reinforcement is calculated per the rules in ASME B31.3, specifically para. 304.3.3 or para. 304.3.4. Note that ASME B31.3 requires that boundaries for a curved closure follow the contour of the closure (versus a chord dimension).
- Rules for multiple openings follow para. 304.3.3 and para. 304.3.4 rules for multiple openings.

4.11 PRESSURE DESIGN OF FLANGES

Most flanges are in accordance with listed standards, such as ASME B16.5, and for larger flanges, ASME B16.47. When a custom flange is required, design by analysis is permitted by para. 304.5.1. ASME B31.3 refers to the rules for flange design contained in ASME BPVC Section VIII, Division 1, Appendix 2, but using the allowable stresses and temperature limits of ASME B31.3. For flanges with metal-to-metal contact outside of the bolt circle, the rules of ASME BPVC Section VIII, Division 1, Appendix Y are referenced.

4.12 PRESSURE DESIGN OF BLIND FLANGES

Most blind flanges are in accordance with listed standards, such as ASME B16.5. When designing a blind flange, the rules of ASME BPVC Section VIII, Division 1, para. UG-34 apply. The procedure in ASME BPVC Section VIII includes consideration of the moment due to the flange bolts, so the required bolt load for boltup and operation must be determined per the flange design rules of Appendix 2 of ASME BPVC Section VIII, Division 1.

4.13 PRESSURE DESIGN OF BLANKS

Blanks are flat plates that get sandwiched between flanges to block flow. A design equation for permanent blanks is provided in para. 304.5.3, as¹⁰

$$t_m = d_g \sqrt{\frac{3P}{16SE}} + c \quad (4.25)$$

where

d_g = inside diameter of gasket for raised or flat face flanges, or the gasket pitch diameter for ring joint and fully retained gasketed flanges

Other terms are as defined in Section 4.2.

Blanks used for test purposes are not subject to these design rules and are often designed to higher allowable stress levels (e.g., 90% of the specified minimum yield strength).

¹⁰ASME B31.3, Eq. (15).

4.14 PRESSURE DESIGN OF REDUCERS

Most reducers in piping systems are in accordance with listed standards. However, pressure design per para. 304.7.2 and detailed design using rules for conical or toriconical closures are permitted. The rules for closures reference ASME BPVC Section VIII, Division 1, as described above in Section 4.10.

4.15 PRESSURE DESIGN OF UNLISTED COMPONENTS

If a component is not in accordance with a listed standard and/or the design rules provided in para. 304 are not applicable, para. 304.7.2 is applicable. This paragraph requires that some calculations be done in accordance with the design criteria provided by the Code and be substantiated by one of several methods. The meat of this paragraph is considered to be the substantiation; the aforementioned calculations are not generally given much consideration. The methods to substantiate the calculations, and thereby the design, include the following:

- Extensive, successful service experience under comparable conditions with similarly proportioned components of the same or like material.
- Experimental stress analysis, such as described in ASME BPVC Section VIII, Division 2, Appendix 6.
- Proof test in accordance with either ASME B16.9, MSS SP-97, or ASME BPVC Section VIII, Division 1, para. UG-101. Note that of these standards, those of B16.9 and MSS SP-97 are more applicable to piping components and have a margin of safety consistent with other components in the Piping Code (factor of three on burst rather than the factor of four in the Pressure Vessel Code).
- Detailed stress analysis (e.g., finite-element method) with results evaluated as described in ASME BPVC Section VIII, Division 2, Appendix 4, Article 4-1. These are the design-by-analysis rules in the Pressure Vessel Code.

Of the above, the methods normally used to qualify new unlisted components are proof testing and detailed stress analysis.

It should be noted that the Code permits interpolation between sizes, wall thicknesses, and pressure classes, and also permits analogies among related materials. Extrapolation is not permitted.

The issue of how to determine that the above has been done in a satisfactory manner has been discussed in detail in B31.3 Section Committee meetings. Earlier editions of the Code required only that proof testing be approved by the Inspector. However, this created concerns that it may be interpreted that the Inspector must witness the proof test, which is not practical when the manufacturer performs proof tests to qualify a line of piping components. Obviously, all the potential future owner's Inspectors could not be gathered for this event. Furthermore, the other methods are of at least equal concern, and their review may be more appropriately done by an engineer rather than an Inspector. As a result of these concerns, the requirement was added that documentation showing compliance with the above must be available for the owner's approval. The review would be by an Inspector or some other qualified individual for the owner.

Although MSS SP-97 and ASME B16.9 provide a clear approach for determining that the rating of a component is equivalent to or better than matching straight pipe, they do not provide clear procedures for determining a rating for a component that may have a unique rating which may differ from matching straight pipe. The procedure generally used here is to establish a pressure-temperature rating by multiplying the proof pressure by the ratio of the allowable stress for the test specimen to the actual tensile strength of the test specimen. An example of this approach is provided by Biersteker et al (1991). In the proposed B31H Standard, this would be reduced by a testing factor depending on the number of tests.

A new standard is under development by ASME that will eventually add to or replace the existing proof test alternatives in para. 304.7.2. This is B31H, *Standard Method to Establish Maximum Allowable Design Pressure for Piping Components*. This Standard provides procedures to determine that a component has a pressure capacity at least as great as a matching straight pipe, or to determine a pressure-temperature rating for a component.

LIMITATIONS ON COMPONENTS AND JOINTS

5.1 OVERVIEW

ASME B31.3 includes limitations on components and joints in the design chapter, Chapter II. These are contained in Part 3, Fluid Service Requirements for Piping Components, and Part 4, Fluid Service Requirements for Piping Joints. Many of these are restrictions for severe cyclic conditions. The present chapter combines the limitations with pressure design and other considerations on a component-by-component basis.

5.2 VALVES

Most valves in ASME B31.3 piping systems are in accordance with the following listed standards:

- ASME B16.9, *Face-to-Face and End-to-End Dimensions of Valves*
- ASME B16.34, *Valves — Flanged, Threaded, and Welding End*
- API 526, *Flanged Steel Pressure Relief Valves*
- API 594, *Wafer and Wafer-Lug Check Valves*
- API 599, *Metal Plug Valves — Flanged and Welding Ends*
- API 600, *Steel Gate Valves — Flanged and Butt-Welding Ends, Bolted, and Pressure Seal Bonnets*
- API 602, *Compact Steel Gate Valves — Flanged, Threaded, Welding and Extended Body Ends*
- API 603, *Class 150, Cast, Corrosion-Resistant, Flanged-End Gate Valves*
- API 608, *Metal Ball Valves — Flanged, Threaded, and Welding Ends*
- API 609, *Butterfly Valves: Double Flanged, Lug- and Wafer-Type*
- AWWA C500, *Metal-Seated Gate Valves for Water Supply Service*
- AWWA C504, *Rubber-Seated Butterfly Valves*
- MSS SP-42, *Class 150 Corrosion-Resistant Gate, Globe, Angle, and Check Valves with Flanged and Buttweld Ends*
- MSS SP-70, *Cast Iron Gate Valves, Flanged and Threaded Ends*
- MSS SP-71, *Cast Iron Swing Check Valves, Flanged and Threaded Ends*
- MSS SP-72, *Ball Valves with Flanged or Buttwelding Ends for General Service*
- MSS SP-80, *Bronze Gate, Globe, Angle, and Check Valves*
- MSS SP-81, *Stainless Steel, Bonnetless, Flanged, Knife Gate Valves*
- MSS SP-85, *Cast Iron Globe and Angle Valves, Flanged and Threaded Ends*

- MSS SP-88, *Diaphragm Type Valves*
- MSS SP-105, *Instrument Valves for Code Applications*

Listed valves are accepted for their specified pressure ratings. Some of these pressure ratings are tabulated in Appendix I. Valves that are not in accordance with one of the listed standards can be accepted as unlisted components in accordance with para. 302.2.3. Two options are permitted for establishing pressure-temperature ratings for unlisted valves. These are given in para. 304.7.2 and ASME B16.34, Appendix F.

The only additional requirement for valves in para. 307 is that bolted bonnet valves must be secured by at least four bolts, except in Category D Fluid Service. Bonnets secured with fewer than four bolts or with U-bolt bonnets are only permitted in Category D Fluid Service.

5.3 FLANGES

Most flanges in ASME B31.3 piping systems are in accordance with the following listed standards:

- ANSI B16.1, *Cast Iron Pipe Flanges and Flanged Fittings*
- ASME B16.5, *Pipe Flanges and Flanged Fittings*
- ASME B16.24, *Bronze Pipe Flanges and Flanged Fittings Classes 150, 300, 400, 600, 900, 1500, and 2500 and Flanged Fittings, Classes 150 and 300*
- ASME B16.36, *Orifice Flanges, Classes 300, 600, 900, 1500, and 2500*
- ASME B16.42, *Ductile Iron Pipe Flanges and Flanged Fittings, Classes 150 and 300*
- ASME B16.47, *Large Diameter Steel Flanges, NPS 26 Through NPS 60*
- AWWA C115, *Flanged Ductile-Iron with Ductile-Iron or Gray-Iron Threaded Flanges*
- AWWA C207, *Steel Pipe Flanges for Water Works Service, Sizes 4 Inch Through 144 Inch (100 mm Through 3,600 mm)*
- MSS SP-44, *Steel Pipe Line Flanges*
- MSS SP-51, *Class 150LW Corrosion Resistant Cast Flanges and Flanged Fittings*
- MSS SP-65, *High Pressure Chemical Industry Flanges and Threaded Stubs for Use with Lens Gaskets*

Listed flanges are accepted for their specified pressure ratings. Some of these pressure ratings are tabulated in Appendix I. Flanges that are not in accordance with one of the listed standards can be accepted as unlisted components in accordance with para. 302.2.3. If the Designer is satisfied that the composition, mechanical properties, method of manufacture, and quality control are comparable to the corresponding characteristics of a listed component, pressure design can be performed in accordance with ASME BPVC, Section VIII, Division 1, Appendix 2 or Y, as applicable. Otherwise, qualification of the flange in accordance with para. 304.7.2 is required. Paragraph 308 provides additional restrictions for flanges.

Slip-on flanges are required to be double-welded when the service is (1) subject to severe erosion, crevice corrosion, or cyclic loading, (2) flammable, toxic, or damaging to human tissue, (3) under severe cyclic conditions, or (4) at temperatures below -101°C (-150°F). Further, the Designer is cautioned that the use of slip-on flanges should be avoided where many large temperature cycles are expected, particularly if the flange is not insulated.

A double-welded slip-on flange has a weld between the pipe and the flange hub and between the pipe and the bore of the flange. A single-welded slip-on flange only has the weld to the flange hub.

For double-welded slip-on flanges in hydrogen service, where hydrogen can diffuse into the annulus between the inner and outer welds, collect, and pressurize it, the flange should be drilled to vent this space. A precautionary consideration, F308.2, mentions this concern.

Slip-on flanges have been substituted for lap joint flanges (both ASME B16.5). However, the flange hub for some sizes is undersized for lap joint applications. As a result, Table 308.2.1 was added to the Code. It limits substitution of slip-on for lap joint flanges to DN 300 (NPS 12) or smaller for PN 20

(Class 150) flanges, and DN 200 (NPS 8) for PN 50 (Class 300) flanges, unless the pressure design has been qualified in accordance with the ASME BPVC, Section VIII, Division 1 rules. In addition, other requirements are provided relative to the lap joint in para. 308.2.1(c).

For flanges attached to the pipe using expanded joints, threaded joints, and socket weld joints, the restrictions on those joints, discussed later in this chapter, apply.

5.4 FITTINGS, BENDS, MITERS, AND BRANCH CONNECTIONS

Most fittings in ASME B31.3 piping systems are in accordance with the following listed standards:

- ASME B16.3, *Malleable Iron Threaded Fittings*
- ASME B16.4, *Gray Iron Threaded Fittings*
- ASME B16.9, *Factory-Made Wrought Steel Butt welding Fittings*
- ASME B16.11, *Forged Fittings, Socket-Welding and Threaded*
- ASME B16.14, *Ferrous Pipe Plugs, Bushings, and Locknuts with Pipe Threads*
- ASME B16.15, *Cast Bronze Threaded Fittings, Classes 125 and 250*
- ASME B16.18, *Cast Copper Alloy Solder Joint Pressure Fittings*
- ASME B16.22, *Wrought Copper and Copper Alloy Solder Joint Pressure Fittings*
- ASME B16.26, *Cast Copper Alloy Fittings for Flared Copper Tubes*
- ASME B16.28, *Wrought Steel Butt welding Short Radius Elbows and Returns*
- ASME B16.39, *Malleable Iron Threaded Pipe Unions, Classes 150, 250, and 300*
- AWWA C110, *Ductile-Iron and Gray-Iron Fittings, 3 Inch Through 48 Inch (75 mm Through 1200 mm), for Water and Other Liquids*
- AWWA C208, *Dimensions for Fabricated Steel Water Pipe Fittings*
- MSS SP-43, *Wrought Stainless Steel Butt welding Fittings*
- MSS SP-75, *Specifications for High Test Wrought Butt welding Fittings*
- MSS SP-79, *Socket-Welding Reducer Inserts*
- MSS SP-83, *Class 3000 Steel Pipe Unions, Socket-Welding and Threaded*
- MSS SP-95, *Swage(d) Nipples and Bull Plugs*
- MSS SP-97, *Integrally Reinforced Forged Branch Outlet Fittings—Socket Welding, Threaded, and Butt welding Ends*
- SAE J513, *Refrigeration Tube Fittings—General Specifications*
- SAE J514, *Hydraulic Tube Fittings*
- SAE J518, *Hydraulic Flange Tube, Pipe, and Hose Connections, Four-Bolt Split Flanged Type*

Listed fittings are accepted for their specified pressure ratings. Note that some fittings are simply specified to have equivalent pressure ratings to matching straight seamless pipe. As discussed in Section 4.1, ASME B31.3 bases this on matching straight seamless pipe with 87.5% of the nominal wall thickness. Some ratings are tabulated in Appendix I. Fittings that are not in accordance with one of the listed standards can be accepted as unlisted components in accordance with para. 302.2.3. The relevant option of this paragraph for fittings is typically para. 304.7.2, because the pressure design equations in para. 304 typically are not relevant.

Branch connections are required to be listed components (para. 306.1.1); fabricated branch connections that are designed per para. 304.3 and welded per para. 311.1 (para. 306.5.1); fittings that are qualified by proof testing per ASME B16.9, MSS SP-97, or BPVC, Section VIII, Division 1, UG-101 [para. 306.1.3(a)]; or unlisted components that are qualified per para. 304.7.2 (para. 306.1.2).

The following fittings are prohibited from use under severe cyclic conditions.

- MSS SP-43 fittings (para. 306.1.4)
- Proprietary “Type C” lap-joint stub-end welding fittings (para. 306.1.4)

- Fittings (para. 306.1.4) other than forged, wrought, with $E_f \geq 0.90$, or cast, with $E_c \geq 0.90$.
- Corrugated or creased bends (para. 306.2.3)
- Miter bends with angle $\alpha > 22.5^\circ$ (para. 306.3.3)
- Fabricated branch connections with the branch set on (rather than set in) the run pipe [i.e., Fig. 328.5.4D, sketches (2) and (4)] are permitted (para. 306.5.2)

The following fittings are only permitted for Category D Fluid Service:

- Miter bends with a single joint angle $\alpha > 45^\circ$.

Limitations for fabricated and flared laps for use with lap joint flanges are provided in para. 306.4. Note that these limitations do not apply to laps that are in accordance with listed standards, such as ASME B16.9. Fabricated laps are constructed by welding; flared laps are constructed by flaring a pipe section. The outside diameter of the lap is required to comply with the dimension specified in ASME B16.9. The lap thickness is required to be at least equal to the nominal wall thickness of the pipe for fabricated laps, and, for flared laps, 95% of the minimum pipe wall thickness (measured or minimum per purchase specification, considering mill tolerance) multiplied by the ratio of the pipe outside radius to the radius at which the lap thickness is measured (this essentially permits thinning due to the flaring process).

For a fabricated lap, the material for the lap must have an allowable stress at least as great as the pipe material and the material must be listed in ASME B31.3, Table A-1, the allowable stress table. Welding must be per ASME B31.3 and typical details are provided in Fig. 328.5.5. Full penetration welds are required (by reference to the applicable requirements of para. 328.5.4, which covers branch connections).

Flared laps are required to have their pressure design qualified in accordance with para. 304.7.2. Thus, they are essentially treated as unlisted components in this regard. In addition, the radius of the fillet of the flared lap is not permitted to exceed 3 mm ($1/8$ in.), and the mating flange is required to be beveled or rounded approximately 3 mm ($1/8$ in.) (by reference to para. 308.2.5). This prevents interference between the lap and the flange at the pipe/lap junction.

5.5 BOLTING

Requirements for bolting are provided in para. 309. Listed bolting is acceptable. The only relevant component standard is ASME B1.1, *Unified Inch Screw Threads (UN and UNR Thread Form)*. Table A-2 in ASME B31.3, Appendix A provides listed bolting material specifications. Unlisted bolting (i.e., not listed in Table A-2) is subject to the requirements of para. 302.3.3 (per para. 309.1.2).

Requirements for flanged joints relative to bolt strength are provided as follows. Low strength is considered to refer to bolts with a specified minimum yield strength of 207 MPa (30 ksi) or less:

(a) Low-strength bolts are not permitted for flanges with ASME B16.5 Class 400 or higher ratings or with flanged joints using metallic gaskets, unless design calculations have been made to show the bolts have sufficient strength to maintain joint tightness. Although not specifically stated in ASME B31.3, this should be done per the bolting design requirements in ASME BPVC, Section VIII, Division 1, Appendix 2.

(b) Certain flanges are considered to be weaker due to the material of construction or the flange design. These are flanges per ANSI B16.1 (cast iron), ANSI B16.24 (bronze), MSS SP-42, and MSS SP-51. These are required to use low-strength bolting to avoid overloading the flange, unless one of the following is satisfied: (i) The flanges are flat face and use a full face gasket. This essentially prevents excessive flange rotation due to bolt load; or (ii) the sequence and torque limits for the boltup are specified, giving appropriate consideration to the loads and the strength of the flange.

(c) Low-strength bolting is prohibited from flange joints under severe cyclic conditions. However, since the stresses in a flanged joint due to piping thermal expansion loads are not normally calculated, it is unclear as to when this would ever be implemented.

The depth of tapped holes is required to be sufficient to provide a length of thread engagement at least seven-eighths times the nominal thread diameter.

5.6 WELDED JOINTS

Welded joints are covered by para. 311 and are required to follow the ASME B31.3 rules for fabrication and examination. In addition, the following specific rules are also provided.

Weld backing rings are generally permitted to be left in. However, if the resulting crevice would be detrimental (e.g., subject to corrosion, vibration, or severe cyclic conditions), para. 311.2.3 recommends removing the ring and grinding the internal joint face smooth. Split-backing rings are prohibited under severe cyclic conditions.

Socket-welded joints are generally permitted, with the following exceptions. They should not be used in services where crevice corrosion (due to the crevice between the pipe and socket) or severe erosion (which can cause local erosion at the gap between the pipe end and socket) may occur. Socket welds larger than DN 50 (NPS 2) are not permitted under severe cyclic conditions. Note that socket welds are generally limited in practice to sizes of DN 50 (NPS 2) or less in any case.

The dimensions of the socket joint are required to conform to either ASME B16.5 for flanges or ASME B16.11 for other socket welding components (except for socket welding drain or bypass attachments to components that conform to Fig. 4 of ASME B16.5).

Weld dimensions are required to comply with the fabrication rules of ASME B31.3 (Figs. 328.5.2B and 328.5.2C). Socket joints are generally part of listed components, so their pressure design is satisfied by the component standard and compliance with the fillet weld size requirements of the fabrication rules of ASME B31.3.

Paragraph 311.2.5 permits fillet welds to be used for the primary welds to attach socket welding components and slip-on flanges. It also permits fillet welds to be used to attach reinforcement and structural attachments and to supplement the strength or reduce the stress concentration of other welds. Note, however, that the wording does not prohibit the use of fillet welds for other applications, and it has been the interpretation of the Committee that they are not prohibited from use in other joints, such as lined pipe sections connected by a metallic sleeve with fillet welds. This is reflected in Interpretation 10-04:

“Question: Does ASME/ANSI B31.3-1987 Edition and its Addenda permit the use of fillet welds as the primary pressure retaining welds for installation of a full encirclement tee type branch connections?”

“Reply: ASME/ANSI B31.3-1987 Edition and its Addenda do not specifically address rules for fillet welding of full encirclement tee type branch connections.”

Seal welds are permitted to be used to prevent leakage of threaded joints. However, the seal weld is not permitted to be considered as contributing to the strength of the joint.

5.7 FLANGED JOINTS

Flanges are discussed in Section 5.3. Additional requirements for flanged joints are provided in para. 312 of ASME B31.3. These discuss conditions where flanges with different strengths are bolted together. This can involve two flanges of the same or different materials or with different ratings, or a nonmetallic and metallic flange bolted together. In both cases, precautions are required to avoid

overloading the weaker flange. In addition, the pressure rating of the joint is limited by the flange with the lower rating.

In either case, precautions are generally required to avoid overtightening the bolts, which could damage the weaker flange. In addition, for metallic-to-nonmetallic flanges, use of full face gaskets is preferred, again, to prevent excessive distortion of the nonmetallic flange due to bolt loads. If full face gaskets are not used with metallic/nonmetallic flange joints, the bolt torque is required to be limited, to avoid overloading the nonmetallic flange.

5.8 THREADED JOINTS

Most threaded joints in ASME B31.3 piping systems are made with taper threads in accordance with ASME B1.20.1. Figure 5.1 shows a taper thread. These threaded joints are generally acceptable for Normal Fluid Service and Category D Fluid Service. Limitations with respect to Category M Fluid Service are discussed in Chapter 16. Paragraph 314.1(a) states that threaded joints should be avoided in any service where crevice corrosion, severe erosion, or cyclic load may occur. Cyclic loading is not defined and this limitation is rather unclear, because all systems are subject to some cycles and threaded joints are commonly used in process piping.

As discussed in the Chapter 12, threaded joints that are to be seal-welded are prohibited from using thread sealing compounds. However, when seal welding is not an intent of the design, but is intended simply to seal leaks that are detected on hydrotest, threaded joints that use thread sealing compound may be seal-welded, provided the sealing compound is removed from the exposed thread. The joint does not have to be undone and remade without the compound.

Threaded joints are prohibited from use in severe cyclic service, except for two conditions. One is the use of threaded components of a specialty nature, which are not subject to external moment loading, such as thermometer wells. However, by the definition of severe cyclic conditions, if there is no moment, there is no Code thermal expansion stress and such a joint cannot be in severe cyclic conditions. The other condition where threaded joints are permitted in severe cyclic conditions is provided in para. 314.2.2, which permits straight thread joints in which the tightness of the joint is provided by a seating surface other than the threads, provided the joint is safeguarded.

ASME B31.3 provides minimum thicknesses for male and female portions of threaded joints. For the female portions, it requires that they have equivalent strength to listed components (in Table 326.1, e.g., ASME B16.11). For the male portion, the minimum thickness is provided in Table 314.2.1, provided here as Table 5.1.

5.9 TUBING JOINTS

Tubing joints are covered by para. 315, and include flared, flareless, and compression-type tube fittings. ASME B31.3 provides some listed tubing joints; however, many tubing joints used in process

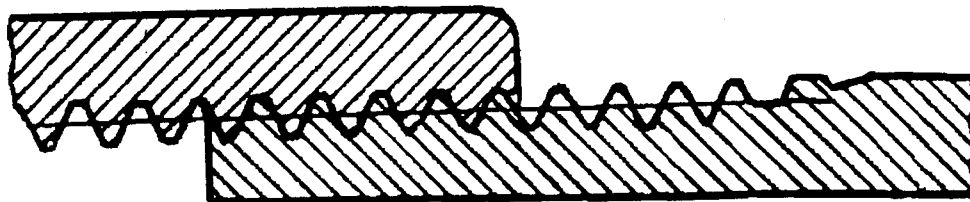


FIG. 5.1
TAPER THREAD

TABLE 5.1
MINIMUM THICKNESS FOR THREADED PIPE¹

Fluid Service	Notch-Sensitive Material	Size Range, in. [Note (2)]		Min. Wall Thickness [Note (3)]
		DN	NPS	
Normal	Yes [Note (4)]	≤40	≤1½	Sch. 80
		50	2	Sch. 40
		65–150	2½–6	Sch. 40
Normal	No [Note (5)]	≤50	≤2	Sch. 40S
		65–150	2½–6	Sch. 40S
Category D	Either	≤300	≤12	Per para. 304.1.1

NOTES:

- (1) Use the greater of ASME B31.3, para. 304.1.1 or thickness shown in this table, which reproduces ASME B31.3, Table 314.2.1.
- (2) For sizes >DN 50 (NPS 2), the joint shall be safeguarded (see ASME B31.3, Appendix G) for a fluid service that is flammable, toxic, or damaging to human tissue.
- (3) Nominal wall thicknesses is listed for Sch. 40 and 80 in ASME B36.10M and for Sch. 40S in ASME B36.19M.
- (4) For example, carbon steel.
- (5) For example, austenitic stainless steel.

pipng are proprietary fittings that are qualified as unlisted components. The following are the listed standards covering tubing joints:

- SAE J513, *Refrigeration Tube Fittings—General Specifications*
- SAE J514, *Hydraulic Tube Fittings*

These listed components are permitted for use in Normal Fluid Service, provided that they are used within the pressure–temperature limitations of the fittings and joint. They are required to be safeguarded when used under severe cyclic conditions; however, it is highly unlikely that this will occur, because it requires the tube joint to be at over 80% of the allowable displacement stress range. Unlisted tube fittings must be qualified in accordance with para. 304.7.2.

5.10 MISCELLANEOUS JOINTS

Caulked and soldered joints are only permitted in Category D Fluid Service. Caulked joints are further limited to a maximum temperature of 93°C (200°F), and provisions are required to prevent disengagement of the joints and resist the effects of longitudinal forces due to internal pressure. With soldered joints, the low melting point of solder is to be considered where possible exposure to fire or elevated temperature is involved.

Brazed and braze-welded joints are permitted to be used in Normal Fluid Service. However, they are required to be safeguarded in fluid services that are flammable, toxic, or damaging to human tissue (see Section 2.5). They are prohibited from use under severe cyclic conditions, and the melting point of the brazing alloys is to be considered where possible exposure to fire is involved.

For other joints, such as bell-type and packed joints, the separation of the joint must be prevented by a means that has sufficient strength to withstand the anticipated conditions of service. Pressure tends to pull these joints apart. Furthermore, if the fluid service is flammable, toxic, or damaging to human tissues, or exposed to temperatures in the creep range, joint separation must be

prevented by mechanical or welded interlocks. Friction within the joint or use of external anchors is not sufficient.

There are provisions that restrict mechanical joints and bell-and-gland-type joints in severe cyclic service. However, since severe cyclic service only applies to those components with a stress exceeding 80% of the allowable displacement stress, it is highly unlikely that these joints would be in severe cyclic service.

DESIGN FOR SUSTAINED AND OCCASIONAL LOADS

6.1 PRIMARY LONGITUDINAL STRESS

The wall thickness of pipe is nearly always selected based on the thickness required for internal pressure and allowances. The piping is then supported sufficiently such that the longitudinal stress (the stress in the axial direction of the pipe) is within Code limits and deflection is within acceptable limits.

Deflection limits are not Code requirements, but are generally accepted practices; a 13-mm ($1/2$ -in.) deflection is a generally accepted guideline for general process plant piping. More stringent limits may be required for lines that must avoid pockets caused by sagging of the line; greater deflection is generally acceptable from a mechanical integrity standpoint, if not an operator confidence standpoint.

It is fortunate that the longitudinal pressure stress is one-half of the hoop stress in a cylinder. What this means is that if the pipe is designed for pressure, at least one-half of the strength in the longitudinal direction remains available for weight and other sustained loads.

The same basic allowable stress as is used for pressure design is used as the limit for longitudinal sustained stresses in piping systems. These sustained and the occasional load stresses should encompass all of the load-controlled, primary-type stresses to which the pipe is subjected. They will fall either in the sustained or the occasional category, depending on duration. Whereas the weight of the pipe and contents are sustained, forces and moments due to wind, earthquake, and phenomena such as dynamic loads due to water hammer would be considered to be occasional loads.

6.2 SUSTAINED LONGITUDINAL STRESS

ASME B31.3 provides a description of what is included in the sustained longitudinal stress, S_L , which is defined as the sum of the longitudinal stresses. Because no equation is provided and the description is not specific with respect to some items, such as the application of stress intensification factors, it is left to some extent to the interpretation of the user and the programmers of piping stress analysis programs.

The description can be found in para. 302.3.5(c), which states that "the sum of longitudinal stresses in any component in a piping system, due to pressure, weight, and other sustained loadings S_L shall not exceed S_h ." S_h is the basic allowable stress at the metal temperature of the operating condition being evaluated.

This has been interpreted by the B31.3 Section Committee to include stresses due to axial loads as well as bending moments. The issue of torsional loads has not been addressed, although it would make sense to include them. However, they are clearly not specifically required to be included. Also, the

issue of stress intensification factors is left unresolved, except that their inclusion is not required by the Code.

Inclusion of stress intensification factors remains a disputed issue. Their origin is in fatigue testing, so their applicability to the collapse mechanisms of pipe due to sustained loads is questionable. Although ASME B31.1 requires that 0.75 times the stress intensification factors be used, ASME B31.3 is silent on the issue. In the absence of explicit guidance, some writers of piping stress analysis programs have elected to use the full value of the stress intensification factors in the analysis of sustained loads for ASME B31.3 piping.

ASME B31.3, para. 302.3.5(c) requires that the thickness of the pipe used in calculating S_L be the nominal thickness minus the mechanical, corrosion, and erosion allowance. Unlike pressure design, it is not required to account for mill tolerance of the piping.

Although it is not a specific Code requirement, what is normally done is to calculate all the forces, moments, and deflections in a piping system using nominal dimensions. This provides appropriate support loads for design. Then, these forces and moments are divided by section properties based on nominal thickness minus the allowances. Inclusion of the metal to be used as corrosion allowance in the weight but not the strength may be viewed as unduly conservative; however, corrosion could be local and at the highest loaded point, such as at a support. The description of this general practice was added to para. 302.3.5(c) in the 2000 Addenda with the addition of the following sentence: "The loads due to weight should be based on the nominal thickness of all system components unless otherwise justified in a more rigorous analysis."

Quality factors, E_j and E_c , are not used in the evaluation of longitudinal stresses due to sustained loads.

6.3 LIMITS OF CALCULATED STRESSES DUE TO OCCASIONAL LOADS

The ASME B31.3 Code is consistent with the structural codes in the treatment of occasional loads. The sum of all longitudinal stress, including both sustained and occasional loads, is limited to 1.33 times the basic allowable stress. This differs from the Pressure Vessel Code, which uses a factor of 1.2. The treatment of earthquake forces in this manner is generally considered to be extremely conservative, and a new Appendix or additional Code document with a different treatment may be forthcoming.

The increase in allowable stress is only permitted as long as the yield strength of the material is not exceeded. This is one of the areas of the Code where use of the higher allowable stress relative to yield for austenitic stainless steels and nickel alloys with similar stress-strain properties can result in additional limits.

Although quality factors are not generally used in the evaluation of stresses due to longitudinal loads, for occasional loads there is an exception, in that the Code requires that the casting quality factor, E_c , be multiplied by the basic allowable stress when evaluating occasional loads.

A recent change has been made to the allowable stress for occasional loads at elevated temperature, that is, at temperatures where the allowable stresses become controlled by time-dependent creep properties of the material. This approved change has not yet appeared in the Code. These creep properties are based on 100,000-hour time periods; they are not relevant to short-term loadings, such as earthquakes. Therefore, for occasional loads of short duration at temperatures above 427°C (800°F), it is permitted to use 90% of the material yield strength at temperature times a strength reduction factor for materials other than cast or ductile iron or materials with nonductile behavior. The yield strength can be from ASME BPVC, Section II, Part D, Table Y, or can be determined in accordance with ASME B31.3, para. 302.3.2(f).

The strength is multiplied by a strength reduction factor. This is included because some materials, low alloys in particular, undergo aging at elevated temperatures, which decreases the yield and tensile

strength over time. Austenitic stainless steels are not subject to this effect, so no reduction is applied to them. A factor of 0.8 is applied to all other alloys in the absence of more applicable data. This factor is based on low-alloy data. Although this is known to be conservative for a variety of materials, the benefit of permitting design for occasional loads based on yield strength greatly outweighs the penalty of the 0.8 factor.

DESIGN CRITERIA FOR THERMAL STRESS

7.1 ALLOWABLE STRESS FOR THERMAL EXPANSION

The allowable stress for thermal expansion and other deformation-induced stresses is substantially higher than for sustained loads. This is due to the difference between load-controlled conditions, such as weight and pressure, and deformation-controlled conditions, such as thermal expansion or end displacements (e.g., due to thermal expansion of attached equipment).

When a load-controlled stress is calculated, it is an actual stress value. It is governed by equilibrium. For example, the stress in a bar when a tensile force is applied to it is the force divided by the area of the bar. This is not the case for thermal stresses. In the case of thermal stresses, it is the value of strain that is known. The elastically calculated stress is simply the strain value times the elastic modulus. This makes essentially no difference until the stress exceeds the yield strength of the material. In that case, the location on the stress-strain curve for the material is determined based on the calculated stress for load-controlled, or sustained, loads. The location on the stress-strain curve for the material is determined based on the calculated strain (or elastically calculated stress divided by elastic modulus) for deformation-controlled (e.g., thermal expansion) loads. This is illustrated in Fig. 7.1. Because the stress analyses are based on the assumption of elastic behavior, it is necessary to discriminate between deformation-controlled and load-controlled conditions in order to properly understand the post-yield behavior.

It is considered desirable for the piping system to behave in a substantially elastic manner so that the elastic stress analysis is valid. Furthermore, having plastic deformation every cycle carries with it uncertainties with respect to strain concentration and can be potentially far more damaging than calculated to be in the elastic analysis. One way to accomplish this would be to limit the total stress range to yield stress. However, this would be overly conservative and result in unnecessary expansion loops and joints. Instead, the concept of shakedown to elastic behavior is used in the Code.

The allowable thermal expansion stress in the Code is designed to result in shakedown to elastic behavior after a few operating cycles. The equation provided in the Code is¹¹

$$S_A = f(1.25S_c + 0.25S_h) \quad (7.1)$$

¹¹ ASME B31.3, Eq. (1a).

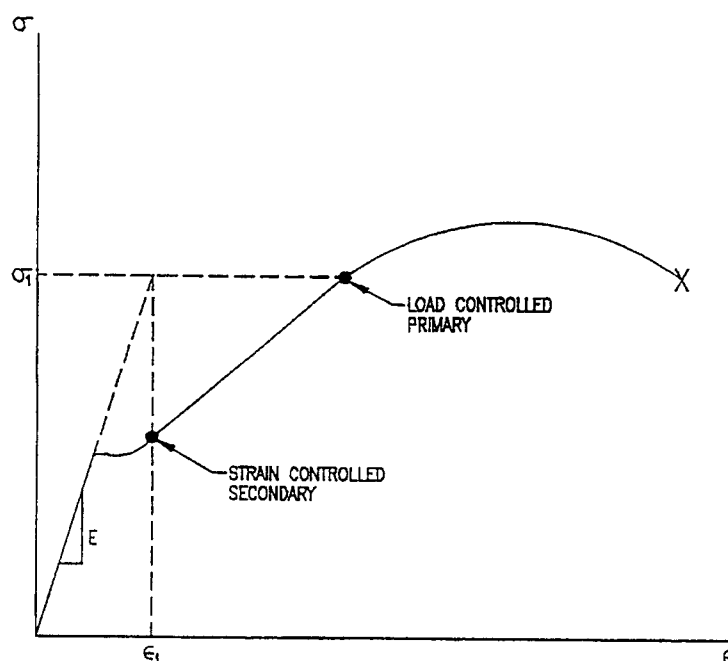


FIG. 7.1
LOAD-CONTROLLED VERSUS DEFORMATION-CONTROLLED BEHAVIOR

where

S_A = allowable displacement stress range

S_c = basic allowable stress at the minimum metal temperature expected during the displacement cycle under analysis

S_h = basic allowable stress at the maximum metal temperature expected during the displacement cycle under analysis

f = stress range reduction factor

This equation assumes that the sustained stress consumes the entire allowable sustained stress, and it is simplified, in that it is not necessary to know the sustained stress in order to determine the allowable thermal expansion stress.

Note that the values of S_c and S_h do not include weld joint quality factors. However, casting quality factors, E_c , must be included.

Equation (1b) of the Code is a more detailed equation, and considers the magnitude of the sustained longitudinal stress,

$$S_A = f[1.25(S_c + S_h) - S_L] \quad (7.2)$$

where

S_L = longitudinal stress due to sustained loadings

The allowable thermal expansion stress range can exceed the yield strength for the material, because both S_c and S_h may be as high as two-thirds of the yield strength. However, it is anticipated that the piping system will shake down to elastic behavior if the stress range is within this limit.

This behavior is illustrated in Fig. 7.2, which is based on the assumption of elastic, perfectly plastic material behavior. Consider, for example, a case where the elastically calculated thermal expansion stress range is two times the yield strength of the material. Because it is a deformation-controlled condition, one must actually move along the strain axis to a value of stress divided by elastic modulus. In the material, assuming elastic, perfectly plastic behavior, the initial start-up cycle goes from point *A* to *B* (yield) to *C* (strain value of twice yield). When the system returns to ambient temperature, the system returns to zero strain and the piping system will unload elastically until it reaches yield stress in the reverse direction. If the stress range is less than twice yield, there is no yielding on the return to ambient temperature. On returning to the operating condition, the system returns from point *D* to point *C* elastically. Thus, the cycling will be between points *D* and *C*, which is elastic. The system has essentially self-sprung and is under stress due to displacement conditions in both the ambient and the operating conditions.

If twice the yield is exceeded, shakedown to elastic cycling does not occur. An example is if the elastically calculated stress range is three times the yield strength of the material. In this case, again referring to Fig. 7.2, the startup goes from point *A* to point *B* (yield) to point *E*. Shutdown results in yielding in the reverse direction, going from point *E* to *F* to *D*. Returning to the operating condition again results in yielding, from point *D* to *C* to *E*. Thus, each operating cycle results in plastic deformation and the system has not shaken down to elastic behavior.

This twice-yield condition was the original consideration. Since the yield strengths in the operating and the ambient conditions are different, the criterion becomes that the stress range must be less than the hot yield strength plus the cold yield strength, which, due to the allowable stress criterion, must be less than 1.5 times the sum of S_c and S_h (Note that the original ASME B31.1 criterion limited the allowable stress to 62.5% of yield, so the original factor that was considered was 1.6.) This 1.5 (1.6 originally) factor was conservatively reduced to 1.25. This total permissible stress range is then reduced by the magnitude of sustained longitudinal stress in order to calculate the permissible thermal expansion stress range. The resulting equation is Eq. (1b) of the Code [Eq. (7.2) here]. Equation (1a) of the Code [Eq. (7.1) here] simply assumes that $S_L = S_h$, the maximum permitted value, and assigns the remainder of the allowable stress range to thermal expansion.

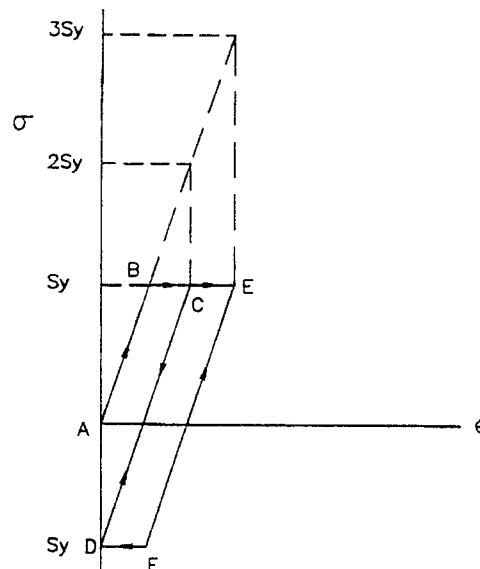


FIG. 7.2
STRESS-STRAIN BEHAVIOR ILLUSTRATING SHAKEDOWN

For A106, Gr. B Pipe at Room Temperature
 $S_h = S_c = 20,000$ psi

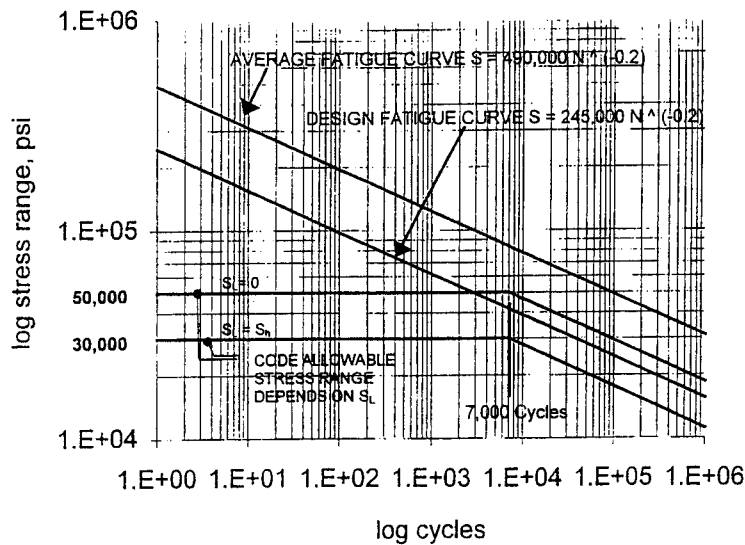


FIG. 7.3
 MARKL FATIGUE CURVE FOR BUTT-WELDED STEEL PIPE

Even though the stress range is limited so as to result in shakedown to elastic behavior, there remains the potential for fatigue failure if there is a sufficient number of cycles. Therefore, the f factor is used to reduce the allowable stress range when the number of cycles exceeds 7000. This is about once per day for 20 years.

Figure 7.3 provides the basic fatigue curve for butt-welded pipe, developed by Markl (1960a) for carbon steel pipe. A safety factor of two on stress was applied to this curve, giving a design fatigue curve. It can be observed that the allowable thermal expansion stress range, prior to application of an f factor, intercepts the design fatigue curve at about 7000 cycles. For higher numbers of cycles, the allowable stress is reduced by the f factor to follow the fatigue curve, per the equation¹²

$$f = 6N^{-0.2} \leq 1.0 \quad (7.3)$$

where

N = equivalent number of full displacement cycles during the expected service life of the piping system (see the next section for how to combine different cycles into an equivalent number of cycles)

As of the 1999 Edition of ASME B31.3, the factor f is not permitted to exceed 1.0. At the present time, increasing the maximum permissible f factor to 1.2 is being considered. An f factor of 1.2 corresponds to 3125 cycles. The rationale for allowing a factor as high as 1.2 is that stresses are permitted to be as high as two times yield when $f = 1.2$. Thus, the desired shakedown behavior is maintained. This change would reduce the conservatism introduced when the original criteria were developed.

¹² ASME B31.3, Eq. (1c).

TABLE 7.1
COMBINATION OF DIFFERENT DISPLACEMENT CYCLES¹

Load Case	Design Cycles	Calc. Stress Range	S_i / S_E	N_{equiv}
1	1,000	30,000	1	1000
2	7,000	10,000	0.33	27
3	20,000	6,000	0.20	6

NOTE:

- (1) Total equivalent number of cycles with stress range of 30,000 is $1033 < 7000$.
The stress range reduction factor $f = 1$.

The equation is considered to be valid to 2,000,000 cycles. Thus, the Code flexibility analysis criteria are not considered appropriate for vibration, where the number of cycles is typically much higher.

The development of this methodology is described by Markl (1960a-d).

7.2 HOW TO COMBINE DIFFERENT DISPLACEMENT CYCLE CONDITIONS

The Designer may have more than one thermal expansion or other displacement cycle conditions to consider. Cycles at lower stress ranges are substantially less damaging than cycles at higher stress ranges. ASME B31.3 uses the cycle with the highest stress range (full displacement cycle), S_E , to compare to the allowable stress range, S_A . However, cycles with lower stress ranges are converted into equivalent numbers of full displacement cycles to determine the f factor. This procedure, described in par. 302.3.5(d), uses the equation¹³

$$N = N_E + \sum (r_i^5 N_i) \quad \text{for } i = 1, 2, \dots, n \quad (7.4)$$

where

N_E = number of cycles of maximum computed displacement stress range, S_E

$r_i = S_i / S_E$

S_i = any computed displacement stress range smaller than S_E

N_i = number of cycles associated with displacement stress range S_i

n = total number of displacement stress conditions to be considered

Based on this procedure, the maximum stress range is limited to S_A , which satisfies the shakedown limit. Lesser cycles are converted into equivalent, with respect to fatigue damage, numbers of cycles at S_E to determine if an f factor less than one is required to protect against fatigue failure.

Table 7.1 provides an example of combining several displacement cycle conditions with different stress ranges and numbers of cycles. Note that due to the sensitivity of fatigue damage to stress (the fifth power in the equation), displacement cycles at significantly lower stress ranges than S_E produce very little damage or a significantly reduced number of equivalent cycles.

¹³ ASME B31.3, Eq. (1d).

Insert 7.1 What About Vibration?

As discussed in Section 7.1, the code flexibility analysis rules are not strictly applicable to vibration. The present limit of 2,000,000 cycles is generally far lower than the expected number of vibration cycles during the lifetime of a system that is vibrating.

Further, ASME B31.3 is a new design code, and typically, piping systems are not designed for vibrating conditions during design. However, there are cases, such as certain reciprocating compressor piping systems, for which detailed vibration assessments are performed, but these are exceptions to the general rule. As such, evaluation of vibration tends to be a post-construction exercise.

Excellent guidance for the evaluation of vibrating piping systems can be found in Part 3 of ASME Standard OM-S/G, ASME OM-3, *Requirements for Preoperational and Initial Start-Up Vibration Testing of Nuclear Power Plant Piping Systems*. The procedures contained therein were developed based on experience with reciprocating compressor systems for gas pipelines. Vibration can be evaluated via the procedures contained therein based on peak measured velocity of the vibrating pipe or calculated stress. A screening velocity criterion that is generally very conservative, 0.5 in./sec peak velocity, is provided. Endurance limit stress ranges are also provided.

Based on the endurance limit stress ranges, endurance limit f factors may be derived. The endurance limit stress ranges are provided for carbon steel and stainless steel. Including all the factors provided in the document gives the endurance limit of 106 MPa (15.4 ksi) for carbon steel. Assuming a typical S_A of 345 MPa (50 ksi) ($1.25S_c + 1.25S_h$), and considering that the Code flexibility analysis equations calculate about one-half of the actual peak stress, we find an endurance limit f factor of 0.15 (at about 10^8 cycles). The endurance limit f factor for stainless steel would be higher, if calculated by the same procedure. Inclusion of endurance limit f factors is presently under consideration. If they were provided, the f factor equation (7.3) [Eq. (1c) of B31.3] would probably be extended from 2,000,000 cycles to this endurance limit.

FLEXIBILITY ANALYSIS

8.1 FLEXIBILITY ANALYSIS

In flexibility analysis, the response of the system to loads is calculated. It is essentially a beam analysis model on pipe centerlines. The fundamental principles include the following:

1. The analysis is based on nominal dimensions of the pipe.
2. The effect of components such as elbows and tees on piping flexibility and stress is considered by inclusion of flexibility factors and stress intensification factors.
3. For thermal stresses, only moment and torsion are typically included. Stresses due to shear and axial loads are generally not significant. However, the Code, via para. 319.2.3(c), directs the analyst to include these stresses in conditions where they may be significant. This paragraph states the following:

“Average axial stresses (over the pipe cross section) due to longitudinal forces caused by displacement strains are not normally considered in the determination of displacement stress range, since this stress is not significant in typical piping layouts. In special cases, however, consideration of average axial displacement stress is necessary. Examples include buried lines containing hot fluids, double wall pipes, and parallel lines with different operating temperatures, connected together at more than one point.”

4. The modulus of elasticity at 21°C (70°F) is normally used in the analysis. As a result, no elastic modulus adjustment is required. For a more detailed discussion, see below, Section 8.8.

Flexibility factors for typical components are included in Appendix D of ASME B31.3. The flexibility factor is the length of straight pipe having the same flexibility as the component divided by the centerline length of the component. It can be used in hand calculations of piping flexibility, and is included in all modern piping stress analysis programs.

Additional flexibility is introduced in the system by elbows. Elbows derive their flexibility from the fact that the cross section ovalizes when the elbow is bent. This ovalization reduces the moment of inertia of the pipe cross section, reducing its stiffness and increasing its flexibility. Note that the presence of a flange at the end of an elbow will reduce the ability of the elbow to ovalize, and thus Appendix D of ASME B31.3 provides reduced flexibilities for elbows with flanges welded to one or both ends.

8.2 WHEN IS FORMAL FLEXIBILITY ANALYSIS REQUIRED?

ASME B31.3 requires a formal flexibility analysis unless certain exemption criteria are met, as provided in para. 319.4.1. A formal flexibility analysis does not mean a computer stress analysis; it can be by simplified, approximate, or comprehensive methods. These vary from simple charts and methods such as Kellogg's guided cantilever method to detailed computer stress analysis of the piping system.

There are three exemptions from formal flexibility analysis:

- A system that duplicates or replaces, without significant change, a system operating with a successful service record
- A system that can be readily judged adequate by comparison with previously analyzed systems
- A system that satisfies the simplified equation (16) of ASME B31.3, reproduced below as Eq. (8.1)

The first exemption is basically a means of grandfathering a successful design. The difficulty comes when trying to determine how long a system must operate successfully in order to demonstrate that the design is acceptable. Considering that some systems may cycle less than one time per year and the design criteria are based on fatigue considerations, the fact that a piping system has not resulted in failure for some period of time provides little indication that it actually complies with the Code or that it will not fail sooner or later. Interpretation 13-14 on this subject reads as follows:

“Question: In accordance with para. 319.4.1, how many years of operation or number of operating cycles are required to qualify a piping system as having a successful service record?”

“Reply: Such determination is the responsibility of the designer. See para. 300(b)(2).”

This matter is left as a judgment call, but should not be blindly used to accept a design without careful consideration.

The second exemption relies on the judgment of an engineer or designer who, based on experience, can determine that a system has adequate flexibility.

The third exemption uses a simplified condition that has limited applicability. The requirements for its use are that the system is of uniform size, has no more than two points of fixation, and has no intermediate restraints. The equation reads¹⁴

$$\frac{Dy}{(L-U)^2} \leq K_1 \quad (8.1)$$

where

D = outside diameter of pipe, mm (in.)

$K_1 = 208,000 S_A / E_a$, (mm/m)², for SI units

$= 0.30 S_A / E_a$, (in./ft)², for U.S. customary units

E_a = reference modulus of elasticity at 21°C (70°F), MPa (ksi)

S_A = allowable displacement stress range per Eq. (1a) of ASME B31.3 [Eq. (7.1) of this book], MPa (ksi)

L = developed length of piping between anchors, m (ft)

U = anchor distance, straight line between anchors, m (ft)

y = resultant of total displacement strains, mm (in.), to be absorbed by the piping system; displacements of equipment to which the pipe is attached (end displacements of the piping system) should be included

This equation tends to be very conservative. However, there are a number of warning statements within the Code regarding limits to its applicability. One such warning is for near-straight sawtooth runs, which are described below in Insert 8.1. In addition, the equation provides no indication of the end reactions, which would need to be considered in any case for load-sensitive equipment.

¹⁴ASME B31.3, Eq. (16).

8.3 WHEN IS COMPUTER STRESS ANALYSIS TYPICALLY USED?

The Code does not indicate when computer stress analysis is required. It is difficult to generalize when a particular piping flexibility problem should be analyzed by computer methods, because this depends on the type of service, actual piping layout and size, and severity of temperature. However, there are quite a few guidelines in use by various organizations that indicate which types of lines should be evaluated by computer in a project. These tend to require that lines at higher combinations of size and temperature or larger lines that are attached to load-sensitive equipment be computer-analyzed. One set of recommended criteria is provided below.

- (a) In the case of general piping systems; according to the following line size/flexibility temperature criteria:
 - All DN 50 (NPS 2) and larger lines with a design differential temperature over 260°C (500°F)
 - All DN 100 (NPS 4) and larger lines with a design differential temperature exceeding 205°C (400°F)
 - All DN 200 (NPS 8) and larger lines with a design differential temperature exceeding 150°C (300°F)
 - All DN 300 (NPS 12) and larger lines with a design differential temperature exceeding 90°C (200°F)
 - All DN 500 (NPS 20) and larger lines at any temperature
- (b) All DN 75 (NPS 3) and larger lines connected to rotating equipment
- (c) All DN 100 (NPS 4) and larger lines connected to air fin heat exchangers
- (d) All DN 150 (NPS 6) and larger lines connected to tankage
- (e) Double-wall piping with a design temperature differential between the inner and the outer pipe greater than 20°C (40°F)

Again it is emphasized that the intent of the above criteria is to identify in principle only typical lines that should be considered at least initially for detailed stress analysis. Obviously, the final decision as to whether or not a computer analysis should be performed should depend on the complexity of the specific piping layout under investigation and the sensitivity of equipment to piping loads.

Just because a line may pass some exemption criteria from computer stress analysis does not mean that it is exempt from other forms of formal analysis, nor that it will always pass the Code criteria if analyzed in detail. The procedure is intended to separate the more trouble-free types of systems from those that are more subject to overload or overstress. The lines exempted from computer stress analysis are considered to be more likely to be properly laid out with sufficient flexibility by the designer.

8.4 STRESS INTENSIFICATION FACTORS

Stress intensification factors are used to relate the stress in a component to the stress in nominal-thickness straight pipe. As discussed in the previous section, the analysis is based on nominal pipe dimensions, so the calculated stress is the stress in straight pipe unless some adjustment is made. The stress can be higher in components such as branch connections.

Stress intensification factors that relate the stress in components to that in butt-welded pipe have been developed from so-called Markl fatigue testing of piping components, which generally follow the procedures developed by Markl (1960a-d).

As mentioned previously, Markl developed a fatigue curve for butt-welded pipe. It was based on displacement-controlled fatigue testing, bending the pipe in a cantilever bending mode. Figure 8.1 shows a Markl-type fatigue test machine. Use of a butt-welded pipe fatigue curve has several practical advantages. One is that the methodology was developed for butt-welded pipe, and the stress analyst typically does not know where the welds will be in the as-constructed system. Using a butt-welded pipe

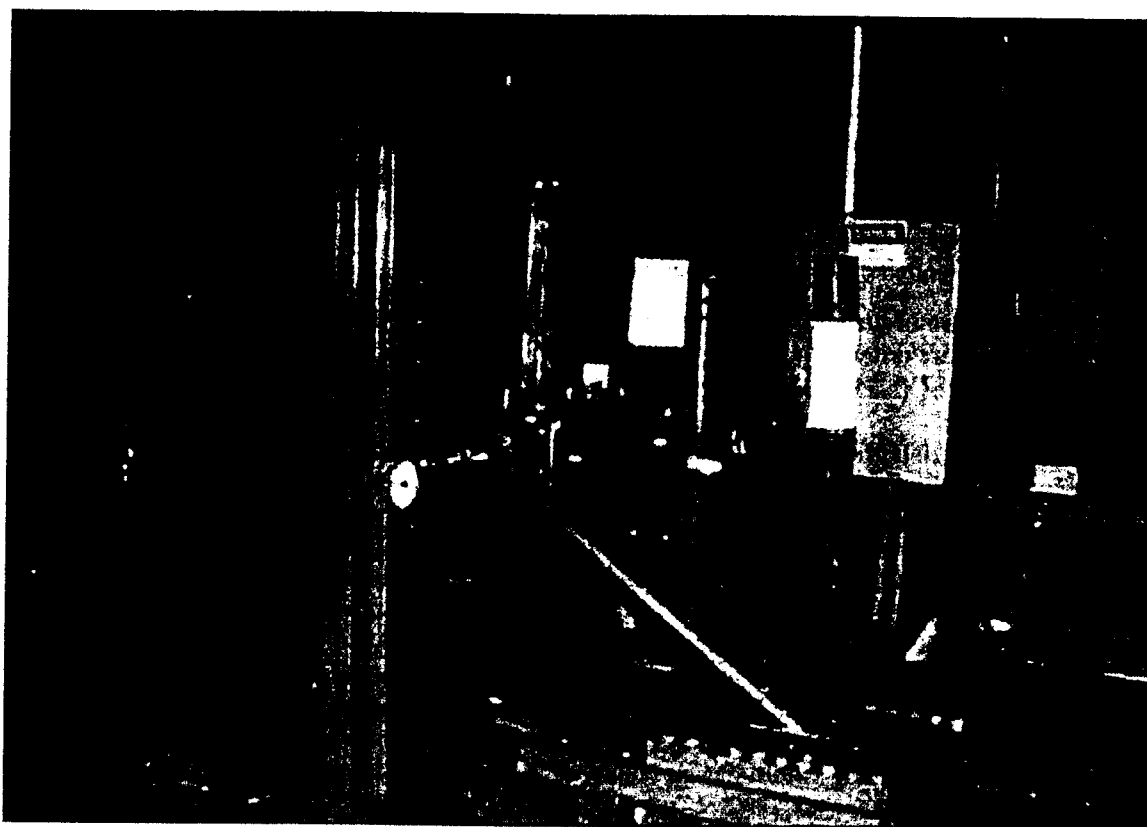


FIG. 8.1
MARKL-TYPE FATIGUE TEST MACHINE (COURTESY OF WFI)

fatigue curve as the baseline fatigue curve allows for the possibility that butt welds could be anywhere in the system. Furthermore, from a testing standpoint, appropriate fatigue curves could not readily be developed for straight pipe without welds in a cantilever bending mode, because the failure will occur at the point of fixity, where effects of the method of anchoring the pipe could significantly affect results.

The stress intensification factors were developed from component fatigue testing. The stress intensification factor is the nominal stress from the butt-welded pipe fatigue curve at the number of cycles to failure in the component test, divided by the nominal stress in the component. The nominal stress in the component is the range of bending moment at the point of failure divided by the section modulus of matching pipe with nominal wall thickness. In a flexibility analysis, it is precisely this nominal stress that is calculated. When the nominal stress is multiplied by the stress intensification factor and then compared to the fatigue curve for butt-welded pipe, one can determine the appropriate number of cycles to failure of the component.

Different stress intensification factors are provided in ASME B31.3 for in-plane and out-plane loads. The direction of these moments are illustrated in Fig. 8.2.

One of the less well known aspects of piping flexibility analysis per the ASME B31 Codes is that in piping stress analysis, the calculated stress range due to bending loads is about one-half of the peak stress range. This is because the stress concentration factor for typical as-welded pipe butt welds is two. Since the stresses are compared to a butt-welded pipe fatigue curve, one-half of the actual peak stresses is calculated. Thus, the theoretical stress, for example, in an elbow due to bending loads is two times what is calculated in a piping flexibility analysis following Code procedures. This is not significant when performing standard design calculations, because the Code procedures are

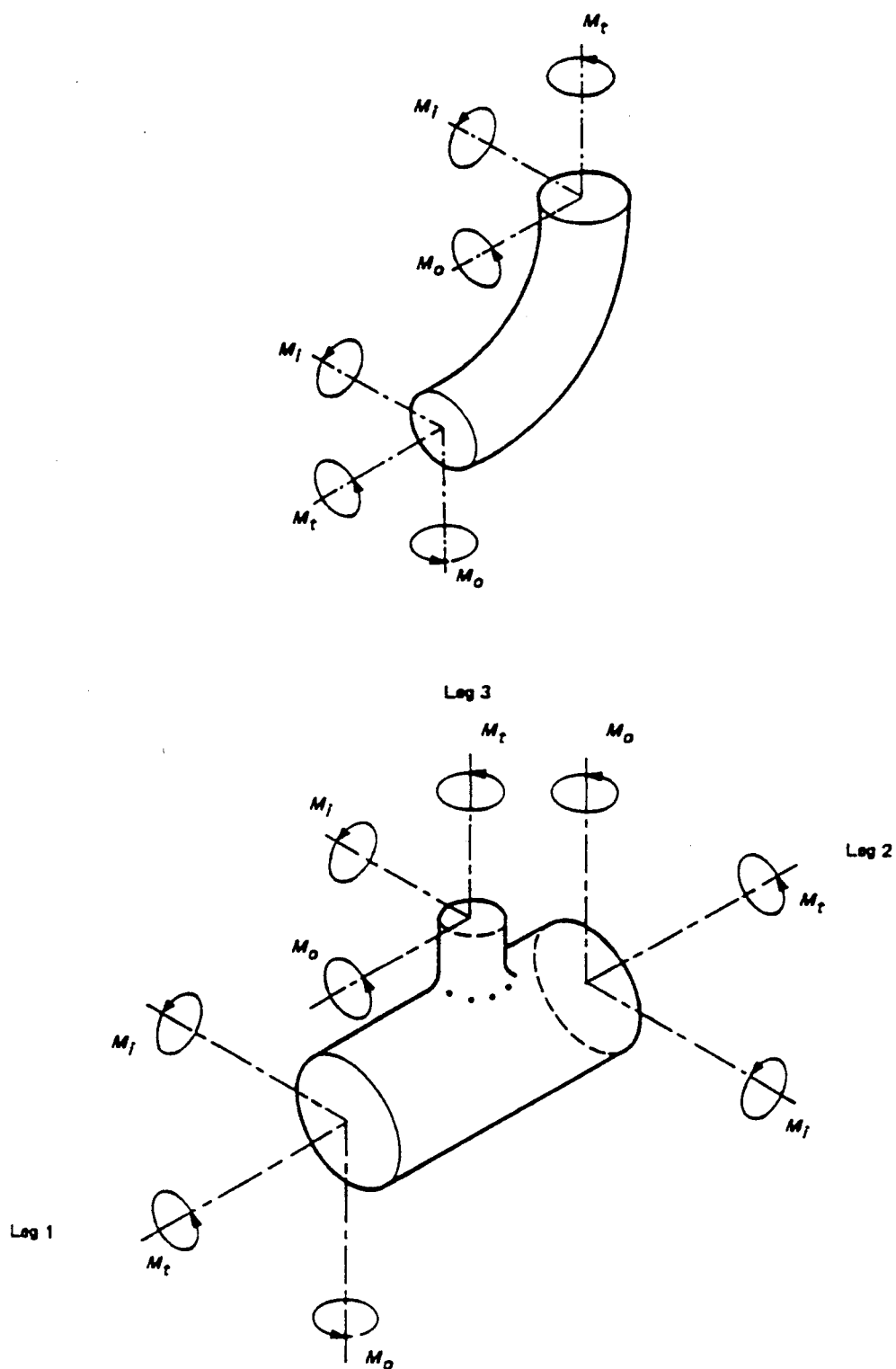


FIG. 8.2
IN-PLANE AND OUT-PLANE BENDING MOMENTS IN BENDS AND BRANCH CONNECTIONS
(ASME B31.3, FIGS. 319.4.4A AND 319.4.4B)

self-consistent. However, it can be very significant when trying to do a more detailed analysis, for example, in a fitness-for-service assessment.

One example occurs in the design of high-pressure piping in ASME B31.3, Chapter IX. This chapter requires a detailed fatigue analysis, using polished bar-type fatigue curves (rather than butt-welded pipe fatigue curves). When calculating stresses due to piping thermal expansion via a flexibility analysis, these calculated stresses must be multiplied by a factor of two in order to arrive at the actual stress range. This should not be confused with the difference between stress range and stress amplitude, which is an additional consideration.

Another commonly misunderstood item relates to where the peak stresses are in an elbow, as this is counterintuitive. With in-plane bending (see Fig. 8.2 for in-plane versus out-plane directions) of an elbow, the highest stresses are not at the intrados or extrados, they are in the elbow sidewalls (crown). They are through-wall bending stresses due to ovalization of the elbow. Again, this is not significant in design, but can be so in failure analysis or fitness-for-service evaluations.

Stress intensification factors were developed for a number of common components by Markl around 1950. More recently, as a result of some findings of nonconservatism and the development of newer products, additional fatigue tests have been performed. Results of these tests and analyses are being considered on an ongoing basis for improvement of the present stress intensification factors contained in Appendix D. Stress intensification and flexibility factors are provided in Appendix D of ASME B31.3, which are for use in the absence of more directly applicable data (see para. 319.3.6). This means that a Designer could use different factors if they are based on more applicable data. The stress intensification factors in Appendix D are based on Committee judgment using available fatigue test data. The B31 Mechanical Design Committee is performing ongoing evaluations of available and new data to improve these stress intensification factors.

A new standard detailing the procedures for performing fatigue testing to develop stress intensification factors is under development. This will be ASME B31J, *Standard Method to Develop Stress Intensification and Flexibility Factors for Piping Components*.

8.5 FLEXIBILITY ANALYSIS EQUATIONS

ASME B31.3 provides the following set of equations for calculating the stress due to thermal expansion loads in piping systems. The displacement stress range, S_E , is calculated from the combination of stresses due to bending and shear stresses due to torsion. These are combined per Mohr's circle,¹⁵

$$S_E = \sqrt{S_b^2 + 4S_t^2} \quad (8.2)$$

where

S_b = resultant bending stress

S_t = torsional stress

There are no torsional stress intensification factors in ASME B31.3, so the shear stress is given by

$$S_t = \frac{M_t}{2Z} \quad (8.3)$$

where

M_t = torsional moment

Z = section modulus of pipe (this is the section modulus of matching nominal pipe)

¹⁵ ASME B31.3, Eq. (17).

For other than reducing outlet branch connections, the resultant bending stress, S_b , is calculated as¹⁶

$$S_b = \frac{\sqrt{(i_i M_i)^2 + (i_o M_o)^2}}{Z} \quad (8.4)$$

where

- M_i = in-plane bending moment
- M_o = out-plane bending moment
- i_i = in-plane stress intensification factor
- i_o = out-plane stress intensification factor

The directions of in-plane and out-plane moments are illustrated in Fig. 8.2.

For reducing branch connections, an equivalent section modulus is used in the calculation of stresses in the branch connection, with Z_e substituted for Z ,

$$Z_e = \pi r_2^2 T_S \quad (8.5)$$

where

- T_S = lesser of $\bar{T}_h$ and $i_i \bar{T}_b$, effective branch wall thickness
- $\bar{T}_b$ = thickness of pipe matching branch
- $\bar{T}_h$ = thickness of pipe matching run of tee or header, exclusive of reinforcing elements
- r_2 = mean branch cross-sectional radius

8.6 COLD SPRING

Cold spring is the deliberate introduction of a cut short in the system to offset future thermal expansion. It is used to reduce loads on equipment; however, it does not affect the strain range. Thus, the cold spring does not affect the calculation of the displacement stress range, S_E , or the allowable stress range, S_A .

The range of loads on equipment due to thermal expansion is also unchanged by cold spring. However, the magnitude of the load at any given operating condition can be changed. Cold spring is typically used to reduce the load in the operating condition. It does this by shifting the load to the nonoperating, ambient condition.

The effectiveness of cold spring is generally considered to be questionable, but it is occasionally the only reasonable means of satisfying equipment load limits. It should not be used indiscriminately. Although it may provide an easy way for an analyst to solve an equipment load problem, there are a number of considerations. Its implementation in the field is generally difficult to achieve accurately. After the plant has been operated, deliberately installed cold spring can be misunderstood to be piping misalignment and "corrected." Additionally, when evaluating an existing piping system that is designed to include cold spring, it is highly questionable what the actual condition is.

Because of difficulties in accomplishing desired cold spring, ASME B31.3 only permits credit for two-thirds of the cold spring that is designed into the system. Equations are provided in para. 319.5.1 for calculating the maximum reaction force or moment, including cold spring, at the operating condition and at the piping installation temperature. The equations are limited to a two-anchor system without

¹⁶ASME B31.3, Eq. (18).

intermediate restraints. Currently, any piping requiring cold spring would most likely be evaluated using computer flexibility analysis. In such an analysis, the load in the operating condition should be calculated using two-thirds of the design cold spring, and the load in the ambient condition should be calculated using the full cold spring.

Insert 8.1 How to Increase Piping Flexibility

There are a number of means of increasing piping flexibility. The intent of this insert is to describe some general concepts in order to better enable the designer to develop increased flexibility in piping systems.

One alternative is to add expansion joints. However, this is usually undesirable. Although it may make the job easier for the analyst, it typically reduces the overall reliability of the piping system. Expansion joints should be the solution of last resort.

Here are a few fundamental concepts in flexibility:

- (a) Flexibility is roughly proportional to the cube of the length of a straight run of pipe. Therefore, longer straight runs are much more flexible than a series of shorter runs chopped up into short lengths and separated by elbows.
- (b) Flexibility can be provided by a length of piping perpendicular to the direction of thermal expansion that must be accommodated.
- (c) In general, the further the center of gravity of the piping system is from a straight line between piping system end points (for a two-anchor system), the lower are the thermal expansion stresses.

These concepts are illustrated in the classic near-sawtooth piping configuration. The question is whether the system in Fig. 8.3 or that in Fig. 8.4 is more flexible. The reader should look and decide before reading the next paragraph.

The thermal stresses in the second figure are significantly lower than those in the first figure. To resolve any doubt, run a flexibility analysis for both cases. The first figure is considered a near-sawtooth pipe configuration. The fact that it has additional elbows does not compensate for the fact that the pipe has been chopped into a number of smaller lengths. By either measure (a) or (c) above, Fig. 8.4 can be visually discerned to be more flexible.

The concepts related above are simple but powerful when understood, in application to flexibility problems.

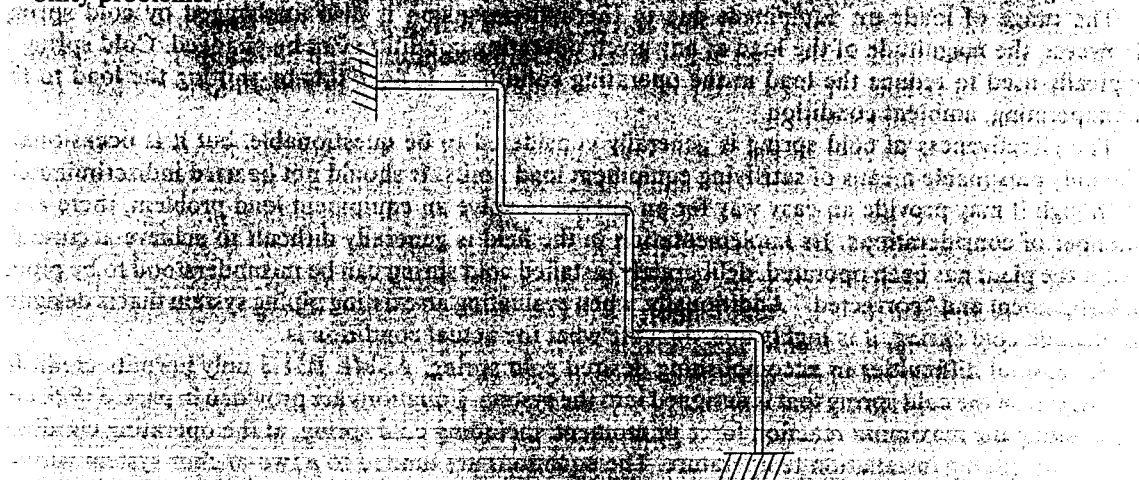


FIG. 8.3
PIPING LAYOUT 1

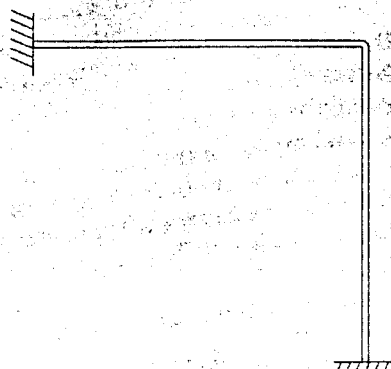


FIG. 8.4
PIPING LAYOUT 2

8.7 ELASTIC FOLLOW-UP/STRAIN CONCENTRATION

The analysis procedures in the Code essentially assume that the strain range in the system can be determined from an elastic analysis. That is, strains are proportional to elastically calculated stresses. The stress range is limited to less than two times the yield stress, in part to achieve this. However, in some systems, strain concentration or elastic follow-up occurs. A classic reference for elastic follow-up due to creep conditions is Robinson (1960).

As an example, consider a cantilevered pipe with a portion adjacent to the fixed end constructed with a reduced-diameter or -thickness pipe or lower-yield-strength material. The elastic analysis assumes that strains will be distributed in the system in accordance with the elastic stiffnesses. However, consider what happens when the locally weak section yields. As the material yields, a greater proportion of additional strain due to displacement occurs in the local region, because its effective stiffness has been reduced by yielding of the material. Thus, there is plastic strain concentration in the local region. In typical systems, this strain concentration is generally not considered to be significant. However, it can be highly significant under specific conditions, such as unbalanced systems. Paragraph 319.2.2(b) provides warnings regarding these conditions.

An example of a two-bar system under axial compression is provided to illustrate elastic follow-up, although the concern in piping is generally bending. However, the axial compression case illustrates the problem.

The problem is illustrated in Fig. 8.5. The elastic distribution of the total displacement, Δ , between bar A and bar B is as follows.

$$\Delta_A = \frac{K_B}{K_A + K_B} \Delta \quad (8.6)$$

$$\Delta_B = \frac{K_A}{K_A + K_B} \Delta \quad (8.7)$$

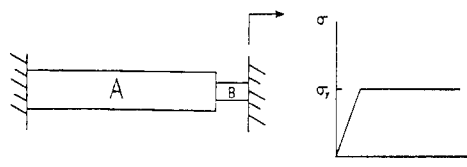


FIG. 8.5
STRAIN CONCENTRATION TWO-BAR MODEL

where

- Δ_A = displacement absorbed by bar A
- Δ_B = displacement absorbed by bar B
- $K_A = A_A E_A / L_A$, elastic stiffness of bar A
- $K_B = A_B E_B / L_B$, elastic stiffness of bar B
- A_A = area of bar A
- A_B = area of bar B
- E_A = elastic modulus of bar A
- E_B = elastic modulus of bar B
- L_A = length of bar A
- L_B = length of bar B
- Δ = total axial displacement imposed on two bars

For elastic, perfectly plastic behavior of the material with a yield stress σ_y , the stress in bar B cannot exceed yield, so the load in bar A cannot exceed $\sigma_y A_B$. Therefore, after bar B starts to yield, the displacement in each bar is given by

$$\Delta_A^{e-p} = \frac{\sigma_y A_A}{K_A} \quad (8.8)$$

$$\Delta_B^{e-p} = \Delta - \Delta_A^{e-p} \quad (8.9)$$

where

- Δ_A^{e-p} = actual displacement absorbed by A , elastic plastic case
- Δ_B^{e-p} = actual displacement absorbed by B , elastic plastic case

The strain concentration is the actual strain in bar B , considering elastic plastic behavior, divided by the elastically calculated strain in bar B . Since strain is displacement divided by length, the strain concentration is given as

$$\text{strain concentration} = \frac{\Delta_B^{e-p}}{\Delta_B} = \frac{[\Delta - (\sigma_y A_B / K_A)] / L_B}{\{[K_A / (K_A + K_B)] \Delta\} / L_B} \quad (8.10)$$

This can be rearranged as

$$\text{strain concentration} = \left(1 + \frac{K_B}{K_A}\right) \left(1 - \frac{\sigma_y A_B}{\Delta K_A}\right) \quad (8.11)$$

If we consider a general region under lower stress or in a stronger condition coupled with a local region under higher stress or with weaker material (e.g., lower σ_y), then the more flexible is the general region (stiffer is the local region), the more severe is the elastic follow-up. This can be seen by considering Eq. (8.11). As the stiffness of bar B increases relative to bar A , the plastic strain concentration becomes more severe.

For a system subject to plastic strain concentration, the simplest solution in design is typically to limit the thermal expansion stress range to less than the yield strength of the material. This avoids plastic strain concentration by keeping the component elastic. Although the stresses due to sustained loads such as weight and pressure usually do not need to be added to those due to thermal expansion when satisfying this limit, proper consideration of this requires a very detailed understanding of the phenomenon. Thus, it is generally preferable to conservatively add the stresses due to sustained loads to the thermal expansion stresses in this type of evaluation.

Under creep conditions, elastic follow-up can have very severe effects. This is due to the implicit assumption that thermal expansion stresses will relax. Near-yield-level stresses cannot be sustained very long at the very high temperatures permitted in ASME B31.3 without rapid creep rupture failures. With elastic follow-up, creep strain in the local region does not result in a corresponding reduction in thermal expansion load/stress. In severe cases of elastic follow-up, rapid creep rupture failures can occur, even though the calculated stresses fall well within Code limits. The most straightforward solution to this condition is to design the system such that the level of thermal expansion stresses in the local region subject to elastic follow-up does not exceed S_h .

8.8 EFFECT OF ELASTIC MODULUS VARIATIONS DUE TO TEMPERATURE

In a typical flexibility analysis, the elastic modulus at a reference temperature of 21°C (70°F) is used. It was changed to this from the elastic modulus at installation temperature in the 1999 Edition, to be simpler to apply and more theoretically correct.

Fatigue, in terms of strain range versus cycles to failure, is generally considered to be independent of temperature at temperatures where creep effects are not significant. Markl fatigue testing was done at ambient temperature, and thus an ambient temperature fatigue curve was developed. If one were to calculate the stress in a flexibility analysis using the elastic modulus at temperature, one would need to divide this by the elastic modulus at temperature and multiply this result by the ambient temperature elastic modulus to compare it to the Markl fatigue curve. Dividing the stress by hot elastic modulus yields the strain, which is then multiplied by ambient temperature elastic modulus to yield the stress range at ambient temperature. This is the procedure used in fatigue analysis per ASME BPVC, Section VIII, Division 2.

It is recognized by the Code Committee that the analysis could be done with the hot modulus of elasticity, with the calculated stress multiplied by the ratio of the cold modulus to the hot modulus, and that in some cases it may be preferable to do this. The following text was recently added [para. 319.2.2(4)]:

“When differences in the elastic modulus within a piping system will significantly affect the stress distribution, the resulting displacement stresses shall be computed based on the actual elastic moduli at the respective operating temperatures for each segment in the system and then multiplied by the ratio of the elastic modulus at ambient temperature to the modulus used in the analysis for each segment.”

SUPPORTS

9.1 OVERVIEW OF SUPPORTS

Requirements for supports and other devices to restrain the piping are provided in Chapter II of ASME B31.3, specifically in Section 321, Piping Support.

ASME B31.3 provides general requirements for piping supports as well as descriptions of conditions for which they must be designed. The support elements (e.g., springs, hanger rods, etc.) are within the scope of ASME B31.3, but the support structures to which they are attached are not. The supports must achieve the objectives in the design of the piping for sustained and occasional loads as well as thermal displacement.

9.2 MATERIALS AND ALLOWABLE STRESS

Pipe support elements may be constructed from a variety of materials, including the materials listed in the allowable stress tables, other metallic materials, steel of unknown specification, and wood and other materials. Some specific requirements are provided in para. 321.1.4 and the allowable stress basis is provided in para. 321.1.3 [para. 321.1.4(c) for steel of unknown specification].

Material that is bonded or welded to the pipe must be compatible with the piping and service and must be of known specification. Material that is not welded directly to the pressure-containing piping component may be of unknown specification. Material of unknown specification may also be welded to a pad or sleeve of known specification that is welded in turn to the pressure boundary.

Steel of unknown specification is assigned a basic allowable stress in tension or compression of 82 MPa (12 ksi) and is limited to metal temperatures in the range of -29°C to 343°C (-20°F to 650°F).

The allowable stress for other materials, except for springs and materials for which the criteria are not applicable (e.g., wood) is required to be in accordance with para. 302.3.1. This makes the allowable stress for supports (via reference to para. 302.3.2 and Appendix A) the same as for other piping components. This includes the limits on shear and bearing, which are 1.6 times the basic allowable stress in tension. However, weld joint quality factors are not used for supports. Structural stability (i.e., buckling) must also be given due consideration for elements in compression. Although there is no specific margin specified for structural elements in compression, a design margin of two (about equal to the margin provided for elastic buckling in the American Institute of Steel Construction Structural Steel Code) to three (consistent with ASME B31.3 external pressure buckling margin) should be considered.

Recommendations are made for cast, ductile, and malleable iron with respect to their use in pipe support elements. Cast iron is not recommended for elements that may be in tension (due to the brittle nature of the material). Cast iron is also not recommended for services where the piping may be subject to impact-type loading resulting from pulsation or vibration. These limitations are not

applied to ductile or malleable iron. However, para. 321.1.4(c) specifically permits use of ductile and malleable iron only for pipe and beam clamps, hanger flanges, clips, brackets, and swivel rings.

9.3 DESIGN OF SUPPORTS

ASME B31.3, Section 321 discusses the types of loads and conditions that must be considered. Detailed criteria are generally not provided, with the exception of the allowable stress. The objectives to be satisfied are described in para. 321.1.1, reproduced below. Layout and design of piping and its supporting elements shall be directed toward preventing the following:

- “(a) piping stresses in excess of those permitted in this Code [ASME B31.3];
- (b) leakage at joints;
- (c) excessive thrusts and moments on connected equipment (such as pumps and turbines);
- (d) excessive stresses in the supporting (or restraining) elements;
- (e) resonance with imposed or fluid-induced vibrations;
- (f) excessive interference with thermal expansion and contraction in piping which is otherwise adequately flexible;
- (g) unintentional disengagement of piping from its supports;
- (h) excessive piping sag in piping requiring drainage slope;
- (i) excessive distortion or sag of piping (e.g., thermoplastics) subject to creep under conditions of repeated thermal cycling;
- (j) excessive heat flow, exposing supporting elements to temperature extremes outside their design limits.”

These are generally requirements for sustained and thermal loads, stated elsewhere in the Code as well as in the section on supports.

The pipe support elements must be designed for all the loads to which they can be subjected, including surge, thermal expansion, and weight. In addition, for supports that can slide, the lateral load due to friction must be considered [paras. 321.2.1(b) and 321.2.2(c)].

The allowable load for all threaded parts (e.g., rods and bolts) is required to be based on the root diameter [para. 321.2.2(b)]. Screw threads are required to be per ANSI B1.1 unless other threads are required for adjustment under heavy loads. Turnbuckles and adjusting nuts are required to have full thread engagement and all threaded adjustments are required to be provided with a means of locking (e.g., locknut) (para. 321.1.5).

General requirements for spring supports, constant-effort supports, counterweight supports, and hydraulic supports are provided in paras. 321.2.3 and 321.2.4.

Paragraph 321.3 requires consideration of stresses in the pipe caused by pipe attachments. This includes local stresses due to lugs, trunions, and supporting elements welded to the pipe or attached by other means. The stated criteria are performance-based (e.g., shall not cause undue flattening of the pipe, excessive localized bending stress, or harmful thermal gradients in the pipe wall). Specific criteria are not provided. However, the design-by-analysis rules of ASME BPVC, Section VIII, Division 2, Appendix 4, would be considered to be an acceptable approach to evaluating these stresses. Insert 9.2 below describes allowable stress criteria of Appendix 4. Approaches to calculating the stresses due to loads on attachments are provided in Bednar (1986), WRC 107 (Wichman et al, 1979), and WRC 297 (Mershon et al, 1984).

Although it is not referred to in the pipe support section, there is a referenced component standard for pipe supports in Table 326.1. This standard is MSS SP-58, *Pipe Hangers and Supports—Materials, Design, and Manufacture*. Piping supports per this standard are acceptable for ASME B31.3 piping systems.

Insert 9.1 Spring Design

Springs are used to provide continued support, while permitting the pipe to move vertically. There are variable-effort springs, which vary supporting force when the pipe moves up or down, and constant-effort springs, which provide a nearly constant supporting force as the pipe moves up or down. Because it is normally desirable to limit the change in load in the spring caused by pipe movement (typically a 25% limit on load change), when the movement becomes large, constant-effort supports are considered.

The three primary considerations in selecting a variable-effort spring are for the spring to provide sufficient support (i.e., be strong enough), sufficient movement capability to accommodate the movement of the pipe, and sufficiently low stiffness to prevent excessive change in load as a result of the pipe movement.

Springs are typically selected from charts. An example is provided in Fig. 9.1. The first step is to find the design load and movement. The design load is used to make the initial selection of spring size. For example, if the load is 3600 lb, a size 13 or size 14 variable spring could be selected. The movement can then be considered by calculating the maximum permissible stiffness or by looking at the spring chart. The maximum stiffness can be determined from the equation

$$K_{\max} = \frac{\text{force} \times \text{variation}}{\text{displacement}}$$

where

$K_{\max}$ = maximum spring stiffness

force = required supporting force from the spring

variation = allowable load variation, expressed as a fraction

displacement = vertical movement of piping

The allowable load variation is typically 25%, or a fraction of 0.25.

Continuing with the same example, for a movement of 1 in. and an allowable variation of 0.25, the maximum allowable spring stiffness would be 900 lb/in. Springs that are more flexible would be acceptable, because their load variation caused by the piping movement would be less than 25%. Looking at the bottom of the chart in Fig. 9.1, we see that a size 14, figure 268 spring would be acceptable. The stiffness of this spring is 800 lb/in., which is less than 900 lb/in. A size 13 spring is also acceptable from a stiffness standpoint.

The spring should be installed at a load such that the supporting force in the operating condition will be the desired supporting force, 3600 lb in this example. If the movement from ambient to operating condition is up, the spring supporting force will be reduced. Thus, the load in the installed, nonoperating or cold condition would need to be higher. The required load would be the desired force in the operating condition plus the spring stiffness times the displacement. In this example, with a size 14, figure 268 spring, the initial force in the cold condition would be 4400 lb [3600 (lb) + 1 (in.) × 800 (lb/in.)]. Noting that this force is within the operating range of a size 14 spring, 2800 lb to 4800 lb, we see that the spring is an acceptable choice. A size 13 spring would not be acceptable, because the required cold load, 4200 lb, exceeds the maximum design load of the spring, 3600 lb.

Another way to approach the problem is to look at the spring charts. Looking at a size 14, figure 268 spring, we can use the displacement scale at the left to determine what the cold setting of the spring will be and if it will be in the appropriate range. Noting that the spring will be moving up (lesser displacement) as it goes from cold to hot, and the movement is 1 in., one can look at the displacement scale and determine that 1 in. down from an operating load setting of 3600 lb

(1 in. on the load scale), the cold load would be 4400 lb (2 in. on the load scale), and it would remain in scale. The next step would be to check load variation; the difference between hot and cold settings divided by the hot setting, and make sure it is within acceptable limits (typically 25%).

SIZE AND SERIES SELECTION

HOW TO USE HANGER SELECTION TABLE:

In order to choose a proper size hanger, it is necessary to know the actual load which the spring is to support and the amount and direction of the pipe line movement from the cold to the hot position:

Find the actual load of the pipe in the load table. As it is desirable to support the actual weight of the pipe when the line is hot, the actual load is the hot load.

To determine the cold load, read the spring scale, up or down, for the amount of expected movement. The chart must be read opposite from the direction of the pipe's movement. The load arrived at is the cold load.

If the cold load falls outside of the working load range of the hanger selected, relocate the actual or hot load in the adjacent column and find the cold load. When the hot and cold loads are both within the working range of a hanger, the size number of that hanger will be found at the top of the column.

Should it be impossible to select a hanger in a particular series such that both loads occur within the working range, consideration should be given to a variable spring hanger with a wider working range or a constant support hanger.

The cold load is calculated by adding (for up movement) or subtracting (for down movement) the product of spring rate times movement to or from the hot load. Cold load = hot load $\pm$ movement $\times$ spring rate.

A key criteria in selecting the size and series of a variable spring is a factor known as variability. This is a measurement of the percentage change in supporting force between the hot and cold positions of a spring and is calculated from the formula:

$$\text{Variability} = \frac{\text{Movement} \times \text{Spring Rate}}{\text{Hot Load}}$$

If an allowable variability is not specified, good practice would be to use 25% as specified by MSS-SP58

load table in pounds: for selection of hanger size

Quadruple	Triple	working range, in.				hanger size																						spring deflection, in.																			
		fig. 88	fig. 268	fig. 82	fig. 98	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	fig. 82	fig. 268	fig. 98																
						7	19	43	63	81	105	141	189	252	336	450	600	780	1020	1350	1800	2400	3240	4500	6000	7980	10610	14100	18750	25000	0	0	0														
						7	20	44	64	84	109	147	197	263	350	468	628	813	1063	1406	1876	2500	3378	4688	6290	8322	11053	14888	19831	26507																	
						8	22	48	68	88	114	153	205	275	364	488	650	845	1105	1463	1950	2600	3510	4775	6500	8655	11485	15273	20313	27089																	
						8	24	48	70	91	118	159	213	284	376	506	675	878	1148	1515	2025	2700	3648	5063	6790	8938	11838	15683	21084	28131																	
0	0	0	0	0	0	10	26	50	74	96	123	165	221	294	392	525	700	910	1180	1575	2100	2800	3760	5296	7000	9220	12280	16456	21875	29173	%	%															
						11	28	52	76	98	127	170	228	305	408	544	725	943	1233	1651	2175	2900	3816	5348	7250	9652	12823	17039	22506	30215																	
						12	30	54	79	101	131	176	236	318	424	563	780	975	1275	1698	2225	3000	4050	5626	7500	9986	13265	17625	23436	31256																	
						12	31	56	81	105	138	184	248	334	448	591	775	1008	1318	1744	2250	3100	4168	5812	7750	10317	13708	18213	24210	32208																	
2	1 1/2	1	1	1	1	14	34	58	84	110	146	198	265	356	476	616	800	1040	1360	1800	2400	3200	4320	6000	8050	10650	14100	18800	25000	33340	%	1	2														
						14	35	59	87	111	144	194	260	347	462	602	785	1023	1343	1800	2400	3200	4320	6000	8050	10650	14100	18800	25000	33482																	
						15	38	61	89	115	149	206	278	369	493	638	820	1058	1385	1850	2450	3250	4375	6075	8125	10725	14175	18900	25100	33542																	
						16	40	63	92	118	153	209	279	380	505	648	833	1073	1408	1880	2475	3275	4390	6090	8145	10745	14220	18950	25150	33592																	
4	2	2	2	2	2	17	41	65	95	122	160	212	284	380	504	647	830	1070	1410	1890	2480	3280	4400	6100	8150	10750	14260	19000	25200	33642																	
						18	43	67	97	125	162	217	291	388	510	654	840	1075	1420	1900	2500	3300	4425	6125	8175	10775	14280	19050	25250	33692																	
						18	46	68	100	128	166	222	298	390	512	657	843	1078	1425	1910	2510	3310	4437	6137	8187	10787	14290	19060	25260	33700	%	1 1/2	3														
						20	47	71	102	132	171	229	307	404	540	691	875	1088	1438	1925	2530	3330	4450	6150	8200	10800	14300	19070	25270	33710																	
6	4 1/2	3	1 1/2	1	1	21	49	73	105	136	175	236	318	420	560	708	900	1100	1450	1950	2550	3350	4475	6175	8225	10825	14325	19100	25300	33760																	
						21	50	74	108	138	178	241	323	424	564	712	908	1105	1455	1960	2560	3360	4487	6187	8237	10837	14337	19110	25310	33770																	
						22	52	76	110	140	182	246	328	430	570	718	918	1115	1465	1970	2570	3370	4500	6200	8250	10850	14350	19120	25320	33780																	
						23	55	78	113	143	186	251	333	436	580	728	928	1125	1475	1980	2580	3380	4512	6212	8262	10862	14362	19132	25332	33792																	
8	6	4	2	1	1	24	56	80	114	146	189	256	347	442	584	738	938	1135	1485	1990	2590	3390	4525	6225	8275	10875	14375	19145	25345	33805	1 1/2	2 1/2	5														
						25	58	82	116	148	192	261	354	451	594	748	948	1145	1495	2000	2600	3400	4537	6237	8287	10887	14387	19157	25357	33817																	
						26	60	84	119	151	195	261	360	463	604	758	958	1155	1495	2005	2608	3408	4540	6240	8290	10890	14390	19160	25360	33820																	
						27	62	86	123	155	200	270	370	464	608	762	962	1158	1500	2012	2612	3412	4543	6243	8293	10893	14393	19163	25363	33823																	
10	7 1/2	5	2 1/2	1 1/2	1 1/2	28	64	88	126	159	202	282	378	484	622	770	970	1168	1510	2020	2620	3420	4550	6250	8300	10900	14400	19170	25370	33830	1 1/2	3 1/2	6														
						28	66	89	129	165	214	288	386	515	666	819	1025	1253	1593	2083	2756	3675	4800	6450	9100	12250	16272	20788	27821	35152																	
						29	68	91	131	169	219	294	394	523	700	858	1025	1255	1625	2125	2815	3715	4850	6475	9125	12280	16335	20850	27900	35204																	
						30	70	93	134	172	223	300	402	536	714	868	1035	1265	1658	2168	2855	3755	4875	6503	9157	12290	16395	20857	27903	35206																	
						31	72	95	137	176	228	308	410	546	728	875	1040	1270	1690	2210	2925	3800	5200	7020	9750	13000	17300	23000	30550	40825	54176	1 1/2	3 1/2														
						spring rate — lb. per in.																																									
						82			30	42	54	70	94	128	168	224	300	400	520	680	900	1200	1600	2160	3000	4000	5320	7080	9400	12500	16670	82															
						268	7	15	15	21	27	35	47	63	84	112	150	200	260	340	450	600	800	1000	1500	2000	2660	3540	4700	6250	8335	268															
						98			7	10	13	17	23	31	42	56	75	100	130	170	225	300	400	540	750	1000	1330	1770	2360	3125	4167	98															
						Triple																																									
							5	7	9	12	16	21	26	37	50	67	87	113	150	200	267	360	500	687	887	1190	1595	2140	2860	3776	2778																
						Quadruple																																									
							4	5	7	9	12	16	21	28	38	50	65	85	113	150	200	270	375	500	685	885	1175	1583	2084																		

Most springs are figure 268. A figure 82 spring is twice as stiff (essentially a half-spring). A figure 98 spring is twice as flexible as a figure 268 spring (essentially two springs stacked). A triple is three times as flexible and a quadruple is four times as flexible. Figure 82 springs are typically used when there are space limitations and the stiffer spring is an acceptable choice. The more flexible springs are used to accommodate greater movement.

When the load variation for variable-support springs is too great, a constant-effort spring can be used. Figure 9.2 provides a typical selection table for constant-effort springs. A spring size is selected based on the desired movement and supporting force. Note that the chart requires a margin be placed on the travel. The greater of 1 in. or 20% is added to the calculated travel, and then rounded up to the nearest $\frac{1}{2}$ in.

If, for example, a constant support with a supporting force of 2100 lb and a calculated movement of 6.2 in. was required, a size 40 spring would be picked from the table in Fig. 9.2. This comes from looking in the column under total travel of 7.5 in. (6.2 in. plus 1 in., rounded up to the nearest $\frac{1}{2}$ in.) for a spring size with a load greater than or equal to 2100 lb.

Insert 9.2. Stress Classification

The design-by-analysis rules of ASME BVPC Section VIII, Division 2, Appendix 4 are used to assess the results of detailed stress analysis of nozzles and attachments to pressure vessels. These rules include consideration of primary, secondary, and peak stresses. Primary stresses are load-controlled stresses and have essentially the same allowable limit for membrane stress as the basic allowable stress provided in Appendix A. A primary stress is one that is necessary for satisfying the laws of equilibrium of external and internal forces and moments.

Primary stresses include primary membrane, primary bending, and local primary stress. Primary membrane stress is limited to the basic allowable stress; it is the average of the stress through the wall thickness. Primary bending stress is rarely a consideration in these evaluations. Although through-wall bending is often present, it is a secondary stress. A typical example of a primary bending stress in a pressure vessel is in a flat head.

Local primary membrane stress is a primary membrane stress that is in a limited region. To be classified local, the meridional (longitudinal in a pipe) extent over which the stress intensity exceeds 1.1 times the basic allowable stress cannot exceed $(Rt)^{1/2}$, where R is the midsurface radius of curvature measured normal to the surface from the axis of rotation and t is the minimum thickness in the region considered.

The limit for local primary membrane stress is 1.5 times the basic allowable stress. This higher stress can be used when the extent of the local stress region can be determined. This typically requires a detailed (e.g., finite-element) analysis.

Secondary stresses are deformation-controlled. The throughwall bending stresses in the vessel wall, calculated using WRC 107 (Wichman et al, 1979), WRC 297 (Mershon et al, 1984), Bednar (1986), finite-element analysis, or similar methods, are considered to be secondary stresses. Local yielding is considered to accommodate the imposed deformation, rather than result in gross failure. The combination of primary plus secondary stress is limited to three times the basic allowable stress (where the allowable stress is the average of the allowable stress at the temperature extremes of the condition being evaluated). At temperatures below the creep regime, this essentially limits the stress range to twice yield, or a shakedown limit (discussed in Section 7.1).

Peak stresses are highly localized and of concern in a fatigue analysis and limited in the ASME BVPC Section VIII, Division 2, by the Appendix 5 fatigue curves. However, these are not calculated in shell analysis, WRC 107 (Wichman et al, 1979), WRC 297 (Mershon et al, 1984), or Bednar (1986), although a stress concentration factor at the welds can be included. However, peak stresses are not generally considered in these evaluations and fatigue life calculations are generally not performed.

figs. 80-V and 81-H

model R

load travel
table

load in pounds for total travel in inches

constant supports

hanger size no.	total travel* in inches															
	1½	2	2½	3	3½	4	4½	5	5½	6	6½	7	7½	8	8½	
1	144	108	86	72	62	54	48	43	39	36	33	31	29	27		
2	173	130	104	87	74	65	58	52	47	43	40	37	35	33		
3	204	163	122	102	87	77	68	61	56	51	47	44	41	38		
4	233	175	140	117	100	88	78	70	64	58	54	50	47	44		
5	260	210	168	140	120	105	93	84	76	70	65	60	56	53		
6	327	245	196	163	140	123	109	98	89	82	75	70	66	61		
7	373	280	224	187	160	140	124	112	102	93	86	80	75	70		
8	451	338	270	225	193	169	150	135	123	113	104	97	90	85		
9	527	396	316	263	228	198	178	158	144	132	122	113	106	99		
10	600	450	360	300	257	225	200	180	164	150	138	129	120	113		
11	787	546	436	363	311	273	242	218	198	182	168	156	146	136		
12	881	636	510	425	365	319	284	255	232	213	196	182	170	160		
13	977	733	596	469	419	367	326	293	267	244	226	209	195	183		
14	1177	883	706	589	505	442	392	353	321	294	272	252	235	221		
15	1373	1030	824	687	599	516	456	412	375	343	317	294	275	258		
16	1573	1180	944	787	674	590	524	472	429	393	363	337	315	295		
17	1863	1420	1136	947	811	710	631	568	516	473	437	406	379	355		
18	2217	1663	1330	1109	950	832	739	665	605	564	512	475	443	416		
19	2540	1905	1524	1270	1099	963	847	762	693	636	586	544	508	478		
20	2925	2205	1760	1460	1255	1105	985	890	815	750	695	645	600	565		
21	3365	2565	2040	1680	1440	1260	1120	1010	925	850	785	730	680	635		
22	3865	2965	2360	1920	1640	1440	1280	1160	1060	980	905	840	785	735		
23	4425	3425	2720	2200	1880	1640	1480	1360	1260	1180	1105	1040	980	930		
24	5045	3945	3140	2560	2160	1880	1680	1560	1460	1380	1305	1240	1180	1130		
25	5725	4525	3620	2960	2480	2160	1960	1840	1740	1660	1585	1520	1460	1410		
26	6465	5165	4160	3400	2840	2480	2240	2120	2020	1940	1865	1800	1740	1690		
27	7265	5865	4760	3920	3280	2880	2640	2520	2420	2340	2265	2200	2140	2090		
28	8125	6625	5420	4500	3760	3280	2960	2840	2740	2660	2585	2520	2460	2410		
29	9045	7445	6140	5120	4280	3760	3440	3320	3220	3140	3065	3000	2940	2890		
30	10025	8325	6920	5800	4880	4320	4000	3880	3780	3700	3625	3560	3500	3450		
31	11065	9265	7760	6560	5560	4960	4640	4520	4420	4340	4265	4200	4140	4090		
32	12165	10265	8660	7360	6260	5660	5340	5220	5120	5040	4965	4900	4840	4790		
33	13325	11325	9620	8220	7020	6420	6100	5980	5880	5800	5725	5660	5600	5550		
34	14545	12545	10800	9400	8200	7600	7280	7160	7060	6980	6905	6840	6780	6730		
35	15825	13825	12080	10680	9480	8880	8560	8440	8340	8260	8185	8120	8060	8010		
36	17165	15165	13420	12020	10820	10220	9900	9780	9680	9600	9525	9460	9400	9350		
37	18565	16565	14820	13420	12220	11620	11300	11180	11080	11000	10925	10860	10800	10750		
38	20025	18025	16280	14880	13680	13080	12760	12640	12540	12460	12385	12320	12260	12210		
39	21545	19545	17800	16400	15200	14600	14280	14160	14060	13980	13905	13840	13780	13730		
40	23125	21125	19380	17980	16780	16180	15860	15740	15640	15560	15485	15420	15360	15310		
41	24765	22765	21020	19620	18420	17820	17500	17380	17280	17200	17125	17060	17000	16950		
42	26465	24465	22720	21320	20120	19520	19200	19080	18980	18900	18825	18760	18700	18650		
43	28225	26225	24480	23080	21880	21280	20960	20840	20740	20660	20585	20520	20460	20410		
44	30045	28045	26300	24900	23700	23100	22780	22660	22560	22480	22405	22340	22280	22230		
45	31925	29925	28180	26780	25580	24980	24660	24540	24440	24360	24285	24220	24160	24110		
46	33865	31865	30120	28720	27520	26920	26600	26480	26380	26300	26225	26160	26100	26050		
47	35865	33865	32120	30720	29520	28920	28600	28480	28380	28300	28225	28160	28100	28050		
48	37925	35925	34180	32780	31580	30980	30660	30540	30440	30360	30285	30220	30160	30110		
49	40045	38045	36300	34900	33700	33100	32780	32660	32560	32480	32405	32340	32280	32230		
50	42225	40225	38480	37080	35880	35280	34960	34840	34740	34660	34585	34520	34460	34410		
51	44465	42465	40720	39320	38120	37520	37200	37080	36980	36900	36825	36760	36700	36650		
52	46765	44765	43020	41620	40420	39820	39500	39380	39280	39200	39125	39060	39000	38950		
53	49125	47125	45380	43980	42780	42180	41860	41740	41640	41560	41485	41420	41360	41310		
54	51545	49545	47800	46400	45200	44600	44280	44160	44060	43980	43905	43840	43780	43730		
55	54025	52025	50280	48880	47680	47080	46760	46640	46540	46460	46385	46320	46260	46210		
56	56565	54565	52820	51420	50220	49620	49300	49180	49080	49000	48925	48860	48800	48750		
57	59165	57165	55420	54020	52820	52220	51900	51780	51680	51600	51525	51460	51400	51350		
58	61825	59825	58080	56680	55480	54880	54560	54440	54340	54260	54185	54120	54060	54010		
59	64545	62545	60800	59400	58200	57600	57280	57160	57060	56980	56905	56840	56780	56730		
60	67325	65325	63580	62180	60980	60380	60060	59940	59840	59760	59685	59620	59560	59510		
61	70165	68165	66420	65020	63820	63220	62900	62780	62680	62600	62525	62460	62400	62350		
62	73065	71065	69320	67920	66720	66120	65800	65680	65580	65500	65425	65360	65300	65250		
63	76025	74025	72280	70880	69680	69080	68760	68640	68540	68460	68385	68320	68260	68210		
64	79045	77045	75300	73900	72700	72100	71780	71660	71560	71480	71405	71340	71280	71230		
65	82125	80125	78380	76980	75780	75180	74860	74740	74640	74560	74485	74420	74360	74310		
66	85265	83265	81520	80120	78920	78320	78000	77880	77780	77700	77625	77560	77500	77450		
67	88465	86465	84720	83320	82120	81520	81200	81080	80980	80900	80825	80760	80700	80650		
68	91725	89725	87980	86580	85380	84780	84460	84340	84240	84160	84085	84020	83960	83910		
69	95045	93045	91300	89900	88700	88100	87780	87660	87560	87480	87405	87340	87280	87230		
70	98425	96425	94680	93280	92080	91480	91160	91040	90940	90860	90785	90720	90660	90610		
71	101865	99865	98120	96720	95520	94920	94600	94480	94380	94300	94225	94160	94100	94050		
72	105365	103365	101620	100220	99020	98420	98100	97980	97880	97800	97725	97660	97600	97550		
73	108925	106925	105180	103780	102580	101980	101660	101540	101440	101360	101285	101220	101160	101110		
74	112545	110545	108800	107400	106200	105600	105280	105160	105060	104980	104905	104840	104780	104730		
75	116225	114225	112480	111080	109880	109280	108960	108840	108740	108660	108585	108520	108460	108410		
76	120065	118065	116320	114920	113720	113120	112800	112680	112580	112500	112425	112360	112300	112250</		

constant supports

load travel table (continued from opposite page) hanger size Nos. 84 to 110 on next page

load in pounds for total travel in inches

hanger size no.	total travel* in inches															
	8	9½	10	10½	11	11½	12	12½	13	13½	14	14½	15	15½	16	
1																
2																
3																
4																
5																
6																
7																
8																
9																
10																
11																
12																
13																
14																
15																
16																
17																
18																
19	423	401	381													
20	480	458	438													
21	477	452	429													
22	519	492	467													
23	561	532	505													
24	602	571	542													
25	647	613	582													
26	691	655	622													
27	739	697	662													
28	807	764	728													
29	878	832	790													
30	949	899	854													
31	1008	956	907													
32	1068	1009	959													
33	1124	1065	1012													
34	1177	1115	1059													
35	1228	1163	1105													
36	1304	1238	1174	1063	1005	962	922	885	851	819	790					
37	1380	1307	1242	1183	1129	1080	1035	994	955	920	887					
38	1458	1379	1310	1248	1191	1139	1092	1048	1008	970	936					
39	1560	1478	1404	1337	1278	1221	1170	1123	1080	1040	1003					
40	1663	1576	1497	1428	1361	1302	1247	1196	1151	1109	1069					
41	1767	1674	1590	1514	1445	1383	1325	1272	1223	1178	1138					
42	1838	1830	1744	1661	1585	1518	1453	1395	1341	1292	1246					
43	2106	1997	1897	1807	1724	1649	1581	1518	1459	1405	1355					
44	2278	2158	2050	1952	1863	1782	1706	1640	1577	1518	1464					
45	2423	2299	2181	2077	1989	1899	1817	1745	1678	1615	1556					
46	2579	2449	2321	2210	2110	2018	1934	1857	1789	1719	1659					
47	2711	2598	2440	2324	2216	2122	2033	1952	1877	1807	1743					
48	2842	2730	2594	2470	2368	2265	2182	2075	1996	1921	1853					
49	3052	2891	2747	2616	2497	2389	2289	2196	2113	2035	1962					
50	3222	3063	2900	2762	2638	2522	2417	2320	2231	2148	2071	2001	1934	1871	1813	
51	3458	3274	3110	2962	2827	2704	2582	2488	2392	2304	2221	2145	2073	2006	1944	
52	3689	3485	3320	3162	3018	2887	2767	2668	2564	2469	2371	2289	2213	2142	2073	
53	3949	3741	3564	3394	3231	3080	2962	2843	2734	2632	2538	2451	2369	2293	2221	
54	4206	3986	3787	3606	3442	3289	3156	3030	2913	2805	2703	2612	2524	2443	2367	
55	4467	4231	4020	3828	3654	3495	3350	3219	3092	2978	2871	2772	2680	2593	2513	
56	4804	4548	4314	4103	4012	3839	3676	3531	3395	3269	3152	3044	2942	2847	2759	
57	5341	5080	4807	4518	4370	4180	4008	3846	3698	3561	3433	3315	3204	3101	3004	
58	5778	5474	5200	4902	4727	4521	4333	4180	4000	3852	3714	3585	3465	3355	3250	
59	6141	5818	5527	5263	5024	4806	4608	4422	4251	4084	3947	3811	3684	3565	3454	
60	6603	6181	5853	5574	5320	5089	4877	4682	4502	4335	4180	4036	3892	3778	3666	
61	6957	6506	6180	5886	5618	5374	5150	4944	4754	4578	4414	4262	4120	3987	3863	
62	7287	6812	6467	6154	5898	5670	5472	5284	5094	4904	4730	4579	4438	4305	4180	
63	7725	7319	6953	6621	6330	6048	5794	5562	5348	5150	4965	4795	4635	4488	4348	
64	8184	7725	7339	6989	6671	6361	6116	5871	5645	5436	5242	5061	4892	4734	4587	
"B"																
average inches	8½	9½	10½	11½	12½	13½	14½	15½	16½	17½	18½	19½	20½	21½	22½	

NOTE: Total Travel equals Actual Travel plus 1 inch or 20% (whichever is greater), rounded up to the nearest ½ inch as applicable.

Constant supports are readily available for travel and loads not listed in this table. Dimensions and lug locations may vary from those shown on the following pages.

ph-121

FIG. 9.2
CONTINUED

fig. 80-V and 81-H
model R

constant supports

See pages ph-120, 121, for sizes 1 to 63

load travel table

load in pounds for total travel in inches

hanger size no.	total travel* in inches															
	4	4 1/4	5	5 1/4	6	6 1/4	7	7 1/4	8	8 1/4	9	9 1/4	10	10 1/4	11	11 1/4
64	19225	17089	15380	13982	12818	11831	10986	10253	9613	9047	8544	8094	7690	7323	6990	6688
65	20100	17866	16080	14618	13400	12370	11486	10720	10050	9459	8933	8463	8040	7657	7308	6991
66	22058	19815	17854	16049	14711	13580	12610	11769	11034	10385	9808	9291	8827	8406	8024	7675
67	24033	21382	19226	17478	16021	14790	13733	12817	12016	11310	10681	10119	9613	9154	8738	8359
68	26000	23111	20800	18909	17333	16000	14857	13868	13000	12236	11555	10947	10400	9904	9454	9043
69	27835	24854	22108	20098	18423	17007	15792	14738	13818	13005	12282	11635	11054	10527	10048	9611
70	29288	26015	23414	21286	19511	18011	16725	15609	14634	13773	13008	12323	11707	11149	10642	10179
71	30800	27466	24720	22473	20589	19016	17657	16480	15450	14542	13733	13010	12360	11770	11235	10747
72	32835	29186	26268	23880	21889	20207	18763	17512	16418	15482	14593	13825	13134	12508	11939	11420
73	34788	30904	27814	25288	23177	21398	19868	18542	17384	16362	15452	14639	13907	13244	12641	12092
74	36700	32822	29580	26891	24466	22585	20972	19573	18350	17271	16311	15452	14660	13980	13344	12764
75	38800	34489	31040	28218	25888	23878	22172	20693	19400	18259	17244	16336	15520	14780	14108	13495
76	40900	36355	32720	29746	27286	25170	23372	21813	20450	19246	18178	17221	16360	15580	14871	14225
77	43000	38222	34400	31273	28686	26462	24572	22833	21500	20236	19111	18105	17200	16380	15635	14955
78	45335	40297	36268	32971	30222	27899	25906	24178	22688	21335	20149	19088	18134	17269	16484	15768
79	47688	42371	38194	34688	31779	29335	27239	25422	23834	22432	21185	20070	19067	18158	17332	16579
80	50000	44444	40000	36364	33332	30770	28572	26666	25000	23530	22222	21052	20000	19046	18180	17390
81	52500	46886	42000	38182	35000	32309	30000	27999	26250	24707	23333	22105	21000	19998	19089	18260
82	55000	48988	44000	40000	36665	33847	31429	29333	27500	25883	24444	23157	22000	20951	20000	19129
83	57500	51111	46000	41819	38332	35386	32868	30666	28750	27060	25555	24210	23000	21903	20907	20000
84			48200	44726	40990	37847	35144	32799	30750	28942	27333	25894	24600	23427	22361	21390
85			52400	47837	43886	40308	37429	34832	32750	30824	29111	27578	26200	24950	23816	22781
86			55400	50384	46186	42818	39572	36832	34625	32588	30777	29157	27700	26379	25179	24085
87			58400	53081	48686	44924	41715	38932	36500	34354	32444	30736	29200	27807	26543	25389
88			61400	55819	51188	47232	43668	40325	38119	36111	34111	32315	30700	29238	27906	26694
89			64000	60000	54898	50771	47144	43998	41250	38825	36666	34736	33000	31426	29997	28694
90					51331	56817	52572	48065	44800	43295	40868	38736	36800	35045	33451	31998
91					57184	62002	57573	53732	50378	47413	44777	42420	40300	38378	36633	35041
92					73500	67848	63001	58799	55125	51884	49000	46420	44100	41998	40087	38345
93					80830	74617	69287	64865	60825	57060	53688	51051	48500	46187	44067	42171
94					87500	81540	75716	70865	66250	62355	58888	55788	53000	50472	48177	46084
95						78930	73665	68663	65002	61369	58156	55250	52615	50222	48040	46040
96						82148	76665	71475	67849	63988	60525	57500	54757	52268	50000	47915
97						85360	79685	74688	70296	66388	62893	59750	56900	54313	51953	49790
98						87500	82665	77500	72943	68888	65261	62000	59043	56358	53909	51665
99							85898	80825	75884	71086	67893	64500	61423	58631	56083	53748
100							87500	83750	78826	74444	70524	67000	63804	60903	58257	55831
101							86875	81767	77221	73156	69500	66185	63178	60430	57914	
102							87500	84708	80000	75787	72000	68686	65448	62604	60000	
103								87500	83610	79210	75250	71681	68402	65430	62706	
104									87221	82829	78508	74756	71367	68256	65414	
105									87500	86500	81750	77851	74311	71062	68122	
106									87500	85000	80846	77265	73908	70831		
107										87500	84469	80628	77125	73914		
108											87500	83992	80342	77000		
109												87448	83846	80163		
110													87500	84950	81330	
"B" dim. size 64-63	3 1/4	4 1/4	4 1/4	5 1/4	5 1/4	6	6 1/4	6 1/4	7 1/4	7 1/4	8 1/4	8 1/4	9 1/4	9 1/4	10 1/4	10 1/4
"B" dim. size 84-110	---	---	4 1/4	4 1/4	5	5 1/4	5 1/4	6 1/4	6 1/4	7 1/4	7 1/4	8 1/4	8 1/4	9 1/4	9 1/4	10

table
contin-
uedtable
contin-
ued

*NOTE: Total Travel equals Actual Travel plus 1 inch or 20% (whichever is greater), rounded up to the nearest 1/2 inch as applicable.
Constant supports are readily available for travel and loads not listed in this table. Dimensions and lug locations may vary from those shown on the following pages.

FIG. 9.2
CONTINUED

constant supports

load travel table (continued from opposite page)

load in pounds for total travel in inches

hanger size no.	total travel* in inches																			
	12½	13	13½	14	14½	15	15½	16	16½	17	17½	18	18½	19	19½	20				
64	8152	5915	5896	5492	5303	5126	4981	4806												
65	6432	6184	5955	5742	5544	5359	5187	5025												
66	7082	6790	6538	6304	6087	5884	5684	5517												
67	7690	7394	7120	6866	6629	6408	6201	6008												
68	8320	8000	7703	7428	7172	6933	6709	6500												
69	8843	8503	8186	7895	7623	7369	7131	6909												
70	9368	9005	8671	8361	8073	7804	7552	7317												
71	9888	9507	9155	8828	8523	8239	7973	7725												
72	10507	10103	9728	9380	9057	8756	8473	8209												
73	11128	10697	10301	9932	9590	9270	8971	8692												
74	11744	11292	10873	10484	10123	9786	9470	9175												
75	12416	11938	11496	11084	10703	10346	10012	9700												
76	13068	12584	12118	11684	11282	10906	10554	10225												
77	13760	13230	12740	12284	11861	11466	11096	10750												
78	14507	13949	13432	12951	12505	12088	11698	11334												
79	15254	14666	14123	13618	13149	12710	12300	11917												
80	16000	15384	14814	14284	13792	13332	12902	12500												
81	16800	16153	15555	14998	14482	14000	13547	13125												
82	17600	16922	16295	15712	15171	14665	14192	13750												
83	18400	17692	17036	16427	15861	15332	14837	14375												
84	19200	18422	17621	16969	16364	15798	15269	14775												
85	20000	20153	19406	18712	18068	17465	16902	16375												
86	22160	21307	20517	19783	19102	18465	17869	17313												
87	23360	22461	21628	20855	20136	19465	18837	18250												
88	24560	23614	22739	21928	21171	20465	19805	19188												
89	26400	25384	24443	23569	22757	21998	21288	20625												
90	29440	28307	27258	26283	25377	24531	23740	23000												
91	32240	31000	29850	28782	27791	26864	25998	25188												
92	35280	33922	32685	31498	30411	29397	28449	27563												
93	38800	37306	35824	34639	33446	32330	31287	30313												
94	42400	40768	39257	37853	36549	35330	34190	33125												
95	44200	42498	40924	39480	38100	36830	35642	34531	33482	32498	31570	30691	29863	29078	28332	27625				
96	48000	46230	44590	43067	41652	39330	37083	35838	34645	33622	32656	31941	31080	30262	29488	28750				
97	47800	45960	44257	42673	41204	39829	38545	37344	36209	35145	34141	33191	32295	31446	30640	29875				
98	49600	47890	46223	44680	43255	41329	40000	38750	37572	36468	35427	34441	33511	32631	31794	31000				
99	51600	49613	47775	46066	44479	42996	41809	40313	39067	37939	36855	35830	34862	33946	33076	32250				
100	53600	51536	49627	47851	46203	44662	43221	41875	40602	39409	38284	37219	36214	35262	34358	33500				
101	55600	53459	51479	49637	47927	46329	44834	43438	42117	40880	39712	38607	37565	36578	35640	34750				
102	57600	56382	53330	51422	49661	47996	46447	45000	43632	42360	41141	39996	38916	37894	36922	36000				
103	60200	57882	55738	53744	51892	50182	48544	47031	45602	44262	42998	41801	40673	39604	38588	37625				
104	62800	60382	58145	56065	54134	52328	50640	49063	47571	46174	44855	43607	42429	41315	40255	39250				
105	65400	62882	60552	58368	56375	54495	52737	51094	49541	48085	46712	45412	44188	43025	41921	40875				
106	68000	65382	62960	60707	58616	56681	54834	53125	51510	50000	48569	47218	45943	44736	43598	42500				
107	70960	68228	65700	63350	61169	59127	57220	55438	53752	52173	50683	49273	47942	46683	45485	44350				
108	73920	71074	68441	65992	63719	61594	59607	57750	55994	54350	52797	51328	49942	48630	47383	46200				
109	76980	74000	71255	68706	66340	64127	62059	60125	58297	56565	54989	53439	52000	50630	49331	48100				
110	80000	76920	74070	71420	68960	66660	64510	62500	60600	58820	57140	55550	54060	52630	51260	50000				
"B" dim size 64-83	11½	12	12½	13	13½	14	14½	15	15½	16	16½	17	17½	18	18½	19	19½	20		
"B" dim size 84-110	10½	10¾/16	11¾/16	11½	12¾/16	12½	13	13½/16	13¾/16	14½	14¾/16	15¾/16	16¾/16	17¾/16	18¾/16	19¾/16	20¾/16			

Constant supports are readily available for travel and loads not listed in this table. Dimensions and lug locations may vary from those shown on the following pages.

FIG. 9.2
CONTINUED

9.4 FABRICATION OF SUPPORTS

ASME does not provide much in the way of requirements for the fabrication of support elements. Supports and restraints that are cold-formed to a centerline radius less than twice the support or restraint thickness are required to be annealed or normalized after forming [para. 321.1.4(a)].

Welded attachments to the pressure boundary are required to comply with the Code requirements for preheating, welding, and heat treatment. These are covered in Chapter V of ASME B31.3.

LOAD LIMITS FOR ATTACHED EQUIPMENT

10.1 OVERVIEW OF EQUIPMENT LOAD LIMITS

It is an ASME B31.3 requirement that piping systems be designed to comply with the load limits of equipment to which they are connected. This equipment can be machinery, pressure vessels, tankage, or air fin heat exchangers, as well as a myriad of other possibilities. The Designer needs to be aware of the applicable load limits. Some are described herein as examples; however, this is not a complete list. The remainder of this chapter does not provide specific Code requirements; it provides general information on load limits for equipment.

10.2 PRESSURE VESSELS

There are several approaches to dealing with load limits for pressure vessels.

- Specify the loads the nozzles are required to be able to accept in the vessel specifications and design the piping to comply with those limits.
- Calculate loads on nozzles, and when they appear high by some rule of thumb, send them to the vessel manufacturer to confirm the nozzles are acceptable for the imposed loads, or perform nozzle stress calculations for the imposed loads.
- Perform nozzle stress calculations to confirm the nozzles are acceptable for the loads, again typically when they exceed some rule-of-thumb value.

No rule of thumb works for every vessel, and a variety may well be in use. One of these is to be concerned when the total stress in the piping, including sustained and thermal loads at the vessel nozzle, exceeds S_h . With a little planning ahead, target allowable loads can be placed in the vessel specifications, which simplifies the entire process.

When a detailed stress calculation of the nozzle, with the imposed loads, is required, it is typical to turn to a design-by-analysis approach, per ASME BPVC, Section VIII, Division 2, Appendix 4. This is done whether the vessel is Division 1 or Division 2. Division 1 does not contain any rules for this evaluation. Stresses are calculated by hand calculations or the finite-element method, also typically by hand.

A variety of hand calculation methods are available, based on work by Bijlaard (1954–1959). WRC 107 (Wichman et al, 1979) and WRC 297 (Mershon et al, 1984) contain procedures for analysis of stresses in nozzles due to external loads in cylinders and spheres. These are somewhat tedious to apply; the procedure is available in commercial programs, including some piping flexibility analysis programs. Bednar (1986) contains a more simplified approach, which is quicker to apply and is based on the same original theory. When calculating the stresses due to external loads one needs to keep in

mind that these must be added to the membrane stresses in the vessel wall due to internal pressure. This is commonly forgotten.

Finite-element analysis of nozzles has become more common, because a variety of finite-element programs have features that make it relatively easy to automatically create such models.

Once the stresses are calculated, they are typically compared to the primary, local primary, and secondary stress limits described in Appendix 4 of ASME BPVC, Section VIII, Division 2. These limits are described in Insert 9.2 of Chapter 9.

10.3 OTHER EQUIPMENT LOAD LIMITS

The load limits for other equipment must come from the equipment manufacturer or, if the equipment is constructed in accordance with a standard that provides acceptable load limits, per those load limits. For equipment in accordance with those standards, the manufacturer is essentially required by the standard to design the equipment for certain minimum loads. The following lists some that are more commonly used:

- Centrifugal pumps, API 610
- Centrifugal compressors, API 617
- Air-cooled heat exchangers, API 661
- Steam turbines, NEMA SM 23
- Steam turbines, API 611
- Steam turbines, API 612
- Rotary-type positive-displacement compressors, API 619

An alternative would be for the purchaser to specify minimum design loads in the purchase specification.

10.4 MEANS OF REDUCING LOADS ON EQUIPMENT

When faced with a situation where excessive loads are calculated to be exerted on equipment, typically the equipment has already been purchased, since the piping stress work is often done after equipment procurement. In such a case, there are a number of alternatives. Some of these are listed as follows in general order of preference:

- If the loads are due to weight, add appropriate support.
- Rerun the analysis using the hot modulus of elasticity to calculate the equipment loads.
- If the loads are due to friction, consider using low-friction slide plates.
- Add restraints to direct thermal expansion and resulting loads away from load-sensitive equipment.
- Add flexibility to the piping system.
- Consider using of cold spring or expansion joints.

REQUIREMENTS FOR MATERIALS

11.1 OVERVIEW OF MATERIAL REQUIREMENTS

The requirements for materials are covered in Chapter III of ASME B31.3. Selection of the appropriate material for a given service is the responsibility of the Designer, and is not covered by the Code. As stated in para. 323.5, "Selection of materials to resist deterioration in service is not within the scope of this Code. It is the designer's responsibility to select materials suitable for the fluid service." Practical considerations make it impossible for the Code to provide guidance relevant to the essentially unlimited process applications covered by the Code. Some precautionary considerations are included in Appendix F based on specific experience or interests of Code Committee members, but this is a very limited list of considerations.

ASME B31.3 contains listed materials, which are simply those that are included in the allowable stress tables and those that are permitted in component standards listed in Table 326.1. However, the Designer is not prevented from using other materials. ASME B31.3 only lists materials in more common use. In addition, the Designer is not generally prevented from using a material at temperatures for which allowable stresses are not provided. Rules are provided in Chapter III for qualification of unlisted materials and for use of materials outside of the temperature limits for which allowable stresses are provided.

Note that the Code typically identifies material by the appropriate American Society for Testing and Materials (ASTM) designation (e.g., A106) rather than the ASME designation (e.g., SA106). An "SA" designation identifies the material as being ASME BPVC material in accordance with specifications listed in Section II, Part A, Ferrous Material Specifications. An "SB" designation identifies the material as being ASME BPVC material in accordance with specifications list in Section II, Part B, Nonferrous Material Specifications. The SA and SB materials typically correspond to ASTM specification designations, and can be used interchangeably in piping systems.

Detailed rules covering notch toughness requirements for materials and exemptions from impact testing are provided in ASME B31.3, Chapter III along with some specific limitations for some materials, such as cast iron.

11.2 QUALIFICATION OF UNLISTED MATERIALS

A listed material is one that is listed in one of the allowable stress tables and/or one that is permitted in a component standard listed in Table 326.1 of ASME B31.3. Use of unlisted materials is covered by para. 323.4, along with use of materials at temperatures above the maximum temperature for which allowable stresses are provided in the Code. In these cases, it is the Designer's responsibility to determine the allowable stresses and determine the suitability of the material for the service.

Per para. 323.1.2, unlisted materials may be used if

“they conform to a published specification covering chemistry, physical and mechanical properties, method and process of manufacture, heat treatment, and quality control,”

and otherwise meet any applicable requirements of ASME B31.3, and allowable stresses are determined in accordance with the ASME B31.3 code rules or on a more conservative basis.

The first requirement can be satisfied if the material is in accordance with some published standard, such as an ASTM standard. This would be the primary vehicle for considering materials in accordance with other (non-U.S.) standards, such as Deutsches Institut für Normung e.V. (DIN) standards.

The allowable stress basis is provided in ASME B31.3, para. 302.3. Using this criterion, allowable stresses can be developed using data obtained from a sound scientific program carried out in accordance with recognized technology [para. 323.2.4(b)]. However, an alternative, simpler approach can be used for many materials. If the material is listed in ASME BPVC, Section II, the allowable stress from that document may be used as follows.

The allowable stresses provided for use with ASME BPVC, Section VIII, Division 2 are based on criteria consistent with ASME B31.3 and can be used directly. The allowable stresses provided for use with ASME BPVC, Section VIII, Division 1 are slightly more conservative for some materials and temperatures and can be used because of this. Essentially, the only difference is that the margin on tensile strength is $1/3.5$ (formerly $1/4$) versus $1/3$.

Recently, the yield and tensile strength tables in ASME BPVC, Section II have been greatly expanded, so that these material properties may be used in the future to directly determine the allowable stresses for unlisted materials in accordance with para. 302.3.

In addition to determining the allowable stress, the Code requires consideration of:

- Applicability and reliability of the data, especially for extremes of the temperature range
- Resistance of the material to deleterious effects of the fluid service and of the environment throughout the temperature range

The latter is actually a requirement for listed materials as well.

11.3 USE OF MATERIALS ABOVE THE HIGHEST TEMPERATURE FOR WHICH ALLOWABLE STRESSES ARE PROVIDED

ASME B31.3 permits the use of materials above the highest temperature for which allowable stresses are provided in Appendix A. Furthermore, it permits the use of materials above the highest temperature permitted in listed standards. For example, ASME B16.5 limits the use of Type 304L stainless steel flanges to temperatures at or below 427°C (800°F); however, ASME B31.3 does not prohibit their use at higher temperatures. In fact, Type 304L flanges can be used satisfactorily at temperatures higher than 815°C (1500°F) if properly designed/rated. Note, however, per para. 302.2.1, the extension of pressure-temperature ratings of a component beyond the ratings of the listed standard is at the Owner's responsibility.

In circumstances such as these, it is the Designer's responsibility to determine the serviceability of the material and the allowable stresses. The allowable stresses are determined the same way as for unlisted materials (described in the previous section).

11.4 LOW-TEMPERATURE SERVICE

The tables of Appendix A of ASME B31.3 provide an allowable stress in a column headed “Min. Temp. to 100°F .” This does not mean the material is limited to temperatures above the minimum temperature, also provided in Appendix A, either directly or by reference to Fig. 323.3.2A.

The minimum temperature simply provides a break point in impact test requirements for most materials. If a material is used below the minimum temperature, the same allowable stresses are used as are provided in the column "Min. Temp. to 100." Although materials typically become increasingly stronger as the temperature is further decreased, the Designer is not permitted to take credit for that.

11.5 WHEN IS IMPACT TESTING REQUIRED?

The Code requires impact testing when, due to various considerations, there is a greater risk of brittle fracture. This risk is due to the combination of material condition, constraint (thickness), toughness, and stress.

Table 323.2.2 of the Code, reproduced here as Table 11.1, sets forth the requirements for impact testing for various materials. Column A provides requirements when the design minimum temperature is at or above the minimum temperature listed in Table A-1 (Appendix A) or Fig. 323.2.2A of ASME B31.3. These are temperatures at which the probability of brittle fracture is considered to be lower. Column B provides requirements for when the design minimum temperature is below the minimum temperature in Table A-1 or Fig. 323.2.2A of ASME B31.3.

The minimum temperature is listed in the allowable stress tables for materials other than carbon steels, carbon steels that are limited to Category D Fluid Service, materials required to be impact-tested by the material specifications as well castings and forgings. Category D Fluid Service must be greater than -29°C (-20°F) by definition, and no impact testing is required for materials used in piping in Category D Fluid Service.

Other carbon steel materials were formerly assumed to be acceptable down to either -29°C (-20°F) or -46°C (-50°F) without impact testing. However, it was recognized that these materials in greater wall thicknesses could fail in a brittle manner. As a result, a thickness dependence of minimum temperature was provided in ASME BPVC, Section VIII, Division 1 in Fig. UCS-66. A similar chart was provided in ASME B31.3 in Fig. 323.2.2A.

Figure 323.2.2A of ASME B31.3, provided here as Fig. 11.1, is similar to the curve of ASME BPVC, Section VIII, Division 1. The primary differences are that it provides more detailed treatment of piping materials, and curve B, which covers many common carbon steel piping materials, was shifted so that the minimum temperature was -29°C (-20°F) through 13-mm ($1/2$ -in.) wall thickness.

The design minimum temperature is the minimum temperature that the metal can experience during service, including conditions such as autorefrigeration (see para. 301.3.1). Generally, when the design minimum temperature is below the minimum temperature specified in the Code, impact testing is required. This consists of using impact-test-qualified weld procedures and impact tested materials, as set forth in Table 323.3.1.

When the temperature is at or above the minimum temperature specified in the Code, there are different requirements listed for the base material [column A(a) of Table 11.1] and the weld metal and heat-affected zone [column A(b)]. Again, when impact testing is required for the weld metal, this is satisfied by impact testing done as part of the weld procedure qualification testing. See Note (2) of Table 11.1. When impact testing of the base material is required, impact-tested material must be used.

The low-stress [41 MPa (6 ksi)] exemption from impact testing under certain conditions that was previously provided was replaced in the 2000 Addenda with a curve similar to AM-218.3 in ASME BPVC, Section VIII, Division 2. This provides a more comprehensive treatment. Note that the Division 2 curve was used rather than the Division 1 curve, because the Division 2 curve is consistent with the ASME B31.3 allowable stress basis (one-third of tensile strength). This curve is provided as Figure 323.2.2B of ASME B31.3, reproduced as Fig. 11.2 here.

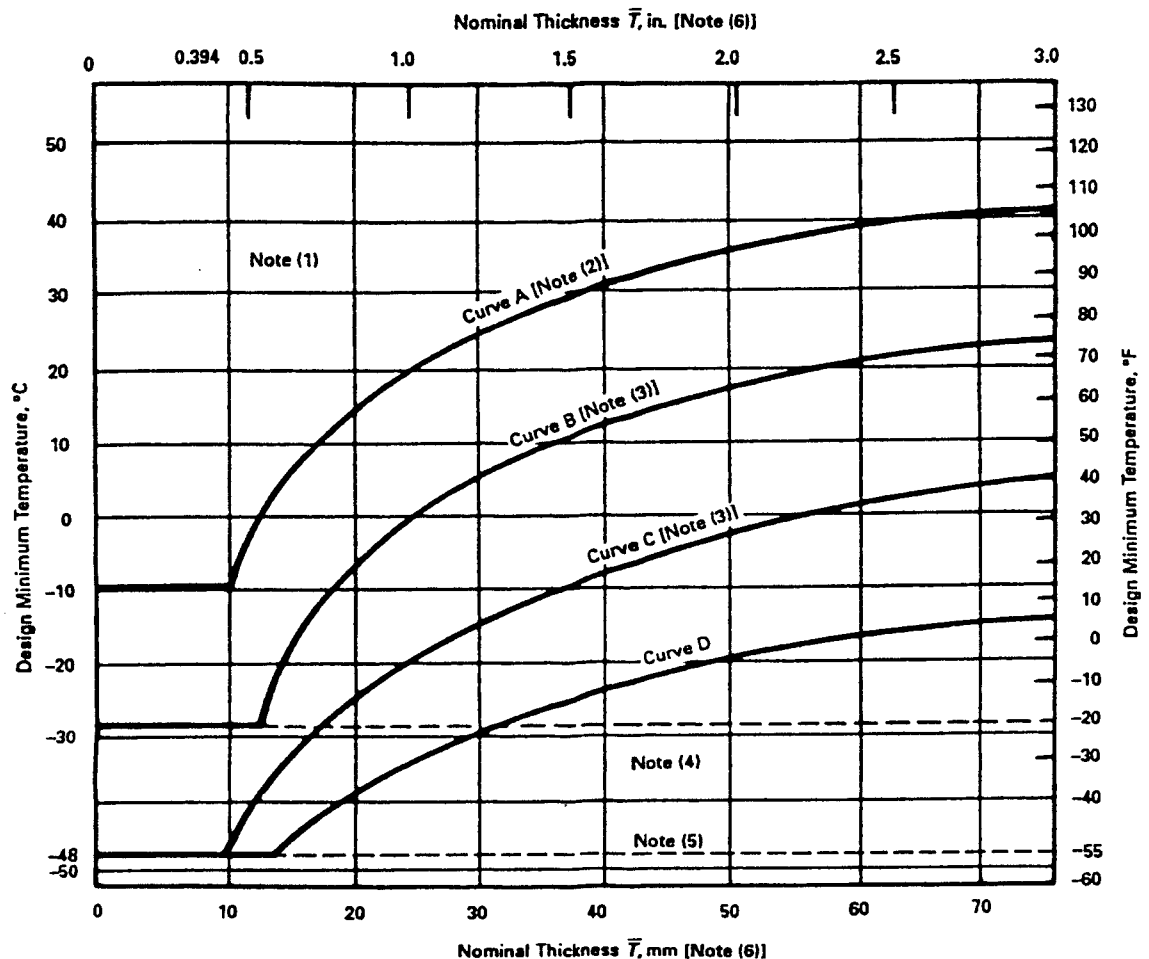
Figure 11.2 provides a reduction in the minimum temperature, without impact testing, for carbon steel material when the stress in the piping is less than the maximum allowable stress. For minimum design metal temperatures of -48°C (-55°F) and above, the temperature reduction provided in

TABLE 11.1
REQUIREMENTS FOR LOW-TEMPERATURE TOUGHNESS TESTS FOR METALS

	Type of Material	Column A Design Minimum Temperature at or Above Min. Temp. in Table A-1 or Fig. 323.2.2A		Column B Design Minimum Temperature Below Min. Temp. in Table A-1 or Fig. 323.2.2A
Listed materials	1 Gray cast iron	A-1 No additional requirements		B-1 No additional requirements
	2 Malleable and ductile cast iron; carbon steel per Note (1)	A-2 No additional requirements		B-2 Materials designated in Box 2 shall not be used.
		(a) Base Metal	(b) Weld Metal and Heat Affected Zone (HAZ) [Note (2)]	
	3 Other carbon steels; low and intermediate alloy steels; high alloy ferritic steels; duplex stainless steels	A-3 (a) No additional requirements	A-3 (b) Weld metal deposits shall be impact tested per para. 323.3 if design min. temp. < -29°C (-20°F), except as provided in Notes (3) and (5), and except as follows: for materials listed for Curves C and D of Fig. 323.2.2A, where corresponding welding consumables are qualified by impact testing at the design minimum temperature or lower in accordance with the applicable AWS specification, additional testing is not required.	B-3 Except as provided in Notes (3) and (5), heat treat base metal per applicable ASTM specification listed in para. 323.3.2; then impact test base metal, weld deposits, and HAZ per para. 323.3 [see Note (2)]. When materials are used at design min. temp. below the assigned curve as permitted by Notes (2) and (3) of Fig. 323.2.2A, weld deposits and HAZ shall be impact tested [see Note (2)].
	4 Austenitic stainless steels	A-4 (a) If: (1) carbon content by analysis > 0.1%; or (2) material is not in solution heat treated condition; then, impact test per para. 323.3 for design min. temp. < -29°C (-20°F) except as provided in Notes (3) and (6)	A-4 (b) Weld metal deposits shall be impact tested per para. 323.3 if design min. temp. < -29°C (-20°F) except as provided in para. 323.2.2 and in Notes (3) and (6)	B-4 Base metal and weld metal deposits shall be impact tested per para. 323.3. See Notes (2), (3), and (6).
	5 Austenitic ductile iron, ASTM A 571	A-5 (a) No additional requirements	A-5 (b) Welding is not permitted	B-5 Base metal shall be impact tested per para. 323.3. Do not use < -196°C (-320°F). Welding is not permitted.
Materials	6 Aluminum, copper, nickel, and their alloys; unalloyed titanium	A-6 (a) No additional requirements	A-6 (b) No additional requirements unless filler metal composition is outside the range for base metal composition; then test per column B-6	B-6 Designer shall be assured by suitable tests [see Note (4)] that base metal, weld deposits, and HAZ are suitable at the design min. temp.
Unlisted	7 An unlisted material shall conform to a published specification. Where composition, heat treatment, and product form are comparable to those of a listed material, requirements for the corresponding listed material shall be met. Other unlisted materials shall be qualified as required in the applicable section of column B.			

NOTES:

- (1) Carbon steels conforming to the following are subject to the limitations in Box B-2; plates per ASTM A 36, A 283, and A 570; pipe per ASTM A 134 when made from these plates; and pipe per ASTM A 53 Type F and API 5L Gr. A25 butt weld.
- (2) Impact tests that meet the requirements of Table 323.3.1, which are performed as part of the weld procedure qualification, will satisfy all requirements of para. 323.2.2, and need not be repeated for production welds.
- (3) Impact testing is not required if the design minimum temperature is below -29°C (-20°F) but at or above -104°C (-155°F) and the Stress Ratio defined in Fig. 323.2.2B does not exceed 0.3 times *S*.
- (4) Tests may include tensile elongation, sharp-notch tensile strength (to be compared with unnotched tensile strength), and/or other tests, conducted at or below design minimum temperature. See also para. 323.3.4.
- (5) Impact tests are not required when the maximum obtainable Charpy specimen has a width along the notch of less than 2.5 mm (0.098 in.). Under these conditions, the design minimum temperature shall not be less than the lower of -48°C (-55°F) or the minimum temperature for the material in Table A-1.
- (6) Impact tests are not required when the maximum obtainable Charpy specimen has a width along the notch of less than 2.5 mm (0.098 in.).



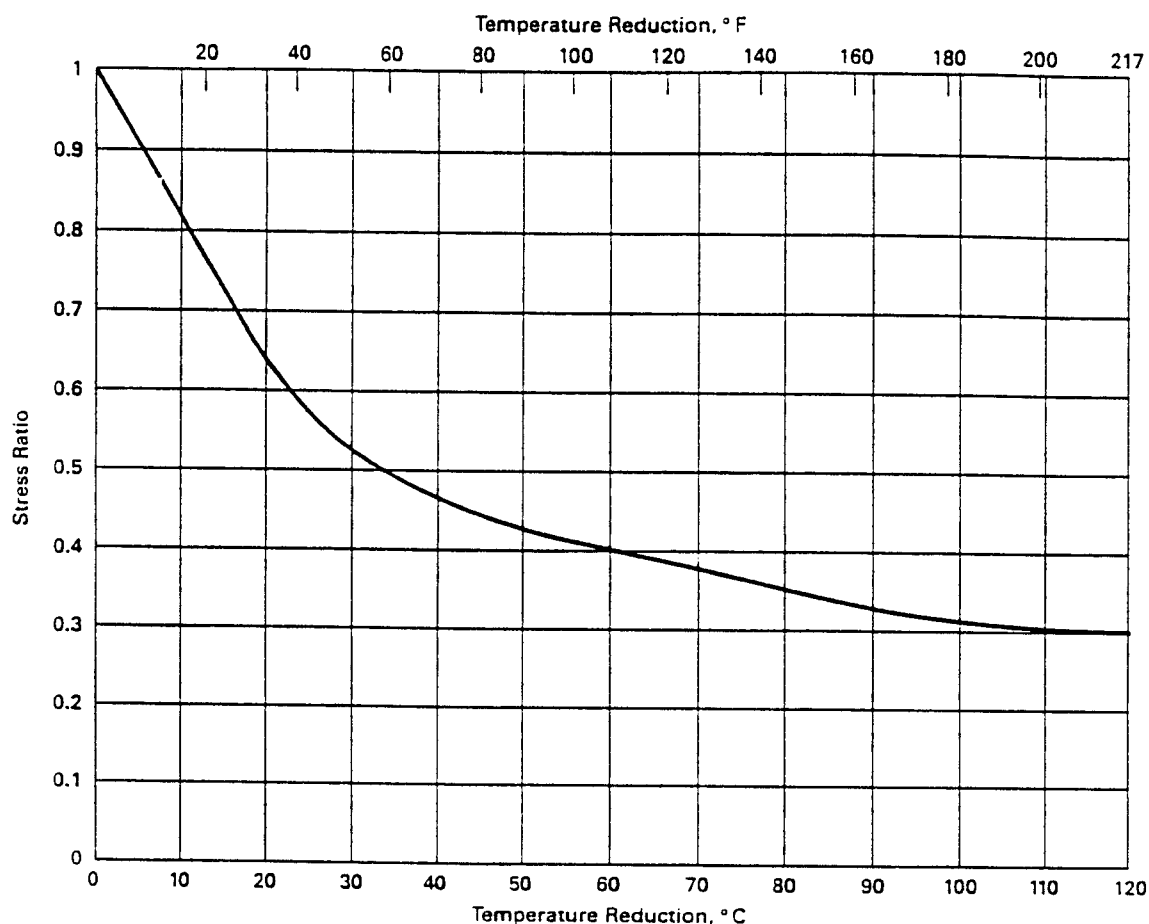
NOTES:

- (1) Any carbon steel material may be used to a minimum temperature of -29°C (-20°F) for Category D Fluid Service.
- (2) X Grades of API 5L, and ASTM A 381 materials, may be used in accordance with Curve B if normalized or quenched and tempered.
- (3) The following materials may be used in accordance with Curve D if normalized:
 - (a) ASTM A 516 Plate, all grades;
 - (b) ASTM A 671 Pipe, Grades CE55, CE60, and all grades made with A 516 plate;
 - (c) ASTM A 672 Pipe, Grades E55, E60, and all grades made with A 516 plate.
- (4) A welding procedure for the manufacture of pipe or components shall include impact testing of welds and HAZ for any design minimum temperature below -29°C (-20°F), except as provided in Table 323.2.2, A-3(b).
- (5) Impact testing in accordance with para. 323.3 is required for any design minimum temperature below -48°C (-55°F), except as permitted by Note (3) in Table 323.2.2.
- (6) For blind flanges and blanks, $\bar{T}$ shall be $1/4$ of the flange thickness.

FIG. 11.1

MINIMUM TEMPERATURE WITHOUT IMPACT TESTING FOR CARBON STEEL MATERIALS (ASME B31.3, FIG. 323.2.2A). SEE ASME B31.3, TABLE A-1 FOR DESIGNATED CURVE FOR A LISTED MATERIAL

Fig. 11.2 may be applied to the minimum permissible temperature. For minimum design metal temperatures below -48°C (-55°F) and at or above -101°C (-155°F), carbon steel is permitted if the stress ratio defined in General Note (a) of Fig. 11.2 is 0.3 or less. Note that this last exemption also applies from -29°C (-20°F) down to -101°C (-155°F), per note (3) of Table 11.1, for austenitic stainless steel, intermediate-alloy steels, high-alloy ferritic steels, and duplex stainless steels.



GENERAL NOTES:

(a) The Stress Ratio is defined as the maximum of the following:

- (1) nominal pressure stress (based on minimum pipe wall thickness less allowances) divided by S at the design minimum temperature;
- (2) for piping components with pressure ratings, the pressure for the condition under consideration divided by the pressure rating at the design minimum temperature;
- (3) combined longitudinal stress due to pressure, dead weight, and displacement strain (stress intensification factors are not included in this calculation) divided by S at the design minimum temperature. In calculating longitudinal stress, the forces and moments in the piping system shall be calculated using nominal dimensions and the stresses shall be calculated using section properties based on the nominal dimensions less corrosion, erosion, and mechanical allowances.

(b) Loadings coincident with the metal temperature under consideration shall be used in determining the Stress Ratio as defined above.

FIG. 11.2
REDUCTION IN MINIMUM DESIGN METAL TEMPERATURE WITHOUT IMPACT TESTING
(ASME B31.3, FIG. 323.2.2B)

The stress ratio must consider both hoop-type and longitudinal stresses. The stress ratio is the maximum of the following:

- (a) Nominal pressure stress (based on minimum pipe wall thickness less allowances) divided by S at the design minimum temperature

- (b) For piping components with pressure ratings, the pressure for the condition under consideration divided by the pressure rating at the design minimum temperature
- (c) Combined longitudinal stress due to pressure, dead weight, and displacement strain (stress intensification factors are not included in this calculation) divided by S at the design minimum temperature; in calculating longitudinal stress, the forces and moments shall be calculated using nominal dimensions and the stresses shall be calculated using section properties based on the nominal dimensions less corrosion, erosion, and mechanical allowances

The calculated stress, which is compared to the allowable stress at the design minimum temperature, is calculated for the loads that are present at the condition being considered. For example, if a system has a temperature range from -80°C to 500°C , it is the thermal stresses at -80°C that would be considered in the stress ratio, not the thermal expansion stress range nor the thermal stress at 500°C . An additional limitation is under consideration that would essentially limit systems subject to these rules to elastic behavior, so that yielding at some higher temperature condition does not self-spring the system, causing unanticipated stresses in the cold condition.

Stress intensification factors are not required to be included when calculating the longitudinal stress in the piping system. The stress is considered to be sufficiently low that even if a crack were to be initiated at some local high-stress region, it would be arrested (stop advancing) when it entered the general low-membrane-stress regions of the piping system.

Two additional requirements are provided for systems to be eligible for the low-stress exemption. They are now only applied down to -48°C (-55°F), but they are equally or more important when the temperature is lower, down to as low as -101°C (-155°F). These requirements are as follows:

- (a) The piping shall be subjected to a hydrostatic test at no less than 1.5 times the design pressure.
- (b) Except for piping with nominal wall thickness of 13 mm ($1/2$ in.) or less, the piping system shall be safeguarded (see ASME B31.1, Appendix G) from external loads, such as maintenance loads, impact loads, and thermal shock.

The first requirement recognizes the benefit of warm pre-stress that a hydrotest provides. This tends to blunt cracks that may be present in the system and leaves crack tips under compression. The second addresses a concern regarding unanticipated loads, such as maintenance loads. There has been, for example, a failure incident involving the use of a wrench to try to open a small-bore drain valve that had been subject to autorefrigeration (breaking the valve off), leading to a fatality.

Another exemption deals with the situation when the material is sufficiently thin. Thinner sections are less susceptible to brittle fracture, because of the reduced constraint. When the maximum obtainable Charpy specimen has a width along the notch of less than 2.5 mm (0.098 in.), impact testing is not required. However, the exemption is limited to design minimum temperatures above -48°C (-55°F) or the minimum temperature listed in ASME B31.3, Table A-1, whichever is less for carbon steel and other alloys listed in the third row of Table 11.1.

The temperatures of -46°C (-50°F) and -101°C (-150°F) were generally changed to -48°C (-55°F) and -104°C (-155°F) with the 2000 Addenda, consistent with the changes to ASME BPVC, Section VIII. There is essentially no difference in the material properties between the prior and revised temperatures, but the change permits inclusion of propylene for the higher threshold and ethylene for the lower threshold.

11.6 IMPACT TEST REQUIREMENTS

Impact testing is required to be performed in accordance with ASTM A370. This is Charpy V-notch impact testing. Each set of impact test specimens consists of three standard bars with the standard

10-mm (0.394-in.) square-cross-section Charpy V-notch configuration. In a standard test, a weight on a pendulum swings through the sample, fracturing it. The initial stored potential energy of the pendulum is consumed partly in fracturing the specimen, and the remaining energy can be measured by how high the pendulum swings after fracturing the specimen (see Fig. 11.3). The initial stored energy minus the energy remaining after breaking the specimen is the energy consumed in fracture, which is a measure of the notch toughness of the material. The specimen must be cooled so that it is at or below the design minimum temperature during the impact test.

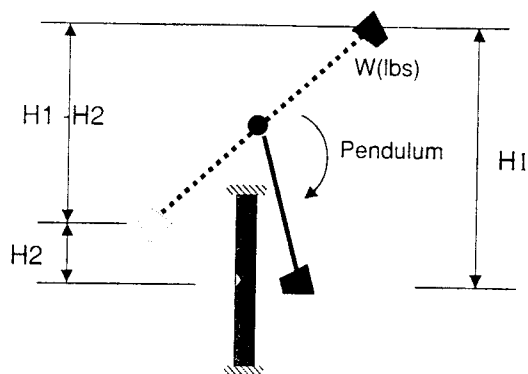
Another measure of notch toughness used in ASME B31.3 for some materials is the lateral expansion of the specimen (after the fracture) opposite the notch. The greater the lateral expansion, the greater is the ductility of the fracture. Figure 11.3 shows a Charpy impact test specimen before and after the test.

Reduced-size specimens may be used, if required, because the material shape or thickness does not permit machining a full-size specimen. However, smaller specimens exhibit improved toughness properties. To compensate for this, the test temperature is reduced for subsize specimens in accordance with Table 323.3.4. For example, if the Charpy impact specimen width along the notch is 2.5 mm (0.098 in.), which is one-fourth of the full size, the test must be performed at a temperature that is 27.8°C (50°F) below the design minimum temperature.

The acceptance criteria for the tests are provided in Table 323.3.5. For carbon and low-alloy steels with specified minimum tensile strengths below 656 MPa (95 ksi), the acceptance criteria are based on energy absorption. A minimum acceptable average for the three specimens and a minimum value for any of the three specimens are provided. For higher strength carbon and low-alloy steels and other materials, the criteria are based on lateral expansion opposite the notch. The minimum acceptable lateral expansion value is 0.38 mm (0.015 in.).

Under limited circumstances, described in para. 323.3.5, retesting is permitted if the initial set of Charpy specimens fails to meet the acceptance criteria.

Impact Testing and Brittle Fracture



Charpy Impact Test

$$C_v = W(H1 - H2) = \text{Energy Absorbed (ft-lbs)}$$

- The Code sets minimum impact energy (C_v) requirements for materials and welds at the Minimum Design Metal Temperature (MDMT).

FIG. 11.3
SCHEMATIC OF CHARPY IMPACT TEST AND TEST SPECIMENS
BEFORE AND AFTER FRACTURE

11.7 FLUID SERVICE LIMITATIONS

Limitations on specific materials with respect to their use for pressure-containing parts are provided in para. 323.4. These include the following for iron materials:

- Temperature limitations with respect to use of ductile iron
- Prohibition of the use of welding for fabrication, repair, or assembly of ductile iron components
- Prohibition of the use of cast, malleable, or high-silicon iron in severe cyclic service
- Pressure-temperature limitations with respect to use of cast or malleable iron in flammable fluid service
- Minimum temperature limit of malleable iron
- Prohibition of the use of high-silicon iron in flammable fluid service.

In all cases, the Code requires that cast iron, malleable iron, and high-silicon iron be safeguarded against excessive heat and thermal shock and mechanical shock and abuse. Lead, tin, and their alloys are prohibited from use in flammable fluid services, because of their low melting points.

Determination of the allowable stresses for cast aluminum alloys that are subject to the heat of welding or thermal cutting is the Designer's responsibility. These are hardened materials, and the welding process results in a reduction of the material strength. Generally, annealed aluminum material properties must be used under these conditions.

11.8 HOW TO USE THE ALLOWABLE STRESS TABLES IN APPENDIX A OF THE CODE

The allowable stresses for use with the metallic materials in accordance with the base Code are listed in Appendix A, Table A-1 for metals other than bolting and Table A-2 for bolting. An illustration of typical pages of allowable stress values is given in Table 11.2.

Materials in Table A-1 are first grouped by general alloy content. They are separated into iron, carbon steel, low and intermediate alloys, stainless steel, copper and copper alloys, nickel and nickel alloys, titanium and titanium alloys, zirconium and zirconium alloys, and aluminum alloys. Within each alloy grouping they are separated into pipes and tubes, plates and sheets, forgings and fittings, bar, and castings, as applicable.

In addition to the designation of the material by alloy content and specification number, additional information is provided. This includes the P-Number or S-Number as well as the specified minimum tensile and yield strengths. All the numbers in this column are P-Numbers unless they are preceded by an S. The specified minimum strengths are from the material specification.

The P-Numbers group alloys for weld procedure qualification purposes, based on composition, weldability, and mechanical properties. These are assigned by ASME BPVC, Section IX. For materials that are not ASME BPVC materials (i.e., not listed in Section II), S-Numbers are assigned. The S-Numbers correspond to the P-Numbers in other respects. See para. 328.2.1(f).

Notes are typically provided. Prior to using a material, the notes should be reviewed. For example, Note (28) for Type 304 stainless steel indicates that the allowable stresses that are listed for temperatures above 1000°F are only valid if the material has a carbon content of 0.04% or higher. There have been instances of systems designed to these allowable stresses without the material procured to the minimum carbon content requirements.

Notes (4) through (7) provide specific guidance on features of the allowable stress tables, as follows.

A single vertical bar in Table A-1 (refer to Table 11.2) indicates that use of the material at the designated temperature is affected by a referenced note. It is a warning. If the bar is to the right of the table entry, it affects the use of the material at a temperature greater than the temperature of that

DOES NOT INCLUDE QUALITY FACTORS

TABLE 11.2

BASIC ALLOWABLE STRESSES IN TENSION FOR METALS¹

Numbers in Parentheses Refer to Notes for Appendix A Tables; Specifications Are ASTM Unless Otherwise Indicated

Material	Spec. No.	P-No. or S-No.	Grade	Notes	Min. Temp., °F (6)	Specified Min. Strength, ksi		Min. Temp. to 100	200	300
		(5)					Tensile			
Carbon Steel (Cont'd)										
Pipes and Tubes (2) (Cont'd)										
...	A 53	1	B	(57)(59)	B	60	35	20.0	20.0	20.0
...	A 106	1	B	(57)						
...	A 333	1	6	(57)						
...	A 334			(57)						
...	A 369			(57)						
...	A 381	S-1	Y35	...	A					
...	API 5L	S-1	B	(57)(59)(77)	B					
...	A 139	S-1	C	(8b)	A	60	42	20.0	20.0	20.0
...	A 139	S-1	D	(8b)	A	60	46			
...	API 5L	S-1	X42	(55)(77)	A	60	42			
...	A 381	S-1	Y42	...	A	60	42			
...	A 381	S-1	Y48	...	A	62	48	20.6	19.7	18.7
...	API 5L	S-1	X46	(55)(77)	A	63	46	21.0	21.0	21.0
...	A 381	S-1	Y46	...	A	63	46	21.0	21.0	21.0
...	A 381	S-1	Y50	...	A	64	50	21.3	20.3	19.3
A 516 Gr. 65	A 671	1	CC65	(57)(67)	B	65	35	21.7	21.3	20.7
A 515 Gr. 65	A 671	1	CB65	(57)(67)	A	65	35	21.7	21.3	20.7
A 515 Gr. 65	A 672	1	B65							
A 516 Gr. 65	A 672	1	C65							
...	A 139	S-1	E	(8b)	A	66	52	22.0	22.0	22.0
...	API 5L	S-1	X52	(55)(77)	A	66	52	22.0	22.0	22.0
...	A 381	S-1	Y52	...	A	66	52	22.0	22.0	22.0
A 516 Gr. 70	A 671	1	CC70	(57)(67)	B	70	38	23.3	23.1	22.5
A 515 Gr. 70	A 671	1	CB70	(57)(67)	A	70	38	23.3	23.1	22.5
A 515 Gr. 70	A 672	1	B70							
A 516 Gr. 70	A 672	1	C70							
...	A 106	1	C	(57)	B	70	40	23.3	23.3	23.3
A 537 Cl. 1 (≤ 2½ in. thick)	A 671	1	CD70	(67)	D	70	50	23.3	23.3	22.9
A 537 Cl. 1 (≤ 2½ in. thick)	A 672	1	D70							
A 537 Cl. 1 (≤ 2½ in. thick)	A 691	1	CMSH70							
...	API 5L	S-1	X56	(51)(55)(71)(77)	A	71	56	23.7	23.7	23.7
...	A 381	S-1	Y56	(51)(55)(71)	A	71	56	23.7	23.7	23.7

READ THESE!!

TIES TO FIG 323.2.2 - DESIGN MIN. TEMP
NORMALLY OK W/O IMPACT TESTING

(Continued)

TABLE 11.2
(Continued)

Basic Allowable Stress <i>S</i> , ksi (1), at Metal Temperature, °F (7)													Grade	Spec. No.
400	500	600	650	700	750	800	850	900	950	1000	1050	1100	Carbon Steel (Cont'd) Pipes and Tubes (2) (Cont'd)	
													B B 6 6 FPB Y35 B	A 53 A 106 A 333 A 334 A 369 A 381 API 5L
20.0	18.9	17.3	17.0	16.5	13.0	10.8	8.7	6.5	4.5	2.5	1.6	1.0		
...	...	...	...	...	...	...	...	...	...	...	...	...		
20.0	...	...	...	...	...	...	...	...	...	...	...	...		
20.0	...	...	...	...	...	...	...	...	...	...	...	...		
17.8	16.9	16.0	15.5	...	...	...	...	...	...	...	...	...	Y48	A 381
21.0	...	...	...	...	...	...	...	...	...	...	...	...	X46	API 5L
21.0	...	...	...	...	...	...	...	...	...	...	...	...	Y46	A 381
18.4	17.4	16.5	16.0	...	...	...	...	...	...	...	...	...	Y50	A 381
20.0	18.9	17.3	17.0	16.8	13.9	11.4	9.0	6.5	4.5	2.5	...	...	CC65	A 671
20.0	18.9	17.3	17.0	16.8	13.9	11.4	9.0	6.5	4.5	2.5	1.6	1.0	CB65	A 671
...	...	...	...	...	...	...	...	...	...	...	...	...	B65	A 672
22.0	...	...	...	...	...	...	...	...	...	...	...	...	C65	A 672
22.0	...	...	...	...	...	...	...	...	...	...	...	...	E	A 139
...	...	...	...	...	...	...	...	...	...	...	...	...	X52	API 5L
21.7	20.5	18.7	18.4	18.3	14.8	12.0	9.3	6.5	4.5	2.5	...	...	Y52	A 381
21.7	20.5	18.7	18.4	18.3	14.8	12.0	9.3	6.5	4.5	2.5	1.6	1.0	CC70	A 671
22.9	21.6	19.7	19.4	19.2	14.8	12.0	...	...	...	...	...	...	CB70	A 671
...	...	...	...	...	...	...	...	...	...	...	...	...	B70	A 672
22.9	22.9	22.6	22.0	21.4	...	...	...	...	...	...	...	...	C70	A 672
...	...	...	...	...	...	...	...	...	...	...	...	...	C	A 106
...	...	...	...	...	...	...	...	...	...	...	...	...	CD70	A 671
23.7	...	...	...	...	...	...	...	...	...	...	...	...	D70	A 672
23.7	...	...	...	...	...	...	...	...	...	...	...	...	CMSH70	A 691
...	...	...	...	...	...	...	...	...	...	...	...	...	X56	API 5L
23.7	...	...	...	...	...	...	...	...	...	...	...	...	Y56	A 381

(Continued)

TABLE 11.2
(Continued)

Material	Spec. No.	P-No. or S-No. (5)	Grade	Notes	Min. Temp., °F (6)	Specified Min. Strength, ksi		Min. Temp. to 100	200	300	400	500	600
						Tensile	Yield						
Stainless Steel (3) (4) (Cont'd)													
Pipes and Tubes (2) (Cont'd)													
18Cr-8Ni pipe	A 430	8	FP304	(26)(31)(36)	-425	70	30	20.0	20.0	20.0	18.7	17.5	16.4
18Cr-8Ni pipe	A 430	8	FP304H	(26)(31)(36)	-325								
18Cr-8Ni tube	A 269	8	TP304	(14)(26)(28)(31)(36)	-425								
18Cr-8Ni pipe	A 312	8	TP304	(26)(28)	-425								
Type 304 A 240	A 358	8	304	(26)(28)(31)(36)	-425								
18Cr-8Ni pipe	A 376	8	TP304	(20)(26)(28)(31)(36)	-425	75	30	20.0	20.0	20.0	18.7	17.5	16.4
18Cr-8Ni pipe	A 376	8	TP304H	(26)(31)(36)	-325								
18Cr-8Ni pipe	A 409	8	TP304	(26)(28)(31)(36)	-425								
18Cr-8Ni pipe	A 312	8	TP304H	(26)	-325								
18Cr-8Ni	A 452	8	TP304H	(26)	-325								
18Cr-10Ni-Mo	A 451	8	CPF8M	(26)(28)	-425	70	30	20.0	20.0	20.0	19.4	18.1	17.1
18Cr-10Ni-Cb	A 452	8	TP347H	...	-325	75	30	20.0	20.0	20.0	20.0	19.9	19.3
20Cr-Cu tube	A 268	10	TP443	(35)	-20	70	40	23.3	23.3	21.4	20.4	19.4	18.4
27Cr tube	A 268	10I	TP446										
25-10Ni-N	A 451	8	CPE20N	(35)(39)	-325	80	40	26.7	26.2	24.9	23.3	22.0	21.4
23Cr-4Ni-N	A 789	10H	S32304	(25)	-60	87	58	29.0	27.9	26.3	25.3	24.9	24.5
23Cr-4Ni-N	A 790												
12%Cr	A 426	6	CPCA-15	(10)(35)	-20	90	65	30.0	...	...	...	...	...
22Cr-5Ni-3Mo	A 789	10H	S31803	(25)	-60	90	65	30.0	30.0	28.9	27.9	27.2	26.9
22Cr-5Ni-3Mo	A 790												
26Cr-4Ni-Mo	A 789	10H	S32900	(25)	-20	90	70	30.0	...	...	...	...	...
26Cr-4Ni-Mo	A 790												
25Cr-8Ni-3Mo-W-Cu-N	A 789	S-10H	S32760	(25)	-60	109	80	36.3	34.4	34.0	34.0	34.0	...
25Cr-8Ni-3Mo-W-Cu-N	A 790												
25Cr-7Ni-4Mo-N	A 789	10H	S32750	(25)	-20	116	80	38.7	35.0	33.1	31.9	31.4	31.2
25Cr-7Ni-4Mo-N	A 790												
Plates and Sheets													
18Cr-10Ni	A 240	8	305	(26)(36)(39)	-325	70	25	16.7	...	...	...	...	...
12Cr-Al	A 240	7	405	(35)	-20	80	25	16.7	15.3	14.8	14.5	14.3	14.0
18Cr-8Ni	A 240	8	304L	(36)	-425	70	25	16.7	16.7	16.7	15.6	14.8	14.0
16Cr-12Ni-2Mo	A 240	8	316L	(36)	-425	70	25	16.7	16.7	16.7	15.5	14.4	13.5
18Cr-Ti-Al	A 240	...	X8M	(35)	-20	65	30	20.0	...	...	...	...	...
18Cr-8Ni	A 167	S-8	3028	(26)(28)(31)(36)(39)	-325	75	30	20.0	20.0	20.0	18.7	17.4	16.4
18Cr-Ni	A 240	8	302	(26)(36)	-325	75	30	20.0	20.0	20.0	18.7	17.4	16.4

(Continued)

TABLE 11.2
(Continued)

Basic Allowable Stress <i>S</i> , ksi (1), at Metal Temperature, °F (7)																		Grade	Spec. No.
650	700	750	800	850	900	950	1000	1050	1100	1150	1200	1250	1300	1350	1400	1450	1500		
BOLD: STRESS = 90% <i>S_y</i>																		Stainless Steel (3) (4) (Cont'd) Pipes and Tubes (2) (Cont'd)	
ITALIC: STRESS EXCEEDS 2/3 <i>S_y</i>																			
16.2	16.0	15.6	15.2	14.9	14.6	14.4	13.8	12.2	9.7	7.7	6.0	4.7	3.7	2.9	2.3	1.8	1.4	FP304 FP304H TP304 TP304 304 TP304 TP304H TP304 TP304H TP304H TP304H CPF8M TP347H	A 430 A 430 A 269 A 312 A 358 A 376 A 376 A 409 A 312 A 312 A 452 A 451 A 452
16.2	16.0	15.6	15.2	14.9	14.6	14.4	13.8	12.2	9.7	7.7	6.0	4.7	3.7	2.9	2.3	1.8	1.4	TP443	A 268
16.2	15.9	15.5	15.1	14.9	14.6	14.3	13.8	12.2	9.8	7.6	6.0	4.7	3.6	2.8	2.2	1.8	1.4	TP446	A 268
16.7	16.2	15.8	15.5	14.7	14.4	14.0	13.4	11.4	9.3	8.0	6.8	5.3	4.0	3.0	2.3	1.9	1.4	CPE20N	A 451
18.9	18.6	18.5	18.2	18.1	18.1	18.1	18.0	17.7	14.2	10.5	7.9	5.9	4.3	3.2	2.5	1.8	1.4		
18.0	17.5	16.9	16.2	15.1	13.0	6.9	4.5	...	...	...	...	...	...	...	...	...	...	S32304	A 789 A 790
21.3	21.2	21.1	21.0	20.8	20.5	...	...	...	...	...	...	...	...	...	...	...	...	CPCA-15	A 426
...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	S31803	A 789 A 790
...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	S32900	A 789 A 790
...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	S32760	A 789 A 790
...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	S32750	A 789 A 790
Plates and Sheets																			
13.8	13.5	11.6	11.1	10.4	9.6	8.4	4.0	...	...	...	...	...	...	...	...	...	...	305	A 240
13.7	13.5	13.3	13.0	12.8	11.9	9.9	7.8	6.3	5.1	4.0	3.2	2.6	2.1	1.7	1.1	1.0	0.9	405	A 240
13.2	12.9	12.6	12.4	12.1	11.8	11.5	11.2	10.8	10.2	8.8	6.4	4.7	3.5	2.5	1.8	1.3	1.0	304L	A 240
...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	316L	A 240
16.1	15.9	15.6	15.2	14.9	14.3	13.7	...	...	...	...	...	...	...	...	...	...	...	X8M	A 240
16.1	15.9	15.6	15.2	14.9	14.3	13.7	...	...	...	...	...	...	...	...	...	...	...	302B	A 167
...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	302	A 240

NOTE:

Specifications are ASTM unless otherwise indicated. For Notes, see ASME B31.1, Notes for Appendix A Tables.

entry. If the bar is to the left of the table entry, it affects the use of the material at a temperature below that of the entry.

A double bar is a prohibition. It prohibits use of the material above or below some temperature. When the double bar is adjacent to a stress value in the table, it means that the use of the material above the corresponding temperature, or above some lower temperature, is prohibited. The placement to the left or right of the table entry has the same meaning as with a single bar. A note indicates the specific temperature limitation. When the double bar is to the left of the "Min. Temp." column, it indicates that use of the material below the minimum temperature is prohibited.

Double bars follow requirements that are listed elsewhere in the Code; they do not add prohibitions that are not listed elsewhere. For example, double bars are used for materials that are only permitted for use in Category D Fluid Service. This is because the definition of Category D Fluid Service requires that the design temperature be between -29°C (-20°F) and 186°C (366°F), inclusive.

For materials that are permitted to exceed two-thirds of the minimum yield strength at temperature, the tables indicate where the values exceed this. Allowable stress values printed in italics exceed two-thirds of the yield strength, but are less than 90% of the expected minimum yield strength at temperature. Values printed in bold are at 90% of the expected minimum yield strength at temperature.

The Designer is permitted to interpolate linearly between temperatures for which allowable stresses are listed. As previously discussed, use of materials above the maximum temperature for which allowable stresses are provided is permitted as long as it is not specifically prohibited (e.g., by a double bar).

Metric versions of the allowable stress tables are being prepared; some features of the allowable stress tables may change when the metric versions are published.

FABRICATION, ASSEMBLY, AND ERECTION

12.1 OVERVIEW OF CHAPTER V OF ASME B31.3

Chapter V of ASME B31.3 covers the base Code rules for fabrication, assembly, and erection. It includes requirements for welding, details for specific types of welded joints, and conditions for preheating, heat treatment, bending and forming, brazing and soldering, assembly, and erection.

12.2 GENERAL WELDING REQUIREMENTS

A variety of welding processes are used with piping. These include shielded metal arc weld (SMAW), gas tungsten arc weld (GTAW or TIG), gas metal arc weld (GMAW or MIG), submerged arc weld (SAW), and flux cored arc weld (FCAW) processes. These are described below in Insert 12.1.

Welding involves a Welding Procedure Specification (WPS), which is qualified by a Procedure Qualification Test, a document included in a Procedure Qualification Record (PQR). Welders are required to pass a Performance Qualification Test in order to be qualified to perform Code welding. This is documented in a Welder Qualification Record (WQR).

Insert 12.1 Arc Welding Processes

Shielded Metal Arc Welding (SMAW)

Principles of Operation

Shielded metal arc welding (SMAW) is the most widely used of all arc welding processes (Fig. 12.1). It is a manual process, which employs the heat of an arc to melt the base metal and the tip of a consumable covered electrode. The bare end of the electrode is clamped in an electrode holder, which is connected to one terminal of a welding power source by a welding lead (cable). The base metal (work) is connected by a cable to the other terminal of the power source. The arc is initiated by touching the electrode tip against the work and then withdrawing it slightly. The heat of the arc melts the base metal in the immediate area, the electrode metal core, and the electrode covering. As the electrode is consumed, shielding of the arc and the weld metal may be provided by gases formed or by slag produced during decomposition of the covering, depending on the electrode type. Also, as the electrode melts, tiny droplets of molten metal are transferred to the molten weld pool by the arc stream. The welder moves the electrode progressively along the joint, and the deposited filler metal and the molten base metal coalesce to form the weld.

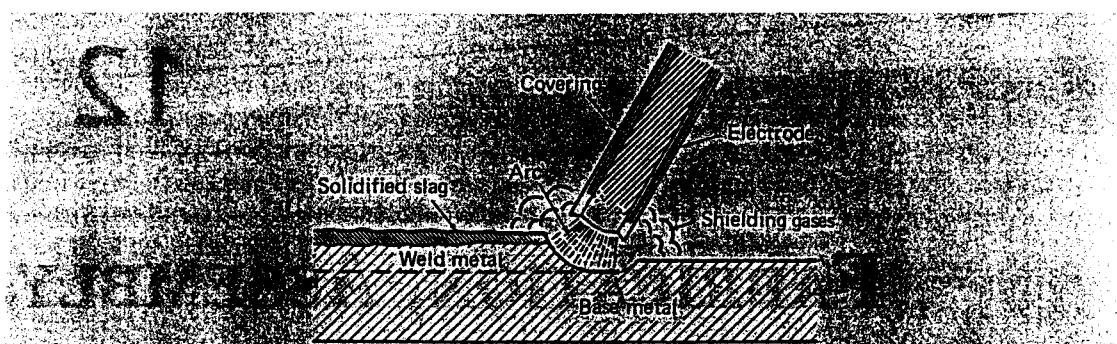


FIG. 12-1
SHIELDED METAL ARC WELDING
(COURTESY OF THE JAMES F. LINCOLN FOUNDATION)

Process Capabilities and Limitations

The SMAW process is suitable for most of the commonly used metals and alloys. However, other processes, such as gas tungsten arc welding (GTAW) and gas metal arc (GMAW) welding, are much preferred for some metals, such as aluminum and aluminum alloys and copper and some copper alloys, and for certain applications, such as root pass welding of groove joints in pipe. The equipment required for SMAW is relatively portable and the process can often be used in areas of limited accessibility.

Because SMAW electrodes are produced in individual straight lengths, they can be consumed only to some minimum length. When this occurs, welding must stop while the electrode is replaced. This results in a loss of arc time; hence, the overall weld metal deposition rate tends to be lower than with a continuous electrode process. However, the deposition rate with SMAW is higher than with GTAW. With SMAW the slag which forms on the weld bead surface usually must be removed from the end of the weld bead before continuing with a new electrode. For many applications, the completed weld surface must be cleaned of slag or oxides, which could contaminate the process stream or cause corrosion. In SMAW the weld quality is highly dependent on the skill of the welder.

Gas Tungsten Arc Welding (GTAW)

Principles of Operation

Gas tungsten arc welding (GTAW) uses a special torch which contains a nonconsumable tungsten or tungsten alloy electrode and provides a concentric flow of inert gas to shield the electrode, arc, and molten metal from the atmosphere (Fig. 12-2). The process is sometimes called TIG for "tungsten inert gas." The heat for welding is produced by an electric arc between the nonconsumable electrode and the part to be welded. Filler metal is added by feeding a bare rod or wire into the zone of the arc and the molten weld metal which is protected by the inert atmosphere. The shielding atmosphere consists of argon, helium, or mixtures thereof. Alternating current is preferred for manual welding of aluminum and magnesium, whereas direct current, electrode negative, is preferred for welding most other materials. A high-frequency pilot arc is usually used for arc starting in order to keep from damaging the electrode or contaminating the weld pool or the workpiece. A weld is produced by applying the arc so that the abutting workpieces and the added filler metal are melted and joined together as the weld metal solidifies.

GTAW equipment is portable, and the process is applicable to most metals in a wide range of thickness and in all welding positions. The process can be used to weld all types of joint geometries and overlays in plate, sheet, pipe, tubing, and structural shapes. It is especially suitable for welding

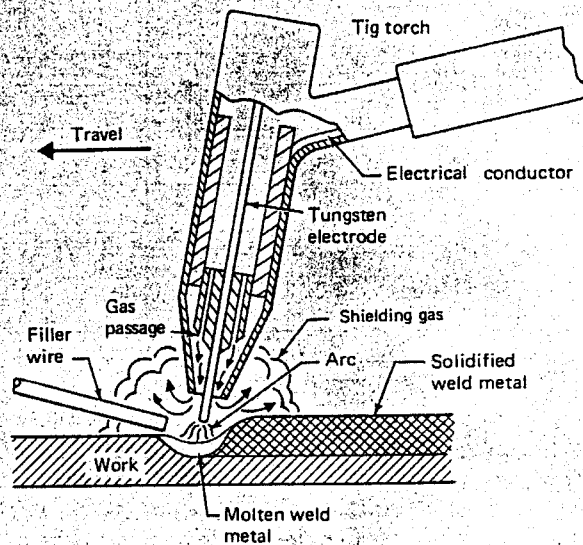


FIG. 12.2
GAS TUNGSTEN ARC WELDING
 (COURTESY OF THE JAMES F. LINCOLN FOUNDATION)

sections of pipe less than 10 mm (3/8 in.) in thickness and also 25.4 mm to 152.4 mm (1 in. to 6 in.) in diameter. Thicker sections can be welded, but it is usually less expensive to use a consumable electrode process, which affords a higher deposition rate. For some applications the process can be used for welding by fusion alone without the addition of filler metal. For welding pipe the combination of GTAW for the root pass followed by either SMAW or GMAW is particularly advantageous. The GTAW process provides a root pass surface which is smooth and uniform on the inside of the pipe, whereas the fill and cap passes are made with a more economical process. High-quality welds can be produced with the GTAW process when adequately trained welders and proper procedures are used.

For GTA welding it is essential to carefully clean the joint surfaces and use clean filler metal in order to avoid weld defects. The GTAW process is slower than consumable electrode processes. Argon and helium shielding gases are relatively expensive. Improper technique which causes transfer of tungsten into the weld or exposure of the hot filler rod to air can result in unsatisfactory welds.

Gas Metal Arc Welding (GMAW)

Principles of Operation

Gas metal arc welding, popularly known as MIG welding, uses a continuous solid-wire consumable electrode for filler metal and an externally supplied gas or gas mixture for shielding (Fig. 12.3). The shielding gas—helium, argon, carbon dioxide, or mixtures thereof—protects the molten metal from reacting with constituents of the atmosphere.

Process Capabilities and Limitations

Gas metal arc welding is operated in semiautomatic, machine, and automatic modes. It is utilized particularly in high-production welding operations. All commercially important metals, such as carbon steel, stainless steel, aluminum, and copper, can be welded with this process in all positions by choosing the appropriate shielding gas, electrode, and welding conditions.

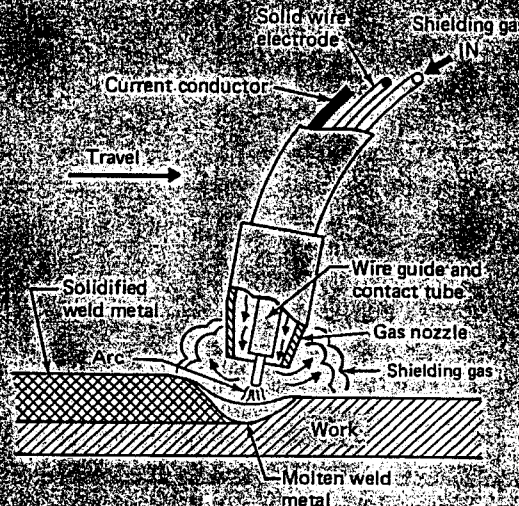


FIG. 12.3

GAS METAL ARC WELDING (COURTESY OF THE JAMES F. LINCOLN FOUNDATION)

Flux-Cored Arc Welding (FCAW)**Principle of Operation**

Flux-cored arc welding uses a continuous tubular wire consumable filler metal. The filler metals are designed principally for joining carbon and low-alloy steels and stainless steels. However, some nickel-base filler metals of this type are also available. The flux core may contain minerals, ferroalloys, and ingredients that provide shielding gases, deoxidizers, and slag-forming materials. There are two basic types of cored electrodes, those designed to be used with an additional external shielding gas (Fig. 12.4), and those of self-shielding type, designed for use without

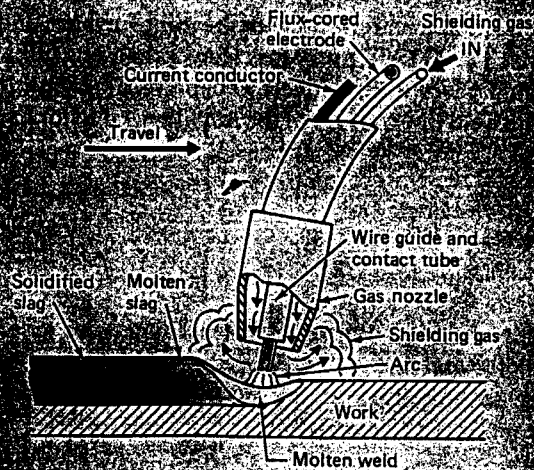


FIG. 12.4

GAS-SHIELDED FLUX-CORED ARC WELDING
(COURTESY OF THE JAMES F. LINCOLN FOUNDATION)

external shielding gas. For carbon and low-alloy steels the external shielding gas is usually carbon dioxide or a mixture of argon and carbon dioxide. For stainless steels, argon and argon-oxygen or argon-carbon dioxide mixtures are often used. Self-shielded FCAW electrodes are designed to generate protective shielding gases from core ingredients, similar to the generation of gases by SMAW electrodes. Self-shielded electrodes do not require external shielding.

Process Capabilities and Limitations

Flux-cored arc welding can be operated in either automatic or semi-automatic modes. Weld metal can be deposited at higher rates, and the welds can be larger and better contoured than those made with solid electrodes (GMAW), regardless of the shielding gas. FCAW lends itself to high-production welding applications, as in a manufacturing plant or fabrication shop.

Some classifications of FCAW electrodes of both the externally shielded and self-shielded types are designed for single-pass welding only. Such electrodes should not be used for multipass welding, because there is a risk of weld cracking. For low-temperature service applications, care should be exercised in selecting a classification of FCAW electrode which is designed and tested to ensure good low-temperature notch ductility.

Submerged Arc Welding (SAW)

Principles of Operation

Submerged arc welding uses an arc (or arcs) maintained between a bare electrode and the work (Fig. 12.5). The feature that distinguishes submerged arc welding from other arc welding processes is the blanket of granular, fusible material that covers the welding area. By common usage the material is termed a flux, although it performs several important functions in addition to those strictly associated with a fluxing agent. Flux plays a central role in achieving the high deposition rates and weld quality that characterize the SAW process in joining and surfacing applications.

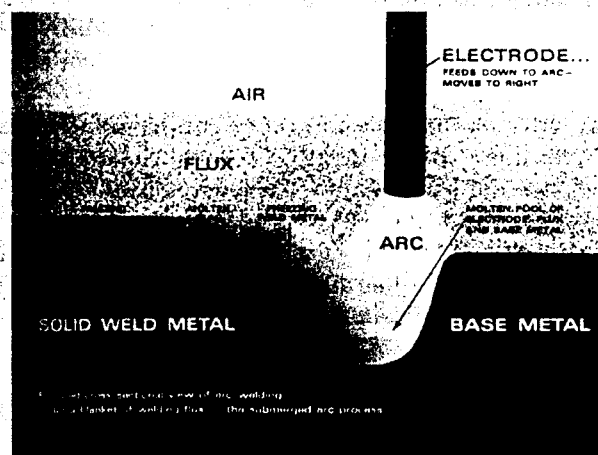


FIG. 12.5
SUBMERGED ARC WELDING (COURTESY OF THE JAMES F. LINCOLN FOUNDATION)

Process Capabilities and Limitations

With proper selection of equipment, SAW can be applied to a wide range of industrial applications. The high quality of welds, the high deposition rates, the deep penetration, and the adaptability to automatic operation make the process particularly suitable for fabrication of large weldments. It is used extensively in ship building, railroad car fabrication, pressure vessel fabrication, pipe manufacturing, and the fabrication of structural members where long welds are required.

SAW is not suitable for all metals and alloys. It is widely used for welding carbon steels, low-alloy structural steels, and stainless steel.

Submerged arc welding can be used for welding butt joints in the flat position; making fillet welds in the flat and horizontal positions; and surfacing in the flat position. With special tooling and fixturing, lap and butt joints can be welded in the horizontal position. The use of this process for welding piping systems is limited to fabrication shops where the pipe can be rolled.

12.3 WELDING PROCEDURE SPECIFICATION

Welds are conducted in accordance with Welding Procedure Specifications (WPSs). A WPS is a written Qualified Welding Procedure prepared to provide direction for making production welds to specified requirements. WPSs or other documents may be used to provide direction to the welder or welding operator to ensure compliance with Code or other specification requirements. A WPS references one or more supporting Procedure Qualification Record(s) (PQR). ASME BPVC, Section IX, QW-482 gives a suggested format for WPSs.

A WPS consists of a number of items, called variables, that define the application of a welding process or combination of processes. Examples of variables are nature of the welding process, base metal type and thickness, filler metal type, welding position, preheat, postheat, electrical characteristics, and type of shielding gas. Variables are divided into three types: essential, nonessential, and supplementary essential.

An essential variable is one that, if changed to a value outside the limits permitted by the original Procedure Qualification Test, requires a new Procedure Qualification Test.

Nonessential variables include items in a WPS that may have to be changed in order to satisfy a particular welding application, but do not affect the properties of the weld. An example is a change in the groove design. A change in a nonessential variable requires revision or amendment of a WPS or a new WPS; however, requalification testing is not required.

A supplementary essential variable is one that comes into consideration when a particular fabrication Code, such as ASME BPVC, Section VIII or ASME B31.3, requires supplementary testing in addition to the customary tensile and bend tests required for procedure qualification (e.g., impact testing for low-temperature service).

ASME BPVC, Section IX, Article II lists the essential, nonessential, and supplementary essential variables for Welding Procedure Specifications and procedure qualification testing for each welding process used for joining and for special applications such as hard-facing overlay and corrosion-resistant overlay.

12.4 WELDING PROCEDURE QUALIFICATION RECORD

The purpose of procedure qualification testing is to determine or demonstrate that the weldment proposed for construction is capable of providing the required properties for its intended application. It is presupposed that the welder or welding operator performing the Welding Procedure Qualification

Test is a skilled worker. That is, the Welding Procedure Qualification Test establishes the properties of the weldment, not the skill of the welder or welding operator.

A PQR includes a record of the welding data used to weld a prescribed test coupon in accordance with a WPS and the results of specified mechanical and other tests. The completed PQR must document all essential and, when required, supplementary essential variables of ASME BPVC, Section IX. ASME BPVC, Section IX, QW-483 gives a suggested format for a PQR.

If any changes are made in a WPS that involve essential variables or, when required, supplementary essential variables, a new Procedure Qualification Test and a new PQR are required in order to document the changes and support a revision of the WPS or issuance of a new WPS.

The details of preparing a weld test assembly or assemblies, testing of the weld, and test results required for procedure qualification are given in ASME BPVC, Section IX. Customary tests for qualification of joint welding procedures are tensile tests and bend tests.

12.5 WELDER PERFORMANCE QUALIFICATION

The Manufacturer or Contractor is responsible for conducting tests to qualify welders and welding operators in accordance with qualified welding procedure specifications that the organization employs in the construction of weldments built in accordance with the ASME Code. The purpose of welder and welding operator qualification tests is to ensure that the manufacturer or Contractor has determined that the welder(s) and welding operator(s) following the procedures are capable of developing the minimum requirements specified for an acceptable weldment. Performance qualification tests are intended to determine the ability of welders and welding operators to make sound welds.

ASME BPVC, Section IX lists and defines essential variables for each welding process for performance qualification of welders and welding operators. It should be noted that the essential variables for performance qualification are in many cases quite different from the variables for procedure qualification discussed in Section 12.3. For the SMAW process, for example, the essential variables for performance qualification are elimination of backing, change in pipe diameter, change in base material P-Number, change in filler metal F-Number, change in thickness of the weld deposit, addition of welding positions beyond that originally qualified, and change in direction of weld progression for vertical welding. A change beyond prescribed limits in an essential variable for performance qualification requires requalification testing and issuance of a new qualification record.

The details of preparing the weld test assembly or assemblies, testing of the weld, and the test results required for welder and welding operator performance qualification are given in ASME BPVC, Section IX. Customary tests for performance qualification are visual and bend tests. However, for some base materials and welding processes there are also alternate provisions for examination of the weld(s) by radiography in lieu of bend tests.

ASME BPVC Section IX, QW-484 gives a suggested format for recording Welder/Welding Operator Performance Qualification (WPQ). A WPQ for a specific process expires when the welder has not welded with that process during a period of 6 months or longer.

Within a 6-month period prior to expiration of a performance qualification a welder who welds using a manual or semiautomatic process maintains all of his or her qualifications for manual or semiautomatic welding with that process. For example, a welder qualified for manual SMAW welding of both carbon steel and stainless steel, but who welds only carbon steel with SMAW during a 6-month period, maintains his or her qualification for SMAW of stainless steel as well as for SMAW of carbon steel. This highlights the importance of keeping timely records of the welding process(es) used during a welder's production welding assignments.

When there is a specific reason to question the ability of a welder or welding operator to make welds that meet the specification, the performance qualification that supports that welding can be revoked. Other qualifications that are not in question remain in effect.

Renewal of performance qualification that has expired may be made for any process by welding a single test coupon of either plate or pipe, of any material, thickness, or diameter, in any position, and by testing that coupon by bending or by radiography (if the latter is permitted by ASME BPVC, Section IX for the process and material involved; see QW-304). Alternatively, where radiography is a permissible method of examination (QW-304), the renewal of qualification may be done on production work.

12.6 PREHEATING

Preheating requirements are provided in para. 330 of ASME B31.3. Preheating is used, along with heat treatment, to minimize the detrimental effects of high temperature and severe thermal gradients in welding, prevent hydrogen cracking, and improve the metalurgical properties. The effect of reducing hydrogen cracking is accomplished by a variety of factors, including driving off moisture, reducing the cooling rate, and increasing the rate of hydrogen diffusion in the material. The reduced cooling rate can also improve material properties; in some materials, too rapid a cooling rate results in brittle microstructural constituents.

The preheat requirements, which are applicable to all types of welding, including tack welds, repair welds, and seal welds of threaded joints, are provided in Table 330.1.1, reproduced here as Table 12.1. The requirements for preheat are a function of the P-Number or S-Number of the base material, the nominal wall thickness, and the specified minimum tensile strength of the base metal. Both required and recommended preheat temperatures are provided. If the ambient temperature is below 0°C (32°F), the recommended preheat temperatures become required preheat temperatures. When welding dissimilar metals, the higher preheat temperature should be used. Preheat for unlisted materials must be as specified in the Welding Procedure Specification.

The preheat zone is required to extend at least 25 mm (1 in.) beyond each edge of the weld. The temperature must be obtained prior to, and maintained during, the welding. The temperature is required to be verified by checking with temperature-indicating crayons, thermocouple pyrometers, or other suitable means.

12.7 HEAT TREATMENT

Post-weld heat treatment is performed to temper the weldment, relax residual stresses, and remove hydrogen. The consequential benefits are avoidance of hydrogen-induced cracking and improved ductility, toughness, corrosion resistance, and dimensional stability.

Heat treatment requirements are provided in para. 331 of ASME B31.3. The Code requires heat treatment after certain welding, bending, and forming operations. Specific requirements for postweld heat treatment are provided in Table 331.1.1 (Table 12.2). This table specifies the heat treatment time and temperature based on the P- or the S-Number, the material chemistry, the wall thickness, and the specified minimum tensile strength of the base material.

Requirements for heat treatment after bending or forming are stated in para. 332.4. These depend on the material P-Number, whether the material is hot- or cold-bent or formed, and, for cold-bent or formed material, the fiber elongation.

Heat treatment in an enclosed furnace, local flame heating, electric resistance heating, electric induction heating, or exothermic chemical reactions are all permitted methods of supplying heat. However, the method is required to provide the required metal temperature, metal temperature uniformity, and temperature control. Similar to preheat, the metal temperature must be verified. The Code requires that at least 25 mm (1 in.) on either side of the weld or bent or formed section be brought up to temperature. AWS D10.10, *Recommended Practices for Local Heating of Welds in Piping and Tubing*, provides useful guidance in the performance of local heat treating.

TABLE 12.1
PREHEAT TEMPERATURES

Base Metal P-No. or S-No. [Note (1)]	Weld Metal Analysis A-No. [Note (2)]	Base Metal Group	Nominal Wall Thickness		Specified Min. Tensile Strength, Base Metal		Min. Temperature	
			mm	in.	MPa	ksi	°C	°F
1	1	Carbon steel	<25 ≥25 All	<1 ≥1 All	≤490 All >490	≤71 All >71	...	10 79 79
3	2, 11	Alloy steels, Cr ≤ 1/2%	<13 ≥13 All	<1/2 ≥1/2 All	≤490 All >490	≤71 All >71	...	10 79 79
4	3	Alloy steels, 1/2% < Cr ≤ 2%	All	All	All	All	149	300
5A, 5B, 5C	4, 5	Alloy steels, 2 1/4% ≤ Cr ≤ 10%	All	All	All	All	177	350
6	6	High alloy steels martensitic	All	All	All	All	...	149 ³
7	7	High alloy steels ferritic	All	All	All	All	...	10
8	8, 9	High alloy steels austenitic	All	All	All	All	...	10
9A, 9B	10	Nickel alloy steels	All	All	All	All	...	93
10	...	Cr-Cu steel	All	All	All	All	149-204	300-400
10I	...	27Cr steel	All	All	All	All	149 ⁴	300 ⁴
11A SG 1	...	8Ni, 9Ni steel	All	All	All	All	...	10
11A SG 2	...	5Ni steel	All	All	All	All	10	50
21-52	...	...	All	All	All	All	...	10

NOTES:

- (1) P-Number or S-Number from BPV Code, Section IX, QW/QB-422.
- (2) A-Number from Section IX, QW-442.
- (3) Maximum interpass temperature 316°C (600°F).
- (4) Maintain interpass temperature between 177°C-232°C (350°F-450°F).

TABLE 12.2
REQUIREMENTS FOR HEAT TREATMENT

Base Metal P-No. or S-No. [Note (1)]	Weld Metal Analysis A-Number [Note (2)]	Base Metal Group	Specified Min.			Metal Temperature Range			Holding Time			Brinell Hardness Min. [Note (4)] Max.
			Nominal Wall Thickness	Strength, Base Metal	Tensile Metal	°C	°F	Nominal Wall [Note (3)]	Min. Time, hr			
mm	in.	MPa	ksi			min/mm	hr/in.					
1	1	Carbon steel	≤19 >19	≤3/4 >3/4	All All	All All	None 593–649	None 1100–1200	...	...	...	...
3	2, 11	Alloy steels, Cr ≤1/2%	≤19 >19 All	≤3/4 >3/4 All	≤490 All >490	≤71 All >71	None 593–718 593–718	None 1100–1325 1100–1325	...	...	...	...
4 ¹⁰	3	Alloy steels, 1/2% < Cr ≤ 2%	≤13 >13 All	≤1/2 >1/2 All	≤490 All >490	≤71 All >71	None 704–746 704–746	None 1300–1375 1300–1375	...	...	...	...
5A, ¹⁰ 5B, ¹⁰ 5C ¹⁰	4, 5	Alloy steels (2 1/4% ≤ Cr ≤10%) ≤3% Cr and ≤0.15% C ≤3% Cr and ≤0.15% C >3% Cr or >0.15% C	≤13 >13 All	≤1/2 >1/2 All	All All All	All All All	None 704–760 704–760	None 1300–1400 1300–1400	...	...	...	...
6	6	High-alloy steels martensitic	All	All	All	All	732–788	1350–1450	2.4	1	2	241
7	7	A 240 Gr. 429 High-alloy steels ferritic	All	All	All	All	621–663	1150–1225	2.4	1	2	241
8	8, 9	High-alloy steels austenitic	All	All	All	All	None	None	...	...	...	...
9A, 9B	10	Nickel alloy steels	≤19 >19	≤3/4 >3/4	All All	All All	None 593–635	None 1100–1175	...	...	...	...
10	...	Cr–Cu steel	All	All	All	All	760–816	1400–1500	1.2	1/2	1/2	...
							[Note (5)]	[Note (5)]				

10H	...	Duplex stainless steel	All	All	All	All	Note (7)	1.2	1/2	1/2	...
10I	...	27Cr steel	All	All	All	663-704 [Note (6)]	1225-1300 [Note (6)]	2.4	1	1	...
11A SG 1	...	8Ni, 9Ni steel	≤51	≤2	All	None	None	...	...	...	...
11A SG 2	...	5Ni steel	>51	>2	All	552-585 [Note (8)]	1025-1085 [Note (8)]	2.4	1	1	...
62	...	Zr R60705	All	All	All	552-585 [Note (8)]	1025-108 [Note (8)]	2.4	1	1	...
						538-593 [Note (9)]	1000-1100 [Note (9)]	Note (9)	Note (9)	1	...

NOTES:

- (1) P-Number or S-Number from BPV Code, Section IX, QW/QB-422.
- (2) A-Number from Section IX, QW-442.
- (3) For holding time in SI metric units use min/mm (minutes per millimeter thickness). For U.S. units, use hours per inch thickness.
- (4) See para. 331.1.7.
- (5) Cool as rapidly as possible after the hold period.
- (6) Cooling rate to 649°C (1200°F) shall be less than 56°C (100°F)/hr; thereafter, the cooling rate shall be fast enough to prevent embrittlement.
- (7) Postweld heat treatment is neither required nor prohibited, but any heat treatment applied shall be as required in the material specification.
- (8) Cooling rate shall be >167°C (300°F)/hr to 316°C (600°F).
- (9) Heat treat within 14 days after welding. Hold time shall be increased by 1/2 hr for each 25 mm (1 in.) over 25 mm thickness. Cool to 427°C (800°F) at a rate ≤278°C (500°F)/hr, per 25 mm (1 in.) nominal thickness, 278°C (500°F)/hr max. Cool in still air From 427°C (800°F).
- (10) See Appendix F, para. F331.1.

Brinell hardness testing after the heat treatment is performed is required for low-alloy steel as a means of quality control, to ensure that the metal has been properly tempered. Hardness testing is performed both on the weld metal and the heat-affected zone. Where hardness testing is required, 10% of the welds, hot-bends, and hot formed components in each furnace heat-treated batch are to be tested. When local heat treatment is performed, 100% are required to be hardness tested.

ASME B31.3, para. 331.2, permits exceptions to the heat treatment requirements of Table 331.1.1 (Table 12.2 here) where warranted based on knowledge or experience of the service conditions:

- Normalizing, or normalizing and tempering, or annealing may be used in lieu of the heat treatment of Table 331.1.1, provided the material properties of the weld and base material meet the specification requirements after heat treatment. This requires approval of the Designer.
- More or less stringent heat treatment may be specified by the Designer. If less stringent heat treatment is performed, the Designer must demonstrate to the Owner that the heat treatment is adequate, and the welding procedure qualification tests must be conducted with the heat treatment that will be used.

One of the reasons for selecting a lower heat treatment temperature is mentioned in para. F331.1 of Appendix F. It states the following.

“Heat treatment temperatures listed in Table 331.1.1 for some P-No. 4 and P-No. 5 materials may be higher than the minimum tempering temperatures specified in the ASTM specifications for the base material. For higher-strength normalized and tempered materials, there is consequently a possibility of reducing tensile properties of the base material, particularly if long holding times at the higher temperatures are used.”

There are differences between the ASME B31.3 heat treatment temperatures and those specified in ASME BPVC, Section VIII, Division 1. For example, the metal temperature range for P4 materials is 704°C to 746°C (1300°F to 1375°F) in ASME B31.3 and 593°C (1100°F) in ASME BPVC, Section VIII, Division 1. In addition, the Pressure Vessel Code does not require hardness testing after heat treatment. This can occasionally cause problems when the Pressure Vessel Code heat treatment is used; the desired tempering, as measure by hardness, may not be achieved. For example, ASME B16.34, which covers valves, refers to ASME BPVC, Section VIII, Division 1 heat treatment requirements rather than ASME B31.3. If the higher Piping Code heat treatment and quality control via hardness testing is required, as it sometimes is for corrosion reasons, it must be specified in the engineering design.

12.8 GOVERNING THICKNESS FOR HEAT TREATMENT

When using Table 331.1.1 (Table 12.2 here), the thickness to be used is generally the thicker of the two components, measured at the joint, that are being joined by welding. For example, if a pipe is welded to a heavier wall valve, but the valve thickness is tapered to the pipe thickness at the welded joint, the governing thickness will be the greater of the valve thickness at the end of the taper at the weld joint (presumably the nominal pipe wall thickness) or the pipe thickness. Two special cases are branch connections and fillet weld joints.

For branch connections, it is the thickness of the weld that is considered. Only half of the thickness of the weld is used as the governing thickness (or, as stated in the Code, the thickness through the weld is compared to twice the minimum material thickness requiring heat treatment in Table 331.1.1). This includes the dimension through the penetration weld joining the run and branch pipes and reinforcement, if any, as well as the cover fillet weld. Specific guidance for determining the weld thickness is provided in para. 331.1.3(a). Note that for the cover fillet, the throat dimension is used. For example, the thickness of the weld in Fig. 12.8(2) (see below) is $(\bar{T}_h + t_c)$.

It is actually required to also consider the thickness of the parts joined by welding in branch connections as well as the weld metal; however, metal added as reinforcement, whether an integral part of a branch fitting or attached by welding, need not be considered. Since branch connections are required by the Code to be full penetration-welded, it is unlikely that a circumstance will arise where the base material is thicker than the weldment. For example, although integrally reinforced branch connection fittings often have substantial thickness, the weld size would govern, because the additional thickness of the fitting is entirely reinforcement.

For fillet welds at slip-on and socket weld joints DN 50 (NPS 2) and smaller, for seal welding of threaded joints in piping DN 50 (NPS 2) and smaller, and for attachment of external nonpressure parts such as lugs, the heat treatment is based on the larger of either the thickness through the weld or the thickness of the parts that are joined by the weld, with certain exceptions. As with branch connections, one bases evaluation of the weld on one-half of the thickness (or compares the thickness through the weld to twice the thickness in Table 331.1.1). For larger pipe, only the base material thickness, not the weld thickness, is considered.

The exceptions permit consideration of the weld only, neglecting the base material thickness. These exceptions, which exempt certain fillet welds from heat treatment (per para. 331.1.3), are as follows:

- P-Number 1 materials with weld throat thickness ≤ 16 mm (5/8 in.)
- P-Number 3, 4, 5, or 10A materials with weld throat thickness ≤ 13 mm (1/2 in.) when the recommended preheat is applied and the specified minimum tensile strength of the base material is less than 490 MPa (71 ksi).
- Ferritic base materials with welds made with filler metal that does not air-harden, such as austenitic material (note that the weld material must also be suitable for the service conditions)

12.9 PIPE BENDS

Pipe may be hot- or cold-bent. For cold-bending of ferritic materials, the temperature must be below the transformation range. For hot-bending, the temperature must be above the transformation range.

The thickness after bending must comply with the design requirements. Per para. 304.2.1, this was required to be the thickness required for straight pipe. However, with the recent addition of the Lorenz equation, as discussed in Section 4.8, the required thickness must be calculated.

When pipe is bent, it tends to ovalize (also termed flattening). Paragraph 332.2 requires that the flattening (the difference between the maximum and minimum diameters at any cross section) not exceed 8%, except when the pipe may be subject to external pressure, in which case the limit is 3%. Although this has often been treated as an absolute limit, it is possible to perform a detailed calculation in the engineering design per para. 300(c)3 to establish a different, perhaps higher, limit. For example, a heavy-wall, small-diameter pipe may have a very substantial margin against buckling even if the 3% flattening is significantly exceeded. See Section 12.7 for heat treatment after bending.

12.10 BRAZING AND SOLDERING

Brazing procedures, brazers, and brazing operators are required to be qualified in accordance with ASME BPVC, Section IX, Part QB. The brazing process is described below, in Insert 12.2. An exception is for piping in Category D Fluid Service with a design temperature not exceeding 93°F, (200°F), for which the Owner can waive the requirements for such qualifications.

Solders are required to follow the procedures in the *Copper Tube Handbook* of the Copper Development Association (1995).

Aside from these requirements, general good practice requirements for brazing and soldering are specified in para. 333 of ASME B31.3.

Insert 12.2 Brazing

Description of Process. Brazing includes a group of metal-joining processes which produce coalescence of materials by heating them to a suitable temperature and by using a filler metal which melts at a temperature above 840°F and below the temperature at which the base metal starts to melt. The brazing filler metal is distributed between closely fitted surfaces of the joint by capillary attraction.

Various types of brazing are defined according to the method of heating employed, for example, dip brazing in a molten metal or salt bath, furnace brazing, induction brazing, infrared brazing, resistance brazing, and torch brazing. Torch brazing is the most commonly used brazing process for process plant construction and maintenance operations.

Torch Brazing: Principles of Operation. Torch brazing is accomplished by heating with one or more torches using a gas fuel (acetylene, propane, city gas, etc.) burned with air, compressed air, or oxygen. Brazing filler metal may be preplaced at the joint in the form of rings, washers, strips, powder, etc.; or it may be fed in wire or rod form by hand. In any case, proper precleaning and fluxing are essential.

Capabilities and Limitations

Brazing can be employed for joining a wide variety of metals and alloys, and appropriate brazing filler metals are available for various applications. To achieve a good joint, the parts must be properly cleaned and must be protected during the heating process, usually by use of a flux in torch brazing, in order to prevent excessive oxidation. The parts must be designed and properly aligned to afford capillary flow of the filler metal. Heat must be applied so as to provide the required temperature range and uniform temperature distribution in the parts.

12.11 BOLTED JOINTS

Proper assembly of bolted joints is essential to avoid leakage during service. This includes not only visible leaks, but also fugitive emissions, which are an important consideration in the United States as a result of environmental regulations. Information on flange bolting is provided in ASME BPVC, Section VIII, Division 1, Appendix S. An ASME guideline on flange boltup procedures, PCC-1, *Guidelines for Pressure Boundary Bolted Flange Joint Assembly*, was published in 2001.

ASME B31.3 provides some good practice guidelines with respect to flange boltup in para. 335. This includes requiring repair or replacement of flanges with damaged gasket seating surfaces, uniformly compressing the gasket during flange boltup, and using only one gasket between seating surfaces.

Paragraph 335.2.3 requires that the bolts extend completely through their threads. However, it provides that the bolt is considered to be acceptably engaged if it is short of being completely through the nut by one thread or less. Thus, if the Owner wishes to have the nuts completely engaged, as many do, this would have to be specified in the engineering design.

Perhaps one of the most frequently violated provisions of the Code is the flange alignment tolerance in para. 335.1.1(c). This requires that, before bolting up, flange faces shall be aligned to the design plane within 1 mm in 200 mm (1/16 in./ft) measured across any diameter, and that bolt holes shall be aligned within 3 mm (1/8 in.) of maximum offset. The first requirement relates to cocking of one

flange relative to the other and the second relates to offset or torsional misalignment. This means that each flange can be misaligned 1 mm in 200 mm relative to the design plane. Thus, the flanges could be misaligned relative to each other by as much as double the amount. Furthermore, the design plane is not required to be the same for each flange (e.g., in a system where there is intentional misalignment to achieve cold spring). However, this would have to be the intention of the engineering design.

This requirement became an issue on a project where the gaps between flanges for small-bore [e.g., DN 50 (NPS 2)] pipe flanges were being measured with feeler gages to check the misalignment. Interpretation 15-07 resulted, with the following question and reply:

“Question: In accordance with ASME B31.3c-1995 Addenda, para. 335.1.1(c), prior to bolting up a flanged joint, may the flange faces be out of alignment from the design plane by more than 1/16 in./ft (0.5%), provided the misalignment is considered in the design of the flanged assembly and attached piping in accordance with para. 300(c)(3)?”

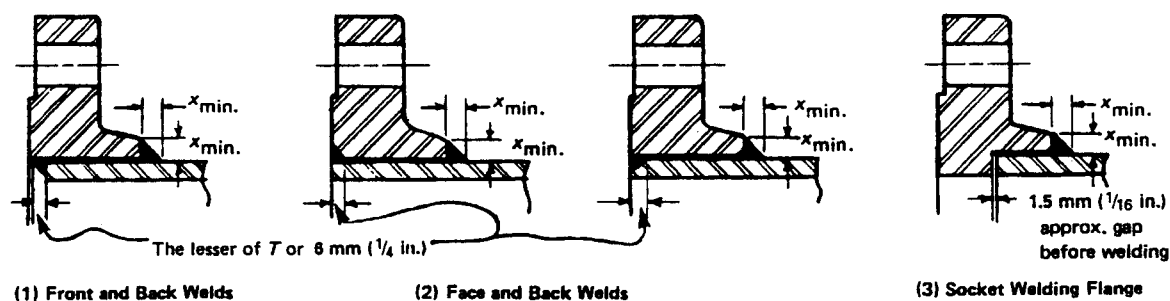
“Reply: Yes.”

Thus, some greater misalignment can be tolerated if it is provided for in the engineering design. It is quite reasonable to expect that greater misalignment than permitted by para. 335.1.1(c) can be accepted in small-bore piping, particularly if it is not connected to load-sensitive equipment. On the other hand, the Code alignment provisions are generally not tight enough for larger piping connected to load-sensitive equipment. An appropriate test of whether the alignment is acceptable is to check the machinery alignment with and without the piping bolted to it.

12.12 WELDED JOINT DETAILS

Welded joint details, including socket weld joints, socket weld and slip-on flanges, and branch connections are provided in ASME B31.3, Chapter V.

Standard details for slip-on and socket welding flange attachment welds are provided in Fig. 328.5.2B, reproduced here as Fig. 12.6. Points worth noting are the fillet weld size, which is 1.4 times the nominal pipe wall thickness, or the thickness of the hub, whichever is less, and the small gap shown between the flange face and the toe of the inside fillet for slip-on flanges. The small gap is intended to avoid damage to the flange face due to welding. It indicates a gap, but there is no specific limit. This differs from ASME BPVC, Section VIII, Division 1, which specifies the gap to be 6 mm (1/4 in.), and some Pipeline Codes that have limits.



$X_{min.}$ = the lesser of $1.4T$ or the thickness of the hub

FIG. 12.6
TYPICAL DETAILS FOR DOUBLE-WELDED SLIP-ON AND SOCKET WELDING FLANGE
ATTACHMENT WELDS (ASME B31.3, FIG. 328.5.2B)

The question arose as to whether a specific limit to the gap between the fillet weld and flange face was appropriate. Studies, including finite-element analysis (Becht et al, 1992) and earlier Markl fatigue testing, indicated that it essentially did not matter how much the pipe was inserted into the flange. Insertion by an amount equal to the hub height was optimal for fatigue life, but there was not a significant difference. To minimize future confusion, inclusion of minimum insertion depth has been recommended and may be specified in a future Edition of ASME B31.3.

The required fillet weld size for socket welds other than socket weld flanges is specified in Fig. 328.5.2C, reproduced here as Fig. 12.7. There are some points worth mentioning in this figure. The first is that the specified fillet weld size is inadequate for general applications and impractical. It will most probably be revised in a future Edition of the Code. It is specified to be $1.25t$, but not less than 3 mm (1/8 in.), where the dimension t is the pressure design thickness, which can be on the order of hundredths of an inch for small-bore piping, even though the pipe may be required to be XS or XXS due to corrosion and mill tolerance considerations. Although a 3-mm (1/8-in.) weld may be perfectly adequate for pressure, it is highly questionable for thermal expansion loads that may be present in heavy-wall, small-bore piping. Furthermore, the welder and Inspector are very unlikely to know the pressure design thickness. Changing this dimension from 1.25 times the pressure design thickness to 1.25 times the nominal thickness of the pipe, or the width of the flat face of the socket, whichever is less, is under consideration.

A second issue that has caused considerable controversy is the 1.5-mm (1/16-in.) approximate gap before welding indicated on the figure. This is a requirement for a gap before welding, so that weld shrinkage will be less likely to cause small cracks in the root of the fillet weld. Whether or not such cracks cause problems is questionable, and fatigue testing has shown that socket welds that are welded after jamming the pipe into the socket have longer fatigue lives than ones welded with a gap. There is no requirement for a gap after welding, and weld shrinkage can close a gap that was present prior to welding.

Some Owners require random radiographic examination to ensure proper socket welding practice. One of the items checked for is the presence of a gap. An argument for doing this is that it is not possible to determine if there was a gap prior to welding unless there is a gap shown by radiography. If desired, the requirement that there be a gap after welding should be specified as an additional requirement of the engineering design. It is not a Code requirement.

Acceptable details for branch connections are provided in Fig. 328.5.4D, reproduced here as Fig. 12.8. ASME B31.3 does not include calculations for required weld sizes for these connections; instead, the minimum weld sizes are specified in this figure.

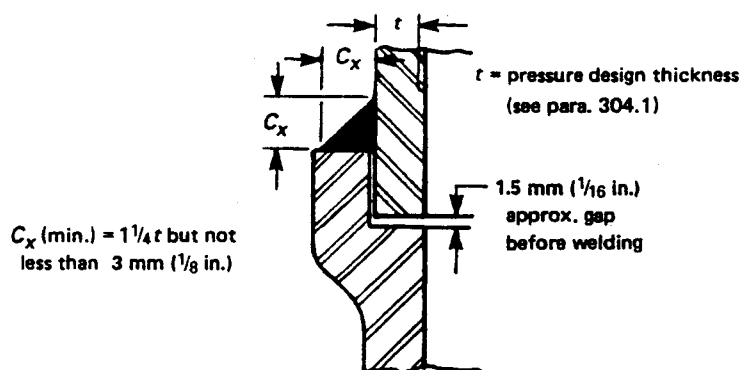
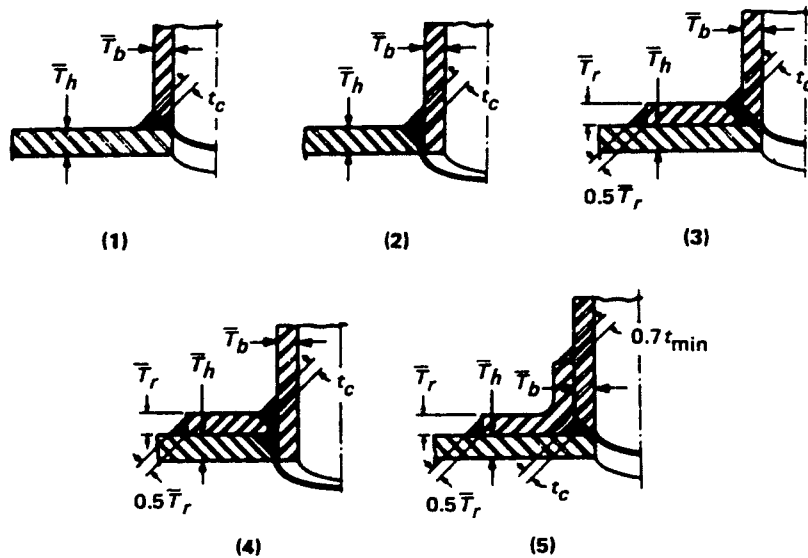


FIG. 12.7
MINIMUM WELDING DIMENSIONS FOR SOCKET WELDING COMPONENTS OTHER THAN FLANGES (ASME B31.3, FIG. 328.5.2C)



GENERAL NOTE: These sketches show minimum acceptable welds. Welds may be larger than those shown here.

FIG. 12.8

ACCEPTABLE DETAILS FOR BRANCH ATTACHMENT WELDS (ASME B31.3, FIG. 328.5.4D)

12.13 MISCELLANEOUS ASSEMBLY REQUIREMENTS

Threaded joints should generally be lubricated with a thread compound or lubricant that is suitable for the service conditions and does not react unfavorably with either the service fluid or the piping material. However, if the joint is intended to be seal-welded, the use of thread compound or tape is prohibited. The material can result in a poor-quality seal weld. There is one exception to this. If a joint is made up with thread tape or compound and it leaks during the leak test, it can be seal-welded without disassembling it, provided that all the material is removed from the exposed threads. As discussed in the Design Section, seal welds are not considered to contribute to the joint strength.

Various good-practice requirements are provided in para. 335 for assembly of straight-threaded joints, tubing joints, caulked joints, expanded joints, and packed joints.

EXAMINATION

13.1 OVERVIEW OF EXAMINATION REQUIREMENTS

ASME B31.3 requires that examination of the piping be performed by the piping Manufacturer (for components only), Fabricator, and/or Erector as a quality control function. These examinations include radiography, for which ultrasonic examination or in-progress examination can be used as substitutes, visual examination, and, under certain conditions, magnetic particle or liquid penetrant examination.

The Examiner is required to have training and experience commensurate with the needs of the specified examination, with records of such qualifications maintained by the employer. Although there are no specific requirements, ASME B31.3 refers to SNT-TC-1A, *Recommended Practice for Nondestructive Testing Personnel Qualification and Certification*, as an acceptable guide.

The owner's Inspector oversees the work performed by the Examiner. It is the Inspector's responsibility to verify that all the required examinations have been completed and to inspect the piping to the extent necessary to be satisfied that it complies with all of the applicable examination requirements of the Code and of the engineering design. Note that the process of inspection does not relieve the Manufacturer, Fabricator, or Erector of their responsibilities for complying with the Code.

The owner's Inspector is also required to be qualified to perform the work. Per para. 340.4(b), the minimum requirement is "not less than 10 years of experience in the design, fabrication, or inspection of industrial pressure piping," with each 20% of completed work toward an engineering degree recognized by the Accreditation Board for Engineering and Technology counting as 1 year of experience, up to a maximum of 5 years.

The owner's Inspector may be an employee of the owner, of an engineering or scientific organization, or of a recognized insurance or inspection company acting as the owner's agent. Unless the owner is the Manufacturer, Fabricator, or Erector, the Inspector cannot be an employee of those organizations. To be so would of course create a conflict of interest.

Requirements for the examination processes are described in ASME BPVC, Section V, with limited exceptions and additions. ASME BPVC, Section V is referenced by ASME B31.3. The required degree of examination and the acceptance criteria for the examinations are provided in Chapter VI of ASME B31.3. Examiners are required to have a written procedure for their examination work (para. 343).

It is not the intent of ASME B31.3 that the examination ensure that the constructed piping system will be free of defects, even ones that are rejectable if found. Nor is it required that welds that are not examined be free of rejectable defects. Instead, the examination work is part of an overall quality assurance procedure. The leak test is used to test the overall system. Interpretation 5-18, issued on May 5, 1987, addresses this point:

"Question (2): Assuming the Engineering Design requires fabrication to ANSI/ASME B31.3, requiring the minimum 5% visual examination and random radiography of girth butt welds, what is the minimum weld quality required by the Code for butt welds which have not been individually examined, but were represented by a successful random visual and radiographic examination?"

"Reply (2): The Code accepts such welds subject to leak testing. Note: The Code assumes that the 5% which were required to be examined will be representative of the entire lot, but a guarantee of such is a contractual matter to be specified by the engineering design."

If a piping system passes the examination requirements of the Code, it is accepted (complies with the Code) if it passes the leak test. If additional examinations are performed that reveal defects, it is beyond Code requirements; the system already complied with the Code, and the issues of whether to repair these defects and who is to pay for the repair is purely a contractual issue. The defect can be left in and the piping system will still comply with the Code. Of course, it would be judicious to at least perform a fitness-for-service evaluation of the defect if it is intended to be left unrepaired.

Records of the examination procedures and examination personnel qualifications are required to be retained for at least 5 years, unless otherwise specified in the engineering design.

13.2 PROGRESSIVE EXAMINATION

ASME B31.3 includes the concept of progressive examination. The concept applies to all random examinations, but will be discussed here in the context of random radiography. In such examinations, the items to be examined (e.g., welds to be radiographed) are separated into lots. This separation is normally done in terms of time periods, with some number of welds included in each lot as the fabrication/erection continues. Some percentage (e.g., 5% of girth welds for Normal Fluid Service) of the items are selected at random and examined. For each item found to be defective, two more are selected from the same lot and given the same type of examination. These two additional samples must be of the same kind (if welded or bonded joints, by the same welder, bonder, or operator). For these additional items, for each one that is found to be defective, two additional items of the same kind must be examined. At that point, if any defects are found, the entire lot that is being randomly examined is rejected and must be 100% examined, repaired, redone, or replaced.

If, at any of the above steps, no further defective work was found, the entire lot would be accepted contingent on successfully passing the leak test. The defects that were found must be repaired and reexamined. Note that once a weld is found to be defective, it must be repaired and reexamined until it passes the examination. However, defects found in the repaired weld do not count as additional defects in the progressive examination procedure.

It would generally be a poor decision, except for rather small projects, to include all of the work in one lot or to wait until the end of the construction to perform the random radiography. One possible result is the rejection of all of the welds. In addition, performing the examinations as the work progresses can help eliminate poorly performing welders earlier in the project. Interpretation 2-25 addresses lot sizes.

"Question (1): A certain piping system has 100 welds made by the same welder and welding procedure. Ten percent radiography was required. Ten welds were radiographed, of which one was defective. The subsequent second and third progression groups also had defective welds. Does ASME B31.3 require radiographic examination, repair, and radiographic reexamination of all the welds other than the 9 found acceptable or of the 10 welds represented by the one original defective weld?"

"Reply (1): The first alternative is correct if the lot size has been established as 100 and the selection is truly random. The second is correct if the lot size has been established as 10 and each is represented by 1 of the 10 welds radiographed at random."

For welds, the work of each welder or welding operator must be included in the random examination. This may affect the selected lot size and/or the percentage of the welds that must be examined. For example, if six welders worked on a lot of 100 welds, with only one welder working on any given weld, a minimum of 6 welds would have to be examined for Normal Fluid Service, or 6%. If there were five or fewer welders, only 5 welds would have to be examined.

Random examination will not detect every defect. It is a quality control procedure that is intended to produce an appropriate level of quality in welds. Part of the effectiveness of the procedure derives from the fact that the welders know that some of their welds will be radiographed, but do not know which ones will be examined.

13.3 TYPES OF EXAMINATION

Visual examination (VT) uses the unaided eye (except for corrective lenses) to inspect the exterior and readily accessible internal surface areas of piping assemblies or components. It does not include nor require remote examination, such as by use of boroscopes. Visual examination is used to check materials and components for conformance to specifications and freedom from defects; fabrication including welds; assembly of threaded, bolted, and other joints; piping during erection; and piping after erection. Furthermore, visual examination can be substituted for radiography, as described later, which is called in-process examination. Requirements for visual examination are provided in ASME BPVC, Section V, Article 9. Records of visual examinations are not required other than those of in-process examination.

Radiographic examination (RT) uses X-ray or gamma-ray radiation to produce a picture of the subject part, including subsurface features, on radiographic film for subsequent interpretation. It is a volumetric examination procedure that provides a means of detecting defects that are not observable on the surface of the material. Requirements for radiographic examination of welds are provided in ASME BPVC, Section V, Article 2.

Ultrasonic examination (UT) detects defects using high-frequency sound impulses. The defects are detected by the reflection of sound waves. Ultrasonic examination is also a volumetric examination method, which can be used to detect subsurface defects. It can be used as an alternative to radiography for weld examination. The requirements for ultrasonic examination of welds are provided in ASME BPVC, Section V, Article 5, with an alternative for basic calibration blocks provided in para. 344.6 of ASME B31.3.

In-process examination involves the visual examination of the entire joining process, as described in para. 344.7 of ASME B31.3. It is applicable to welding and brazing for metals and bonding for nonmetals. Since radiographic examination is not considered to provide useful results in brazing and bonding, in-process examination is used for these instead of radiography. For welding, it is permitted as a substitute for radiographic examination if specified in the engineering design or specifically authorized by the Inspector. In-process examination, however, is not as effective a quality control procedure as random radiography, and it should only be used for welds when special circumstances warrant.

Liquid-penetrant examination (PT) detects surface defects by spreading a liquid dye penetrant on the surface, removing the dye after sufficient time has passed for the dye to penetrate into any surface defect, and applying a thin coat of developer to the surface, which draws the dye from defects. The defects are observable by the contrast between the color of the dye penetrant and the color of the developer. Liquid-penetrant examination is used for the detection of surface defects, for the examination of socket welds and branch connections in severe cyclic service that cannot be radiographed, and for the examination of all welds, including structural attachment welds, that are not radiographed when the alternative leak test (ASME B31.3, para. 345.9) is used. In addition, liquid-penetrant examination of metallic bellows is required by Appendix X, para. X302.2.2. The requirements for liquid-penetrant examination of welds and components other than castings are provided in ASME BPVC, Section V, Article 6.

Magnetic-particle examination (MT) employs either electric coils wound around the part or prods to create a magnetic field. A magnetic powder is applied to the surface, and defects are revealed by patterns the powder forms in response to the magnetic field disturbances caused by defects. This technique reveals surface and shallow subsurface defects. As such, it can provide more information than liquid-penetrant examination. However, its use is limited to magnetic materials. Magnetic-particle examination is an alternative to liquid-penetrant examination wherever such an examination is required in ASME B31.3 (except in the case of metallic bellows). The requirements for magnetic-particle

examination of welds and components other than castings are provided in ASME BPVC, Section V, Article 7.

Hardness testing is required after heat treatment under some circumstances, as specified in Table 331.1.1 of ASME B31.3. Hardness testing is not required for carbon steel (P-1), ferritic and austenitic stainless steel (P-7, P-8), high-nickel alloys (P-9A, P-9B), as well as some less commonly used alloys. It is required in some circumstances for low- and intermediate-alloy steels. For welds, the hardness check includes both the weld and the heat-affected zone. It is a quality control procedure to make sure that the heat treatment was effective.

In addition to the above, ASME B31.3 permits increasing the quality factor for castings with supplementary examination of the casting. Liquid-penetrant, magnetic-particle, ultrasonic, and radiographic examinations may be used in these supplementary examinations. In this case, the requirements for the examination and acceptance criteria are stated and/or referenced in para. 302.3.3 of ASME B31.3.

13.4 REQUIRED EXAMINATION

The required examination depends on the category of fluid service. Different degrees of examination are required for Category D, Normal, and Category M Fluid Services. More examination is required for more hazardous services. In addition, piping under severe cyclic conditions and piping that will be subjected to an alternative leak test require even more examination. Table 13.1 summarizes the required examination for metallic piping.

13.5 ACCEPTANCE CRITERIA

The acceptance criteria for the various examination techniques are provide in Table 341.3.2 for Normal Fluid Service, severe cyclic conditions, and Category D Fluid Service. The acceptance criteria for Category M Fluid Service is the same as for Normal Fluid Service.

The difference between the acceptance criteria for severe cyclic conditions and Normal Fluid Service relates to the potential for fatigue failure. For example, although some undercutting is permitted for Normal Fluid Service, none is permitted for severe cyclic conditions.

13.6 EXAMINATION OF NONMETALLICS

Nonmetallic piping systems have the same visual examination requirements as the base Code, as follows:

- VT requirements for normal fluid service follow the base Code.
- VT requirements for Category D Fluid Service follow the base Code.
- VT requirements for Category M Fluid Service follow the base Code requirements for Normal Fluid Service. This is an inconsistency, and it is recommended that the VT requirements for metallic piping in Category M Fluid Service be followed (i.e., 100% VT of fabrication and mechanical joints).

Because radiography is not a generally accepted practice for nonmetallics, 5% in-process examination is substituted for 5% random radiography for Normal Fluid Service. Again, for nonmetallics in Category M Fluid Service, the same 5% in-process examination requirement applies. Increasing this to 20% is not required.

TABLE 13.1
REQUIRED EXAMINATION FOR METALLIC PIPING

	Normal Service	Severe Cyclic Conditions [Note (1)]	Category D	Category M
Materials and components [Note (2)]	Random VT	Random VT	VT to the extent necessary [Note (11)]	Random VT
Fabrication [Note (3)]	Min. 5% VT	100% VT	VT to the extent necessary [Note (11)]	100% VT
Longitudinal welds [Note (4)]	100% VT	100% VT	VT to the extent necessary [Note (11)]	100% VT
Bolted, threaded mechanical joints [Note (5)]	Random VT [Note (13)]	100% VT	VT to the extent necessary [Note (11)]	100% VT
During erection [Note (6)]	Random VT	Random VT	VT to the extent necessary [Note (11)]	Random VT
Erected piping [Note (7)]	VT	100% VT	VT to the extent necessary [Note (11)]	VT
Circumferential butt and miter welds [Note (8)]	5% random RT [Note (14)]	100% RT [Note (12)]	VT to the extent necessary [Note (11)]	20% random RT [Note (12)]
Brazed joints	5% IP	NP	VT to the extent necessary [Note (11)]	NP
Soldered joints	NP	NP	VT to the extent necessary [Note (11)]	NP
Socket welds	NA	100% MT/PT	NA	NA
Nonradiographed branch connections [Note (9)]	NA	100% MT/PT	NA	NA
Structural attachments [Note (10)]	NA	NA	NA	NA

LEGEND:

NP = not permitted

RT = radiographic examination (UT is always an acceptable alternative and IP is an acceptable alternative if specified in the engineering design or approved by the inspector)

UT = ultrasonic examination

MT = magnetic-particle examination

PT = liquid-penetrant examination

VT = visual examination

IP = in-process examination

NA = no additional requirements (beyond otherwise specified VT)

NOTES:

- (1) "Severe cyclic conditions" essentially only applies to Normal Fluid Service. There are no provisions in ASME B31.3 for Category M Fluid Service under severe cyclic conditions; Category D piping is highly unlikely to be under severe cyclic conditions and there are no provisions in ASME B31.3 to cover that event.
- (2) Sufficient materials and components to satisfy the examiner that they conform to specifications and are free from defects.
- (3) For welds, each welder's or welding operator's work must be represented.
- (4) Except those in components made in accordance with a listed specification.
- (5) Assembly of threaded, bolted, and other joints.
- (6) Examination during erection of piping, including checking of alignment, supports, and cold spring.
- (7) Examination of erected piping for evidence of defects that would require repair or replacement, and for other evident deviations from the intent of the design.
- (8) Circumferential butt and miter groove welds.
- (9) Branch connection welds that are not or cannot be radiographed.
- (10) 100% PT or MT is required if the alternative leak test is to be performed.
- (11) Visual examination to the extent necessary to satisfy the examiner that components, materials, and workmanship conform to the requirements of ASME B31.3 and the engineering design.
- (12) In-progress examination, if used, must be supplemented by other nondestructive examination.
- (13) 100% VT is required if pneumatic testing is to be performed.
- (14) 100% RT is required if the alternative leak test is to be performed.

PRESSURE TESTING

14.1 OVERVIEW OF PRESSURE TEST REQUIREMENTS

ASME B31.3 requires leak testing of all piping systems other than Category D systems. For piping in Category D Fluid Service, the piping may (at the Owner's option) be put in service without a leak test and examined for leakage during the initial operation of the system. This is an initial service leak test. For all other piping, the following options are available:

- Hydrostatic test
- Pneumatic test
- Hydropneumatic test
- Alternative leak test

An alternative leak test is only permitted by ASME B31.3 when:

- Exposure of the piping to water via a hydrostatic test would damage the linings or internal insulation, or contaminate a process that would be hazardous, corrosive, or inoperative in the presence of moisture; and
- A pneumatic test is considered by the owner to entail an unacceptable risk due to the potential release of stored energy in the system (the danger of a pneumatic test increases with the pressure and contained volume); or
- A hydrostatic test or pneumatic test would present the danger of brittle fracture due to low metal temperature during the test

The leak test is required to be conducted after any heat treatment has been completed. If repairs or additions are made following the leak test, the affected piping must be retested unless the repairs or additions are minor and the owner waives retest.

All joints, except those previously tested, must be left uninsulated and exposed for the leak test. ASME B31.3 permits painting of the joints prior to the test except when the joint is to be sensitive leak-tested; however, paint can effectively seal small leaks to extremely high pressures. These small leaks can then, in fact, start leaking in service if the paint film fails, such as due to the presence of solvents in the line.

Piping subassemblies may be tested either separately or as assembled piping. This permits, for example, testing subassemblies and insulating them prior to leak testing the entire system.

Test records are required, per para. 345.2.7 of ASME B31.3, but need not be retained after completion of the test if a certification by the Inspector that the piping has satisfactorily passed pressure testing as required by ASME B31.3 is retained.

14.2 HYDROSTATIC TEST

A hydrostatic test is generally the preferred alternative, because it is conducted at a higher pressure, which has beneficial effects, such as crack blunting and warm pre-stressing, and entails substantially less risk than a pneumatic test. These reduce the risk of crack growth and brittle fracture after the hydrotest when the pipe is placed in service. The test is generally conducted at a pressure, P_T , of 1.5 times the design pressure, P , times a temperature correction factor. The temperature correction factor compensates for the fact that the test may be conducted at a lower temperature, where the material has a higher strength than at the design condition. The equation is¹⁷

$$P_T = 1.5 P \frac{S_T}{S} \quad (14.1)$$

where

S_T = allowable stress at test temperature

S = allowable stress at design temperature

The ratio of S_T/S in the above equation cannot be less than 1 and cannot exceed 6.5. The value 6.5 is used for ratios that are higher than that.

If the test pressure determined in the above equation results in a nominal pressure stress or longitudinal stress in excess of the yield strength of the material at the test pressure, the test pressure may be reduced to a value that brings the maximum of those stresses to yield stress level.

If the piping leak test is combined with a pressure vessel, it is also permitted to reduce the test pressure if it is necessary to do so to avoid overpressuring the vessel when it is not practicable to isolate the piping from the vessel for the purpose of testing them separately and if the owner approves. In that case, the test pressure may be reduced to the vessel test pressure, but must be at least 77% of the hydrotest pressure required by Eq. (14.1).

The springs should generally be left with their travel stops in place through the hydrotest. Also, if the line normally contains vapor or a fluid with a lower density than water, the need for supplemental temporary supports must be considered due to the higher fluid weight than in normal operation.

The pressure is required to be held for at least 10 minutes and as long as is required to completely examine the piping system. All joints must be visually inspected for leakage. There is no provision in ASME B31.3 for substituting a monitoring of pressure decay for the 100% visual examination of the pipe joints during the hydrotest.

14.3 PNEUMATIC TEST

A pneumatic test is considered to potentially entail a significant hazard due to the amount of stored energy in the compressed gas. A rupture could result in an explosive release of this energy. For example, an explosion of 60 m (200 ft) of NPS 36 line containing air at 3500 kPa (500 psi) can create a blast wave roughly equivalent to 35 kg (80 lb) of TNT. The hazard is proportional to both the volume and the pressure.

Because of this concern, the pneumatic test pressure is specified to be at 1.1 times the design pressure. Further, specific precautions are required. These include provision of an adequate pressure relief device to prevent pressurizing the line to too high a pressure. An intermediate hold at the lesser of one-half of the test pressure or 170 kPa (25 psi) with a visual inspection of all joints is required prior to bringing the pressure gradually up to the full test pressure.

¹⁷ASME B31.3, Eq. (24).

14.4 HYDROPNEUMATIC TEST

A hydropneumatic test entails having part of the system filled with water and part of the system filled with pressurized air. Basically, the portions of the system filled with water can be tested at a higher pressure than would be permitted for a pneumatic test. The portions of the system filled with compressed gas are subjected to the limitations and requirements for a pneumatic test.

14.5 ALTERNATIVE LEAK TEST

An alternative leak test is permitted, with the owner's approval, when neither a hydrostatic nor a pneumatic leak test would be possible or safe. An alternative leak test contains the following required elements:

- A detailed inspection of all weld joints, including 100% radiography of all circumferential and longitudinal seam welds and liquid-penetrant or magnetic-particle examination of all welds, including structural attachment welds, that cannot be radiographed
- A formal flexibility analysis
- A sensitive leak test

14.6 SENSITIVE LEAK TEST

A sensitive leak test is a required part of an alternative leak test and is required for Category M Fluid Service piping. This is a test performed at low pressures to test for leaks. Methods for performing this type of test are described in ASME BPVC, Section V, Article 10. One approach is to pressurize the piping system with air and check for leaks using a bubble test solution.

The test pressure for the sensitive leak test is required to be at least 105 kPa (15 psi) or 25% of the design pressure, whichever is less. Similar to a pneumatic test, the pressure is to be increased gradually up to an intermediate hold at the lesser of one-half of the test pressure or 170 kPa (25 psi), at which point a preliminary check for leaks must be made. The pressure is then gradually increased to the test pressure, at which point the sensitive leak test is performed.

In addition to the above Code-required circumstances, a sensitive leak test is used by some owners as a supplemental tightness check of other piping systems prior to placing them in operation and/or to check for leaks in piping systems assembled from shop-fabricated and tested subassemblies.

14.7 JACKETED AND VACUUM PIPING

Piping that is designed for external pressure conditions is generally tested with internal pressure. The test pressure is required to be the greater of 1.5 times the design differential pressure or 105 kPa (15 psi).

Jacketed or other double-wall piping requires leak testing of both the inner pipe and the jacket. The test of the inner pipe is based on the more severe of the internal pressure or the jacket pressure condition. The inner pipe welds must be visible for this leak test, which creates a very significant consideration in the design of these systems. Permitting use of a pressure change test (e.g., per ASME BPVC, Section V, Article 10, Appendix VI) for these types of systems is being considered. The jacket is tested as normal piping based on its design pressure, unless otherwise specified in the engineering design. The inner pipe is normally designed to carry the jacket pressure with no internal pipe pressure, without buckling.

14.8 INITIAL SERVICE LEAK TEST

For piping in Category D Fluid Service, ASME B31.3 permits an initial service leak test in lieu of other leak tests, such as hydrostatic or pneumatic tests. In this test, the system is pressurized with the process fluid and the joints are inspected for leaks.

If the process fluid is a vapor, provisions for gradual increase in pressure, with an inspection at an intermediate pressure, apply, similar to pneumatic test requirements. For piping filled with liquid, ASME B31.3 requires that the pressure be gradually increased to the operating pressure. All joints and connections must be observed for leakage.

14.9 REQUIREMENTS FOR NONMETALLIC PIPING

The leak test rules in the base Code, described in the prior paragraphs, are generally applicable to nonmetallic piping, with a few exceptions. The hydrotest pressure for nonmetallics other than thermoplastics (e.g., RTR, fiberglass pipe) and metallic piping lined with nonmetals is 1.5 times the design pressure, but not more than 1.5 times the maximum rated pressure of the lowest rated component in the system. There is no temperature correction factor. It is particularly important not to overpressure fiberglass piping systems. Excessive hydrotest pressures in fiberglass systems have caused subsequent failures in service. The overload condition can damage the material without evidence of a leak during the test itself. This damage has led to subsequent failures in service.

For thermoplastic piping, a temperature correction factor is used, the same as for metallic piping, except that the allowable stresses for thermoplastics in Appendix B of ASME B31.3 are used.

For metallic piping lined with nonmetals, the Code test pressures for the metallic portion of the pipe apply.

Some thermoplastics can behave in a brittle manner when they fail under compressed gas service. The Code specifically prohibits polyvinyl chloride (PVC) and chlorinated polyvinyl chloride (CPVC) from pneumatic testing, because of this hazard. The owner's approval is required for pneumatic testing of any other nonmetallic piping system.

The alternative leak test is not permitted for nonmetallic piping and piping lined with nonmetals.

NONMETALLIC PIPING

15.1 ORGANIZATION

The rules for nonmetallic piping and piping lined with nonmetals are given in Chapter VII of ASME B31.3. The paragraphs in that chapter follow the same numbering as the base Code, Chapters I through VII, but start with the letter A. If requirements located elsewhere in the Code apply to these piping systems, they are referenced by a paragraph in Chapter VII.

The behavior of nonmetallic piping is more complex than that of metallic piping, and the design criteria are significantly less well developed. As a result, designers are left to use their best judgment in many circumstances. For example, although a formal flexibility analysis is required, no methods are provided for performing one.

For metallic piping lined with nonmetals, the base Code requirements for metallic piping are applied to the metallic portion. Supplemental rules are provided for the nonmetallic lining. Background on the Chapter VII rules is provided by Short (1989, 1992).

15.2 DESIGN CONDITIONS

The base Code requirements with respect to design pressure and temperature are generally applicable. However, no credit is permitted for ambient cooling for uninsulated components. In other words, the design temperature is not permitted to be less than the fluid temperature.

15.3 ALLOWABLE STRESS

Various nonmetals have different established methods of determining allowable stresses. Some limited allowable stress values are provided in ASME B31.3, Appendix B for thermoplastic and reinforced thermosetting resin pipe. For the most part, allowable stresses or pressure ratings must be determined from tests performed by the manufacturer. Allowable pressures are provided for reinforced concrete pipe in Table B-4 and borosilicate glass pipe in Table B-5.

The methods of determining the allowable stresses in thermoplastics and reinforced thermosetting resins are provided in ASTM specifications, as follows.

(a) *Thermoplastic*: Hydrostatic design stress (HDS) is determined in accordance with ASTM D2837. Note that the strength is determined based on time-dependent properties (long-term tests extrapolated to longer design times 100,000 hours and 50 years). This is because creep is significant for this material even at ambient temperature. In addition, the strength of this material is highly sensitive to temperature.

(b) *Reinforced Thermosetting Resin (Laminated)*: Design stress (DS) is taken as one-tenth of the minimum tensile strengths specified in Table 1 of ASTM C582. This is also called hand layup. The

strength of RTR is not particularly temperature-sensitive in the range of application, so this allowable stress is considered to be valid from -29°C (-20°F) through 82°C (180°F).

(c) *Reinforced Thermosetting Resin and Reinforced Plastic Mortar (Filament-Wound and Centrifugally Cast)*: The hydrostatic design stress (HDS) used in design is the hydrostatic design basis stress (HDBS) times a service factor, F (often taken as 0.5), which is selected in accordance with ASTM D2992. The HDBS is determined in accordance with ASTM D2992. The Code states that the HDBS is considered to be valid only at 23°C (73°F). However, the strength of reinforced thermosetting resin piping is relatively temperature-insensitive within the normal range of use temperatures. The HDBS is determined from long-term testing.

15.4 PRESSURE DESIGN

The philosophy of the base Code with respect to metallic piping applies to nonmetallic piping. The primary differences are that the table of listed components for nonmetallic piping is Table A326.1 rather than Table 326.1, and there are substantially fewer pressure design equations provided in the Code.

Listed components with established ratings are accepted at those ratings. Listed components without established ratings, but with allowable stresses listed, can be rated using the pressure design rules of para. A304; however, these are very limited. In the cases of listed components without allowable stresses or unlisted components, components must be rated per para. A304.7.2.

The variations permitted in the base Code (para. 302.2.4) are not permitted for nonmetallic piping. They are not permitted to exceed the maximum permissible pressure, even during relief, considering accumulation. They are permitted for metallic piping with nonmetallic lining, provided the lining material is established to be suitable for the increased conditions through prior successful service experience or tests under comparable conditions.

The equations that are available for sizing nonmetallic components are very limited in ASME B31.3. The equations are for straight pipe, flanges, and blind flanges. The use of the referenced flange design method (per ASME BPVC, Section VIII, Division 1, Appendix 2) is questionable for many non-metallics. As a result, for pressure design, most nonmetallic piping components must be either per a listed standard (i.e., listed in Table A326.1) or qualified per A304.7.2.

Paragraph A304.7.2, on pressure design of unlisted components and joints, differs from the base Code in that neither experimental stress analysis nor numerical analysis (e.g., finite-element analysis) are listed as an acceptable alternative for qualifying components. The two methods that are considered acceptable for substantiating the pressure design are (1) extensive successful service experience under comparable design conditions with similarly proportioned components made of the same or like material and (2) a performance test. The performance test must include the effects of time, because failure of nonmetallic components can be time-dependent.

For straight pipe, Eq. (26a) for thermoplastic, Eq. (26b) for RTR laminated pipe, and Eq. (26c) for RTR filament-wound and RPM centrifugally cast pipe are provided in ASME B31.3. They are of the form

$$t = \frac{PD}{2SF + P} \quad (15.1)$$

where

D = outside diameter of pipe

F = service factor, which is only used for filament-wound and centrifugally cast pipe

P = internal design gage pressure

S = design stress from applicable table in Appendix B of ASME B31.3

t = pressure design thickness

15.5 LIMITATIONS ON COMPONENTS AND JOINTS

Fluid service requirements for nonmetallic piping components are covered in ASME B31.3, Chapter VII, Part 3. Fluid service requirements for nonmetallic piping joints are covered in Chapter VII, Part 4. For the most part, the requirements are similar to the base Code requirements, with relevant paragraphs on nonmetallic components and joints substituted for paragraphs on their metallic counterparts.

15.6 FLEXIBILITY AND SUPPORT

Rules regarding flexibility and support for nonmetallic piping are provided in ASME B31.3, Chapter VII, Part 5. ASME B31.3 does not provide detailed rules for evaluation of nonmetallic piping systems for thermal expansion. However, it requires a formal flexibility analysis when the following exemptions from formal flexibility analysis are not met:

- The system duplicates, or replaces without significant change, a system operating with a successful service record.
- The system can readily be judged adequate by comparison with previously analyzed systems.
- The system is laid out with a conservative margin of inherent flexibility, or employs joining methods or expansion joint devices, or a combination of these methods, in accordance with Manufacturer's instructions.

As in metallic piping, a formal analysis is not necessarily a computer analysis. It can be any appropriate method, including charts and simplified calculations. The objectives in the design of a piping system for thermal expansion are the same as for metallic piping systems. Specifically, they are to prevent:

- Failure of piping or supports from overstrain or fatigue
- Leakage at joints
- Detrimental stresses or distortion in piping or in connected equipment (e.g., pumps) resulting from excessive thrusts and moments in the piping (ASME B31.3, para. A319.1.1).

One of the significant differences from metallic systems is that fully restrained designs are commonly used, that is, systems where the thermal expansion is offset by elastic compression/extension of the piping between axial restraints. This is possible due to the relatively low elastic modulus of plastic piping. The resulting loads are generally reasonable for the design of structural anchors. Note, however, that in a computer flexibility analysis of such systems, the axial load component of thermal expansion stress must be included. See Section 8.1 for a discussion of stresses due to axial loads in flexibility analysis.

Other significant differences from metallic piping include the following:

- (a) Most RTR and RPM systems are nonisotropic. That is, the material properties are different in different directions, depending on the orientation of the reinforcing fibers.
- (b) Axial extension of the pipe due to internal pressure can be significant and should be considered. Note that fiber-wound RTR pipe can either extend or contract due to internal pressure, depending on the orientation of the reinforcing fibers.
- (c) Plastic materials creep at ambient temperature. For example, a plastic pipe that is fully restrained and is compressed as it heats up can experience compressive creep strain during operation. When it cools back to ambient temperature, this can result in tension in the pipe and a load reversal on the restraints.

(d) In plastic piping, particularly RTR systems, the limiting component is often a fitting or a joint. For such systems, the result of the flexibility analysis can be an evaluation of the loads versus the allowable loads on components, rather than a comparison of stress with allowable stress.

(e) Material properties, even for nominally the same material, are often Manufacturer-specific. Thus, design of plastic systems generally requires interaction and consultation with the Manufacturer of the pipe and information on the resin. This is particularly so for RTR and RPM piping, which also include the consideration of the fiber reinforcing.

(f) Stress intensification factors have not been developed for nonmetallic piping. For many non-metallic components (RTR in particular), the design is manufacturer-specific, so the development of industry standard stress intensification factors is problematic.

In general, design of RTR and RPM piping considers the material to be brittle. Thus, there is essentially no difference between stresses due to thermal expansion and those due to weight or pressure. An allowable stress that is comparable to that permitted for pressure is commonly used as an allowable value for the total weight plus pressure plus thermal expansion stress. For laminated RTR, the allowable pressure stress is one-tenth of the tensile strength, and a commonly used allowable value for longitudinal stress due to combined loads is one-fifth of the tensile strength.

Although the behavior of thermoplastics is generally not brittle, so that the shakedown concepts of metallic piping may be applicable, this technology is not developed. Thus, the allowable value for longitudinal stresses due to combined loads is often taken conservatively as the allowable stress for internal pressure.

Some methods have been developed for evaluation of plastic piping. An industry survey is provided in WRC 415 (Short et al, 1996).

With respect to support, ASME B31.3 highlights some specific concerns for nonmetallic piping in para. A321.5. In nonmetallic piping that has limited ductility (e.g., generally with RTR and always with borosilicate glass), avoidance of point loads can be critical to system performance. Although local loads may be accommodated in ductile systems by local plastic deformation, such loads can result in brittle fracture in materials that are brittle. Another consideration is that deformation can accumulate over time due to creep. Support spacing must be sufficiently close to avoid excessive long-term sagging due to creep.

15.7 MATERIALS

Thermoplastic materials may only be used for flammable fluid service when they are underground. In any use other than Category D Fluid Service, thermoplastic piping is required to be safeguarded (see Section 2.5 for information on safeguarding). Poly(vinyl chloride) and chlorinated poly(vinyl chloride) are prohibited from compressed air or other compressed gas service due to the potential for brittle failure.

Reinforced plastic mortar (RPM) piping is required to be safeguarded when used in other than Category D Fluid Service. Reinforced thermosetting resin (RTR) piping may be used in toxic or flammable fluid service, but requires safeguarding. It is generally acceptable for other services, subject to suitability of the material.

Borosilicate glass and porcelain are brittle materials and therefore are required to be safeguarded against large, rapid temperature changes (i.e., thermal shock). Further, these materials are required to be safeguarded when used in toxic or flammable fluid services.

Note that the above limitations only apply when the material is used as the outer pressure-containing element. They do not apply to materials used as linings within metallic piping.

Recommended maximum and minimum temperatures are generally provided. However, similar to the requirements of the base Code, the maximum temperature may be exceeded (unless there is a specific prohibition) if the Designer verifies the serviceability of the material at the temperature. If a material

is to be used at a temperature below the minimum temperature listed in ASME B31.3, Appendix B, the Designer must have some test results at or below the lowest use temperature that ensure that the materials and bonds will have adequate toughness and are suitable at the design minimum temperature. Unlike metallic materials, for nonmetallics, specific tests such as Charpy are not specified.

15.8 BONDING OF PLASTICS

One of the key elements to successful construction of a plastic piping system is the joints. ASME B31.3 requires a formal process of developing, documenting, and qualifying bonding procedures and personnel performing the bonding. The joints in plastic (RTR, RPM) and thermoplastic piping are called bonds. The requirements are similar to the requirements for qualification of welds and welders.

The first step is to have a documented bonding procedure specification (BPS). The specification must document the procedures for making the joint, as set forth in para. A328.2.1. This procedure must be qualified by a bonding procedure qualification test. Once it is so qualified, it may be used by personnel to bond nonmetallic ASME B31.3 piping systems. Those bonders, however, must also be qualified to perform the work.

It is the responsibility of the employer of the bonder who will be joining plastic piping to have a BPS for the joining process that has been qualified by that employer by testing. Per ASME B31.3, para. A328.2.1(b), the BPS is required to specify at least the following:

- All materials and supplies (including storage requirements)
- Tools and fixtures (including proper care and handling)
- Environmental requirements (e.g., temperature, humidity, and methods of measurement)
- Joint preparation
- Dimensional requirements and tolerances
- Cure time
- Protection of work
- Tests and examinations other than those required by ASME B31.3, para. A328.2.5
- Acceptance criteria for the completed test assembly

The bonding procedure must be qualified by test, as described below, to be used by the organization for which the bonder works. The BPS and records of the BPS qualifications must be maintained by the employer of the bonders and be available for review by the Owner or Owner's agent and the Inspector.

The qualification test is the same for the bonding procedure and the bonder. The bonder must fabricate and assemble at least one pipe-to-pipe joint and one pipe-to-fitting joint and must pressure-test it. There are two options for the pressure test. The first is a burst test in accordance with ASTM D1599. This requires increasing the pressure uniformly at a rate that will result in failure in 60 seconds to 70 seconds. The acceptance criteria when using this burst test procedure to qualify the bond is that the failure must initiate outside of the joint (the joint must not be the weak link). The second option is a hydrostatic test with a duration of 1 hour with no leakage or separation of joints.

The hydrostatic test pressure depends on the material of construction. For thermoplastics, it is calculated based on the short-term (about 1 minute) and long-term (50 years) burst strengths. These are interpolated by ASME B31.3, Eq. (27) in a manner appropriate to evaluation of thermoplastic creep data to arrive at a 1-hour strength. This is used to test the pipe. Prior rules that used an arbitrary factor of three times the manufacturer's rated pressure did not work, because quite often the pipe would not satisfy this test, so the bond could not be tested. The factor for RTR and RPM remains at three times the manufacturer's allowable pressure for the components being joined. ASME B31.3, Eq. (27) is

$$P_T = 0.80 \bar{T} \left(\frac{S_S + S_H}{D - \bar{T}} \right) \quad (15.2)$$

where

D = outside diameter of pipe

P_T = hydrostatic test pressure

S_H = mean long-term hydrostatic design strength (LTHS) in accordance with ASTM D2837; twice the 23°C (73°F) HDB design stress from Table B-1 if listed, otherwise from manufacturer's data

S_s = mean short-term burst stress in accordance with ASTM D1599; from Table B-1 if listed, otherwise from manufacturer's data

$\bar{T}$ = nominal thickness of pipe

To qualify the BPS, at least one of each joint type covered by the BPS must be included in the test(s). With respect to size, if the largest joint is 100 mm (NPS 4) or smaller, the test assembly is required to be the largest size to be joined. If the largest pipe to be joined is greater than 100 mm (NPS 4), the size needs to be 100 mm (NPS 4) or 25% of the largest pipe to be joined, whichever is greater.

The same as for welding, the employer of the bonder is responsible for performing the bonding procedure qualification test, qualifying bonders, and maintaining records of the specifications and test. Under certain circumstances as described in ASME B31.3, including approval of the (owner's) Inspector, use of a BPS qualified by others and bonders qualified by others is permitted. If a bonder or bonding operator has not used a specific bonding process for a period of 6 months, requalification is required.

Again, similar to welding, the bonds that are made are required to be identified with a symbol that indicates which joints are made by which bonder. As an alternative, appropriate records that provide this information may be used instead of a physical mark on each joint.

General requirements are provided in ASME B31.3, para. A328.5 for hot-gas-welded joints in thermoplastic piping, solvent-cemented joints in thermoplastic piping, heat-fusion joints in thermoplastic piping, electrofusion joints in thermoplastic piping, adhesive joints in RTR and RPM piping, and butt-and-wrapped joints in RTR and RPM piping. Descriptions of some bonding processes are provided below in Insert 15.1.

Insert 15.1 Bonding Processes

Thermoplastic Piping

A thermosetting plastic material is one that is polymerized (cured) by application of heat or chemical means and is thus changed into an infusible or insoluble final product. Fiberglass pipe is an example of a reinforced thermosetting resin (RTR) pipe. Unlike thermoplastic materials, they cannot be readily reshaped by heating them to their melting range once they are cured. The thermosetting plastic products will usually degrade once they are heated to such temperatures. The two types of thermosetting resin for piping are (1) RTR, which is composed of a fibrous reinforcement material such as glass or carbon fiber in a thermosetting plastic resin such as epoxy, polyester, furan, or phenolic, and (2) reinforced plastic mortar (RPM), which also includes a filler material such as sand. Bonding processes for thermoplastic pipe include adhesive joints and various types of butt-and-wrap joints.

Adhesive Joints

Principles of Operation

Adhesive joining for RTR piping uses adhesive to join the pipes (Fig. 15.1). The pipe joint usually has a bell-and-spigot or a tapered-joint configuration. Depending on the adhesive used and the piping materials, the joint may require curing by one of several methods. These include

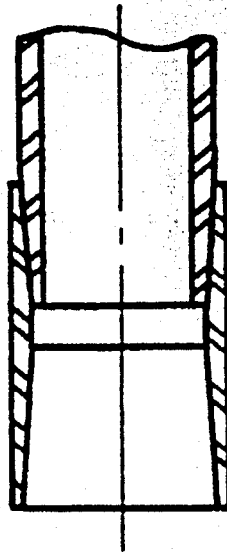


FIG. 15.1
FULLY TAPERED THERMOSETTING ADHESIVE JOINT (ASME B31.3, FIG. A328.5.6)

using chemical heat wraps, applying heat to the pipe joint by other means, or allowing the joint to cure naturally.

Process Limitations

The application of the adhesive to the surfaces to be joined and subsequent assembly of the surfaces should result in a continuous bond. These joints require the pipe to overlap in order to provide a sufficient area of glued surfaces. As a result, in assembling a piping system, the piping may need to have sufficient flexibility for the final pipe joint or joints to be made. Strength of the glued joint may be less than that of the pipe, although the strength of a properly made tapered joint can be as high as that of the pipe.

Butt-and-Wrap Joints

Principles of Operation

Butt-and-wrap pipe joints may be applied to a variety of RTR piping (Fig. 15.2). This joining technique can be applied to pipe joints by butting two pipes end to end or where the joints have a bell-and-spigot arrangement. In either case, layers of resin-impregnated glass fiber cloth (or other reinforcing fiber consistent with the pipe) overwrap the pipe joint area. The cloths are applied in layers on the joint, building it up. Ideally, the finished joint should be capable of withstanding internal pressure, longitudinal force, and bending moments.

Cuts of the pipe should always be sealed so the fiber reinforcement can be protected from contacting the service fluid. Branch connections may be made using similar hand layup techniques.

Process Limitations

The strength of the pipe joint may be designed such that it equals or exceeds that of the pipe. However, the joints can also be a weak link in the system, with butt-and-wrap strength (using

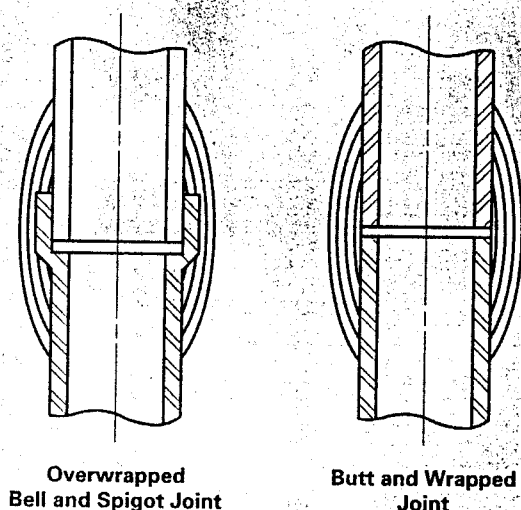


FIG. 15.2
THERMOSETTING WRAPPED JOINTS (ASME B31.3, FIG. A328.5.7)

square-ended pipes butted together) is sometimes considered to be one-half the strength of the pipe.

Thermoplastic Piping

Thermoplastic piping is typically unreinforced and composed of thermoplastic material such as polyethylene (PE), high-density polyethylene (HDPE), polypropylene (PP), ethylene-chlorotrifluoroethylene (ECTFE), ethylene-tetrafluoroethylene (ETFE), poly(vinyl chloride) (PVC), chlorinated poly(vinyl chloride) (CPVC), acrylonitrile-butadiene-styrene (ABS), and poly(vinylidene fluoride) (PVDF). These materials soften when heated and can be shaped and fused in the heated state and cooled without degradation of the material. As a result, fusion bonding of thermoplastic is a possible and often preferred option for bonding.

Solvent-Cement Joining

Principles of Operation

Solvent-cement joints are used on some common types of thermoplastic piping materials, including PVC, CPVC, and ABS (Fig. 15.3). Joining methods are covered by ASTM standards. Usually, the pipe and socket fitting to be joined have a slight interference fit. Proper joining relies on softening and fusing of the pipe and fitting material and should result in a bond that is stronger than the pipe. Both parts of the joint to be made are first cleaned by wiping with an appropriate cleaning agent. The joint to be made is prepared by applying a suitable primer before the solvent cement is applied. The primers are described in an ASTM specification that applies to the material being solvent-cemented. Primer and cement are applied to both surfaces to be joined according to recommended procedures. A continuous bond must be produced, with a small bead of excess cement appearing on the outer limit of the joint, when the fully wetted male end is inserted into the fully wetted female socket.

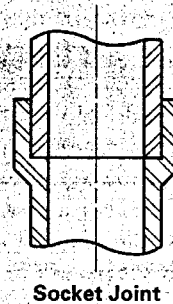


FIG. 15.3

THERMOPLASTIC SOLVENT-CEMENTED JOINT (ASME B31.3, FIG. A328.5.3)

Process Limitations

Some materials, such as PP, cannot be suitably joined using solvent-cement techniques. Further, a good solvent-cement joint must have a slight interference fit between the outside diameter of the pipe and the inside diameter of the mating socket. The diametrical clearance between the pipe and the entrance of the fitting socket should never exceed 0.04 in. With a large gap, heavier cement, which has a greater film thickness to bridge the gap, is used. However, such joints can end up relying on the shear strength of the glue rather than a bonding of the mating plastic parts, which is significantly weaker than proper solvent-cement joints with interference fits.

Hot Gas Welding

Principles of Operation

Hot gas welding is used exclusively for thermoplastic materials; it is not suitable for thermosetting resin materials (Fig. 15.4). It employs hot air or inert gas to melt the base material and tip of the welding rod. The melted welding rod fuses with the base material to form a weld joint.

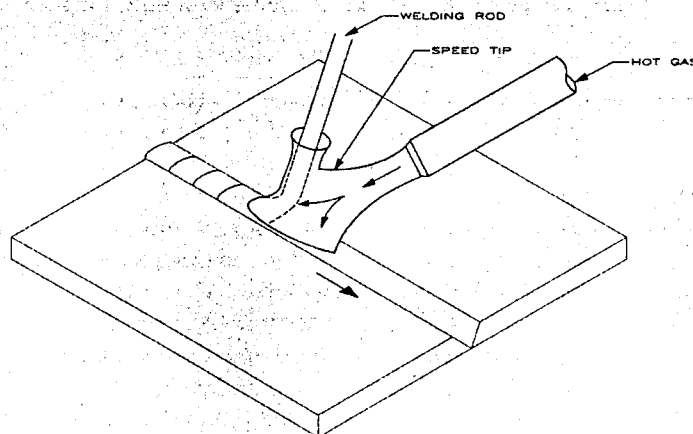


FIG. 15.4

HOT GAS WELDING

Although it is common practice to use partial-penetration welds, it is essential to use full-penetration welds to properly develop the joint strength, which will be weaker than the base material in any case.

Extrusion welding is similar to hot gas welding except that the weldment is extruded from a screw-driven nozzle. The equipment required for extrusion welding is much larger than that for hot gas welding. The nozzle must be large enough to hold an extrusion screw and a motor must be attached to drive the screw. As a result, extrusion welding requires a great deal of space to conduct the welding; approximately 3 ft of clearance in all directions is the usual requirement.

The advantage of extrusion welding over hot gas welding is the application of a thick weld bead in a single pass (usually the equivalent to 5 to 12 passes of hot gas welding). The result is a substantially lower degree of residual stress and lesser effects of oxidation.

Process Limitations

Proper joining by this technique is more difficult for some materials, such as PVC, and should not be used for those materials except in special circumstances. In those circumstances, special precautions, such as use of an inert gas such as nitrogen for the hot gas, may be necessary.

Due to a number of factors, including imperfections, residual stress, and oxidation, the strength of hot-gas-welded joints and extrusion-welded joints is lower than that of the base material. According to German Standard DVS 2203, the long-term strength of a hot-gas-welded joint is 40% of that of the base material for HDPE, PP, PVC, and PVDF. The long-term strength of extrusion welds is, according to the same standard, 60% of that of the base material for HDPE and PP. The relatively common practice of making partial-penetration welds using hot gas welding techniques results in substantial further reduction of joint strength.

Heat Fusion

Principles of Operation

Heat fusion relies on using a suitable heating element to bring the mating surfaces up to the melting temperature and then forcing the two surfaces together to cause them to flow and fuse. It is the most common technique for bonding all polyolefins, such as HDPE and PE, and fluoropolymers, such as PVDF, ECTFE, and ETFE. Available techniques include heat-element butt fusion, heat-element socket fusion, electrical resistance fusion, and branch fabrication techniques such as sidewall fusion.

In heat-element butt fusion, the mating pipe ends are forced against a heating element with a specific pressure to bring the mating surfaces up to melting temperature (Fig. 15.5). The heating element is removed and then the mating pipe ends are forced against each other with a specified pressure for a specified duration of time.

In heat-element socket fusion, a heating element is inserted into the socket and the pipe end is inserted into a heating element (Fig. 15.6). After the mating surfaces are brought up to melting temperature, the pipe is inserted into the socket, again with a specified force and duration. There is an interference fit and the pipe fuses to the socket.

In electric resistance socket fusion, electric heating element embedded in the socket fitting wall brings the surfaces up to melting temperature, again accomplishing the same objective (Fig. 15.7).

In sidewall (saddle) fusion techniques (Fig. 15.7), a hole is cut in the sidewall of the pipe with a bevel and the branch pipe is prepared with a mating bevel. A heating element is used to heat the end of the branch and mating surfaces in the run pipe. Joining is similar to other heat fusion techniques. Other heat fusion techniques based on the same principles are also used in branch and reinforcement bonding.

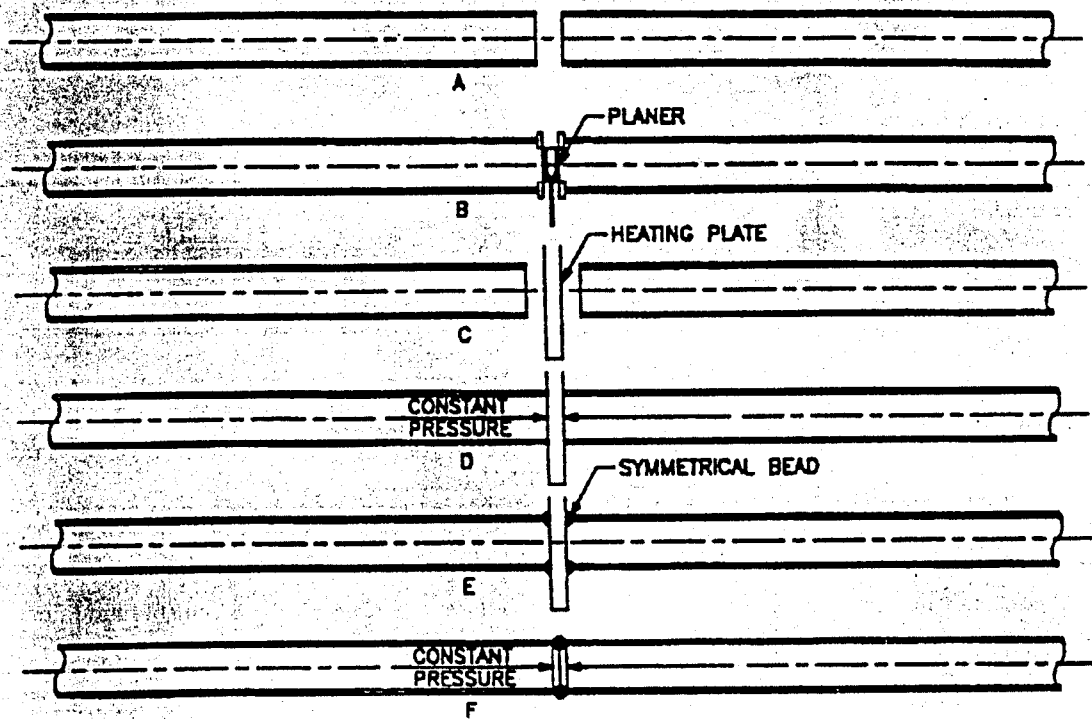


FIG. 15.5
STEPS FOR HEAT-ELEMENT BUTT FUSION (COURTESY OF CHRIS ZIU)

Process Limitations

Heat fusion provides superior joints to hot gas and extrusion welding. However, special equipment is required to heat the surfaces and draw the mating components together with the appropriate force (pressure on the surfaces being bonded). The equipment can be large, so sufficient working space is required. Also, the mating parts must be physically separated to provide space for the heating element and then pressed together. As such, sufficient pipe flexibility or other means must be provided to make the final joint in piping systems between anchors or other fixed points.

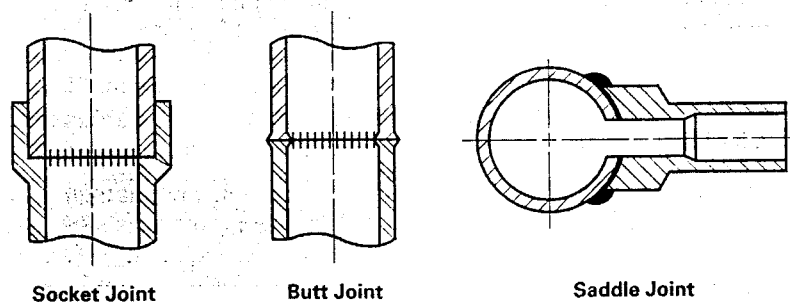


FIG. 15.6
THERMOPLASTIC HEAT FUSION JOINTS (ASME B31.3, FIG. A328.5.4)

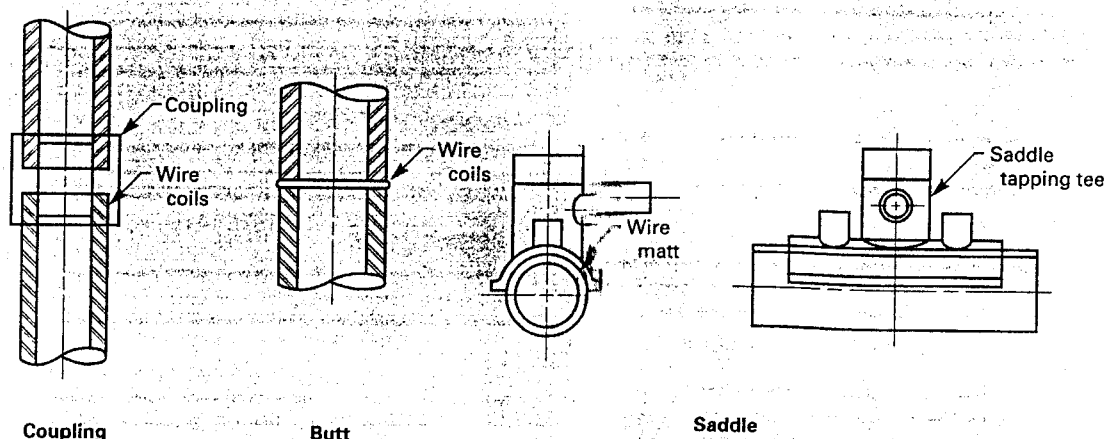


FIG. 15.7
THERMOPLASTIC ELECTROFUSION JOINTS (ASME B31.3, FIG. A328.5.5)

According to German Standard DVS 2203, the long-term strength of a properly made heat fusion butt joint is 80% of that of the base material for HDPE and PP and 60% of that for PVC and PVDF.

15.9 OTHER FABRICATION, ASSEMBLY, AND ERECTION REQUIREMENTS

Requirements for fabrication of piping lined with nonmetals (para. A329), joining of nonplastic piping (e.g., borosilicate glass) (para. A334), and assembly and erection of nonmetallic piping (para. A335) are provided in ASME B31.3. These are not comprehensive requirements, but instead they address some specific considerations that apply to nonmetallic piping in addition to the base Code requirements.

Welding of metallic piping lined with nonmetals generally follows the base Code requirements for welding of metallic piping. Precautions are provided in ASME B31.3, para. A329. For example, precautions are required to avoid damage to the nonmetallic lining. If such damage occurs, it must be repaired. Further, qualification of a welder or welding operator for a WPS for lined pipe is specific to the lining; a different qualification test is required for each lining material. This, of course, only applies to pipe that has already been lined, not welding of piping prior to lining it.

15.10 EXAMINATION AND TESTING

The nondestructive examination techniques for nonmetallic piping are not nearly as well developed as for metallic piping. As a result, the techniques that are used are visual and in-process examination. Specific examination requirements for nonmetallic piping are included with the discussion of examination for metallic piping in Section 13.6.

Leak testing requirements for nonmetallic piping are covered in Section 14.9.

CATEGORY M PIPING

16.1 WHEN TO USE THE RULES FOR CATEGORY M FLUID SERVICE

The rules in Chapter VIII of ASME B31.3 are used when the owner designates a piping system to be in Category M Fluid Service. The owner is guided in the classification for the piping system by the definition of Category M Fluid Service in Chapter I of ASME B31.3. This definition is the Code rule relative to classification. A guide to the application of these rules is provided in ASME B31.3, Appendix M, which contains a flow chart to assist the owner in classifying fluid services.

The definition of Category M Fluid Service is as follows. Note that for purposes of emphasis, it has been broken into subparts, all of which must be satisfied for the service to meet the definition of Category M (ASME B31.3, para. 300.2):

“a fluid service
in which the potential for personnel exposure is judged to be significant
and in which a single exposure of a very small quantity
of a toxic fluid,
caused by leakage,
can produce serious irreversible harm to persons on breathing or bodily contact,
even when prompt restorative measures are taken.”

Note that the Code considers many very hazardous fluid services to be Normal Fluid Service. The design and construction rules for Normal Fluid Service are suitable for hazardous services. Category M provides a higher level. If higher integrity piping is desired by the owner, even though the fluid does not meet the definition of Category M, the owner can still specify the additional design, construction, examination, and testing requirements that are provided in Chapter VIII. Hydrofluoric acid is one example of a fluid for which many owners specify more stringent requirements than are provided in the Code for Normal Fluid Service, although it would be considered Normal Fluid Service.

A key part of the definition that is often neglected are the words “in which the potential for personnel exposure is judged to be significant.” Thus, a piping system for which the exposure of personnel to the fluid is judged to be insignificant would not satisfy the definition of Category M. Two different piping systems containing the same fluid may be judged to be in different fluid services, one Normal and the other Category M. Thus, there cannot be a list of chemical compounds that states the fluid service irrespective of the conditions of installation.

Another consideration that is often missed is that the definition requires the fluid to be toxic. Thus, simply dangerous fluids are not necessarily Category M. Further, exposure to a very small quantity is required to cause serious irreversible harm. Thus, H_2S would not be considered to be Category M.

Examples of systems for which personnel exposure may be judged to be insignificant include double-containment piping with leak detection and piping systems that people may not be exposed to by virtue of isolation or other means of personnel protection. As such, chemicals such as phosgene and methyl isocyanate, which may be classified as Category M in a single-wall piping system, may be classified as Normal Fluid Service in a double-containment piping system.

Another key consideration is that only the owner has the right and responsibility to select the fluid service.

16.2 ORGANIZATION OF CHAPTER VIII OF ASME B31.3

Chapter VIII follows the same paragraph numbering as the base Code, Chapters I through VI, except that in Chapter VIII the paragraphs start with M for metallic piping and MA for nonmetallic piping.

To determine the rules for piping systems in Category M Fluid Service, one simply refers to Chapter VIII. If rules elsewhere in the Code apply, they are referred to in Chapter VIII. In general, Chapter VIII refers to the base Code for metallic piping and to Chapter VII for nonmetallic piping.

Chapter VIII makes no provision for severe cyclic conditions, as stated in para. M300(e). Severe cyclic conditions should be avoided by design in these systems. This simply requires, for systems with greater than 7000 equivalent cycles, the inclusion of enough flexibility to reduce the thermal expansion stress range to 80% or less of the allowable value. If, for some reason, this is not feasible, ASME B31.3 requires that the engineering design specify any necessary provisions.

16.3 OVERVIEW OF METALLIC RULES

The rules for metallics prohibit the use of certain components considered to have lower integrity, and they require additional design considerations, additional examination, and additional testing. The measures are intended to result in a piping system that is less likely to leak.

The following are highlights of some of these requirements. It is not an all-inclusive list; refer to ASME B31.3 for the complete requirements.

- The presumptive degree of ambient cooling (e.g., 5% for uninsulated pipe) provided in the base Code is not permitted. Instead, the design metal temperature, if less than the fluid temperature, must be substantiated by heat transfer calculations confirmed by tests or by experimental measures.
- Increased pressure and/or temperature for short-term variations (allowances for variations) are not permitted.
- Lower integrity piping and components are prohibited.
- Special consideration is required for prevention of valve stem leakage to the environment. Specific requirements are provided for valve bonnet or cover plate closures (para. M307.2).
- Single-welded slip-on, expanded-joint, and threaded (with certain exceptions) joint flanges are prohibited (para. M308.2).
- Expanded joints are prohibited (para. M313).
- Additional limitations are provided for threaded joints (para. M314).
- Joints such as caulked (para. M316), soldered and brazed (para. M317), and bell-type joints (para. M318) are prohibited.
- Pipe supports are required to be constructed of listed materials (para. M321).
- Specific provisions are given for instrument piping [e.g., limiting tubing to 15-mm (5/8-in.)-diameter maximum, accessible block valves required to be available to isolate the instrument piping from pipeline] (para. M322.3).

- The design pressure is not permitted to be exceeded by more than 10% during pressure relief (para. M322.6.3).
- The low-stress exemption from impact testing is not permitted (para. M323.2).
- Cast iron and ductile iron are not permitted for pressure-containing parts, and lead and tin may only be used as linings (para. M322.4.2).
- Less stringent heat treatments than required in ASME B31.3 Table 331.1.1 are not permitted (para. M331).
- Additional examination is required. Although the acceptance criteria of the base Code are applicable, the amount of radiography is increased from 5% to 20% and random visual examination is generally increased to 100% visual examination. See Section 13.4 for specific requirements.
- In addition to the testing required for Normal Fluid Service, an additional sensitive leak test is required to ensure that the piping is free of small leaks. See Section 14.6 for specific requirements.

16.4 OVERVIEW OF NONMETALLIC RULES

The rules for nonmetallics prohibit the use of certain components considered to have lower integrity, and they require additional design considerations, additional examination, and additional testing. The measures are intended to result in a piping system that is less likely to leak. These rules generally refer to Chapter VII.

The following are highlights of some of these requirements. It is not an all-inclusive list; refer to ASME B31.3 for the complete requirements.

- The piping is not permitted to exceed the design pressure under any circumstance, including pressure relief conditions; this is the same as the rule for nonmetallic piping (para. MA302.2.4).
- Nonmetallic fabricated branch connections are prohibited (para. MA306.5).
- Nonmetallic valves and specialty components are prohibited (para. MA307).
- Hot-gas-welded, heat-fusion, solvent-cemented, and adhesive-bonded joints are not permitted, except in linings (para. MA311.2).
- Expanded, nonmetallic threaded, and caulked joints are prohibited (paras. MA313, MA314, MA316).
- Thermoplastics and reinforced plastic mortar are permitted only as linings and, for thermoplastics, gaskets (para. MA323.4.2).
- The examination and testing rules of Chapter VII apply (i.e., more stringent rules are not added for Category M Fluid Service) (Chapter VIII, Part 20); revisions to this, to require more stringent examination and testing, are being considered.

16.5 GENERAL COMMENTS

The rules of ASME B31.3, Chapter VIII are intended to provide greater assurance of leak-tightness. An example is the requirement that a sensitive-leak test be performed in addition to the standard leak test (e.g., hydrotest). The ASME B31.3 Section Committee does not generally spend a great deal of time on this section. In this author's opinion, the rules are not sufficient for the extremely dangerous chemicals that fit the definition of Category M. Additional precautions should be considered, such as 100% radiography or double-containment (which could be considered to remove the piping system from Category M).

HIGH-PRESSURE PIPING

17.1 SCOPE OF CHAPTER IX OF ASME B31.3

Chapter IX of ASME B31.3 only applies when the owner specifies its use. It applies to piping in High-Pressure Fluid Service. Note that the definition of High-Pressure Fluid Service simply requires that the owner specify use of Chapter IX. Some guidance is provided in K300 (a), which states that "High pressure is considered to be pressure in excess of that allowed by the ASME B16.5 PN 420 (Class 2500) rating for the specified design temperature and material group." This is not a requirement, and the base Code may be satisfactorily used at pressures higher than ASME B16.5, PN 420 (Class 2500). However, the base Code rules become increasingly conservative and, in fact, impossible to use as the pressure approaches the allowable stress (including quality factors). See Section 17.3 for a discussion of pressure design of straight pipe. By the same token, Chapter IX may be used at lower pressures; it has been used with high-strength steels with pressures as low as 28,000 kPa (5,000 psi).

The rules provide a combination of considerations. Although reduced wall thicknesses and provisions that are specific to the needs of high pressure (e.g., not including thread depth as an allowance under specific conditions) are provided, additional material toughness, analysis during design, inspection, and testing are required. For background on these rules see Sims (1986).

Chapter IX makes no provision for Category D and M Fluid Service classifications. Considering the pressure, Category D is not applicable. There are no provisions for Category M Fluid Service. The concept of severe cyclic conditions is not used in Chapter IX; however, the rules are more stringent than the base Code rules for severe cyclic service. Therefore, the additional consideration of severe cyclic service is unnecessary.

The criteria consider limit load failure and fatigue. Elevated-temperature creep effects are not included; thus, the use of Chapter IX is limited to temperatures below the creep regime for the materials of construction. Nonmetallic parts, other than gaskets and packing, are not permitted.

This section provides an overview of Chapter IX. See the ASME B31.3 Code for specific requirements.

ASME BPVC, Section VIII, Division 3, *Pressure Vessels, Alternative Rules for Construction of High Pressure Vessels*, was completed after Chapter IX. As a result, there are various references to the requirements of ASME, BPVC, Section VIII, Division 2 that have been changed to either include Division 3 as an acceptable alternative or to simply require the Division 3 rules rather than the Division 2 rules. The Division 3 rules are generally more applicable, because they were developed for high-pressure equipment.

17.2 ORGANIZATION OF CHAPTER IX OF ASME B31.3

Chapter IX follows the same paragraph numbering as the base Code, Chapters I through VI, except that the paragraphs in Chapter IX start with K. To determine the rules for high-pressure piping

systems, simply refer to Chapter IX. If rules elsewhere in the Code apply, they are referenced in Chapter IX.

17.3 PRESSURE DESIGN OF HIGH-PRESSURE PIPING

Beyond a certain pressure, it is not possible to design piping in accordance with the basic wall thickness equation in the base Code, Eq. (3a). When Y is assumed equal to zero (note that Y approaches zero as the inside diameter approaches zero; see definition of Y in para. 304.1.1), the required wall thickness is equal to the outside radius of the pipe when the pressure is equal to the allowable stress times the quality factor (SE ; see Section 4.2). However, heavy wall pipe has substantial pressure capacity beyond the point where the circumferential stress at the bore reaches yield. Further, for high internal pressure, the radial stresses due to the surface traction of internal pressure significantly affect yielding of the material on the inside of the pipe, in accord with the Von Mises or the Tresca yield theory.

Equation (34) in Chapter IX provides the required thickness for high-pressure straight pipe. Rather than being based on maximum circumferential stress, as in the base Code, it is based on limit load pressure. It provides a margin of two against through-thickness yielding when used in conjunction with the allowable stresses in Appendix K. The equations for calculation of required wall thickness are [Eqs. (35a) and (35b) in the Code provide the allowable pressure based on available thickness]¹⁸

$$t_m = t + c \quad (17.1)$$

$$t = \frac{D - 2c_o}{2} \left[1 - \exp \left(\frac{-1.155P}{S} \right) \right] \quad (17.2)$$

$$t = \frac{d + 2c_I}{2} \left[\exp \left(\frac{1.155P}{S} \right) - 1 \right] \quad (17.3)$$

where

D = outside diameter of the pipe (maximum permitted by specifications)

S = allowable stress from Table K-1 of ASME B31.3

$c = c_I + c_o$

c_I = sum of internal mechanical plus corrosion and erosion allowances

c_o = sum of external mechanical plus corrosion and erosion allowances including, except under certain conditions, thread depth

d = inside diameter of the pipe (maximum permitted by specifications)

t = pressure design wall thickness

t_m = minimum required wall thickness including mechanical, corrosion, and erosion allowances

No quality factor is included, because the minimum permitted quality factor in Chapter IX is 1.0. When the mechanical allowances are not specified to be internal or external, they are to be assumed to be internal.

The external thread on pipe can be neglected in the mechanical allowances when, for straight-threaded connections, the following are satisfied:

- The thread depth does not exceed 20% of the wall thickness
- The ratio of outside to inside diameter, D/d , is greater than 1.1
- The internally threaded attachment provides adequate reinforcement

¹⁸Equations (17.1) through (17.3) correspond to ASME B31.3 Eqs. (33), (34a), and (34b), respectively.

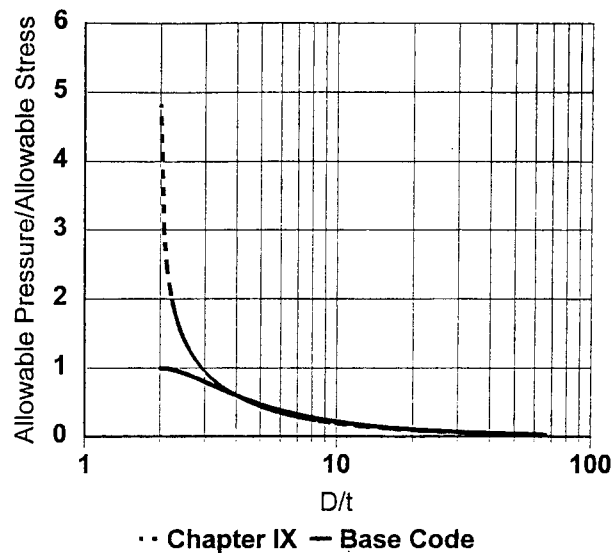


FIG. 17.1
COMPARISON OF WALL THICKNESS REQUIRED BY BASE CODE PRESSURE DESIGN
RULES [ASME B31.3, EQ. (3A)] VERSUS THOSE OF ASME B31.3, CHAPTER IX [ASME
B31.3, EQS. (34A), (34B)]

- The thread plus the undercut area, if any, does not extend beyond the reinforcement for a distance more than the nominal wall thickness of the pipe

This recognizes that the threaded fitting, typically a threaded flange, to which the pipe is attached can effectively reinforce the pipe for internal pressure under certain conditions.

Figure 17.1 provides a comparison of the base Code pressure design rules with those of Chapter IX. It provides the ratio of allowable pressure to allowable stress as a function of the D/t ratio. It can be observed that the base Code provides a higher allowable pressure for D/t ratios greater than about four. However, because the allowable stress in Chapter IX is based on yield strength and not tensile strength, there will be a greater advantage with respect to wall thickness if Chapter IX is used for steels with high yield-to-tensile strength ratios. This is because the base Code allowable stress would be controlled by one-third the tensile strength.

Provisions similar to, but more limited than those of the base Code are provided for design of specific types of components, and listed components are accepted. The listed components are given in Table K3261.

The comparable paragraph to 304.7.2 is K304.7.2, Unlisted Components and Elements. The options for proving the pressure design of unlisted components are more limited than in the base Code. Only extensive service experience or performance testing is permitted for qualifying components. The performance testing must substantiate both the static pressure design and the fatigue life.

The option of permitting finite-element analysis for qualification of components is under consideration.

Short-term variations above the design pressure and temperature are not permitted, except for accumulation during pressure-relieving events.

17.4 EXTERNAL PRESSURE

Buckling due to external pressure is not generally a concern for high-pressure piping. However, the limit pressure determined using Eqs. (34a), (34b), (35a), and (35b) of ASME B31.3 is not always

conservative for external pressure on straight pipe, considering buckling, and under rare circumstances when external pressure does not cause axial compression of the pipe. In the latter case, collapse can be predicted by triaxial stress states and yield theory.

When $D/t < 3.33$ and at least one end of the pipe is exposed to full external pressure, which produces compressive axial stress, Eqs. (34a), (34b), (35a), and (35b) of ASME B31.3 for internal pressure can be used. In all other circumstances, the base Code rules for external pressure design of straight pipe are used.

17.5 DESIGN FOR SUSTAINED AND OCCASIONAL LOADS

The criterion for sustained loads is the same as in the base Code. The longitudinal stress due to pressure, weight, and other sustained loads must be less than S_h .

The criterion for occasional loads is more conservative than in the base Code. A factor of 1.2 times the allowable stress (the same as ASME BPVC, Section VIII, Division 1) is used, rather than a factor of 1.33.

17.6 DESIGN FOR THERMAL EXPANSION AND FATIGUE

Flexibility analysis is conducted similar to the base Code. However, only the more conservative equation for S_A , which assumes that $S_L = S_h$, is used [Eq. (32) of ASME B31.3]. The allowable stresses from Appendix K are used, rather than the allowable stresses from Appendix A. These will be higher when tensile strength controls the allowable stresses in Appendix A.

Chapter IX (para. 304.8) requires a detailed fatigue analysis in addition to the flexibility analysis provided in the base Code. For this analysis, the allowable stress amplitude from the fatigue curves in ASME BPVC, Section VIII, Division 2, Appendix 5 are used. Note that the rules of ASME BPVC, Section VIII, Division 3 may be more appropriate, but are not yet listed as an alternative or requirement.

The fatigue analysis should include the typically high strains due to internal pressure, in particular those at the bore of the pipe, as well as the stress due to thermal expansion. Since the calculated stress is compared to the polished bar fatigue curve rather than a butt-welded pipe fatigue curve, the stresses calculated in accordance with the flexibility analysis rules of ASME B31.3 in most cases need to be multiplied by a factor of two. All components with stress intensification factors greater than one and pipe at girth welds should have the stresses multiplied by two. The reason for doing this is explained in Section 8.4.

The requirements for fatigue analysis pose some challenging problems to the Designer. Some of these are highlighted below.

(a) The Designer must consider both pressure and displacement cycles. Thus, a load histogram and a procedure such as rainflow counting is appropriate for determining the variety of stress ranges and number of cycles at each stress range for which the piping must be designed. For example, there may be normal pressure cycles, which may be more numerous than thermal displacement cycles, and also pressure pulsations. There will be some number of cycles at a maximum stress range, which could be the normal pressure plus pulsation plus displacement stress, as well as many cycles with smaller stress ranges to consider.

(b) Fatigue analysis per ASME BPVC, Section VIII, Division 2 deals with stresses at a point, whereas the flexibility analysis rules of ASME B31.3 do not inform the Designer as to the location and direction of stresses that are calculated. For example, the stresses in an elbow due to in-plane bending are through-wall bending and greatest in the circumferential direction and should be directly added to the pressure stress, whereas the stresses due to bending in straight pipe are gross bending on the pipe section and longitudinal and should be combined with the pressure stress to determine Tresca stress intensity.

(c) The stresses due to internal pressure vary through the thickness and this should be considered in determining what pressure stress to combine with the deflection stresses (e.g., bore stress should be considered for an elbow and the outside stress should be considered for a threaded joint).

(d) The radial stress through the wall of the pipe [compressive and equal to internal pressure on the inside surface and generally considered to be zero on the outside surface (for a pipe under internal pressure)] can be a significant component of the Tresca stress intensity.

All of these mean that a Designer tackling a fatigue analysis for Chapter IX piping should be an expert, intimately familiar with the stress distributions in thick-wall piping components.

ASME B31.3 provides the equation below for the stress intensity on the inside surface of straight pipe due to internal pressure. Note that this is not likely to be the controlling location in the system. As such, its usefulness is limited. Also note that this equation is only considered to be valid if the calculated stress per this equation does not exceed three times the allowable stress from Table K-1 (this is two times the yield strength). If this stress is exceeded, an inelastic analysis is required to determine the strain range due to internal pressure. This provision assumes that the pipe will shake down to elastic action if the stress intensity range is less than two times the yield strength. The equation is¹⁹

$$S = \frac{PD^2}{2(T-c)[D-(T-c)]} \quad (17.4)$$

where

D = outside diameter

P = pressure [definition in nomenclature, Appendix J, is not correct, because a variety of pressure conditions may be considered]

S = stress intensity on the inside surface of the pipe [definition in nomenclature, Appendix J, is not correct]

c = sum of mechanical allowances

17.7 MATERIALS

The allowable stress is provided in Appendix K, Table K-1. It is two-thirds of the material yield strength, to be consistent with the pressure design equation. For solution-heat-treated austenitic stainless steels and certain nickel alloys with similar stress-strain behavior, the minimum of two-thirds the specified minimum yield strength and 90% of the yield strength at temperature is used. Similar to the base Code, this is because the material has significant strength beyond the nominal 0.2% offset yield stress. As in the base Code, these higher stress values provided for stainless steel and similar materials are not recommended for flanges and similar components where slight deformation can cause leakage or other malfunction.

As in the base Code (as of the 2000 Addenda), materials may be used above the maximum temperature for which allowable stresses are provided. Previously, this had been prohibited because of concerns such as temper embrittlement. Per para. K323.2.1, there must be no prohibition in the Code, the Designer must verify the serviceability of the material, and the temperature must be below that at which creep properties would govern the allowable stress that would be determined using the base Code allowable stress criteria.

The impact test requirements are a very important part of the material requirements in Chapter IX for high-pressure piping. Essentially all high-pressure piping materials and welds must be impact-tested

¹⁹ASME B31.1, Eq. (37).

to determine that they have sufficient notch toughness for any temperature condition at which stresses exceed 41 MPa (6 ksi). Impact test requirements are provided in para. K323.3. Transverse specimens are required, unless the component size or shape does not permit cutting transverse (to the longitudinal axis of the pipe, or the maximum direction of elongation during rolling, or the direction of major working during forging) specimens. In that case, longitudinal specimens may be used; however, the required impact energy absorption is higher.

For materials, at least one set of impact tests per lot is required. For impact tests on welds, significantly more testing is required than in the base Code. Whereas the base Code only requires impact testing as part of the weld procedure qualification, Chapter IX requires impact testing for each welder and welding procedure, type of electrode, filler metal, and flux to be used. For tests on welds, separate tests are not required for each lot of material. Test specimens for the welds and heat-affected zones are required.

The minimum permissible temperature for a material is the minimum temperature at which an impact test that satisfies the Code requirements was performed. The only exception to this is the 41-MPa (6-ksi) exemption, but that exemption may only be used down to -46°C (-50°F). Impact testing, regardless of stress, is required for use at temperatures below that temperature.

17.8 FABRICATION

Requirements for qualification of welding procedures and welders or welding operators follow ASME BPVC, Section IX. However, there are additional requirements and limitations, including the following:

- Qualification test weldments are required to be made using the same specification and type or grade of base metal(s) and the same specification and classification of filler metal(s) as will be used in production welding.
- Qualification by testing weldments made with materials with the same *P*-Number is not permitted.
- Test weldments are required to be subjected to essentially the same heat treatment, including cooling rate and cumulative time at temperature, as the production welds.
- Mechanical testing is required for all performance qualification tests; testing by radiography is not permitted.

Branch connection fittings are required to be designed to permit 100% radiography. Acceptable details are provided in Fig. K328.5.4 of ASME B31.3.

General fabrication requirements pertaining to end preparation, alignment, welding, preheat, and postweld heat treatment are provided. They are largely similar to those in the base Code, with some variations. For example, the recommended preheat temperature per Table 330.1.1 in the base Code is required for Chapter IX.

17.9 EXAMINATION

In general, Chapter IX requires 100% examination. This includes 100% visual examination of materials and components; fabrication; threaded, bolted, and other joints; piping erection; and pressure-containing threads.

All girth, longitudinal, and branch connection welds are required to be 100% examined by radiography. Ultrasonic and in-process examination are not acceptable alternatives. The acceptance criteria for welds are provided in Table K341.3.2, reproduced here as Table 17.1.

TABLE 17.1
ACCEPTANCE CRITERIA FOR WELDS

Criteria (A–E) for Types of Welds, and for Required Examination Methods [Note (1)]						
Type of Imperfection	Type of Weld					
	Methods		Girth Groove	Longitudinal Groove [Note (2)]	Fillet [Note (3)]	Branch Connection [Note (4)]
	Visual	100% Radiography				
Crack	X	X	A	A	A	A
Lack of fusion	X	X	A	A	A	A
Incomplete penetration	X	X	A	A	A	A
Internal porosity	...	X	B	B	NA	B
Slag inclusion or elongated indication	...	X	C	C	NA	C
Undercutting	X	X	A	A	A	A
Surface porosity or exposed slag inclusion	X	...	A	A	A	A
Concave root surface (suck-up)	X	X	D	D	NA	D
Surface Finish	X	...	E	E	E	E
Reinforcement or internal protrusion	X	...	F	F	F	F

GENERAL NOTE: X = required examination; NA = not applicable; ... = not required.

Criterion Value Notes

Criterion		Acceptable Value Limits [Note (5)]	
Symbol	Measure		
A	Extent of imperfection	Zero (no evident imperfection)	
B	Size and distribution of internal porosity	See BPV Code, Section VIII, Division 1, Appendix 4	
C	Slag inclusion or elongated indication		
	Individual length	$\leq \bar{T}_W/4$ and ≤ 4 mm ($5/32$ in.)	
	Individual width	$\leq \bar{T}_W/4$ and ≤ 2.5 mm ($3/32$ in.)	
	Cumulative length	$\leq \bar{T}_W$ in any 12 $\bar{T}_W$ weld length	
D	Depth of surface concavity	Total joint thickness including weld reinforcement, $\geq \bar{T}_W$	
E	Surface roughness	≤ 12.5 μm R_a (500 $\mu\text{in.}$ R_a per ASME B46.1)	
F	Height of reinforcement or internal protrusion [Note (6)] in any plane through the weld shall be within the limits of the applicable height value in the tabulation at the right. Weld metal shall be fused with and merge smoothly into the component surface.	Wall Thickness $\bar{T}_W$ mm (in.)	External Weld Reinforcement or Internal Weld Protrusion
		≤ 13 ($1/2$)	1.5 ($1/16$)
		$> 13; \leq 51$ (2)	3 ($1/8$)
		> 51	4 ($5/32$)

NOTES:

- (1) Criteria given are for required examination. More stringent criteria may be specified in the engineering design.
- (2) Longitudinal welds include only those permitted in paras. K302.3.4 and K305. The radiographic criteria shall be met by all welds, including those made in accordance with a standard listed in Table K326.1 or in Appendix K.
- (3) Fillet welds include only those permitted in para. 311.2.5(b).
- (4) Branch connection welds include only those permitted in para. K328.5.4.
- (5) Where two limiting values are given, the lesser measured value governs acceptance. $\bar{T}_W$ is the nominal wall thickness of the thinner of two components joined by a butt weld.
- (6) For groove welds, height is the lesser of the measurements made from the surfaces of the adjacent components. For fillet welds, height is measured from the theoretical throat; internal protrusion does not apply. Required thickness t_m shall not include reinforcement or internal protrusion.

17.10 TESTING

All elements, including all components and welds, of a high-pressure piping system, except for bolting and gaskets used during final system assembly, are required to be subjected to a full-pressure test. This pressure test need not be performed on the installed piping system, but can be done on piping subassemblies prior to erection. The full-pressure test is the base Code hydrotest pressure (1.5 times the design pressure times the temperature correction factor; see Section 14.2), except that there is no limitation on the temperature correction factor. Furthermore, if this test is performed as a pneumatic test, the same full pressure as for a hydrotest is required. However, the test pressure can be reduced, if necessary, to limit the pressure to that at which through-thickness yielding (calculated based on specified minimum yield stress) occurs in any component.

After the piping is assembled, an additional leak test of the installed piping system is required. This is conducted at 110% of the design pressure. However, if the full-pressure test described in the preceding paragraph was performed on the installed piping, the additional leak test is not required.

17.11 RECORDS

Chapter IX contains more substantive requirements for record transfer to the owner and the retention of records than the base Code. For example, the base Code does not contain requirements relative to documentation of the engineering design. However, Chapter IX requires records of the following to be provided to the owner or the (owner's) Inspector:

- The engineering design
- Material certifications
- Procedures used for fabrication, welding, heat treatment, examination, and testing
- Repair of materials including the procedure used for each and location of repairs
- Performance qualifications for welders and welding operators
- Qualification of examination personnel
- Records of examination of pipe and tubing for longitudinal defects as specified in paras. K344.6.4 and K344.8.3

These records are required to be retained by the owner for at least 5 years after they have been received.

BELLOWS EXPANSION JOINTS

18.1 OVERVIEW OF BELLOWS REQUIREMENTS

Prior to 1988, coverage of metallic bellows expansion joints in ASME B31.3 was very limited. Bellows were required to be designed in accordance with the rules for pressure design of unlisted components (par. 304.7.2) and the Standards of the Expansion Joint Manufacturers Association, Inc. (EJMA Standards). Appendix X was added in 1988, and has undergone several significant revisions in subsequent years. It provides specific Code coverage for metallic bellows expansion joints by adopting, with exceptions and additions, a widely used industry standard for expansion joints, the EJMA Standards.

The overall philosophy of Appendix X was to adopt the EJMA Standards with modifications that the Code Committee felt were necessary to make the EJMA Standards into Code requirements suitable for ASME B31.3 piping. This differs somewhat from the approach taken in ASME BPVC, Section VIII, Division 1, Appendix 26 (formerly Appendix BB), Pressure Vessel and Heat Exchanger Expansion Joints, where the design equations from the EJMA Standards were directly incorporated into the Code. The background behind these bellows stress analysis equations and comparison with detailed analysis and test results are provided in Becht (1985), Becht and Skopp (1981a), Osweiler (1989), and Broyles (1994).

The most significant of these modifications to the EJMA Standards were in requirements for pressure design, fatigue design, and hydrotest. It should be noted that both the EJMA Standards and the ASME B31.3 Code are living documents. For example, EJMA revised the pressure design requirements in the Sixth Edition, issued in 1993. This Edition was adopted by ASME B31.3 in the 1994 Addenda. Some of the pressure design requirements in Appendix X could now be dropped, because they are now adequately addressed in the EJMA Standards. However, they have been left in for emphasis of certain considerations.

18.2 RULES FOR PRESSURE DESIGN

It was shown by Becht (1980) and Becht and Skopp (1981b) that the design rules in the Fifth Edition of the EJMA Standards for internal pressure were, in some cases, nonconservative. In fact, it was possible to design a bellows that would fail at a pressure lower than the design pressure. As a result, both ASME BPVC, Section VIII, Division 1 and ASME B31.3 contain an allowable stress basis that differs from the 5th Edition of the EJMA Standards.

The source of the nonconservatism was an inconsistency between the allowable stress basis and empirical test data. Based on test data, the EJMA Standards included a design approach that permitted a pressure-induced bending stress which was significantly greater than the specified minimum yield strength of the material. It permitted a stress as high as 2.86 times the ASME BPVC, Section VIII, Division 1 allowable stress in tension, which could be as high as 1.9 times the minimum yield strength (or 2.6 times for austenitic stainless steel).

The reason that bellows generally do not fail at this high level of stress is that the material, in the as-formed condition, is cold-worked and therefore has a yield strength two or more times higher than the specified minimum yield strength upon which the Code-allowable stress is based. The bellows that were tested in developing the pressure stress limits in the EJMA Standards were all in the as-formed condition, and therefore had the benefit of a higher than specified minimum yield strength. Thus, a high multiple of the Code-allowable stress, which was based on specified minimum yield strength values, was found to be acceptable.

If a bellows is annealed, as is sometimes required for corrosive or high-temperature service, the benefit of the cold work in terms of increased yield strength is no longer available, and the EJMA pressure design equations became nonconservative. Figure 18.1 illustrates how the root in a test bellows bulged outward to failure when the meridional stress due to internal pressure exceeded 1.5 times the actual yield strength in an annealed bellows (Becht and Skopp, 1981b).

The ASME B31.3 approach is to require the meridional membrane plus bending stresses for annealed bellows to be limited to 1.5 times the basic allowable stress in tension. This took care of the known nonconservatism for the case of annealed bellows.

The degree of conservatism for other materials in the as-formed condition was not known and depended on the degree of strengthening that occurs in the forming process. However, the Committee did not want to create excessively restrictive rules that would unduly impact bellows designs that have historically worked. Also, for bellows, a balance must be maintained between design for pressure and deflection. Therefore, no specific stress limits were provided for bellows in the as-formed condition (not heat-treated). The design defaulted to the rules in the EJMA Standards. However, the shop leak test was changed so that it was essentially a proof test of the bellows, so the pressure capacity of a bellows design is proven by the shop leak test.

Note that the philosophy in ASME BPVC, Section VIII, Division 1, Appendix 26 has the same intent. The meridional membrane plus bending stress due to internal pressure is limited to 1.5 times the Code-allowable stress. However, "alternate factors may be used if substantiated by test data (e.g., by hydrotest for design temperatures below the creep range)."

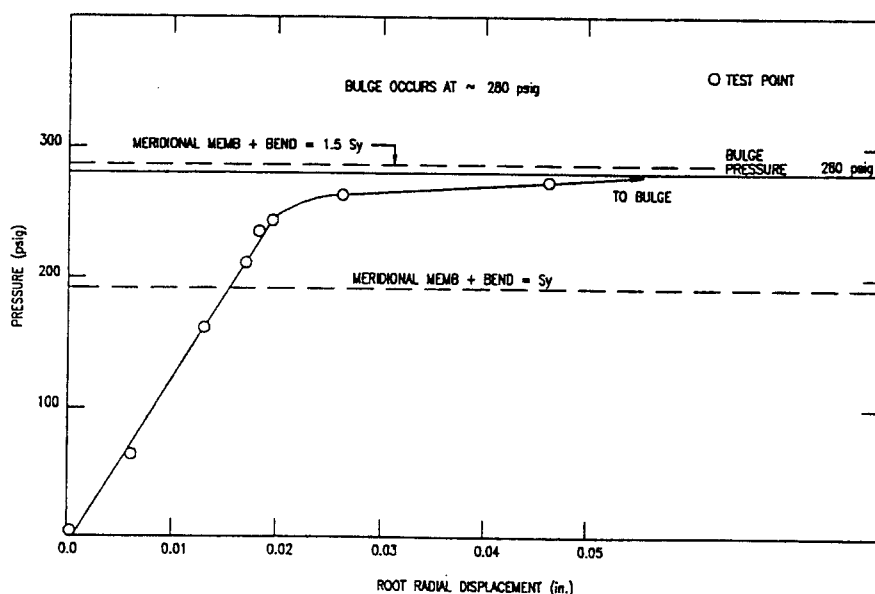


FIG. 18.1
BELLOWS PRESSURE TEST DATA

New design equations have been developed by Broyles (1989, 1994) that address this nonconservatism in the pressure design equations; these equations are now in the Sixth Edition of the EJMA Standards. The 6th Edition, as well as subsequent editions, of the EJMA Standards has been adopted by ASME B31.3. The specific stress limit for annealed bellows could have been dropped from Appendix X, because it is now in the EJMA Standards. However, they were left in place for additional emphasis.

18.3 FATIGUE DESIGN

With respect to Code rules, the fatigue curves for bellows contained in the EJMA Standards are deficient, in that they were developed to reflect average bellows performance. In a design code, a design curve that will provide sufficient margin of safety is required.

The advantage of fatigue curves of the type included in the EJMA Standards is that they were based on component testing. There are a number of complications in bellows performance, such as probable conservatism in the deflection stress equation for reinforced bellows, strain concentration that occurs when bellows are deflected beyond the elastic limit, and the effects of tolerances on bellows geometry, to which they can be very sensitive (Becht, 1985). These effects are inherently included in empirical fatigue curves, and need not be determined analytically. This simplifies the consideration of a number of complex phenomena. Fatigue curves were included in the first edition of Appendix X, which were the EJMA curves for unreinforced and reinforced bellows, reduced by what was considered to be an appropriate factor.

18.4 UNREINFORCED-BELLOWS FATIGUE CURVE DEVELOPMENT

Further investigation (Becht, 1989a) revealed that some fatigue test data for bellows fell below the original ASME B31.3 bellows fatigue curve and, further, that the slope of the EJMA curve did not appear to best reflect the data. In fact, a slope where stress range (actually, strain range times elastic modulus) is a function of $N^{-0.5}$, per the Manson-Coffin relation between plastic strain range and cycles to failure (N), appeared to better represent the data. Note that this slope is the basis for the fatigue curves in ASME BPVC, Section VIII, Division 2 and was the final slope adopted in ASME BPVC, Section VIII, Division 1, Appendix 26, and is widely accepted for most material.

A bellows fatigue database was provided to the Committee by the EJMA. The bellows were deflection-cycled, with varying amounts of internal pressure. Only the deflection stress was considered in developing the fatigue curve; including the steady-state pressure stress would nonconservatively bias the design curve on the high (nonconservative) side. Because the pressure stresses were noncyclic, they were not considered to significantly affect the fatigue life. Because the alternating stress was generally greater than the yield stress, the mean stress is effectively zero, so there is no mean stress correction on fatigue. Note, however, that cyclic pressure stresses must be included in design (the fatigue test pressure was noncyclic).

Some of the test bellows contained significantly more pressure than is permitted by the EJMA Standards. This high internal pressure can cause ratchet (Becht et al, 1981; Yamashita et al, 1989) and other problems, and was considered to distort the fatigue test results. Therefore, from this data, bellows tests where the stress due to internal pressure was more than 50% greater than the allowable stress were eliminated.

A best-fit curve using an equation of the form $S_T = AN_C^{-0.5} + B$, or, as presented in the Code,

$$N_C = \left(\frac{A}{S_T - B} \right)^2 \quad (18.1)$$

where

N_C = number of cycles to failure

S_T = calculated bellows stress range

A = material constant

B = material constant

was developed for the data. This equation is of the form provided by Langer (1961), which is based on the Manson–Coffin equation, but adapted for use with elastically calculated stress instead of plastic strain. The curve was then adjusted down in stress until it lower-bounded the data. Finally, the maximum margin (lowest curve) developed from factors of 1.25 on stress or 2.6 on cycles was applied to develop a design curve.

The factors of 1.25 on stress and 2.6 on cycles were based on ASME BPVC, Section VIII, Division 2, Appendix 6, Experimental Stress Analysis, and reflected the use of a number of tests greater than five. The cycle factor governs below 40,000 cycles and the stress factor governs above 40,000 cycles. Thus, there is a break in the curve at 40,000 cycles.

Figure 18.2 shows the fatigue test data, the lower bound curve, and the resulting design curve, including the safety margins, for unreinforced, as-formed, austenitic stainless steel bellows. Note that the actual desired fatigue life is intended to be used in conjunction with this curve, without the large factors of safety often included in the specified fatigue life for bellows. Although these factors may have been required when using an average fatigue curve, as provided in the EJMA Standards, for design, they are not required when using the design fatigue curve in the ASME B31.3 Code. Further, excessive conservatism in design for deflection can reduce the conservatism of the design with respect to internal pressure capacity.

The manner in which temperature is considered to affect the fatigue strength was considered in the same fashion as in ASME BPVC, Section VIII, Division 2. Because the correction factor for temperature would be the ratio of the elastic modulus for the fatigue test condition to that used in calculating the stresses, this was simplified by requiring that the calculation of stresses be done with the elastic modulus at ambient temperature (the fatigue test condition), 195×10^3 MPa (28.3×10^6 psi) for austenitic stainless steel.

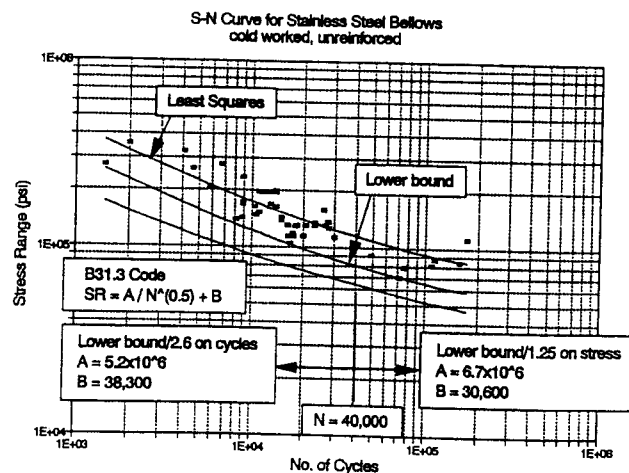


FIG. 18.2
FATIGUE DATA FOR UNREINFORCED AS-FORMED STAINLESS STEEL BELLOWS

18.5 REINFORCED-BELLOWS FATIGUE CURVE DEVELOPMENT

The deflection stress equations for reinforced bellows include an adjustment to account for the effect of internal pressure. Internal pressure is considered to reduce the effective convolution height by causing a greater interaction between the bellows sidewalls and the reinforcing ring. The basis for this adjustment is not documented, and it may not be accurate. However, because the empirical fatigue curve was developed with the same stress equations, the basis is self-consistent.

A best-fit fatigue curve was also developed for reinforced-bellows fatigue data, but was not considered to represent the data well. One possible cause was that the high-cycle data tended to have low internal pressure, whereas the low-cycle data tended to have high internal pressure. Therefore, inaccuracy in how pressure stress is calculated to affect deflection stress would distort the fatigue data (e.g., if pressure was calculated to cause a greater increase in deflection stress than actually occurs, the calculated cyclic stress in the low-cycle region with tests at a higher internal pressure would be shifted up relative to those tested at lower internal pressure in the high-cycle region).

Given the poor representation of the data and the questions regarding the stress equation, the best-fit curve for reinforced bellows was not used. Instead, the same curve as was developed for unreinforced bellows was used for reinforced bellows, except that it was shifted by a factor on stress until it lower-bounded the reinforced-bellows fatigue test data. The same factors as were applied to the unreinforced curve to develop a design curve were then applied to the reinforced curve.

Figure 18.3 shows the fatigue test data, the lower bound curve, and the resulting design fatigue curve for reinforced, as-formed, austenitic stainless steel bellows.

18.6 FATIGUE DESIGN FOR OTHER ALLOYS

All of the fatigue data used in developing the fatigue curve was from tests of as-formed (not heat-treated) austenitic stainless steel bellows. However, the Code does not prohibit the use of other alloys, and construction with alloys such as Inconel 625 and others is fairly common. Unfortunately, the bellows fatigue data for these other alloys was not available. Also, a simple comparison of polished bar fatigue properties was not considered sufficient, because the bellows fatigue curve was based on experimental data with as-formed bellows with cold-worked material.

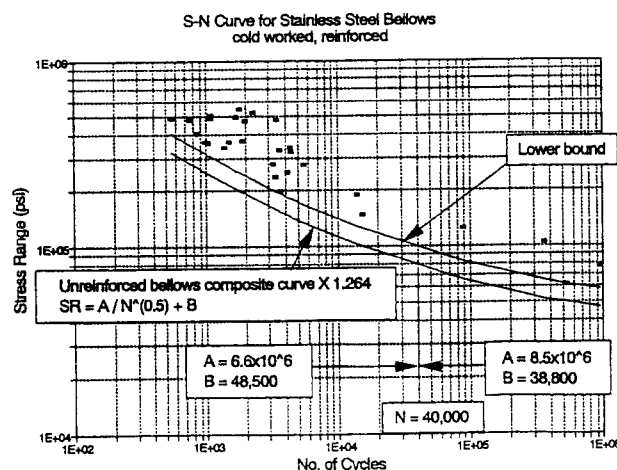


FIG. 18.3
FATIGUE DATA FOR REINFORCED AS-FORMED STAINLESS STEEL BELLOWS

Appendix X of ASME B31.3 initially referred to ASME BPVC, Section VIII, Division 2, Appendix 6 for development of fatigue curves for other alloys. However, this was not strictly applicable. Therefore, specific rules for developing the allowable stress range for other alloys based on bellows testing were developed and incorporated in the 1993 Edition, Addenda a. It was assumed that the slope of the fatigue curve for other alloys is the same. The fatigue data from tests on other alloys is simply used to effectively adjust the fatigue curve downward or upward (a minimum of five tests is required to adjust the fatigue curve upward). This adjustment takes the form of a material factor applied to the allowable stress range.

Specifically, the rules require a minimum of two bellows fatigue tests for each material, differing in stress range by a minimum factor of two. The allowable stress range determined from the fatigue chart for as-formed austenitic stainless steel bellows included in Appendix X is then multiplied by a material factor, X , derived from the fatigue tests. This material factor is based on the relationship between the fatigue test data and the Code fatigue curve as well as the number of fatigue tests. The procedure is illustrated in Fig. 18.4.

Because the fatigue curve in the Code is based on falling a minimum of a factor of 1.25 on stress or a factor of 2.6 on cycles below the lowest fatigue test data point, the material factor is developed on the same basis. The Code fatigue curve is essentially adjusted back up by these same factors to regenerate the lower bound curve and is compared to the bellows fatigue test data.

The minimum ratio of the total stress range of the test bellows to that from the lower-bound fatigue curve for as-formed austenitic stainless steel bellows, $R_{\min.}$, is used. This factor is multiplied by a K_s factor taken from Section ASME BPVC, VIII, Division 2, Appendix 6, which reflects the statistical variation of test results. The K_s factor varies from a minimum of 0.90 for two tests to a maximum of 1.0 for five tests.

An alternate way to determine $R_{\min.}$ would be to directly divide the calculated stress range for each fatigue test by the stress range from the lower-bound fatigue curve at the number of cycles to failure for the test. These equations are

$$S^{tb} = \frac{8.4 \times 10^6}{N_{ct}^{0.5}} + 38,300 \quad \text{for unreinforced bellows} \quad (18.2)$$

$$S^{tb} = \frac{10.6 \times 10^6}{N_{ct}^{0.5}} + 48,500 \quad \text{for reinforced bellows} \quad (18.3)$$

where

N_{ct} = number of cycles to failure of the bellows

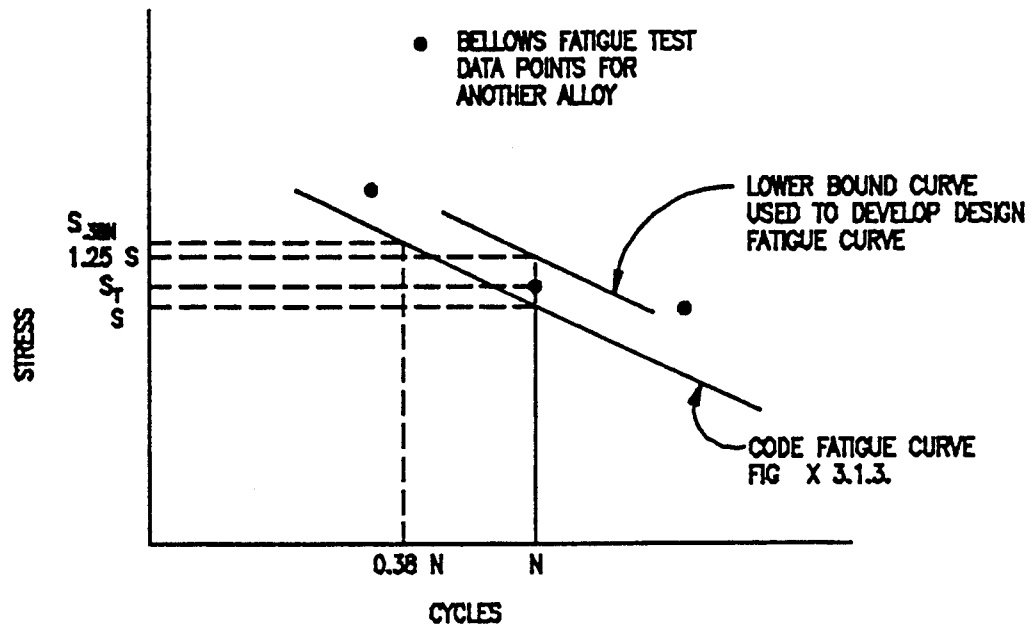
S^{tb} = stress range from the curve that lower-bounds the bellows fatigue test data

Although not stated in ASME B31.3, failure is generally considered to be the development of a through-thickness crack.

The product of K_s and $R_{\min.}$ is used to adjust the allowable total stress range provided in the fatigue curve for as-formed austenitic stainless steel bellows to result in the allowable cyclic stress range for bellows constructed with the different material. The intent of the material factor is to permit future inclusion of material factors for specific alloys should bellows fatigue test data on other alloys be made available to the Code Committee.

18.7 CORRELATION TESTING

It should be noted that the EJMA Standards require manufacturers to perform testing to correlate bellows performance with the equations provided. This includes a minimum of 25 fatigue tests on



$$S_{REFERENCE} = \min \left\{ \begin{array}{l} 1.25 S \\ S_{MIN} \end{array} \right.$$

$$R_{MIN} = \text{MINIMUM RATIO OF } \frac{S_T}{S_{REFERENCE}}, \text{ CONSIDERING ALL FATIGUE TESTS}$$

$$X = K_s R_{MIN}$$

$$S_T = \text{CALCULATED STRESS RANGE FOR TESTED BELLOWS}$$

$$N = \text{NUMBER OF CYCLES TO FAILURE IN TEST}$$

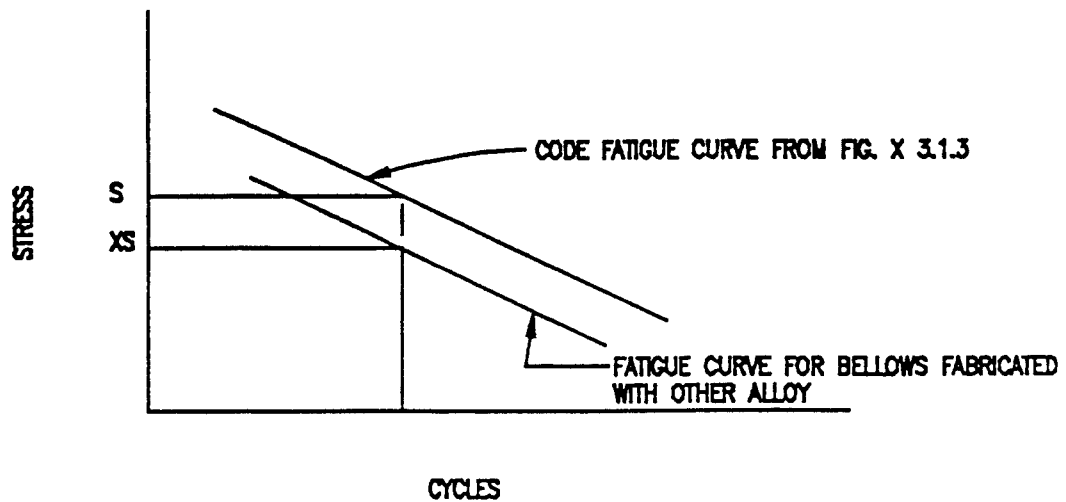


FIG. 18.4
PROCEDURE FOR CONSTRUCTING FATIGUE CURVES FOR OTHER MATERIALS

bellows of varying diameters, thicknesses, and convolution profiles. Appendix X reinforces this by stating, in paragraph X3.1.3(b):

“The manufacturer shall substantiate the method of calculation and its relationship to fatigue life by correlation testing as required by Sec. C-5.1.8 of the EJMA Standards. The minimum of 25 tests (each, for reinforced and unreinforced bellows) required by EJMA shall include the effects of all conditions necessary to validate the correlation between the design equations and the finished product, and shall represent the effects of the following variables as applicable: bellows diameter, thickness, convolution profile, manufacturing process, material, final heat treatment, and single-ply verses multi-ply construction.”

18.8 ELEVATED-TEMPERATURE DESIGN

The ASME B31.3 Code and ASME BPVC, Section VIII, Division 1 do not address elevated-temperature design of bellows. Design of bellows in the creep regime can often be avoided in piping by designing the expansion joint such that the bellows element operates at a significantly lower temperature than the process temperature. Where this is not possible, additional design considerations beyond the rules provided in the Code are required. In ASME B31.3, the Designer is referred to para. 304.7.2, Unlisted Components and Elements. However, even there, no specific guidance is provided for elevated-temperature design, and design for deflection stresses, a critical aspect of bellows design, is not covered.

The EJMA Standards started incorporating specific rules in the Seventh Edition that address the design of bellows in the creep range of the material of construction. For bellows displacement conditions, elevated-temperature cyclic testing is required.

Hold times are critical in creep fatigue. For example, a 1-hour hold time each cycle for Type 304 stainless steel at 550°C (1022°F) was shown to reduce cyclic life by as much as a factor of four compared to zero hold time (Yamamoto et al, 1986; Kobatake et al, 1986). Significantly greater factors have been shown in polished bar tests at higher temperatures. Some guidance for elevated-temperature design of metallic bellows expansion joints is provided by Becht (1989a). Criteria developed in the Japanese Liquid Metal Fast Breeder Reactor Program are described by Tsukimori et al (1989). Broyles (1994) developed a simplified design method based on elevated-temperature bellows creep fatigue testing, which was incorporated in the Seventh Edition of the EJMA Standards.

18.9 SHOP TEST OF EXPANSION JOINTS

Because of concerns regarding the EJMA pressure design rules, the leak test for bellows was changed so that it is effectively a proof test. The purpose was to demonstrate a minimum design margin of 1.5 against gross deformations, with specific concerns with respect to in-plane squirm and high meridional bending stresses due to internal pressure. Therefore, even pneumatic tests are required to be performed at or above 1.5 times the design pressure. Specifically, including the temperature correction factor, the test pressure is given by the equation

$$P_T = \frac{1.5 P S_T}{S} \quad (18.4)$$

where

P_T = minimum test gage pressure

S_T = allowable stress at test temperature

S = allowable stress at design temperature

P = design gage pressure

However, an exception to the above is required, because a problem can be caused by the temperature adjustment to the test pressure (Becht, 1989b). It can cause bellows that would perform satisfactorily in service to fail by buckling during the test. The test pressure in the Code is normally increased by the ratio of the allowable stress at the test temperature to the allowable stress at the design temperature. However, the strength properties on which the allowable stress is based vary to a much greater extent with temperature than the elastic modulus, which governs column squirm-type buckling of bellows. Therefore, the temperature adjustment for strength properties will overadjust the test pressure with respect to stability. Because of this, the Code provides that the test pressure need not exceed

$$\text{Minimum } P_T = \frac{1.5 P_S E_A}{E} \quad (18.5)$$

where

P_S = limiting design gauge pressure based on column instability

E_A = Young's modulus at test temperature

E = Young's modulus at design temperature

The test pressure is required to be a minimum of 1.5 times the design pressure.

Hardware on the expansion joint that is intended to resist pressure thrust forces must be included in the test and used to restrain the expansion joint without additional axial restraint during the pressure test. This essentially qualifies the hardware. It is to a certain extent more severe than actually necessary, because the temperature correction factor applied to the test pressure may not be applicable to the hardware, which may operate at temperatures closer to ambient. Therefore, the hardware must be designed with the test pressure condition in mind.

18.10 FIELD TEST OF PIPING SYSTEMS WITH EXPANSION JOINTS

One of the problems in the piping system caused by insertion of an expansion joint is the need to carry pressure thrust loads (end loads) in hardware on the expansion joint or restraints on the piping system. These same restraints must carry the pressure thrust of the expansion joint in the hydrotest condition, if the expansion joint is installed in the system during the test. The temperature correction factor for the system test pressure can result in a pressure thrust force that would fail system restraints. This can be readily observed from the following example.

Consider a system with an allowable stress of 10 at the design temperature and an allowable stress of 20 at the test temperature. The system test pressure could then be three times the design pressure of the system ($1.5 \times 20/10$). During the test, the pipe may be at a stress as high as 1.5 times the allowable stress at the test temperature; the result will generally be a test at yield stress or less for the piping. Consider that a bellows is included in the system and structural members are used to restrain the pressure thrust forces in the piping. These members would generally be designed for ambient temperature operation in accordance with the American Institute of Steel Construction *Manual of Steel Construction*. Axial stress in the structural members caused by these pressure thrust forces may be 60% of yield stress at the design pressure. Stress at the system test pressure could then be 1.8 (3 times 0.6) times the material's yield strength. Thus, the system structural restraints, properly designed for the design condition, could fail at the test pressure condition due to the temperature correction factor, which is not really applicable to the ambient temperature structural members.

However, because the resistance of the structural members to the pressure thrust force is essential for system integrity, and because inadequate design for pressure thrust has caused significant system

failures, the structural members should be tested as part of the system leak test. Therefore, the ASME B31.3 Code requires that expansion joints in systems in which anchors are used to restrain the pressure thrust force be included in the system pressure test, without temporary restraints, up to 150% of the design pressure. The structural members should be acceptable for this test load. For continuation of the test to above 150% of the design pressure, the expansion joint may be provided with temporary restraints or removed from the system. Expansion joints should never be subjected to a test pressure above the shop test pressure.

18.11 FUTURE DEVELOPMENTS

Recent work has shown that the fatigue behavior of bellows can be better characterized when the fatigue data is partitioned between low-profile convolutions, which are not subject to significant plastic strain concentration, and high-profile convolutions, which are. In fact, the reinforced- and unreinforced-bellows data for low-profile convolutions appears to follow the same fatigue curve. These findings may result in future changes to the fatigue design curves in ASME B31.3 (Becht, 1999, 2000a–c, 2001).

Insert 18.1 Pressure Thrust Considerations

One of the key characteristics of expansion joints in design is pressure thrust. The failure to properly consider pressure thrust has perhaps led to most of the system-design-error-related failures of piping systems with expansion joints.

Pressure thrust is simply the internal pressure in the pipe pushing on the ends, at changes of direction such as elbows and tees, and on partial surfaces such as reducers, orifice plates, and pressure letdown valves. It is the force that causes the $Pr/2t$ stress discussed in Chapter 4, Insert 4.2. In a straight pipe without expansion joints, it is simply the internal pressure times the internal area of the pipe. The inclusion of an expansion joint does not so much create a pressure thrust as cut the means that normally carries it, the pipe. Because the bellows or slip joint is not normally capable of carrying this pressure thrust load, some other means must be provided. This can be hardware, such as tie rods, hinges, or gimbals, provided with the expansion joint, or external anchors. Failure to provide such restraint typically results in a failed expansion joint.

The actual magnitude of the pressure thrust load in a metallic bellows expansion joint is actually higher than the pressure times the internal area of the pipe. This is because the included area is larger at the expansion joint. The pressure thrust force is the pressure times the bellows effective area, which is the area included in a circle defined by the mean diameter of the bellows (roughly the average of the inside and the outside diameters of the bellows). Note that the pressure times the internal area of the pipe acts at changes in cross section, such as at changes in direction. The difference between the bellows pressure thrust and this force acts at the ends of the bellows. When the bellows is under internal pressure, this force compresses the pipe on either end of the bellows if the load is taken by some external anchor remote from the expansion joint. As a result, guides may be required to prevent buckling of the pipe.

In some systems, it is important to understand the locations of the forces and how they are carried. For example, consider a pipe connected to a vessel nozzle. The pipe extends straight from the vessel to an elbow. Without an expansion joint in that straight run, there is a pressure thrust load of pressure times the pipe inside area, pulling (with internal pressure) on the nozzle. The net load on the vessel is zero, because this load is offset by the pressure acting on the wall of the vessel opposite the nozzle. If an expansion joint is placed in the straight run, with an external anchor at the change of direction, the situation is different. The load on the vessel nozzle will very

likely be significantly reduced. It will be the pressure times the difference between the effective bellows area and the pipe inside area. Further, it will be compression rather than tension. The vessel, however, will see an unbalanced load equal to the pressure times the effective bellows area. This is the sum of the compressive load acting on the nozzle and the unbalance pressure acting on the vessel wall opposite the nozzle. Note that under conditions of vacuum, the pressure thrust acts in reverse, tending to compress rather than stretch a bellows.

Detailed guidance on design of piping systems with expansion joints can be found in the Standards of the Expansion Joint Manufacturers Association.

APPENDIX I

PROPERTIES OF PIPE AND PRESSURE RATINGS OF LISTED PIPING COMPONENTS

I-1 THREAD AND GROOVE DEPTHS

The following table provides the nominal thread depth, h , from Table 2 of ANSI/ASME B1.20.1—1983. It is applicable for pressure design of pipe with American Standard Taper Pipe Thread (NPT). It is 0.8 times the thread pitch. The thread depth must be considered as a mechanical allowance in the pressure design of straight pipe in accordance with the base Code.

Nominal Pipe Size	Height of Thread, h (in.)
$\frac{1}{16}$	0.02963
$\frac{1}{8}$	0.02963
$\frac{1}{4}$	0.04444
$\frac{3}{8}$	0.04444
$\frac{1}{2}$	0.05714
$\frac{3}{4}$	0.05714
1	0.06957
$1\frac{1}{4}$	0.06957
$1\frac{1}{2}$	0.06957
2	0.06957
$2\frac{1}{2}$	0.100000
3	0.100000
$3\frac{1}{2}$	0.100000
4	0.100000
5	0.100000
6	0.100000
8	0.100000
10	0.100000
12	0.100000
14 O.D.	0.100000
16 O.D.	0.100000
18 O.D.	0.100000
20 O.D.	0.100000
24 O.D.	0.100000

The following table provides standard cut groove depths for IPS pipe from Victaulic®. Roll grooving does not remove metal from the pipe and therefore need not be considered in the allowances in pressure design. The cut groove depth must be considered in the allowances in pressure design.

Nominal Size (in.)	Groove Depth, <i>D</i> (in.)	Minimum Allowable Pipe Wall Thickness, <i>T</i> (in.)
3/4	0.056	0.113
1	0.063	0.133
1 1/4	0.063	0.140
1 1/2	0.063	0.145
2	0.063	0.154
2 1/2	0.078	0.188
3 O.D.	0.078	0.188
3	0.078	0.188
3 1/2	0.083	0.188
4	0.083	0.203
4 1/4 O.D.	0.083	0.203
4 1/2	0.083	0.203
5 1/4 O.D.	0.083	0.203
5 1/2 O.D.	0.083	0.203
5	0.084	0.203
6 O.D.	0.085	0.219
6 1/4 O.D.	0.109	0.249
6 1/2 O.D.	0.085	0.219
6	0.085	0.219
8 O.D.	0.092	0.238
8	0.092	0.238
10 O.D.	0.094	0.250
10	0.094	0.250
12 O.D.	0.109	0.279
12	0.109	0.279
14 O.D.	0.109	0.281
15 O.D.	0.109	0.312
16 O.D.	0.109	0.312
18 O.D.	0.109	0.312
20 O.D.	0.109	0.312
22 O.D.	0.172	0.375
24 O.D.	0.172	0.375
30 O.D.	0.250	0.625

The groove depth, *D*, is the nominal groove depth which is required to be uniform. The thickness, *T*, is the minimum pipe wall thickness that may be cut grooved. Note that the pressure rating of the fitting, provided by the manufacturer, may govern the pressure design. Also note that the pressure rating of Victaulic® couplings on roll and cut grooved standard wall pipes are the same.

I-2 PROPERTIES OF PIPE

The following table provides properties for nominal pipe. The schedule numbers are provided as follows.

- Pipe schedule numbers per ASME B36.10, Welded and Seamless Wrought Steel Pipe.
- Pipe nominal wall thickness designation per ASME B36.10.
- Pipe schedule numbers per ASME B36.19, Stainless Steel Pipe.

The weight per foot of pipe is based on carbon steel.

PROPERTIES OF PIPE (Courtesy of Anvil International)

The following formulas are used in the computation of the values shown in the table:

$$\begin{aligned} \uparrow \text{weight of pipe per foot (pounds)} &= 10.6802t(D - t) \\ \text{weight of water per foot (pounds)} &= 0.3405d^2 \\ \text{square feet outside surface per foot} &= 0.2618D \\ \text{square feet inside surface per foot} &= 0.2618d \\ \text{inside area (square inches)} &= 0.785d^2 \\ \text{area of metal (square inches)} &= 0.785(D^2 - d^2) \\ \text{moment of inertia (inches}^4\text{)} &= 0.0491(D^4 - d^4) \\ &= A_m R_g^2 \\ \text{section modulus (inches}^3\text{)} &= \frac{0.0982(D^4 - d^4)}{D} \\ \text{radius of gyration (inches)} &= 0.25 \sqrt{D^2 + d^2} \\ A_m &= \text{area of metal (square inches)} \\ d &= \text{inside diameter (inches)} \\ D &= \text{outside diameter (inches)} \\ R_g &= \text{radius of gyration (inches)} \\ t &= \text{pipe wall thickness (inches)} \end{aligned}$$

† The ferritic steels may be about 5% less, and the austenitic stainless steels about 2% greater than the values shown in this table which are based on weights for carbon steel.

***schedule numbers**

Standard weight pipe and schedule 40 are the same in all sizes through 10-inch; from 12-inch through 24-inch, standard weight pipe has a wall thickness of 3/8-inch.

Extra strong weight pipe and schedule 80 are the same in all sizes through 8-inch; from 8-inch through 24-inch, extra strong weight pipe has a wall thickness of 1/2-inch.

Double extra strong weight pipe has no corresponding schedule number.

a: ANSI B36.10 steel pipe schedule numbers

b: ANSI B36.10 steel pipe nominal wall thickness designation

c: ANSI B36.19 stainless steel pipe schedule numbers

nominal pipe size outside diameter, in.	schedule number*			wall thick- ness, in.	inside diam- eter, in.	inside area, sq. in.	metal area, sq. in.	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 modul- us, in. ³	radius gyra- tion, in.
	a	b	c											
1/8 0.405	—	—	10S	0.049	0.307	0.0740	0.0548	0.106	0.0804	0.186	0.0321	0.00088	0.00437	0.1271
	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
	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
1/4 0.540	—	—	10S	0.065	0.410	0.1320	0.0970	0.141	0.1073	0.330	0.0572	0.00279	0.01032	0.1694
	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
	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
3/8 0.675	—	—	10S	0.065	0.545	0.2333	0.1246	0.177	0.1427	0.423	0.1011	0.00586	0.01737	0.2169
	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
	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
1/2 0.840	—	—	5S	0.065	0.710	0.3959	0.1583	0.220	0.1859	0.538	0.171	0.0120	0.0285	0.2750
	—	—	10S	0.083	0.674	0.357	0.1974	0.220	0.1765	0.671	0.1547	0.01431	0.0341	0.2692
	40	Std	40S	0.109	0.622	0.304	0.2503	0.220	0.1628	0.851	0.1316	0.01710	0.0407	0.2613
	80	XS	80S	0.147	0.546	0.2340	0.320	0.220	0.1433	1.088	0.1013	0.02010	0.0478	0.2505
	160	—	—	0.187	0.466	0.1706	0.383	0.220	0.1220	1.304	0.0740	0.02213	0.0527	0.2402
	—	XXS	—	0.294	0.252	0.0499	0.504	0.220	0.0660	1.714	0.0216	0.02425	0.0577	0.2192
3/4 1.050	—	—	5S	0.065	0.920	0.665	0.2011	0.275	0.2409	0.684	0.2882	0.02451	0.0467	0.349
	—	—	10S	0.083	0.884	0.614	0.2521	0.275	0.2314	0.857	0.2661	0.02970	0.0566	0.343
	40	Std	40S	0.113	0.824	0.533	0.333	0.275	0.2157	1.131	0.2301	0.0370	0.0706	0.334
	80	XS	80S	0.154	0.742	0.432	0.435	0.275	0.1943	1.474	0.1875	0.0448	0.0853	0.321
	160	—	—	0.218	0.614	0.2961	0.570	0.275	0.1607	1.937	0.1284	0.0527	0.1004	0.304
	—	XXS	—	0.308	0.434	0.1479	0.718	0.275	0.1137	2.441	0.0641	0.0579	0.1104	0.2840
1 1.315	—	—	5S	0.065	1.185	1.103	0.2553	0.344	0.310	0.868	0.478	0.0500	0.0760	0.443
	—	—	10S	0.109	1.097	0.945	0.413	0.344	0.2872	1.404	0.409	0.0757	0.1151	0.428
	40	Std	40S	0.133	1.049	0.864	0.494	0.344	0.2746	1.679	0.374	0.0874	0.1329	0.421
	80	XS	80S	0.179	0.957	0.719	0.639	0.344	0.2520	2.172	0.311	0.1056	0.1606	0.407
	160	—	—	0.250	0.815	0.522	0.836	0.344	0.2134	2.844	0.2261	0.1252	0.1903	0.387
	—	XXS	—	0.358	0.599	0.2818	1.076	0.344	0.1570	3.659	0.1221	0.1405	0.2137	0.361
1 1/4 1.660	—	—	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
	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
	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	0.4580	0.2839	0.342	0.506
	—	XXS	—	0.382	0.896	0.0631	1.534	0.434	0.2346	5.214	0.2732	0.341	0.411	0.472
1 1/2 1.900	—	—	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

PROPERTIES OF PIPE (CONTINUED)
(Courtesy of Anvil International)

nominal pipe size outside diameter, in.	schedule number*			wall thick- ness, in.	inside diam- eter, in.	inside area, sq. in.	metal area, sq. in.	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 modul- us, in. ³	radius gyra- tion, in.
	a	b	c											
1½ 1.900	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
	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
	—	—	—	0.525	0.850	0.567	2.267	0.497	0.223	7.710	0.246	0.6140	0.6470	0.5200
	—	—	—	0.650	0.600	0.283	2.551	0.497	0.157	8.678	0.123	0.6340	0.6670	0.4980
2 2.375	—	—	5S	0.065	2.245	3.96	0.472	0.622	0.588	1.604	1.716	0.315	0.2652	0.817
	—	—	10S	0.109	2.157	3.65	0.776	0.622	0.565	2.638	1.582	0.499	0.420	0.802
	40	Std	40S	0.154	2.067	3.36	1.075	0.622	0.541	3.653	1.455	0.666	0.561	0.787
	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
	—	—	—	0.562	1.251	1.229	3.199	0.622	0.328	10.882	0.533	1.442	1.2140	0.6710
	—	—	—	0.687	1.001	0.787	3.641	0.622	0.262	12.385	0.341	1.5130	1.2740	0.6440
2½ 2.875	—	—	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
	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
	80	XS	80S	0.276	2.323	4.24	2.254	0.753	0.608	7.661	1.837	1.925	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	2.464	4.03	0.753	0.464	13.70	1.067	2.872	1.998	0.844
	—	—	—	0.675	1.525	1.826	4.663	0.753	0.399	15.860	0.792	3.0890	2.1490	0.8140
	—	—	—	0.800	1.275	1.276	5.212	0.753	0.334	17.729	0.554	3.2250	2.2430	0.7860
3 3.500	—	—	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
	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
	80	XS	80S	0.300	2.900	6.61	3.02	0.916	0.759	10.25	2.864	3.90	2.226	1.136
	160	—	—	0.437	2.626	5.42	4.21	0.916	0.687	14.32	2.348	5.03	2.876	1.094
	—	XXS	—	0.600	2.300	4.15	5.47	0.916	0.602	18.58	1.801	5.99	3.43	1.047
	—	—	—	0.725	2.050	3.299	6.317	0.916	0.537	21.487	1.431	6.5010	3.7150	1.0140
	—	—	—	0.850	1.800	2.543	7.073	0.916	0.471	24.057	1.103	6.8530	3.9160	0.9840
3½ 4.000	—	—	5S	0.083	3.834	11.55	1.021	1.047	1.004	3.47	5.01	1.960	0.980	1.385
	—	—	10S	0.120	3.760	11.10	1.463	1.047	0.984	4.97	4.81	2.756	1.378	1.372
	40	Std	40S	0.226	3.548	9.89	2.680	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
	—	XXS	—	0.636	2.728	5.845	6.721	1.047	0.716	22.850	2.530	9.8480	4.9240	1.2100
4 4.500	—	—	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
	—	—	—	0.188	4.124	13.357	2.547	1.178	1.082	8.560	5.800	5.8500	2.6000	1.5250
	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
	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
	—	—	—	0.500	3.500	9.621	6.283	1.178	0.916	21.360	4.160	12.7710	5.6760	1.4250
	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
	—	—	—	0.800	2.900	6.602	9.294	1.178	0.759	31.613	2.864	16.6610	7.4050	1.3380
5 5.563	—	—	—	0.925	2.650	5.513	10.384	1.178	0.694	35.318	2.391	17.7130	7.8720	1.3060
	—	—	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
	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
	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	30.0	10.80	1.760
	—	XXS	—	0.750	4.063	12.97	11.34	1.456	1.064	38.55	5.62	33.6	12.10	1.722
	—	—	—	0.875	3.813	11.413	12.880	1.456	0.998	43.810	4.951	36.6450	13.1750	1.6860
	—	—	—	1.000	3.563	9.966	14.328	1.456	0.933	47.734	4.232	39.1110	14.0610	1.6520

PROPERTIES OF PIPE (CONTINUED)
(Courtesy of Anvil International)

nominal pipe size outside diameter, in.	schedule number*			wall thick- ness, in.	inside diam- eter, in.	inside area, sq. in.	metal area, sq. in.	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 modul- us, in. ³	radius gyra- tion, in.
	a	b	c											
6 6.625	—	—	5S	0.109	6.407	32.2	2.231	1.734	1.677	5.37	13.98	11.85	3.58	2.304
	—	—	10S	0.134	6.357	31.7	2.733	1.734	1.664	9.29	13.74	14.40	4.35	2.295
	—	—	—	0.129	6.187	30.100	4.410	1.734	1.620	15.020	13.100	22.6600	6.8400	2.2700
	40	Std	40S	0.280	6.065	28.89	5.58	1.734	1.588	18.97	12.51	28.14	8.50	2.245
	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	59.0	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
	—	—	—	1.000	4.625	16.792	17.662	1.734	1.211	60.076	7.284	72.1190	21.7720	2.0200
	—	—	—	1.125	4.375	15.025	19.429	1.734	1.145	66.084	6.517	76.5970	23.1240	1.9850
8 8.625	—	—	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
	—	—	—	0.219	8.187	52.530	5.800	2.258	2.150	19.640	22.900	51.3200	11.9000	2.9700
	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
	60	—	—	0.406	7.813	47.9	10.48	2.258	2.045	35.64	20.79	88.8	20.58	2.909
	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
8 8.625	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
	160	—	—	0.906	6.813	36.5	21.97	2.258	1.784	74.69	15.80	165.9	38.5	2.748
	—	—	—	1.000	6.625	34.454	23.942	2.258	1.734	81.437	14.945	177.1320	41.0740	2.7190
	—	—	—	1.125	6.375	31.903	26.494	2.258	1.669	90.114	13.838	190.6210	44.2020	2.6810
10 10.750	—	—	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	14.30	3.74
	—	—	—	0.219	10.312	83.52	7.24	2.815	2.70	24.63	36.2	100.46	18.69	3.72
	20	—	—	0.250	10.250	82.5	8.26	2.815	2.683	28.04	35.8	113.7	21.16	3.71
	30	—	—	0.307	10.136	80.7	10.07	2.815	2.654	34.24	35.0	137.5	25.57	3.69
	40	Std	40S	0.365	10.020	78.9	11.91	2.815	2.623	40.48	34.1	160.8	29.90	3.67
	60	XS	80S	0.500	9.750	74.7	16.10	2.815	2.553	54.74	32.3	212.0	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
	—	—	—	0.875	9.000	63.62	27.14	2.815	2.36	92.28	27.6	333.46	62.04	3.50
	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
	—	—	—	1.250	8.250	53.45	37.31	2.815	2.16	126.82	23.2	428.17	79.66	3.39
	—	—	—	1.500	7.750	47.15	43.57	2.815	2.03	148.19	20.5	478.59	89.04	3.31
12 12.750	—	—	5S	0.156	12.438	121.4	6.17	3.34	3.26	20.99	52.7	122.2	19.20	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
	40	—	—	0.406	11.938	111.9	15.74	3.34	3.13	53.53	48.5	300	47.1	4.37
	—	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
	—	—	—	0.750	11.250	99.40	28.27	3.34	2.94	96.2	43.1	510.7	80.1	4.25
	100	—	—	0.843	11.064	96.1	31.5	3.34	2.897	107.20	41.6	562	88.1	4.22
	—	—	—	0.875	11.000	95.00	32.64	3.34	2.88	110.9	41.1	578.5	90.7	4.21
	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
	—	—	—	1.250	10.250	82.50	45.16	3.34	2.68	153.6	35.8	755.5	118.5	4.09
	160	—	—	1.312	10.126	80.5	47.1	3.34	2.651	160.27	34.9	781	122.6	4.07

PROPERTIES OF PIPE (CONTINUED)
(Courtesy of Anvil International)

nominal pipe size outside diameter, in.	schedule number*			wall thick- ness, in.	inside diam- eter, in.	inside area, sq. in.	metal area, sq. in.	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 modul- us, in. ³	radius gyra- tion, in.
	a	b	c											
14 14.000	—	—	5S	0.156	13.688	147.20	6.78	3.67	3.58	23.0	63.7	162.6	23.2	4.90
	—	—	10S	0.188	13.624	145.80	8.16	3.67	3.57	27.7	63.1	194.6	27.8	4.88
	—	—	—	0.210	13.580	144.80	9.10	3.67	3.55	30.9	62.8	216.2	30.9	4.87
	—	—	—	0.219	13.562	144.50	9.48	3.67	3.55	32.2	62.6	225.1	32.2	4.87
	10	—	—	0.250	13.500	143.1	10.80	3.67	3.53	36.71	62.1	255.4	36.5	4.86
	—	—	—	0.281	13.438	141.80	12.11	3.67	3.52	41.2	61.5	285.2	40.7	4.85
	20	—	—	0.312	13.376	140.5	13.42	3.67	3.50	45.68	60.9	314	44.9	4.84
	—	—	—	0.344	13.312	139.20	14.76	3.67	3.48	50.2	60.3	344.3	49.2	4.83
	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
	—	—	—	0.469	13.062	134.00	19.94	3.67	3.42	67.8	58.0	456.8	65.3	4.79
	—	XS	—	0.500	13.000	132.7	21.21	3.67	3.40	72.09	57.5	484	69.1	4.78
	60	—	—	0.593	12.814	129.0	24.98	3.67	3.35	84.91	55.9	562	80.3	4.74
	—	—	—	0.625	12.750	127.7	26.26	3.67	3.34	89.28	55.3	589	84.1	4.73
	80	—	—	0.750	12.500	122.7	31.2	3.67	3.27	106.13	53.2	687	98.2	4.69
	100	—	—	0.937	12.126	115.5	38.5	3.67	3.17	130.73	50.0	825	117.8	4.63
16 16.000	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	1028	146.8	4.53
	160	—	—	1.406	11.188	98.3	55.6	3.67	2.929	189.12	42.6	1117	159.6	4.48
	—	—	5S	0.165	15.670	192.90	8.21	4.19	4.10	28	83.5	257	32.2	5.60
	—	—	10S	0.188	15.624	191.70	9.34	4.19	4.09	32	83.0	292	36.5	5.59
	10	—	—	0.250	15.500	188.7	12.37	4.19	4.06	42.05	81.8	384	48.0	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
	40	XS	—	0.500	15.000	176.7	24.35	4.19	3.93	82.77	76.5	732	91.5	5.48
	60	—	—	0.656	14.688	169.4	31.6	4.19	3.85	107.50	73.4	933	116.6	5.43
	80	—	—	0.843	14.314	160.9	40.1	4.19	3.75	136.46	69.7	1157	144.6	5.37
	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	223.84	58.6	1760	220.0	5.17
	160	—	—	1.593	12.814	129.0	72.1	4.19	3.35	245.11	55.9	1894	236.7	5.12
18 18.000	—	—	5S	0.165	17.670	245.20	9.24	4.71	4.63	31	106.2	368	40.8	6.31
	—	—	10S	0.188	17.624	243.90	10.52	4.71	4.61	36	105.7	417	46.4	6.30
	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	17.00	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
	60	—	—	0.750	16.500	213.8	40.6	4.71	4.32	138.17	92.7	1515	168.3	6.10
	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
20 20.000	—	—	5S	0.188	19.634	302.40	11.70	5.24	5.14	40	131.0	574	57.4	7.00
	—	—	10S	0.218	19.564	300.60	13.55	5.24	5.12	46	130.2	663	66.3	6.99
	10	—	—	0.250	19.500	298.60	15.51	5.24	5.11	52.73	129.5	757	75.7	6.98
	20	Std	—	0.375	19.250	291.0	23.12	5.24	5.04	78.60	126.0	1114	111.4	6.94
	30	XS	—	0.500	19.000	283.5	30.6	5.24	4.97	104.13	122.8	1457	145.7	6.90
	40	—	—	0.593	18.814	278.0	36.2	5.24	4.93	122.91	120.4	1704	170.4	6.86
	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

PROPERTIES OF PIPE (CONTINUED)
(Courtesy of Anvil International)

nominal pipe size outside diameter, in.	schedule number*			wall thick- ness, in.	inside diam- eter, in.	inside area, sq. in.	metal area, sq. in.	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 modul- us, in. ³	radius gyra- tion, in.
	a	b	c											
20 20.000	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
22 22.000	—	—	5S	0.188	21.624	367.3	12.88	5.76	5.66	44	159.1	766	69.7	7.71
	—	—	10S	0.218	21.564	365.2	14.92	5.76	5.65	51	158.2	885	80.4	7.70
	10	—	—	0.250	21.500	363.1	17.18	5.76	5.63	58	157.4	1010	91.8	7.69
	20	Std	—	0.375	21.250	354.7	25.48	5.76	5.56	87	153.7	1490	135.4	7.65
	30	XS	—	0.500	21.000	346.4	33.77	5.76	5.50	115	150.2	1953	177.5	7.61
	—	—	—	0.625	20.750	338.2	41.97	5.76	5.43	143	146.6	2400	218.2	7.56
	—	—	—	0.750	20.500	330.1	50.07	5.76	5.37	170	143.1	2829	257.2	7.52
	60	—	—	0.875	20.250	322.1	58.07	5.76	5.30	197	139.6	3245	295.0	7.47
	80	—	—	1.125	19.750	306.4	73.78	5.76	5.17	251	132.8	4029	366.3	7.39
	100	—	—	1.375	19.250	291.0	89.09	5.76	5.04	303	126.2	4758	432.6	7.31
	120	—	—	1.625	18.750	276.1	104.02	5.76	4.91	354	119.6	5432	493.8	7.23
	140	—	—	1.875	18.250	261.6	118.55	5.76	4.78	403	113.3	6054	550.3	7.15
	160	—	—	2.125	17.750	247.4	132.68	5.76	4.65	451	107.2	6626	602.4	7.07
24 24.000	10	—	—	0.250	23.500	434	18.65	6.28	6.15	63.41	188.0	1316	109.6	8.40
	20	Std	—	0.375	23.250	425	27.83	6.28	6.09	94.62	183.8	1943	161.9	8.35
	—	XS	—	0.500	23.000	415	36.9	6.28	6.02	125.49	180.1	2550	212.5	8.31
	30	—	—	0.562	22.876	411	41.4	6.28	5.99	140.80	178.1	2840	237.0	8.29
	—	—	—	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
	—	—	5S	0.218	23.564	436.1	16.29	6.28	6.17	55	188.9	1152	96.0	8.41
	—	—	—	0.875	22.250	388.6	63.54	6.28	5.83	216	168.6	4256	354.7	8.18
	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
	160	—	—	2.343	19.314	293	159.4	6.28	5.06	541.94	127.0	9460	788	7.70
26 26.000	—	—	—	0.250	25.500	510.7	19.85	6.81	6.68	67	221.4	1646	126.6	9.10
	10	—	—	0.312	25.376	505.8	25.18	6.81	6.64	86	219.2	2076	159.7	9.08
	—	Std	—	0.375	25.250	500.7	30.19	6.81	6.61	103	217.1	2478	190.6	9.06
	20	XS	—	0.500	25.000	490.9	40.06	6.81	6.54	136	212.8	3259	250.7	9.02
	—	—	—	0.625	24.750	481.1	49.82	6.81	6.48	169	208.6	4013	308.7	8.98
	—	—	—	0.750	24.500	471.4	59.49	6.81	6.41	202	204.4	4744	364.9	8.93
	—	—	—	0.875	24.250	461.9	69.07	6.81	6.35	235	200.2	5458	419.9	8.89
	—	—	—	1.000	24.000	452.4	78.54	6.81	6.28	267	196.1	6149	473.0	8.85
28 28.000	—	—	—	1.125	23.750	443.0	87.91	6.81	6.22	299	192.1	6813	524.1	8.80
	—	—	—	0.250	27.500	594.0	21.80	7.33	7.20	74	257.3	2098	149.8	9.81
	10	—	—	0.312	27.376	588.6	27.14	7.33	7.17	92	255.0	2601	185.8	9.79
	—	Std	—	0.375	27.250	583.2	32.54	7.33	7.13	111	252.6	3105	221.8	9.77
	20	XS	—	0.500	27.000	572.6	43.20	7.33	7.07	147	248.0	4085	291.8	9.72
	30	—	—	0.625	26.750	562.0	53.75	7.33	7.00	183	243.4	5038	359.8	9.68
	—	—	—	0.750	26.500	551.6	64.21	7.33	6.94	218	238.9	5964	426.0	9.64
	—	—	—	0.875	26.250	541.2	74.56	7.33	6.87	253	234.4	6865	490.3	9.60
30 30.000	—	—	—	1.000	26.000	530.9	84.82	7.33	6.81	288	230.0	7740	552.8	9.55
	—	—	—	1.125	25.750	520.8	94.98	7.33	6.74	323	225.6	8590	613.6	9.51
	—	—	5S	0.250	29.500	683.4	23.37	7.85	7.72	79	296.3	2585	172.3	10.52
	10	—	10S	0.312	29.376	677.8	29.19	7.85	7.69	99	293.7	3201	213.4	10.50
	—	Std	—	0.375	29.250	672.0	34.90	7.85	7.66	119	291.2	3823	254.8	10.48
30 30.000	20	XS	—	0.500	29.000	660.5	46.34	7.85	7.59	158	286.2	5033	335.5	10.43
	30	—	—	0.625	28.750	649.2	57.68	7.85	7.53	196	281.3	6213	414.2	10.39

PROPERTIES OF PIPE (CONTINUED)
(Courtesy of Anvil International)

nominal pipe size outside diameter, in.	schedule number*			wall thick- ness, in.	inside diam- eter, in.	inside area, sq. in.	metal area, sq. in.	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 modul- us, in. ³	radius gyra- tion, in.
	a	b	c											
30 30.000	40	—	—	0.750	28.500	637.9	68.92	7.85	7.46	234	276.6	7371	491.4	10.34
	—	—	—	0.875	28.250	620.7	80.06	7.85	7.39	272	271.8	8494	566.2	10.30
	—	—	—	1.000	28.000	615.7	91.11	7.85	7.33	310	267.0	9591	639.4	10.26
	—	—	—	1.125	27.750	604.7	102.05	7.85	7.26	347	262.2	10653	710.2	10.22
32 32.000	—	—	—	0.250	31.500	779.2	24.93	8.38	8.25	85	337.8	3141	196.3	11.22
	10	—	—	0.312	31.376	773.2	31.02	8.38	8.21	106	335.2	3891	243.2	11.20
	—	Std	—	0.375	31.250	766.9	37.25	8.38	8.18	127	332.5	4656	291.0	11.18
	20	XS	—	0.500	31.000	754.7	49.48	8.38	8.11	168	327.2	6140	383.8	11.14
	30	—	—	0.625	30.750	742.5	61.59	8.38	8.05	209	321.9	7578	473.6	11.09
	40	—	—	0.688	30.624	736.6	67.68	8.38	8.02	230	319.0	8298	518.6	11.07
	—	—	—	0.750	30.500	730.5	73.63	8.38	7.98	250	316.7	8990	561.9	11.05
	—	—	—	0.875	30.250	718.3	85.52	8.38	7.92	291	311.6	10372	648.2	11.01
	—	—	—	1.000	30.000	706.8	97.38	8.38	7.85	331	306.4	11680	730.0	10.95
	—	—	—	1.125	29.750	694.7	109.0	8.38	7.79	371	301.3	13023	814.0	10.92
34 34.000	—	—	—	0.250	33.500	881.2	26.50	8.90	8.77	90	382.0	3773	221.9	11.93
	10	—	—	0.312	33.376	874.9	32.99	8.90	8.74	112	379.3	4680	275.3	11.91
	—	Std	—	0.375	33.250	867.8	39.61	8.90	8.70	135	376.2	5597	329.2	11.89
	20	XS	—	0.500	33.000	855.3	52.62	8.90	8.64	179	370.8	7385	434.4	11.85
	30	—	—	0.625	32.750	841.9	65.53	8.90	8.57	223	365.0	9124	536.7	11.80
	40	—	—	0.688	32.624	835.9	72.00	8.90	8.54	245	362.1	9992	587.8	11.78
	—	—	—	0.750	32.500	829.3	78.34	8.90	8.51	266	359.5	10829	637.0	11.76
	—	—	—	0.875	32.250	816.4	91.01	8.90	8.44	310	354.1	12501	735.4	11.72
	—	—	—	1.000	32.000	804.2	103.67	8.90	8.38	353	348.6	14114	830.2	11.67
	—	—	—	1.125	31.750	791.3	116.13	8.90	8.31	395	343.2	15719	924.7	11.63
36 36.000	—	—	—	0.250	35.500	989.7	28.11	9.42	9.29	96	429.1	4491	249.5	12.64
	10	—	—	0.312	35.376	982.9	34.95	9.42	9.26	119	426.1	5565	309.1	12.62
	—	Std	—	0.375	35.250	975.8	42.01	9.42	9.23	143	423.1	6664	370.2	12.59
	20	XS	—	0.500	35.000	962.1	55.76	9.42	9.16	190	417.1	8785	488.1	12.55
	30	—	—	0.625	34.750	948.3	69.50	9.42	9.10	236	411.1	10872	604.0	12.51
	40	—	—	0.750	34.500	934.7	83.01	9.42	9.03	282	405.3	12898	716.5	12.46
	—	—	—	0.875	34.250	920.6	96.50	9.42	8.97	328	399.4	14903	827.9	12.42
	—	—	—	1.000	34.000	907.9	109.96	9.42	8.90	374	393.6	16851	936.2	12.38
	—	—	—	1.125	33.750	894.2	123.19	9.42	8.89	419	387.9	18763	1042.4	12.34
	—	—	—	—	—	—	—	—	—	—	—	—	—	—
42 42.000	—	—	—	0.250	41.500	1352.6	32.82	10.99	10.86	112	586.4	7126	339.3	14.73
	—	Std	—	0.375	41.250	1336.3	49.08	10.99	10.80	167	579.3	10627	506.1	14.71
	20	XS	—	0.500	41.000	1320.2	65.18	10.99	10.73	222	572.3	14037	668.4	14.67
	30	—	—	0.625	40.750	1304.1	81.28	10.99	10.67	276	565.4	17373	827.3	14.62
	40	—	—	0.750	40.500	1288.2	97.23	10.99	10.60	330	558.4	20689	985.2	14.59
	—	—	—	1.000	40.000	1256.6	128.81	10.99	10.47	438	544.8	27080	1289.5	14.50
	—	—	—	1.250	39.500	1225.3	160.03	10.99	10.34	544	531.2	33233	1582.5	14.41
	—	—	—	1.500	39.000	1194.5	190.85	10.99	10.21	649	517.9	39181	1865.7	14.33

I-3 PRESSURE RATINGS OF STANDARD COMPONENTS

Pressure ratings provided in listed standards (those listed in Table 326.1 of ASME B31.3) are provided below. They are grouped by Iron (I-3.1), Steel (I-3.2), and Copper and Copper Alloys (I-3.3).

I-3.1 Iron

ASME B16.1 covers Cast Iron Pipe Flanges and Flanged Fittings, Classes 25, 125, and 250. It provides both dimensional information and pressure-temperature ratings. Sizes from NPS 1 to NPS 96 are included. Class A or B relates to class of iron. The pressure-temperature ratings are provided in the following table, Table 1 in ASME B16.1.

ASME B16.1 NONSHOCK GAGE PRESSURE-TEMPERATURE RATINGS

Temperature (°F)	Class 25 (1) ASTM A 126		Class 125 ASTM A 126				Class 250 (1) ASTM A 126			
	Class A		Class A	Class B			Class A	Class B		
	NPS 4-36	NPS 42-96	NPS 1-12	NPS 1-12	NPS 14-24	NPS 30-48	NPS 1-12	NPS 1-12	NPS 14-24	NPS 30-48
-20 to 150	45	25	175	200	150	150	400	500	300	300
200	40	25	165	190	135	115	370	460	280	250
225	35	25	155	180	130	100	355	440	270	225
250	30	25	150	175	125	85	340	415	260	200
275	25	25	145	170	120	65	325	395	250	175
300	—	—	140	165	110	50	310	375	240	150
325	—	—	130	155	105	—	295	355	230	125
353 (2)	—	—	125	150	100	—	280	335	220	100
375	—	—	—	145	—	—	265	315	210	—
406 (3)	—	—	—	140	—	—	250	290	200	—
425	—	—	—	130	—	—	—	270	—	—
450	—	—	—	125	—	—	—	250	—	—
Hydrostatic Shell Test Pressures (4)										
100	70	40	270	300	230	230	600	750	450	450

GENERAL NOTES:

(a) Pressure is in lb/sq in. gage.

(b) NPS is nominal pipe size.

(1) Limitations:

a. Class 25. When Class 25 cast iron flanges and flanged fittings are used for gaseous service, the maximum pressure shall be limited to 25 psig. Tabulated pressure-temperature ratings above 25 psig for Class 25 cast iron flanges and flanged fittings are applicable for nonshock hydraulic service only.

b. Class 250. When used for liquid service, the tabulated pressure-temperature ratings in NPS 14 and larger are applicable to Class 250 flanges only and not to Class 250 fittings.

(2) 353°F (max.) to reflect the temperature of saturated steam at 125 psig.

(3) 406°F (max.) to reflect the temperature of saturated steam at 250 psig.

(4) Hydrostatic tests are not required unless specified by the user.

ASME B16.3 covers Malleable Iron Threaded Fittings, Classes 150 and 300. It provides both dimensional information and pressure-temperature ratings. It includes sizes from NPS 1/8 to NPS 6. The pressure-temperature ratings are provided in the following table, Table 1 in ASME B16.3.

ASME B16.3 PRESSURE-TEMPERATURE RATINGS

Temperature (°F)	Class 150 (psig)	Class 300 (psig)		
		Sizes (1/4-1)	Sizes (1 1/4-2)	Sizes (2 1/2-3)
-20 to 150	300	2000	1500	1000
200	265	1785	1350	910
250	225	1575	1200	825
300	185	1360	1050	735
350	150 (1)	1150	900	650
400	—	935	750	560
450	—	725	600	475
500	—	510	450	385
550	—	300	300	300

NOTE:

(1) Permissible for service temperature up to 366°F, reflecting the temperature of saturated steam at 150 psig.

ASME B16.4 covers Gray Iron Threaded Fittings, Classes 125 and 250. It provides both dimensional information and pressure-temperature ratings. It includes sizes from NPS 1/4 to NPS 12. The pressure-temperature ratings are provided in the following table, Table 1 in ASME B16.4

ASME B16.4 PRESSURE-TEMPERATURE RATINGS

Temperature (°F)	Class 125 (psi)	Class 250 (psi)
-20 to 150	175	400
200	165	370
250	150	340
300	140	310
350	125 (1)	300
400	—	250 (2)

NOTES:

- (1) Permissible for service temperature up to 353°F reflecting the temperature of saturated steam at 125 psig.
- (2) Permissible for service temperature up to 406°F reflecting the temperature of saturated steam at 250 psig.

ASME B16.39 covers Malleable Iron Threaded Pipe Unions, Classes 150, 250, and 300. It provides both dimensional information and pressure-temperature ratings. It covers sizes from NPS 1/8 to NPS 4. The pressure-temperature ratings are provided in the following table, Table 1 in ASME B16.39. Unions with copper or copper alloy seats are not to be used for temperatures in excess of 232°C (450°F).

ASME B16.39 PRESSURE-TEMPERATURE RATINGS

Temperature (°F)	Pressure (psig)		
	Class 150	Class 250	Class 300
-20 to 150	300	500	600
200	265	455	550
250	225	405	505
300	185	360	460
350	150	315	415
400	110	270	370
450	75	225	325
500	—	180	280
550	—	130	230

ASME B16.42 covers Ductile Iron Pipe Flanges and Flanged Fittings, Classes 150 and 300. It provides both dimensional information and pressure-temperature ratings. It covers sizes from NPS 1 to NPS 24. The pressure-temperature ratings are provided in the following table, Table 1 in ASME B16.42.

ASME B16.42 PRESSURE-TEMPERATURE RATINGS

Temperature (°F)	Working Pressure (psi gage)	
	Class 150	Class 300
-20 to 100	250	640
200	235	600
300	215	565
400	200	525
500	170	495
600	140	465
650	125	450

I-3.2 Steel

ASME B16.5 covers Pipe Flanges and Flanged Fittings, NPS $\frac{1}{2}$ through NPS 24. It provides both dimensional information and pressure-temperature ratings. The pressure temperature ratings provided in ASME B16.5 are the commonly used pressure classes. These include Classes 150, 300, 400, 600, 900, 1500, and 2500. Tables of pressure-temperature ratings are provided for groups of materials. Examples for commonly used carbon steel and austenitic stainless steel materials are provided in the following tables, Tables 2-1.1 and 2-2.1 in ASME B16.5.

ASME B16.5 PRESSURE-TEMPERATURE RATINGS FOR GROUPS 1.1 THROUGH 3.17 MATERIALS

RATINGS FOR GROUP 1.1 MATERIALS

Nominal Designation	Forgings	Castings	Plates
C-Si	A 105 (1)	A 216 Gr. WCB (1)	A515 Gr. 70 (1)
C-Mn-Si	A 350 Gr. LF2 (1)		A 516 Gr. 70 (1)(2) A 537 Cl. 1 (3)
C-Mn-Si-V	A 350 Gr. LF6 Cl. 1 (4)		

NOTES:

- (1) Upon prolonged exposure to temperatures above 800°F, the carbide phase of steel may be converted to graphite. Permissible, but not recommended for prolonged use above 800°F.
- (2) Not to be used over 850°F.
- (3) Not to be used over 700°F.
- (4) Not to be used over 500°F.

Class Temperature (°F)	Working Pressures by Classes (psig)						
	150	300	400	600	900	1500	2500
-20 to 100	285	740	990	1480	2220	3705	6170
200	260	675	900	1350	2025	3375	5625
300	230	655	875	1315	1970	3280	5470
400	200	635	845	1270	1900	3170	5280
500	170	600	800	1200	1795	2995	4990
600	140	550	730	1095	1640	2735	4560
650	125	535	715	1075	1610	2685	4475
700	110	535	710	1065	1600	2665	4440
750	95	505	670	1010	1510	2520	4200
800	80	410	550	825	1235	2060	3430
850	65	270	355	535	805	1340	2230
900	50	170	230	345	515	860	1430
950	35	105	140	205	310	515	860
1000	20	50	70	105	155	260	430

RATINGS FOR GROUP 2.1 MATERIALS

Nominal Designation	Forgings	Castings	Plates
18Cr-8Ni	A 182 Gr. F304 (1) A 182 Gr. F304H	A 351 Gr. CF3 (2) A 351 Gr. CF8 (1)	A 240 Gr. 304 (1) A 240 Gr. 304H

NOTES:

(1) At temperatures over 1000°F, use only when the carbon content is 0.04% or higher.

(2) Not to be used over 800°F.

Class Temperature (°F)	Working Pressures by Classes (psig)						
	150	300	400	600	900	1500	2500
-20 to 100	275	720	960	1440	2160	3600	6000
200	230	600	800	1200	1800	3000	5000
300	205	540	720	1080	1620	2700	4500
400	190	495	660	995	1490	2485	4140
500	170	465	620	930	1395	2330	3880
600	140	435	580	875	1310	2185	3640
650	125	430	575	860	1290	2150	3580
700	110	425	565	850	1275	2125	3540
750	95	415	555	830	1245	2075	3460
800	80	405	540	805	1210	2015	3360
850	65	395	530	790	1190	1980	3300
900	50	390	520	780	1165	1945	3240
950	35	380	510	765	1145	1910	3180
1000	20	320	430	640	965	1605	2675
1050	—	310	410	615	925	1545	2570
1100	—	255	345	515	770	1285	2145
1150	—	200	265	400	595	995	1655
1200	—	155	205	310	465	770	1285
1250	—	115	150	225	340	565	945
1300	—	85	115	170	255	430	715
1350	—	60	80	125	185	310	515
1400	—	50	65	90	145	240	400
1450	—	35	45	70	105	170	285
1500	—	25	35	55	80	135	230

ASME B16.9 covers **Factory-Made Wrought Buttwelding Fittings**. It includes sizes up to NPS 24 for most fittings and up to NPS 48 for some. Pressure-temperature ratings are not provided; rather, the allowable pressure ratings for the fittings are stated to be the same as for straight seamless pipe of equivalent material (as shown by comparison of composition and mechanical properties in the respective material specifications) in accordance with the rules established in the applicable section of ASME B31. ASME B31.3 presently requires that the calculation be based on 87.5% of the wall thickness of matching straight seamless pipe.

ASME B16.11 covers **Forged Fittings, Socket-Welding and Threaded**. It includes sizes from NPS 1/8 to NPS 4. Pressure-temperature ratings are not provided; rather, the allowable pressure ratings for the fittings are stated to be the same as for straight seamless pipe of equivalent material (as shown by comparison of composition and mechanical properties in the respective material specifications) in accordance with the rules established in the applicable section of ASME B31. The correlation between fitting class and matching pipe is provided in the following table, Table 2 in ASME B16.11.

ASME B16.34 covers **Valves—Flanged, Threaded and Welding End**. It provides both dimensional information and pressure-temperature ratings. Ratings for both Standard Class and Special Class

**ASME B16.11 CORRELATION OF FITTINGS CLASS WITH
SCHEDULE NUMBER OR WALL DESIGNATION OF PIPE
FOR CALCULATION OF RATINGS**

Class Designation of Fitting	Type of Fitting	Pipe Used For Rating Basis (1)	
		Schedule No.	Wall Designation
2000	Threaded	80	XS
3000	Threaded	160	—
6000	Threaded	—	XXS
3000	Socket-Welding	80	XS
6000	Socket-Welding	160	—
9000	Socket-Welding	—	XXS

NOTE:

- (1) This table is not intended to restrict the use of pipe of thinner or thicker wall with fittings. Pipe actually used may be thinner or thicker in nominal wall than that shown in this table. When thinner pipe is used, its strength may govern the rating. When thicker pipe is used (e.g., for mechanical strength), the strength of the fitting governs the rating.

valves are provided. Standard Class valves have the same pressure-temperature ratings as provided in ASME B16.5 for flanges, with the addition of Class 4500. Special Class valves have a higher pressure-temperature rating. Since the flanges are essentially governed by ASME B16.5, flanged valves are limited to Standard Class, and to Class 2500 and less. Special Class valves are subjected to additional examination requirements. The pressure-temperature ratings provided in ASME B16.34 are the commonly used pressure classes plus Class 4500. Tables of pressure-temperature ratings are provided for groups of materials. An example for commonly used carbon steel material is provided in the following table, Table 2-1.1 in ASME B16.34.

ASME B16.47 covers Large Diameter Steel Flanges, NPS 26 through NPS 60. It provides both dimensional information and pressure-temperature ratings. The pressure-temperature ratings are the same as in ASME B16.5, with the addition of Class 75 for Series B flanges. Two flange dimensions are provided; Series A flange dimensions were based on MSS SP 44 and Series B were based on API 605.

API 600 covers Steel Gate Valves—Flanged and Butt-Welding Ends. The requirements are essentially supplementary to ASME B16.34 and the pressure-temperature ratings are the same as in ASME B16.34.

ASME B16.34 RATINGS FOR GROUP 1.1 MATERIALS

A 105 (1)(2)	A 515 Gr. 70 (1)	A 675 Gr. 70 (1)(3)(4)	A 672 Gr. B70 (1)
A 216 Gr. WCB (1)	A 516 Gr. 70 (1)(5)	A 696 Gr. C	A 672 Gr. C70 (1)
A 350 Gr. LF2 (1)	A 537 Cl. 1 (6)		

NOTES:

- (1) Upon prolonged exposure to temperatures above 800°F, the carbide phase of steel may be converted to graphite. Permissible, but not recommended for prolonged use above 800°F.
- (2) Only killed steel shall be used above 850°F.
- (3) Lead grade shall not be used where welded or in any application above 500°F.
- (4) For service temperatures above 850°F, it is recommended that killed steels containing not less than 0.10% residual silicon be used.
- (5) Not to be used over 850°F.
- (6) Not to be used over 700°F.

STANDARD CLASS

	Working Pressures by Classes, psig							
Temperature (°F)	150	300	400	600	900	1500	2500	4500
	Standard Class							
-20 to 100	285	740	990	1,480	2,220	3,705	6,170	11,110
200	260	675	900	1,350	2,025	3,375	5,625	10,120
300	230	655	875	1,315	1,970	3,280	5,470	9,845
400	200	635	845	1,270	1,900	3,170	5,280	9,505
500	170	600	800	1,200	1,795	2,995	4,990	8,980
600	140	550	730	1,095	1,640	2,735	4,560	8,210
650	125	535	715	1,075	1,610	2,685	4,475	8,055
700	110	535	710	1,065	1,600	2,665	4,440	7,990
750	95	505	670	1,010	1,510	2,520	4,200	7,560
800	80	410	550	825	1,235	2,060	3,430	6,170
850	65	270	355	535	805	1,340	2,230	4,010
900	50	170	230	345	515	860	1,430	2,570
950	35	105	140	205	310	515	860	1,545
1000	20	50	70	105	155	260	430	770

SPECIAL CLASS

	Working Pressures by Classes, psig							
Temperature (°F)	150	300	400	600	900	1500	2500	4500
	Special Class							
-20 to 100	290	750	1,000	1,500	2,250	3,750	6,250	11,250
200	290	750	1,000	1,500	2,250	3,750	6,250	11,250
300	290	750	1,000	1,500	2,250	3,750	6,250	11,250
400	290	750	1,000	1,500	2,250	3,750	6,250	11,250
500	290	750	1,000	1,500	2,250	3,750	6,250	11,250
600	275	715	950	1,425	2,140	3,565	5,940	10,690
650	270	700	935	1,400	2,100	3,495	5,825	10,485
700	265	695	925	1,390	2,080	3,470	5,780	10,405
750	240	630	840	1,260	1,890	3,150	5,250	9,450
800	200	515	685	1,030	1,545	2,570	4,285	7,715
850	130	335	445	670	1,005	1,670	2,785	5,015
900	85	215	285	430	645	1,070	1,785	3,215
950	50	130	170	260	385	645	1,070	1,930
1000	25	65	85	130	195	320	535	965

API 602 covers Compact Steel Gate Valves—Flanged, Threaded, Welding, and Extended-Body Ends. It includes sizes from NPS 1/4 to NPS 4. It provides requirements for valves with pressure-temperature ratings of Classes 150, 300, 600, 800, and 1500. All except Class 800 use the same pressure-temperature ratings as listed in ASME B16.34. The pressure-temperature ratings for Class 800 are provided in the following table, Table 2 in API 602.

MSS SP-42 covers Class 150 Corrosion-Resistant Gate, Globe, Angle and Check Valves With Flanged and Butt-weld Ends. The pressure-temperature ratings are provided in the following table, Table 4 in MSS SP-42.

API 602 PRESSURE-TEMPERATURE RATINGS FOR CLASS 800 GATE VALVES

		Material Group Number											
		1.1		1.2		1.3		1.9		1.10		1.13	
Service Temperature (1)	°F	A 105 (2)		A 350-LF3 (3)		A 352-LC2 (3)		A 352-LC3 (3)		A 352-LCB (3)		A 182-F5	
		psig	MPa	psig	MPa	psig	MPa	psig	MPa	psig	MPa	psig	MPa
-20 to 100	-29 to 38	1975	13.62	2000	13.79	1855	12.79	2000	13.79	2000	13.79	2000	13.79
200	93.5	1800	12.41	2000	13.79	1750	12.06	1900	13.10	1910	13.17	2000	13.79
300	149	1750	12.07	1940	13.38	1700	11.72	1795	12.38	1805	12.45	1940	13.38
400	204.5	1690	11.65	1880	12.96	1645	11.34	1755	12.10	1730	11.93	1880	12.96
500	260	1595	11.00	1775	12.24	1550	10.69	1710	11.79	1705	11.76	1775	12.24
600	315.5	1460	10.07	1615	11.14	1420	9.79	1615	11.14	1615	11.14	1615	11.14
650	343.5	1430	9.86	1570	10.82	1395	9.62	1570	10.82	1570	10.82	1570	10.82
700	371	1420	9.79	—	—	—	—	1515	10.45	1515	10.45	1515	10.45
750	399	1345	9.27	—	—	—	—	1420	9.79	1420	9.79	1420	9.79
800	426.5	1100	7.58	—	—	—	—	1355	9.34	1355	9.34	1355	9.34
850	454.5	715	4.93	—	—	—	—	1300	8.96	1300	8.96	1170	8.07
900	482	460	3.17	—	—	—	—	1200	8.27	1200	8.27	940	6.48
950	510	275	1.90	—	—	—	—	1005	6.93	1005	6.93	695	4.79
1000	538	140	0.96	—	—	—	—	595	4.10	715	4.93	510	3.52
1050	565.5	—	—	—	—	—	—	365	2.52	530	3.65	375	2.59
1100	593.5	—	—	—	—	—	—	255	1.76	300	2.07	275	1.90
1150	621	—	—	—	—	—	—	140	0.96	275	1.90	185	1.28
1200	649	—	—	—	—	—	—	95	0.65	145	1.00	120	0.83

GENERAL NOTES:

psig = pounds per square inch gauge; MPa = megapascals.

(1) For a material shown in this table that is acceptable for low temperature service the pressure rating for a service at any temperature below -20°F (-29°C) shall be no greater than the rating shown in this table for -20°F (-29°C-8°C).

(2) Permitted but not recommended for prolonged use above about 800°F (425°C).

(3) Not to be used over 650°F (345°C).

(4) Permitted but not recommended for prolonged use above about 1100°F (595°C).

(5) Not to be used over 1100°F (595°C).

API 602 PRESSURE-TEMPERATURE RATINGS FOR CLASS 800 GATE VALVES (CONTINUED)

		Material Group Number							
		2.1		2.2		2.3		2.5	
Service Temperature	°F	A 182-F304		A 182-F316		A 182-F304L		A 182 F347H	
		A 351-CF3 (6)	A 351-CF8	A 351-CF3M (7)	A 351-CF8M	A 182-F316L	A 182-F316L	A 351-CF8C	A 351-CF8C
	°C	psig	MPa	psig	MPa	psig	MPa	psig	MPa
-20 to 100	-29 to 38	1920	13.24	1920	13.24	1600	11.03	1920	13.24
200	93.5	1600	11.03	1655	11.41	1350	9.31	1695	11.69
300	149	1410	9.72	1495	10.31	1210	8.34	1570	10.82
400	204.5	1255	8.65	1370	9.45	1100	7.58	1480	10.20
500	260	1165	8.03	1275	8.79	1020	7.03	1380	9.51
600	315.5	1105	7.62	1205	8.31	960	6.62	1310	9.03
650	343.5	1090	7.52	1185	8.17	935	6.45	1280	8.83
700	371	1075	7.41	1150	7.93	915	6.31	1250	8.62
750	399	1060	7.31	1130	7.79	895	6.17	1230	8.48
800	426.5	1050	7.24	1105	7.62	875	6.03	1215	8.38
850	454.5	1035	7.14	1080	7.45	860	5.93	1185	8.17
900	482	1025	7.07	1050	7.24	—	—	1150	7.93
950	510	1000	6.89	1030	7.10	—	—	1030	7.10
1000	538	860	5.93	970	6.69	—	—	970	6.69
1050	565.5	825	5.69	960	6.62	—	—	960	6.62
1100	593.5	685	4.72	860	5.93	—	—	860	5.93
1150	621	520	3.58	735	5.07	—	—	735	5.07
1200	649	415	2.86	550	3.79	—	—	460	3.17
1250	676.5	295	2.03	485	3.34	—	—	330	2.28
1300	704.5	220	1.51	365	2.52	—	—	250	1.72
1350	732	165	1.14	275	1.90	—	—	180	1.24
1400	760	130	0.90	200	1.38	—	—	140	0.97
1450	788	95	0.66	155	1.07	—	—	110	0.76
1500	815.5	65	0.45	110	0.76	—	—	95	0.66

(6) Not to be used over 800 °F (425°C).

(7) Not to be used over 850 °F (455°C).

MSS SP-42 PRESSURE-TEMPERATURE RATINGS

Service Temperature (°F)	Maximum Working Pressure (psig)				
	CF8, CF3, 304	CF8M, CF3M, 316	304L, 316L	CF8C, 347	CN7M
-20 to 100	275	275	230	275	230
150	253	255	213	265	215
200	230	235	195	255	200
250	218	225	185	243	195
300	205	215	175	230	190
350	198	205	168	215	190
400	190	195	160	200	190
450	180	183	153	185	180
500	170	170	145	170	170

GENERAL NOTE:

This table gives data in U.S. customary units, the temperature being indicated in °F and the actual values being represented in metric units.

MSS SP-42 PRESSURE-TEMPERATURE RATINGS

Service Temperature (°C)	Maximum Working Pressure (bar)				
	CF8, CF3, 304	CF8M, CF3M, 316	304L, 316L	CF8C, 347	CN7M
-29 to 38	19.0	19.0	15.9	19.0	15.9
50	18.3	18.4	15.3	18.7	15.4
100	15.7	16.0	13.3	17.4	13.7
150	14.1	14.8	12.0	15.8	13.1
200	13.2	13.6	11.1	14.0	13.1
250	12.0	12.0	10.2	12.1	12.0
300	10.2	10.2	9.8	10.3	10.2
260	11.7	11.7	10.0	11.7	11.7

MSS SP-51 covers Class 150LW Corrosion-Resistant Cast Flanges and Flanged Fittings. The pressure-temperature ratings are provided in the following table, Table 1 in MSS SP-51.

I-3.3 Copper and Copper Alloys

ASME B16.15 covers Cast Bronze Threaded Fittings, Classes 125 and 250. It includes sizes from NPS 1/8 to NPS 4. The pressure-temperature ratings for fittings per this standard are provided in the following table, Table 1 in ASME B16.15.

MSS SP-51 PRESSURE-TEMPERATURE RATINGS

Temperature (°C)	Pressure Rating (Bar)	Temperature (°F)	Pressure Rating (psig)
-29 to 38	6.9	-20 to 100	100
50	6.8	200	90
100	6.1	300	80
150	5.5	400	70
200	4.9	500	60
260	4.1		

ASME B16.15 PRESSURE-TEMPERATURE RATINGS

Temperature (°F)	Class 125 (psi)	Class 250 (psi)
-20 to 150	200	400
200	190	385
250	180	365
300	165	335
350	150	300
400	125	250

ASME B16.18 covers Cast Copper Alloy Solder Joint Pressure Fittings. It also includes solder × threaded fittings. It provides both dimensional information and information on the permissible pressure-temperature. Sizes that are included range from Standard Water Tube 1/4 to 12. The pressure ratings of these fittings are required to be the same as ASTM B88 Type L Copper Water Tube, annealed. However, the pressure rating cannot exceed the rating of the solder joint, provided in the following table, Table A1 in ASME B16.18, Annex A.

ASME B16.22 covers Wrought Copper and Copper Alloy Solder Joint Pressure Fittings. It also includes solder × threaded fittings. It provides both dimensional information and pressure-temperature ratings. Sizes included range from Standard Water Tube 1/4 to 8. The pressure rating of the fittings is the lower of the rating shown in the following table, Table 1 in B16.22, and the rating of the solder joint, shown in the table on the following page from ASME B16.18.

ASME B16.22 RATED INTERNAL WORKING PRESSURE (1) FOR COPPER FITTINGS (PSI)

Standard Water Tube Size (2)	-20 to 100°F	150°F	200°F	250°F	300°F	350°F	400°F
1/4	912	775	729	729	714	608	456
3/8	779	662	623	623	610	519	389
1/2	722	613	577	577	565	481	361
5/8	631	537	505	505	495	421	316
3/4	582	495	466	466	456	388	291
1	494	420	395	395	387	330	247
1 1/4	439	373	351	351	344	293	219
1 1/2	408	347	327	327	320	272	204
2	364	309	291	291	285	242	182
2 1/2	336	285	269	269	263	224	168
3	317	270	254	254	248	211	159
3 1/2	304	258	243	243	238	202	152
4	293	249	235	235	230	196	147
5	269	229	215	215	211	179	135
6	251	213	201	201	196	167	125
8	270	230	216	216	212	180	135

NOTES:

- (1) The fitting pressure rating applies to the largest opening of the fitting.
 (2) For size designation of fittings, see Section 4 of ASME B16.22.

ASME B16.24 covers Cast Copper Alloy Pipe Flanges and Flanged Fittings, Classes 150, 300, 400, 600, 900, 1500, and 2500. It provides both dimensional information and pressure-temperature ratings. Sizes included range from NPS 1/2 to NPS 12. The pressure-temperature rating depends upon the alloy, and is provided in the following two tables, Tables 1 and 2 in ASME B16.24.

ASME B16.18 PRESSURE-TEMPERATURE RATINGS

Joining Material	Working Temperature		Maximum Working Gage Pressure									
			Size 1/8 thru 1(1)		Size 1 1/4 thru 2(1)		Size 2 1/2 thru 4(1)		Size 5 thru 8(1)		Size 10 and 12(1)	
	°F	°C	psi	bar	psi	bar	psi	bar	psi	bar	psi	bar
50-50 Tin-Lead Solder(2)	100	38	200	14	175	12	150	10	135	9	100	7
	150	66	150	10	125	9	100	7	90	6	70	5
	200	93	100	7	90	6	75	5	70	5	50	3
	250	120	85	6	75	5	50	3	45	3	40	3
95-5 Tin-Antimony Solder(3)	100	38	500	35	400	28	300	20	270	19	150	10
	150	66	400	28	350	24	275	19	250	17	150	10
	200	93	300	20	250	17	200	14	180	13	140	10
	250	120	200	14	175	12	150	10	135	9	110	8
Joining(4) Materials Melting at or above 1000° (540°C)												

Pressure-temperature ratings consistent with the materials and procedures employed

GENERAL NOTES:

1 bar = 10⁵ Pa.

(1) Standard water tube sizes.

(2) ANSI/ASTM B32 Alloy Grade 50A.

(3) ANSI/ASTM B32 Alloy Grade 95TA.

(4) These joining materials are defined as brazing alloys by the American Welding Society.

ASME B16.26 covers Cast Copper Alloy Fittings for Flared Copper Tubes. They are rated for a maximum cold water service pressure of 175 psig.

**ASME B16.24 PRESSURE-TEMPERATURE RATINGS FOR ALLOYS
C83600 AND C92200**

Service Temperature (°F)	Working Pressure (psig)			
	Class 150		Class 300	
	ASTM B 62 C83600	ASTM B 61 C92200	ASTM B 62 C83600	ASTM B 61 C92200
-20 to 150	225	225	500	500
175	220	220	480	490
200	210	215	465	475
225	205	210	445	465
250	195	205	425	450
275	190	200	410	440
300	180	195	390	425
350	165	180	350	400
400	—	170	—	375
406	150	—	—	—
450	135 (1)	160	280 (1)	350
500	—	150	—	325
550	—	140	—	300
Test Pressure	350	350	750	750

NOTE:

(1) Some codes (e.g., ASME Boiler and Pressure Vessel Code, Section I; ASME B31.1; ASME B31.5) limit the rating temperature of the indicated material to 406°F.

ASME B16.24 PRESSURE-TEMPERATURE RATINGS FOR ASTM B 148, ALLOY C95200

Service Temperature (°F)	Working Pressure (psig)						
	Class 150	Class 300	Class 400	Class 600	Class 900	Class 1500	Class 2500
-20 to 100	195	515	685	1030	1545	2575	4290
150	165	430	570	855	1285	2140	3570
200	155	400	535	800	1205	2005	3340
250	145	385	510	770	1150	1920	3200
300	140	370	495	740	1110	1850	3085
350	140	365	490	735	1100	1835	3060
400	140	365	485	725	1090	1820	3030
450	140	360	480	725	1085	1805	3010
500	140	360	480	720	1080	1800	3000
Test Pressure	300	775	1050	1550	2325	3875	6450

APPENDIX II

GUIDELINES FOR COMPUTER FLEXIBILITY ANALYSIS

Prior to initiating a computer piping flexibility analysis, the analyst should gather as much information as they can to perform the job, in an organized manner. This information includes the following.

1. Piping system operating and design conditions. The analyst should have an understanding of the various potential combinations of operating conditions that may occur. There may be various combinations of lines and vessels at different temperature conditions. For example, with parallel heat exchangers, operating with one heat exchanger in operation and hot, and the other blocked off and at an ambient temperature condition, may be one of the potential modes of operation. Note that while it is not a Code requirement that the flexibility analysis be conducted using the design temperature, it may be the requirement of the owner that it be done in that manner.
2. Determine if there are steam out or other conditions that may cause different temperatures (e.g. steam tracing temperature).
3. Standard piping classes or other information indicating the materials used to construct the piping system.
4. Dimensional information on the piping system, in the form of isometric, orthographic, or other format.
5. Information on attached equipment. This typically includes outline drawings of vessels, rotating equipment, etc. Load limits for the equipment. Operating conditions/temperatures for attached equipment to consider thermal growth. Thermal movements of fired equipment are typically provided on the equipment drawings.
6. If supports and restraints have been designed, drawings or other information to determine the types of supports, to appropriately model them.
7. If the piping system is existing, photographs, walkdown, or other means to confirm the supports used in construction. It is not unusual for the supports found in the field to differ from those shown on the original design drawings.
8. Allowable stresses. Code of construction.
9. Weights of piping components.
10. Density of fluid.
11. Insulation thickness and type (including properties such as density, if standard insulation type densities are not provided by the flexibility analysis program).
12. Information on expansion joints, if present in the system, including pressure thrust area and stiffnesses.
13. Information on refractory lining, if present in the system, to determine the effect of the refractory on the stiffness of the piping. The increase in stiffness can effect equipment loads and load sensitive components in a system (e.g. a hot walled slide valve in a refractory lined system).

14. Wind, seismic (seismic may be governed by a building code, e.g. UBC, BOCA, IBC) and other occasional loads, such as waterhammer, that require consideration.
15. Any special requirements, such as natural frequency limitations.
16. For vapor lines, information on how the hydrotest is to be conducted. Must the piping be designed to carry the weight of water during a hydrotest?

The first step in conducting a flexibility analysis is to gather this information. It is then used to construct a pipe stress isometric. The stress isometric is used to organize the information for input into the computer program. The following information should generally be on the piping stress isometric. Note that while it is certainly possible to input a pipe stress model without preparing a piping stress isometric, the chance of making errors is greatly increased.

1. A global coordinate system with the positive direction indicated for the x, y and z reference axes used in the stress analysis. The positive y axis is typically assigned to the vertical up direction. The plant north direction shown with respect to this global reference frame. It may be helpful on a project to consistently model plant north as the positive x axis and east as the positive z axis.
2. Node points used in the analysis, labeled and located. Dimensions between node points, resolved into components parallel to each of the three global axes.
3. Locations, function, and lines of action of all supports and restraints. Spring hanger stiffnesses, hot and cold load settings, and design travel. When known, spring hanger model number and manufacturer.
4. Piping design parameters such as pipe size, thickness or schedule, piping material, corrosion/erosion allowance, mechanical allowances, insulation, refractory density, valve and flange (and other piping component) weights and locations.
5. The general arrangement of equipment to which the piping is attached.

The final stress report should also contain the following information. Having this information gathered together makes it possible to check the analysis with relative ease, and provides complete documentation of the basis for the work performed.

1. Equipment outline drawings for all vessels, heat exchangers, fired heaters, rotating equipment, tankage, etc. to which the piping is attached. These drawings should include, as a minimum, basic overall dimensions, material, nozzle locations and details where piping is attached, and equipment support locations. As a minimum, references to the appropriate drawings should be provided.
2. Pressures, temperatures and content specific gravities for all load cases considered. Data on any pipe sections with high temperature gradients, thermal bowing, or transients should be included as well.
3. Expansion joint stiffness and movements (axial, transverse, and rotational), bellows dimensional information and overall expansion joint assembly weight.
4. Any dynamic load data used in the analysis.
5. Tabulations showing compliance of loads on rotating and other load sensitive equipment with their allowables.
6. Program input and output showing all analysis results for the specified load cases as well as a record of the software used, its version number, and record of its setup or configuration. This may be an electronic record.
7. Program generated plots showing principal nodes (annotated as appropriate).
8. Assumptions/supporting calculations for stress intensification factors when values other than program defaults are used.
9. Support stiffness calculations, when appropriate.
10. If the analysis is also a pressure design check, appropriate calculations for wall thickness, area replacement, etc.

The following is a checklist of considerations.

1. Is thermal movement of attached equipment included?
2. If design temperature is being used for the flexibility analysis, is the thermal movement of the attached equipment consistent?
3. Have design temperatures of the equipment been improperly used in lieu of operating temperatures? This can occur when pressure vessels are given higher design temperatures when the allowable stress does not change up to that temperature (e.g. 650F) even though the expected metal temperature may be substantially less. In that case, the design temperature has nothing to do with the expected metal temperature for determining thermal movement.
4. Is the pressure thrust in the piping properly considered in loads on attached equipment (e.g. pressure thrust typically must be added to the reported force in the piping, for conducting a nozzle analysis)?
5. Are there conditions that may cause reversing moments in the system, such that the range between these conditions must be considered in determining the stress range?
6. Has friction been included, when significant?
7. Has the effect of friction on sliding support loads been considered?
8. If friction has been included in the piping stress analysis, has an analysis been run without friction? In general, friction is not something that is relied on. The harmful effects of friction are considered, but the benefit is not. Thus, in some cases analyses should be run with and without friction, to determine which is worse.
9. Are the loads on attached equipment within the allowable limits?
10. May the flexibility of the structural steel or other supporting elements significantly effect the results?
11. Has thermal expansion of the pipe support, or equipment support, been considered?
12. Has corrosion allowance been included?
13. Is the geometry correct? Check the plots.
14. Are longitudinal stresses due to internal pressure included in the S_L calculation?
15. Are the correct valve and other component weights included?
16. Has the weight of valve actuators been included?
17. Does the flange weight include the weight of the bolting?
18. Has the effect of settlement (e.g. around tankage) been considered?
19. Have all combinations of operating conditions been considered?
20. Have environmental loads (e.g. wind, seismic, ice) been considered?
21. Has the correct minimum temperature for the location been considered in the analysis, in determining the range of temperatures the piping may be exposed to, or has a default value been used?
22. Are springs properly modeled?
23. If gaps at supports are included in the model, are adequate controls/documentation in place to assure that they will be installed in that manner in the field, and checked?
24. Has the effect of welding trunions on elbows (on their flexibility) been considered?
25. Is there a potential for elastic followup or other strain concentration condition?
26. Is radial thermal expansion of the line potentially significant (e.g. bottom supported large diameter hot lines)?
27. If the piping lifts off of supports modeled as nonlinear supports, has the lack of that support been considered in calculating the sustained stress for that operating condition?
28. Has pressure thrust of expansion joints been considered?
29. Has thermal expansion and the action of expansion joint hardware been considered?
30. Have axial load due to thermal expansion been considered in the thermal expansion stress range, if significant?

31. Have the effect of any flanges (or heavy walled components) welded adjacent to elbows been considered with respect to elbow stress intensification factors and flexibility?
32. Have the stresses in elbows and bends been considered at more than one location on the elbow or bend? These should typically be at least checked at the ends and the midpoint.
33. Has deflection between supports been considered, when necessary? Note that deflection is only calculated, reported, and plotted based on values at node points. Midspan nodes may be necessary.
34. On large diameter lines, has the effect of pipe radius on geometry been considered? Default modeling typically assumes the pipe spans between centerlines; however with short branch runs between large diameter lines, or between a large diameter line and some point of restraint, the assumption can be significantly unconservative.
35. Have sufficient means to direct the movements of a piping system been incorporated. Remember, ideal conditions do not typically exist, so floating systems without any intermediate points of fixity generally are not desirable.
36. Have any unlisted components been considered for pressure and fatigue?
37. On large D/t piping, have limitations to the SIF's in Appendix D been considered?
38. Should pressure stiffening of bends be considered in the analysis?
39. Are loads on the flanges high? Has the impact of loads on flange leakage been considered?
40. Should the change in pipe length due to internal pressure be considered? This depends upon the hoop stress, axial stress (which may be effected by the presence of expansion joints) and the material of construction (e.g. reinforced thermosetting resin piping may extend or contract due to internal pressure, depending upon the fiber wind angle).

APPENDIX III

USEFUL INFORMATION FOR FLEXIBILITY ANALYSIS

Weights of Piping Materials
Weights of Flanged Valves
Weights of Flange Bolting
Dimensions of Fittings (ASME B16.9)
Dimensions of Welding Neck Flanges (ASME B16.5)
Dimensions of Valves (ASME B16.10)
Coefficient of Thermal Expansion (ASME B31.3)

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

The tabulation of weights of standard piping materials has been arranged for convenience of selection of data that formerly consumed considerable time to develop. For special materials, the three formulae listed below for weights of tubes, weights of contents of tubes, and weights of piping insulation will be helpful.

$$\text{Weight of tube} = F \times 10.68 \times T \times (D - T) \text{ lb/ft}$$

T = wall thickness in inches

D = outside diameter in inches

F = relative weight factor

The weight of tube furnished in this piping data is based on low carbon steel weighing 0.2833 pounds per cubic inch.

Relative Weight Factor F

Aluminum	0.34
Brass	1.09
Cast Iron	0.92
Copper	1.14
Ferritic stainless steel	0.95
Austenitic stainless steel	1.02
Steel	1.00
Wrought iron	0.99

Weight of contents of a tube -

$$G \times .3405 \times (D - 2T)^2 \text{ lb/ft}$$

G = specific gravity of contents

T = tube wall thickness in inches

D = tube outside diameter in inches

The weight per foot of steel pipe is subject to the following tolerances:

SPECIFICATION		TOLERANCE	
ASTM A-53 ASTM A-120	STD WT	+5%	-5%
	XS WT	+5%	-5%
	XXS WT	+10%	-10%
ASTM A-106	SCH 10-120	+6.5%	-3.5%
	SCH 140-160	+10%	-3.5%
ASTM A-335	12" and under	+6.5%	-3.5%
	over 12"	+10%	-5%
ASTM A-312 ASTM A-376	12" and under	+6.5%	-3.5%
API 5L	All sizes	+6.5%	-3.5%

The weight of welding tees and laterals are for full size fittings. The weights of reducing fittings are approximately the same as for full size fittings.

The weights of welding reducers are for one size reduction, and are approximately correct for other reductions.

Weights of valves of the same type may vary because of individual manufacturer's designs. Listed valve weights are approximate only. Specific valve weights should be used when available.

Where specific insulation thicknesses and densities differ from those shown, refer to "Weight of Piping Insulation" formula below.

Weight of piping insulation -

$$I \times .0218 \times T \times (D + T) \text{ lb/ft}$$

I = insulation density in pounds per cubic foot

T = insulation thickness in inches

D = outside diameter of pipe in inches

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

1.313" O.D. 1" PIPE

PIPE	Schedule No.	5S	10S	40	80	160												
	Wall Designation			Std.	XS		XXS											
	Thickness — In.	0.065	0.109	.133	.179	.250	.358											
	Pipe — Lbs/Ft	0.868	1.404	1.68	2.17	2.84	3.66											
	Water — Lbs/Ft	0.478	0.409	.37	.31	.23	.12											
      	L.R. 90° Elbow	.2 .3	.4 .3	.4 .3	.4 .3	.6 .3	1.0 .3											
	S.R. 90° Elbow			.3 .2														
	L.R. 45° Elbow	.1 .2	.3 .2	.3 .2	.3 .2	.4 .2	.5 .2											
	Tee	.4 .4	.6 .4	.8 .4	.9 .4	1.1 .4	1.3 .4											
	Lateral	.7 1.1	1.2 1.1	1.7 1.1	2.5 1.1													
	Reducer	.2 .2	.4 .2	.3 .2	.4 .2	.5 .2	.5 .2											
	Cap	.1 .3	.1 .3	.3 .3	.3 .3	.4 .3	.5 .3											
	Temperature Range °F	100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200						
INSULATION	85% Magnesia Calcium Silicate	Nom. Thick., In.	1	1	1½	2	2											
		Lbs./Ft	.57	.57	.97	1.54	1.54											
	Combina- tion	Nom. Thick., In.						2½	2½	2½	3	3	3					
		Lbs/Ft						3.30	3.30	3.30	4.70	4.70	4.70					
   	Pressure Rating psi	Cast Iron		Steel								Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation. Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot. Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes. To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe. Valve weights are approximate. When possible, obtain weights from the manufacturer. Cast iron valve weights are for flanged end valves; steel weights for welding end valves. All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.						
		125	250	150	300	400	600	900	1500	2500								
	Screwed or Slip-On	2.3 1.5	4 1.5	2.5 1.5	4 1.5	5 1.5	5 1.5	12 1.5	12 1.5	15 1.5								
	Welding Neck			3 1.5	5 1.5	7 1.5	7 1.5	12 1.5	12 1.5	16 1.5								
	Lap Joint			2.5 1.5	4 1.5	5 1.5	5 1.5	12 1.5	12 1.5	15 1.5								
	Blind	2.5 1.5	5 1.5	2.5 1.5	5 1.5	5 1.5	5 1.5	12 1.5	12 1.5	15 1.5								
   	S.R. 90° Elbow						15 3.7		28 3.8									
	L.R. 90° Elbow																	
	45° Elbow						14 3.4		26 3.6									
	Tee						20 5.6		39 5.7									
    	Flanged Bonnet Gate				20 1.2		25 1.5		80 4.3									
	Flanged Bonnet Globe or Angle								84 3.5									
	Flanged Bonnet Check																	
	Pressure Seal Bonnet — Gate						31 1.7	31 1.7										
	Pressure Seal Bonnet — Globe																	

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

1.660" O.D. 1 1/4" PIPE

PIPE	Schedule No.	5S	10S	40	80	160							
	Wall Designation			Std.	XS		XXS						
	Thickness — In.	.065	.109	.140	.191	.250	.382						
	Pipe — Lbs./Ft	1.11	1.81	2.27	3.00	3.77	5.22						
	Water — Lbs./Ft	.80	.71	.65	.56	.46	.27						
WELDING FITTINGS	 L.R. 90° Elbow	.3 .3	.5 .3	.6 .3	.8 .3	1.0 .3	1.3 .3						
	 S.R. 90° Elbow			.4 .2									
	 L.R. 45° Elbow	.2 .2	.3 .2	.3 .2	.5 .2	.6 .2	.7 .2						
	 Tee	.7 .5	1.1 .5	1.6 .5	1.6 .5	1.9 .5	2.4 .5						
	 Lateral	1.1 1.2	1.9 1.2	2.4 1.2	3.8 1.2								
	 Reducer	.3 .2	.4 .2	.5 .2	.6 .2	.7 .2	.8 .2						
	 Cap	.1 .3	.1 .3	.4 .3	.4 .3	.6 .3	.6 .3						
Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
INSULATION	 85% Magnesia Calcium Silicate	Nom. Thick., In.	1	1	1 1/2	2	2	2 1/2	2 1/2	2 1/2	3	3	3
		Lbs./Ft	.65	.65	1.47	1.83	1.83	2.65	2.65	2.65	3.58	3.58	3.58
	 Combination	Nom. Thick., In.						2 1/2	2 1/2	2 1/2	3	3	3
		Lbs./Ft						3.17	3.17	3.17	5.76	5.76	5.76
FLANGES	 Screwed or Slip-On	Pressure Rating psi	Cast Iron		Steel								
			125	250	150	300	400	600	900	1500	2500		
	 Welding Neck		2.5 1.5	4.8 1.5	3.5 1.5	5 1.5	7 1.5	7 1.5	13 1.5	13 1.5	23 1.5		
	 Lap Joint				3 1.5	7 1.5	8 1.5	13 1.5	13 1.5	25 1.5			
	 Blind		2.8 1.5	5.5 1.5	3.5 1.5	4 1.5	7 1.5	7 1.5	13 1.5	13 1.5	23 1.5		
FLANGED FITTINGS	 S.R. 90° Elbow				17 3.7		18 3.8		33 3.9				
	 L.R. 90° Elbow				18 3.9								
	 45° Elbow				15 3.4		16 3.5		31 3.7				
	 Tee				23 5.6		28 5.6		49 5.9				
VALVES	 Flanged Bonnet Gate				40 4		60 4.2		97 4.6				
	 Flanged Bonnet Globe or Angle												
	 Flanged Bonnet Check				21 4								
	 Pressure Seal Bonnet — Gate							38 1.8	38 1.1				
	 Pressure Seal Bonnet — Globe												

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welded end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

1.900" O.D. 1 1/2" PIPE

PIPE	Schedule No.	5S	10S	40	80	160							
	Wall Designation			Std.	XS		XXS						
	Thickness — In.	.065	.109	.145	.200	.281	.400	.525	.650				
	Pipe — Lbs/Ft	1.27	2.09	2.72	3.63	4.86	6.41	7.71	8.68				
	Water — Lbs/Ft	1.07	.96	.88	.77	.61	.41	.25	.12				
WELDING FITTINGS	L.R. 90° Elbow	.4 .4	.8 .4	.9 .4	1.2 .4	1.5 .4	2.0 .4						
	S.R. 90° Elbow			.6 .3	.8 .3								
	L.R. 45° Elbow	.3 .2	.5 .2	.5 .2	.7 .2	.8 .2	.1 .2						
	Tee	.9 .6	1.5 .6	2 .6	2.4 .6	3.0 .6	3.7 .6						
	Lateral	1.3 1.3	2.1 1.3	3.3 1.3	5.5 1.3								
	Reducer	.3 .2	.6 .2	.6 .2	.8 .2	1.0 .2	1.2 .2						
	Cap	.1 .3	.2 .3	.4 .3	.5 .3	.7 .3	.8 .3						
INSULATION	Temperature Range °F	100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
	85% Magnesia Calcium Silicate	Nom. Thick., In.	1	1	1 1/2	2	2	2 1/2	2 1/2	2 1/2	3	3	3
	Combination	Lbs./Ft	.84	.84	1.35	2.52	2.52	3.47	3.47	3.47	4.52	4.52	4.52
		Nom. Thick., In.					2 1/2	2 1/2	2 1/2	3	3	3	
FLANGES	Pressure Rating psi												
	Screwed or Slip-On	3 1.5	6 1.5	3.5 1.5	6 1.5	9 1.5	9 1.5	19 1.5	19 1.5	30 1.5			
	Welding Neck			4.5 1.5	8 1.5	12 1.5	12 1.5	19 1.5	19 1.5	34 1.5			
	Lap Joint			3.5 1.5	6 1.5	9 1.5	9 1.5	19 1.5	19 1.5	30 1.5			
	Blind	4 1.5	6 1.5	3.5 1.5	8 1.5	10 1.5	10 1.5	19 1.5	19 1.5	31 1.5			
FLANGED FITTINGS	S.R. 90° Elbow	9 3.7		12 3.7	23 3.8		26 3.9		46 4				
	L.R. 90° Elbow	12 4		13 4	24 4								
	45° Elbow	8 3.4		11 3.4	21 3.5		23 3.5		39 3.7				
	Tee	15 5.6		20 5.6	30 5.7		37 5.8		70 6				
VALVES	Flanged Bonnet Gate	27 6.8			55 4.2		70 4.5		125 5				
	Flanged Bonnet Globe or Angle				40 4.2		45 4.2		170 5				
	Flanged Bonnet Check			30 4.1	35 4.1		40 4.2		110 4.5				
	Pressure Seal Bonnet — Gate							42 1.9	42 1.2				
	Pressure Seal Bonnet — Globe												

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation. Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

2.375" O.D. 2" PIPE

PIPE	Schedule No.	5S	10S	40	80	160							
	Wall Designation			Std.	XS		XXS						
	Thickness — In.	.065	.109	.154	.218	.343	.436	.562	.687				
	Pipe — Lbs./Ft	1.60	2.64	3.65	5.02	7.44	9.03	10.88	12.39				
	Water — Lbs./Ft	1.72	1.58	1.46	1.28	.97	.77	.53	.34				
WELDING FITTINGS	 L.R. 90° Elbow	.6 .5	1.1 .5	1.5 .5	2.1 .5	3.0 .5	4.0 .5						
	 S.R. 90° Elbow			1 .3	1.4 .3								
	 L.R. 45° Elbow	.4 .2	.6 .2	.9 .2	1.1 .2	1.6 .2	2.0 .2						
	 Tee	1.1 .6	1.8 .6	2.9 .6	3.7 .6	4.9 .6	5.7 .6						
	 Lateral	1.9 1.4	3.2 1.4	5 1.4	7.7 1.4								
	 Reducer	.4 .3	.9 .3	.9 .3	1.2 .3	1.6 .3	1.9 .3						
	 Cap	.2 .4	.3 .4	.6 .4	.7 .4	1.1 .4	1.2 .4						
	Temperature Range °F	100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
INSULATION	85% Magnesia Calcium Silicate	Nom. Thick., In.	1	1	1½	2	2	2½	2½	2½	3	3	3
		Lbs./Ft	1.01	1.01	1.71	2.53	2.53	3.48	3.48	4.42	4.42	4.42	5.59
	Combination	Nom. Thick., In.						2½	2½	2½	3	3	3
		Lbs./Ft						4.28	4.28	5.93	5.93	5.93	7.80
FLANGES	Pressure Rating psi	Cast Iron		Steel									
		125	250	150	300	400	600	900	1500	2500			
	 Screwed or Slip-On	5 1.5	7 1.5	6 1.5	9 1.5	11 1.5	11 1.5	32 1.5	32 1.5	49 1.5			
	 Welding Neck			7 1.5	11 1.5	14 1.5	14 1.5	32 1.5	32 1.5	53 1.5			
	 Lap Joint			6 1.5	9 1.5	11 1.5	11 1.5	32 1.5	32 1.5	48 1.5			
	 Blind	5 1.5	8 1.5	5 1.5	10 1.5	12 1.5	12 1.5	32 1.5	32 1.5	50 1.5			
FLANGED FITTINGS	 S.R. 90° Elbow	14 3.8	20 3.8	19 3.8	29 3.8		35 4		83 4.2				
	 L.R. 90° Elbow	16 4.1	27 4.1	22 4.1	31 4.1								
	 45° Elbow	12 3.4	18 3.5	16 3.4	24 3.5		33 3.7		73 3.9				
	 Tee	21 5.7	32 5.7	27 5.7	41 5.7		52 6		129 6.3				
	 Flanged Bonnet Gate	37 6.9	52 7.1	40 4	65 4.2		80 4.5		190 5				
VALVES	 Flanged Bonnet Globe or Angle	30 7	64 7.3	30 3.8	45 4		85 4.5		235 5.5				
	 Flanged Bonnet Check	26 7	51 7.3	35 3.8	40 4		60 4.2		300 5.8				
	 Pressure Seal Bonnet — Gate								150 2.5				
	 Pressure Seal Bonnet — Globe								165 3				

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

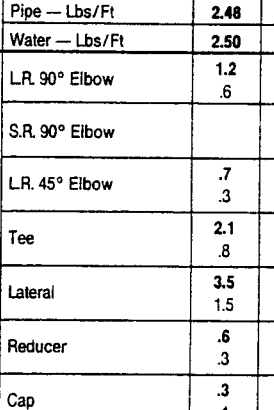
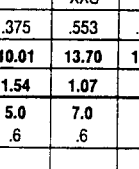
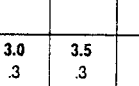
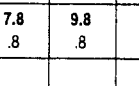
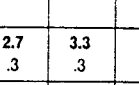
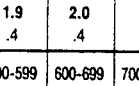
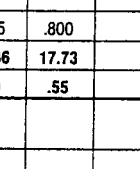
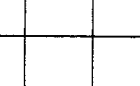
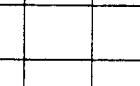
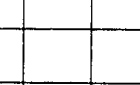
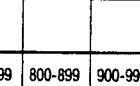
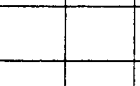
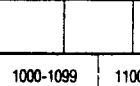
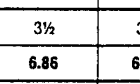
Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

2.875" O.D. 2 1/2" PIPE

PIPE	Schedule No.	5S	10S	40	80	160							
	Wall Designation			Std.	XS		XXS						
	Thickness — In.	.083	.120	.203	.276	.375	.553	.675	.800				
	Pipe — Lbs/Ft	2.48	3.53	5.79	7.66	10.01	13.70	15.86	17.73				
	Water — Lbs/Ft	2.50	2.36	2.08	1.84	1.54	1.07	.79	.55				
 WELDING FITTINGS	L.R. 90° Elbow	1.2 .6	1.8 .6	3.0 .6	3.8 .6	5.0 .6	7.0 .6						
	S.R. 90° Elbow			2.2 .4	2.5 .4								
	L.R. 45° Elbow	.7 .3	1.0 .3	1.6 .3	2.1 .3	3.0 .3	3.5 .3						
	Tee	2.1 .8	3.0 .8	5.2 .8	6.4 .8	7.8 .8	9.8 .8						
	Lateral	3.5 1.5	4.9 1.5	9.0 1.5	13 1.5								
	Reducer	.6 .3	1.2 .3	1.6 .3	2.0 .3	2.7 .3	3.3 .3						
	Cap	.3 .4	.4 .4	.9 .4	.1 .4	1.9 .4	2.0 .4						
INSULATION	Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200
	 85% Magnesia Calcium Silicate	Nom. Thick., In.	1	1	1½	2	2	2½	2½	3	3	3½	3½
		Lbs./Ft	1.14	1.14	2.29	3.23	3.23	4.28	4.28	5.46	5.46	6.86	6.86
	 Combination	Nom. Thick., In.						2½	2½	3	3	3½	3½
Lbs./Ft							5.20	5.20	7.36	7.36	9.58	9.58	
 FLANGES	Pressure Rating psi	Cast Iron			Steel								
		125	250	150	300	400	600	900	1500	2500			
	 Screwed or Slip-On	7 1.5	12.5 1.5	8 1.5	14 1.5	17 1.5	17 1.5	46 1.5	46 1.5	69 1.5			
		 Welding Neck			11 1.5	16 1.5	22 1.5	22 1.5	46 1.5	46 1.5	66 1.5		
	 Lap Joint				8 1.5	14 1.5	16 1.5	16 1.5	45 1.5	45 1.5	67 1.5		
		 Blind	7.8 1.5	10 1.5	8 1.5	16 1.5	19 1.5	19 1.5	45 1.5	45 1.5	70 1.5		
 FLANGED FITTINGS	 S.R. 90° Elbow	20 3.8	33 3.9	27 3.8	42 3.9		50 4.1		114 4.4				
		 L.R. 90° Elbow	24 4.2		30 4.2	47 4.2							
	 45° Elbow		18 3.5	31 3.6	22 3.5	35 3.6		46 3.8		99 3.9			
		 Tee	31 5.7	49 5.8	42 5.7	61 5.9		77 6.2		169 6.6			
 VALVES	 Flanged Bonnet Gate	50 7	82 7.1	60 4	100 4.2		105 4.6		275 5.2				
		 Flanged Bonnet Globe or Angle	43 7.1	87 7.4	50 4	70 4.1		120 4.6		325 5.5			
	 Flanged Bonnet Check		36 7.1	71 7.4	40 4	50 4		105 4.6		320 5.5			
		 Pressure Seal Bonnet — Gate								215 2.5			
	 Pressure Seal Bonnet — Globe									230 2.8			

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

3.500" O.D. 3" PIPE

PIPE	Schedule No.	5S	10S	40	80	160							
	Wall Designation			Std.	XS		XXS						
	Thickness — In.	.083	.120	.216	.300	.438	.600	.725	.850				
	Pipe — Lbs/Ft	3.03	4.33	7.58	10.25	14.32	18.58	21.49	24.06				
	Water — Lbs/Ft	3.78	3.61	3.20	2.86	2.35	1.80	1.43	1.10				
WELDING FITTINGS	 L.R. 90° Elbow	1.7 .8	2.5 .8	4.7 .8	6.0 .8	8.5 .8	11 .8						
	 S.R. 90° Elbow			3.3 .5	4.1 .5								
	 L.R. 45° Elbow	.9 .3	1.3 .3	2.5 .3	3.3 .3	4.5 .3	5.5 .3						
	 Tee	2.7 .8	3.9 .8	7.0 .8	10 .8	12.2 .8	14.8 .8						
	 Lateral	4.5 1.8	6.4 1.8	12.5 1.8	18 1.8								
	 Reducer	.8 .3	1.5 .3	2.1 .3	2.8 .3	3.7 .3	4.6 .3						
	 Cap	.5 .5	.7 .5	1.4 .5	1.8 .5	3.5 .5	3.6 .5						
Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
INSULATION	 85% Magnesia Calcium Silicate	Nom. Thick., In.	1	1	1½	2	2	2½	3	3	3	3½	3½
		Lbs./Ft	1.25	1.25	2.08	3.01	3.01	4.07	5.24	5.24	5.24	6.65	6.65
	 Combination	Nom. Thick., In.						2½	3	3	3	3½	3½
		Lbs./Ft						5.07	6.94	6.94	6.94	9.17	9.17
FLANGES	 Pressure Rating psi		Cast Iron		Steel								
			125	250	150	300	400	600	900	1500	2500		
	 Screwed or Slip-On	8.6 1.5	15.8 1.5	9 1.5	17 1.5	20 1.5	20 1.5	37 1.5	61 1.5	102 1.5			
	 Welding Neck			12 1.5	19 1.5	27 1.5	27 1.5	38 1.5	61 1.5	113 1.5			
	 Lap Joint			9 1.5	17 1.5	19 1.5	19 1.5	36 1.5	60 1.5	99 1.5			
	 Blind	9 1.5	17.5 1.5	10 1.5	20 1.5	24 1.5	24 1.5	38 1.5	61 1.5	105 1.5			
FLANGED FITTINGS	 S.R. 90° Elbow	25 3.9	44 4	32 3.9	53 4		67 4.1	98 4.3	150 4.6				
	 L.R. 90° Elbow	29 4.3		40 4.3	63 4.3								
	 45° Elbow	21 3.5	39 3.6	28 3.5	46 3.6		60 3.8	93 3.9	135 4				
	 Tee	38 5.9	62 6	52 5.9	81 6		102 6.2	151 6.5	238 6.9				
VALVES	 Flanged Bonnet Gate	66 7	112 7.4	70 4	125 4.4		155 4.8	260 5	410 5.5				
	 Flanged Bonnet Globe or Angle	56 7.2	87 7.6	60 4.3	95 4.5		155 4.8	225 5	495 5.5				
	 Flanged Bonnet Check	46 7.2	100 7.6	60 4.3	70 4.4		120 4.8	150 4.9	440 5.8				
	 Pressure Seal Bonnet — Gate							208 3	235 3.2				
	 Pressure Seal Bonnet — Globe							135 2.5	180 3				

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

4.000" O.D. 3 1/2" PIPE

PIPE	Schedule No.	5S	10S	40	80	160							
	Wall Designation			Std.	XS		XXS						
	Thickness — In.	.083	.120	.226	.318	.636							
	Pipe — Lbs./Ft	3.47	4.97	9.11	12.51	22.85							
	Water — Lbs./Ft	5.01	4.81	4.28	3.85	2.53							
WELDING FITTINGS	 L.R. 90° Elbow	2.4 .9	3.4 .9	6.7 .9	8.7 .9	15 .9							
	 S.R. 90° Elbow			4.2 .6	5.75 .6								
	 L.R. 45° Elbow	1.2 .4	1.7 .4	3.3 .4	4.4 .4	8 .4							
	 Tee	3.4 .9	4.9 .9	10.3 .9	13.8 .9	20.2 .9							
	 Lateral	6.2 1.8	8.9 1.8	17.2 1.8	25 1.8								
	 Reducer	1.2 .3	2.1 .3	3.0 .3	4.0 .3	6.8 .3							
	 Cap	.6 .6	.8 .6	2.1 .6	2.8 .6	5.5 .6							
Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
INSULATION	 85% Magnesia Calcium Silicate	Nom. Thick., In.	1	1	1 1/2	2	2 1/2	2 1/2	3	3	3 1/2	3 1/2	3 1/2
		Lbs./Ft	1.83	1.83	2.77	3.71	4.88	4.88	6.39	6.39	7.80	7.80	7.80
	Combina-tion	Nom. Thick., In.						2 1/2	3	3	3 1/2	3 1/2	3 1/2
		Lbs./Ft						6.49	8.71	8.71	10.8	10.8	10.8
FLANGES	 Pressure Rating psi	Cast Iron		Steel									
		125	250	150	300	400	600	900	1500	2500			
	 Screwed or Slip-On	11 1.5	20 1.5	13 1.5	21 1.5	27 1.5	27 1.5						
	 Welding Neck			14 1.5	22 1.5	32 1.5	32 1.5						
	 Lap Joint			13 1.5	21 1.5	26 1.5	26 1.5						
FLANGED FITTINGS	 Blind	13 1.5	23 1.5	15 1.5	25 1.5	35 1.5	35 1.5						
	 S.R. 90° Elbow	33 4		49 4			82 4.3						
	 L.R. 90° Elbow			54 4.4									
	 45° Elbow	29 3.6		39 3.6			75 3.6						
	 Tee	51 6	103 6.2	70 6			133 6.4						
VALVES	 Flanged Bonnet Gate	82 7.1	143 7.5	90 4.1	155 4.5		180 4.8	360 5	510 5.5				
	 Flanged Bonnet Globe or Angle	74 7.3	137 7.7				160 4.7						
	 Flanged Bonnet Check	71 7.3	125 7.7				125 4.7						
	 Pressure Seal Bonnet — Gate						140 2.5	295 2.8	380 3				
	 Pressure Seal Bonnet — Globe												

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

4.500" O.D. 4" PIPE

PIPE	Schedule No.	5S	10S		40	80	120		160				
	Wall Designation				Std.	XS				XXS			
	Thickness — In.	.083	.120	.188	.237	.337	.438	.500	.531	.674	.800	.925	
	Pipe — Lbs/Ft	3.92	5.61	8.56	10.79	14.98	18.96	21.36	22.51	27.54	31.61	35.32	
	Water — Lbs/Ft	6.40	6.17	5.80	5.51	4.98	4.48	4.16	4.02	3.38	2.86	2.39	
WELDING FITTINGS	 L.R. 90° Elbow	3.0 1	4.3 1		8.7 1	12.0 1			18.0 1	20.5 1			
	 S.R. 90° Elbow				6.7 .7	8.3 .7							
	 L.R. 45° Elbow	1.5 .4	2.2 .4		4.3 .4	5.9 .4			8.5 .4	10 .4			
	 Tee	3.9 1	5.7 1		13.5 1	16.4 1			22.8 1	26.6 1			
	 Lateral	6.6 2.1	10.0 2.1		20.5 2.1	32 2.1							
	 Reducer	1.2 .3	2.4 .3		3.6 .3	4.8 .3			6.6 .3	8.2 .3			
	 Cap	.8 .3	1.2 .3		2.5 .5	3.4 .5			6.5 6.5	6.6 6.6			
	Temperature Range °F												
	INSULATION 85% Magnesia Calcium Silicate Combina-tion												
	Nom. Thick., In.	1	1	1½	2	2½	2½	3	3	3½	3½	4	
	Lbs./Ft	1.62	1.62	2.55	3.61	4.66	4.66	6.07	6.07	7.48	7.48	9.10	
	Nom. Thick., In.						2½	3	3	3½	3½	3½	
	Lbs./Ft						6.07	8.30	8.30	10.6	10.6	10.6	
FLANGES	Pressure Rating psi	Cast Iron			Steel								
		125	250	150	300	400	600	900	1500	2500			
	 Screwed or Slip-On	14 1.5	24 1.5	15 1.5	26 1.5	32 1.5	43 1.5	66 1.5	90 1.5	158 1.5			
	 Welding Neck			17 1.5	29 1.5	41 1.5	48 1.5	648 1.5	90 1.5	177 1.5			
	 Lap Joint			15 1.5	26 1.5	31 1.5	42 1.5	64 1.5	92 1.5	153 1.5			
FLANGED FITTINGS	 Blind	16 1.5	27 1.5	19 1.5	31 1.5	39 1.5	47 1.5	67 1.5	90 1.5	164 1.5			
	 S.R. 90° Elbow	43 4.1	69 4.2	59 4.1	85 4.2	99 4.3	128 4.4	185 4.5	254 4.8				
	 L.R. 90° Elbow	50 4.5		72 4.5	98 4.5								
	 45° Elbow	38 3.7	62 3.8	51 3.7	78 3.8	82 3.9	119 4	170 4.1	214 4.2				
	 Tee	66 6.1	103 6.3	86 6.1	121 6.3	153 6.4	187 6.6	262 6.8	386 7.2				
VALVES	 Flanged Bonnet Gate	109 7.2	188 7.5	100 4.2	175 4.5	195 5	255 5.1	455 5.4	735 6				
	 Flanged Bonnet Globe or Angle	97 7.4	177 7.8	95 4.3	145 4.8	215 5	230 5.1	415 5.5	800 6				
	 Flanged Bonnet Check	80 7.4	146 7.8	80 4.3	105 4.5	160 4.8	195 5	320 5.6	780 6				
	 Pressure Seal Bonnet — Gate						215 2.8	380 3	520 4				
	 Pressure Seal Bonnet — Globe							240 2.7	290 3				

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

5.563" O.D. 5" PIPE

PIPE	Schedule No.	5S	10S	40	80	120	160						
	Wall Designation			Std.	XS			XXS					
	Thickness — In.	.109	.134	.258	.375	.500	.625	.750	.875	1.000			
	Pipe — Lbs./Ft	6.35	7.77	14.62	20.78	27.04	32.96	38.55	43.81	47.73			
	Water — Lbs./Ft	9.73	9.53	8.66	7.89	7.09	6.33	5.62	4.95	4.23			
WELDING FITTINGS	 L.R. 90° Elbow	6.0 1.3	7.4 1.3	16.0 1.3	21.4 1.3		33 1.3	34 1.3					
	 S.R. 90° Elbow	4.2 .8	5.2 .8	10.4 .8	14.5 .8								
	 L.R. 45° Elbow	3.1 .5	3.8 .5	8.3 .5	10.5 .5		14 .5	18 .5					
	 Tee	9.8 1.2	12.0 1.2	19.8 1.2	26.9 1.2		38.5 1.2	43.4 1.2					
	 Lateral	15.3 2.5	18.4 2.5	31 2.5	49 2.5								
	 Reducer	2.5 .4	4.3 .4	5.9 .4	8.3 .4		12.4 .4	14.2 .4					
	 Cap	1.3 .7	1.6 .7	4.2 .7	5.7 .7		11 .7	11 .7					
	Temperature Range °F	100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
INSULATION	 85% Magnesia Calcium Silicate	Nom. Thick., In.	1	1½	1½	2	2½	2½	3	3½	3½	4	4
		Lbs./Ft	1.86	2.92	2.92	4.08	5.38	5.38	6.90	8.41	8.41	10.4	10.4
	 Combination	Nom. Thick., In.						2½	3	3½	3½	4	4
		Lbs./Ft						7.01	9.30	11.8	11.8	14.9	14.9
FLANGES	 Pressure Rating psi		Cast Iron		Steel								
			125	250	150	300	400	600	900	1500	2500		
	 Screwed or Slip-On	17 1.5	28 1.5	18 1.5	32 1.5	37 1.5	73 1.5	100 1.5	162 1.5	259 1.5			
	 Welding Neck			22 1.5	36 1.5	49 1.5	78 1.5	103 1.5	162 1.5	293 1.5			
	 Lap Joint			18 1.5	32 1.5	35 1.5	71 1.5	98 1.5	179 1.5	253 1.5			
	 Blind	21 1.5	35 1.5	23 1.5	39 1.5	50 1.5	78 1.5	104 1.5	172 1.5	272 1.5			
FLANGED FITTINGS	 S.R. 90° Elbow	55 4.3	91 4.3	80 4.3	113 4.3	123 4.5	205 4.7	268 4.8	435 5.2				
	 L.R. 90° Elbow	65 4.7		91 4.7	128 4.7								
	 45° Elbow	48 3.8	80 3.8	66 3.8	98 3.8	123 4	180 4.2	239 4.3	350 4.5				
	 Tee	84 6.4	139 6.5	119 6.4	172 6.4	179 6.8	304 7	415 7.2	665 7.8				
VALVES	 Flanged Bonnet Gate	138 7.3	264 7.9	150 4.3	265 4.9	310 5.3	455 5.5	615 6	1340 7				
	 Flanged Bonnet Globe or Angle	138 7.6	247 8	155 4.3	215 5	355 5.2	515 5.8	555 5.8	950 6				
	 Flanged Bonnet Check	118 7.6	210 8	110 4.3	165 5	185 5	350 5.8	570 6	1150 7				
	 Pressure Seal Bonnet — Gate						350 3.1	520 3.8	865 4.5				
	 Pressure Seal Bonnet — Globe							280 4	450 4.5				

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

6.625" O.D. 6" PIPE

PIPE	Schedule No.	5S	10S		40	80	120	160				
	Wall Designation				Std.	XS			XXS			
	Thickness — In.	.109	.134	.219	.280	.432	.562	.718	.864	1.000	1.125	
	Pipe — Lbs./Ft	5.37	9.29	15.02	18.97	28.57	36.39	45.3	53.2	60.01	66.08	
	Water — Lbs./Ft	13.98	13.74	13.10	12.51	11.29	10.30	9.2	8.2	7.28	6.52	
WELDING FITTINGS	 L.R. 90° Elbow	8.9 1.5	11.0 1.5		22.8 1.5	32.2 1.5	43 1.5	55 1.6	62 1.5			
	 S.R. 90° Elbow	6.1 1	7.5 1		16.6 1	22.9 1	30 1					
	 L.R. 45° Elbow	4.5 .6	5.5 .6		11.3 .6	16.4 .6	21 .6	26 .6	30 .6			
	 Tee	13.8 1.4	17.0 1.4		31.3 1.4	39.5 1.4		59 1.4	68 1.4			
	 Lateral	16.7 2.9	20.5 2.9		42 2.9	78 2.9						
	 Reducer	3.3 .5	5.8 .5		8.6 .6	12.6 .5		18.8 .5	21.4 .5			
	 Cap	1.6 .9	1.9 .9		6.4 .9	9.2 .9	13.3 .9	17.5 .9	17.5 .9			
	Temperature Range °F	100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200
	 85% Magnesia Calcium Silicate	Nom. Thick., In.	1	1½	2	2	2½	3	3	3½	3½	4
		Lbs./Ft	2.11	3.28	4.57	4.57	6.09	7.60	7.60	9.82	9.82	11.5
	 Combination	Nom. Thick., In.						3	3	3½	3½	4
		Lbs./Ft						10.3	10.3	13.4	13.4	16.6
FLANGES	Pressure Rating psi	Cast Iron		Steel								
		125	250	150	300	400	600	900	1500	2500		
	 Screwed or Slip-On	20 1.5	38 1.5	22 1.5	45 1.5	54 1.5	95 1.5	128 1.5	202 1.5	396 1.5		
	 Welding Neck			27 1.5	48 1.5	67 1.5	96 1.5	130 1.5	202 1.5	451 1.5		
	 Lap Joint			22 1.5	45 1.5	52 1.5	93 1.5	125 1.5	208 1.5	387 1.5		
	 Blind	26 1.5	48 1.5	29 1.5	56 1.5	71 1.5	101 1.5	133 1.5	197 1.5	418 1.5		
FLANGED FITTINGS	 S.R. 90° Elbow	71 4.3	121 4.4	90 4.3	147 4.4	184 4.6	275 4.8	375 5	566 5.3			
	 L.R. 90° Elbow	88 4.9		126 4.9	182 4.9							
	 45° Elbow	63 3.8	111 3.9	82 3.8	132 3.9	149 4.1	240 4.3	320 4.3	487 4.6			
	 Tee	108 6.5	188 6.6	149 6.5	218 6.6	280 6.9	400 7.2	565 7.5	839 8			
	 Flanged Bonnet Gate	172 7.3	359 8	190 4.3	360 5	435 5.5	620 5.8	835 6	1595 7			
VALVES	 Flanged Bonnet Globe or Angle	184 7.8	345 8.2	185 4.4	275 5	415 5.3	645 5.8	765 6	1800 7			
	 Flanged Bonnet Check	154 7.8	286 8.2	150 4.8	200 5	360 5.4	445 6	800 6.4	1730 7			
	 Pressure Seal Bonnet — Gate						580 3.5	750 4	1215 5			
	 Pressure Seal Bonnet — Globe							730 4	780 5			

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

8.625" O.D. 8" PIPE

PIPE	Schedule No.	5S	10S		20	30	40	60	80	100	120	140	160
	Wall Designation						Std.		XS				
	Thickness — In.	.109	.148	.219	.250	.277	.322	.406	.500	.593	.718	.812	.906
	Pipe — Lbs/Ft	9.91	13.40	19.64	22.36	24.70	28.55	35.64	43.4	50.9	60.6	67.8	74.7
	Water — Lbs/Ft	24.07	23.59	22.90	22.48	22.18	21.69	20.79	19.8	18.8	17.6	16.7	15.8
WELDING FITTINGS	 L.R. 90° Elbow	15.4 2	21.0 2				44.9 2		70.3 2				120 2
	 S.R. 90° Elbow	6.6 1.3	14.3 1.3				34.5 1.3		50.2 1.3				
	 L.R. 45° Elbow	8.1 .8	11.0 .8				22.8 .8		32.8 .8				56 .8
	 Tee	18.4 1.8	25.0 1.8				60.2 1.8		78 1.8				120 1.8
	 Lateral	25.3 3.8	41.1 3.8				76 3.8		140 3.8				
	 Reducer	4.5 .5	7.8 .5				13.9 .5		20.4 .5				32.1 .5
	 Cap	2.1 1	2.8 1				11.3 1		16.3 1				32 1
INSULATION	Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200
	 85% Magnesia Calcium Silicate	Nom. Thick., In.	1½	1½	2	2	2½	3	3½	3½	4	4	4½
		Lbs./Ft	4.13	4.13	5.64	5.64	7.85	9.48	11.5	11.5	13.8	13.8	16.0
	 Combination	Nom. Thick., In.						3	3½	3½	4	4	4½
		Lbs./Ft						12.9	16.2	16.2	20.4	20.4	23.8
FLANGES	 Pressure Rating psi	Cast Iron		Steel									
		125	250	150	300	400	600	900	1500	2500			
	 Screwed or Slip-On	29 1.5	60 1.5	33 1.5	67 1.5	82 1.5	135 1.5	207 1.5	319 1.5	601 1.5			
	 Welding Neck			42 1.5	76 1.5	104 1.5	137 1.5	222 1.5	334 1.5	692 1.5			
	 Lap Joint			33 1.5	77 1.5	79 1.5	132 1.5	223 1.5	347 1.5	587 1.5			
FLANGED FITTINGS	 Blind	43 1.5	79 1.5	48 1.5	90 1.5	115 1.5	159 1.5	232 1.5	363 1.5	649 1.5			
	 S.R. 90° Elbow	113 4.5	194 4.7	157 4.5	238 4.7	310 5	435 5.2	639 5.4	995 5.7				
	 L.R. 90° Elbow	148 5.3		202 5.3	283 5.3								
	 45° Elbow	97 3.9	164 4	127 3.9	203 4	215 4.1	360 4.4	507 4.5	870 4.8				
	 Tee	168 6.8	289 7.1	230 6.8	337 7.1	445 7.5	610 7.8	978 8.1	1465 8.6				
VALVES	 Flanged Bonnet Gate	251 7.5	583 8.1	305 4.5	505 5.1	703 6	960 6.3	1180 6.6	2740 7				
	 Flanged Bonnet Globe or Angle	317 8.4	554 8.6	475 5.4	505 5.5	610 5.9	1130 6.3	1160 6.3	2865 7				
	 Flanged Bonnet Check	302 8.4	454 8.6	235 5.2	310 5.3	475 5.6	725 6	1140 6.4	2075 7				
	 Pressure Seal Bonnet — Gate						925 4.5	1185 4.7	2345 5.5				
	 Pressure Seal Bonnet — Globe							1550 4	1680 5				

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

10.750" O.D. 10" PIPE

PIPE	Schedule No.	5S	10S		20	30	40	60	80	100	120	140	160
	Wall Designation						Std.		XS				
	Thickness — In.	.134	.165	.219	.250	.307	.365	.500	.593	.718	.843	1.000	1.125
	Pipe — Lbs/Ft	15.15	18.70	24.63	28.04	34.24	40.5	54.7	64.3	76.9	89.2	104.1	115.7
	Water — Lbs/Ft	37.4	36.9	36.2	35.77	34.98	34.1	32.3	31.1	29.5	28.0	26.1	24.6
WELDING FITTINGS		L.R. 90° Elbow	29.2 2.5	36.0 2.5			84 2.5	112 2.5					230 2.5
		S.R. 90° Elbow	20.3 1.7	24.9 1.7			62.2 1.7	74 1.7					
		L.R. 45° Elbow	14.6 1	18.0 1			42.4 1	53.8 1					109 1
		Tee	30.0 2.1	37.0 2.1			104 2.1	132 2.1					222 2.1
		Lateral	47.5 4.4	70.0 4.4			124 4.4	200 4.4					
		Reducer	8.1 .6	14.0 .6			23.2 .6	31.4 .6					58 .6
		Cap	3.8 1.3	4.7 1.3			20 1.3	26.3 1.3					59 1.3
	INSULATION	Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099
		Nom. Thick., In.	1½	1½	2	2½	2½	3	3½	3½	4	4	4½
		Lbs./Ft	5.20	5.20	7.07	8.93	8.93	11.0	13.2	13.2	15.5	15.5	18.1
Combination		Nom. Thick., In.						3	3½	3½	4	4	4½
	Lbs/Ft						15.4	19.3	19.3	23	23	27.2	
FLANGES		Pressure Rating psi	Cast Iron		Steel								
		125	250	150	300	400	600	900	1500	2500			
		Screwed or Slip-On	45 1.5	93 1.5	50 1.5	100 1.5	117 1.5	213 1.5	293 1.5	528 1.5	1148 1.5		
		Welding Neck			59 1.5	110 1.5	152 1.5	225 1.5	316 1.5	546 1.5	1291 1.5		
		Lap Joint			50 1.5	110 1.5	138 1.5	231 1.5	325 1.5	577 1.5	1120 1.5		
FLANGED FITTINGS		Blind	66 1.5	120 1.5	77 1.5	146 1.5	181 1.5	267 1.5	338 1.5	599 1.5	1248 1.5		
		S.R. 90° Elbow	182 4.8	306 4.9	240 4.8	343 4.9	462 5.2	747 5.6	995 5.8				
		L.R. 90° Elbow	237 5.8		290 5.8	438 5.8							
		45° Elbow	152 4.1	256 4.2	185 4.1	288 4.2	332 4.3	572 4.6	732 4.7				
		Tee	277 7.2	446 7.4	353 7.2	527 7.4	578 7.8	1007 8.4	1417 8.7				
VALVES		Flanged Bonnet Gate	471 7.7	899 8.3	455 4.5	750 5	1035 6	1575 6.9	2140 7.1	3690 8			
		Flanged Bonnet Globe or Angle	541 9.1	943 9.1	485 4.5	855 5.5	1070 6	1500 6.3	2500 6.8	4160 8			
		Flanged Bonnet Check	453 9.1	751 9.1	370 6	485 6.1	605 6.3	1030 6.8	1350 7	2280 7.5			
		Pressure Seal Bonnet — Gate						1450 4.9	1860 5.5	3150 6			
		Pressure Seal Bonnet — Globe							1800 5	1910 6			

INSULATION

85% Magnesia Calcium Silicate

Combination

FLANGES









FLANGED FITTINGS







VALVES











Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

12.750" O.D. 12" PIPE

PIPE	Schedule No.	5S	10S	20	30		40		60	80	120	140	160
	Wall Designation					Std.		XS					
	Thickness — In.	.156	.180	.250	.330	.375	.406	.500	.562	.687	1.000	1.125	1.312
	Pipe — Lbs./Ft	20.99	24.20	33.38	43.8	49.6	53.5	65.4	73.2	88.5	125.5	139.7	160.3
	Water — Lbs./Ft	52.7	52.2	51.10	49.7	49.0	48.5	47.0	46.0	44.0	39.3	37.5	34.9
WELDING FITTINGS	 L.R. 90° Elbow	51.2 3	57.0 3			122 3		156 3					375 3
	 S.R. 90° Elbow	33.6 2	38.1 2			82 2		104 2					
	 L.R. 45° Elbow	25.5 1.3	29.0 1.3			60.3 1.3		78 1.3					182 1.3
	 Tee	46.7 2.5	54.0 2.5			162 2.5		180 2.5					360 2.5
	 Lateral	74.7 5.4	86.2 5.4			180 5.4		273 5.4					
	 Reducer	14.1 .7	20.9 .7			33.4 .7		43.6 .7					94 .7
	 Cap	6.2 1.5	7.1 1.5			29.5 1.5		38.1 1.5					95 1.5
	Temperature Range °F	100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
INSULATION	 85% Magnesia Calcium Silicate	Nom. Thick., In.	1½	1½	2	2½	3	3	3½	4	4	4½	4½
		Lbs./Ft	6.04	6.04	8.13	10.5	12.7	12.7	15.1	17.9	17.9	20.4	20.4
	 Combination	Nom. Thick., In.						3	3½	4	4	4½	4½
		Lbs./Ft						17.7	21.9	26.7	26.7	31.1	31.1
FLANGES	 Pressure Rating psi		Cast Iron		Steel								
			125	250	150	300	400	600	900	1500	2500		
	 Screwed or Slip-On	58 1.5	123 1.5	71 1.5	140 1.5	164 1.5	261 1.5	388 1.5	820 1.5	1611 1.5			
	 Welding Neck			87 1.5	163 1.5	212 1.5	272 1.5	434 1.5	843 1.5	1919 1.5			
	 Lap Joint			71 1.5	164 1.5	187 1.5	286 1.5	433 1.5	902 1.5	1583 1.5			
FLANGED FITTINGS	 Blind	95 1.5	165 1.5	117 1.5	209 1.5	261 1.5	341 1.5	475 1.5	928 1.5	1775 1.5			
	 S.R. 90° Elbow	257 5	430 5.2	345 5	509 5.2	669 5.5	815 5.8	1474 6.2					
	 L.R. 90° Elbow	357 6.2		485 6.2	624 6.2			1598 6.2					
	 45° Elbow	227 4.3	360 4.3	282 4.3	414 4.3	469 4.5	705 4.7	1124 4.8					
	 Tee	387 7.5	640 7.8	513 7.5	754 7.8	943 8.3	1361 8.7	1928 9.3					
VALVES	 Flanged Bonnet Gate	687 7.8	1298 8.5	635 4	1015 5	1420 5.5	2155 7	2770 7.2	4650 8				
	 Flanged Bonnet Globe or Angle	808 9.4	1200 9.5	710 5	1410 5.5								
	 Flanged Bonnet Check	674 9.4	1160 9.5	560 6	720 6.5		1410 7.2	2600 8	3370 8				
	 Pressure Seal Bonnet — Gate						1975 5.5	2560 6	4515 7				
	 Pressure Seal Bonnet — Globe												

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

14" O.D. 14" PIPE

PIPE	Schedule No.	5S	10S	10	20	30	40		60	80	120	140	160
	Wall Designation					Std.		XS					
	Thickness — In.	.156	.188	.250	.312	.375	.438	.500	.593	.750	1.093	1.250	1.406
	Pipe — Lbs/Ft	23.0	27.7	36.71	45.7	54.6	63.4	72.1	84.9	106.1	150.7	170.2	189.1
	Water — Lbs/Ft	63.7	63.1	62.06	60.92	59.7	58.7	57.5	55.9	53.2	47.5	45.0	42.6
WELDING FITTINGS	 L.R. 90° Elbow	65.6 3.5	78.0 3.5			157 3.5		200 3.5					
	 S.R. 90° Elbow	43.1 2.3	51.7 2.3			108 2.3		135 2.3					
	 L.R. 45° Elbow	32.5 1.5	39.4 1.5			80 1.5		98 1.5					
	 Tee	49.4 2.8	59.6 2.8			196 2.8		220 2.8					
	 Lateral	94.4 5.8	113 5.8			218 5.8		340 5.8					
	 Reducer	25.0 1.1	31.2 1.1			63 1.1		83 1.1					
	 Cap	7.6 1.7	9.2 1.7			35.3 1.7		45.9 1.7					
INSULATION	Temperature Range °F	100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
	85% Magnesia Calcium Silicate	Nom. Thick., In.	1½	1½	2	2½	3	3	3½	4	4	4½	4½
		Lbs./Ft	6.16	6.16	8.38	10.7	13.1	13.1	15.8	18.5	18.5	21.3	21.3
	Combination	Nom. Thick., In.					3	3½	4	4	4½	4½	
FLANGES							18.2	22.8	27.5	27.5	32.4	32.4	
	Pressure Rating psi	Cast Iron		Steel									
		125	250	150	300	400	600	900	1500	2500			
	 Screwed or Slip-On	90 1.5	184 1.5	95 1.5	195 1.5	235 1.5	318 1.5	480 1.5	1016 1.5				
	 Welding Neck			130 1.5	217 1.5	277 1.5	406 1.5	642 1.5	1241 1.5				
	 Lap Joint			119 1.5	220 1.5	254 1.5	349 1.5	477 1.5	1076 1.5				
	 Blind	125 1.5	239 1.5	141 1.5	267 1.5	354 1.5	437 1.5	574 1.5					
FLANGED FITTINGS	 S.R. 90° Elbow	360 5.3	617 5.5	497 5.3	632 5.5	664 5.7	918 5.9	1549 6.4					
	 L.R. 90° Elbow	480 6.6	767 6.6	622 6.6	772 6.6								
	 45° Elbow	280 4.3	497 4.4	377 4.3	587 4.4	638 4.6	883 4.8	1246 4.9					
	 Tee	540 8	956 8.4	683 8	968 8.3	1131 8.6	1652 8.9	2318 9.6					
VALVES	 Flanged Bonnet Gate	921 7.9	1762 8.8	905 4.9	1525 6	1920 6.3	2960 7	4170 8	6425 8.8				
	 Flanged Bonnet Globe or Angle	1171 9.9											
	 Flanged Bonnet Check	885 9.9		1010 5	1155 5.2								
	 Pressure Seal Bonnet — Gate						2620 6	3475 6.5	6380 7.5				
	 Pressure Seal Bonnet — Globe												

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

16" O.D. 16" PIPE

PIPE	Schedule No.	5S	10S	10	20	30	40	60	80	100	120	140	160
	Wall Designation					Std.		XS					
	Thickness — In.	.165	.188	.250	.312	.375	.500	.656	.843	1.031	1.218	1.438	1.593
	Pipe — Lbs/Ft	28.0	32.0	42.1	52.4	62.6	82.8	107.5	136.5	164.8	192.3	223.6	245.1
	Water — Lbs/Ft	83.5	83.0	81.8	80.5	79.1	76.5	73.4	69.7	66.1	62.6	58.6	55.9
WELDING FITTINGS	 L.R. 90° Elbow	89.8 4	102.0 4			208 4	270 4						
	 S.R. 90° Elbow	59.7 2.5	67.7 2.5			135 2.5	177 2.5						
	 L.R. 45° Elbow	44.9 1.7	51.0 1.7			104 1.7	136 1.7						
	 Tee	66.8 3.2	75.9 3.2			250 3.2	278 3.2						
	 Lateral	127.0 6.7	144.0 6.7			275 6.7	431 6.7						
	 Reducer	31.3 1.2	35.7 1.2			77 1.2	102 1.2						
	 Cap	10.1 1.8	11.5 1.8			44.3 1.8	57 1.8						
Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
INSULATION	 85% Magnesia Calcium Silicate	Nom. Thick., In.	1½	1½	2	2½	3	3	3½	4	4	4½	4½
		Lbs./Ft	6.90	6.90	9.33	12.0	14.6	14.6	17.5	20.5	20.5	23.6	23.6
	 Combination	Nom. Thick., In.						3	3½	4	4	4½	4½
		Lbs/Ft						20.3	25.2	30.7	30.7	36.0	36.0
FLANGES	 Pressure Rating psi	Cast Iron		Steel									
		125	250	150	300	400	600	900	1500	2500			
	 Screwed or Slip-On	114 1.5	233 1.5	107 1.5	262 1.5	310 1.5	442 1.5	559 1.5	1297 1.5				
	 Welding Neck			141 1.5	288 1.5	351 1.5	577 1.5	785 1.5	1597 1.5				
	 Lap Joint			142 1.5	282 1.5	337 1.5	476 1.5	588 1.5	1372 1.5				
	 Blind	174 1.5	308 1.5	184 1.5	349 1.5	455 1.5	603 1.5	719 1.5					
FLANGED FITTINGS	 S.R. 90° Elbow	484 5.5	826 5.8	656 5.5	958 5.8	1014 6	1402 6.3	1886 6.7					
	 L.R. 90° Elbow	684 7	1036 7	781 7	1058 7								
	 45° Elbow	374 4.3	696 4.6	481 4.3	708 4.6	839 4.7	1212 5	1586 5					
	 Tee	714 8.3	1263 8.7	961 8.3	1404 8.6	1671 9	2128 9.4	3054 10					
VALVES	 Flanged Bonnet Gate	1254 8	2321 9	1190 5	2015 7	2300 7.2	3675 7.9	4950 8.2	7875 9				
	 Flanged Bonnet Globe or Angle												
	 Flanged Bonnet Check	1166 10.5				1225 6							
	Pressure Seal Bonnet — Gate						3230 7			8130 8			
	Pressure Seal Bonnet — Globe												

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

18" O.D. 18" PIPE

PIPE	Schedule No.	5S	10S	10	20		30		40	60	80	120	160
	Wall Designation					Std.		XS					
	Thickness — In.	.165	.188	.250	.312	.375	.438	.500	.562	.750	.937	1.375	1.781
	Pipe — Lbs./Ft	31.0	36.0	47.4	59.0	70.6	82.1	93.5	104.8	138.2	170.8	244.1	308.5
	Water — Lbs./Ft	106.2	195.7	104.3	102.8	101.2	99.9	98.4	97	92.7	88.5	79.2	71.0
WELDING FITTINGS	 L.R. 90° Elbow	114.0 4.5	129.0 4.5			256 4.5		332 4.5					
	 S.R. 90° Elbow	75.7 2.8	85.7 2.8			176 2.8		225 2.8					
	 L.R. 45° Elbow	57.2 1.9	64.5 1.9			132 1.9		168 1.9					
	 Tee	83.2 3.6	94.7 3.6			282 3.6		351 3.6					
	 Lateral	157.0 7.5	179.0 7.5			326 7.5		525 7.5					
	 Reducer	42.6 1.3	48.5 1.3			94 1.3		123 1.3					
	 Cap	12.7 2.1	14.5 2.1			57 2.1		75 2.1					
	Temperature Range °F	100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
INSULATION	 85% Magnesia Calcium Silicate	Nom. Thick., In.	1½	1½	2	2½	3	3	3½	4	4	4½	4½
		Lbs./Ft	7.73	7.73	10.4	13.3	16.3	16.3	19.3	22.6	22.6	25.9	25.9
	 Combination	Nom. Thick., In.						3	3½	4	4	4½	4½
		Lbs./Ft						22.7	28.0	33.8	33.8	39.5	39.5
FLANGES	Pressure Rating psi	Cast Iron		Steel									
		125	250	150	300	400	600	900	1500	2500			
	 Screwed or Slip-On	125 1.5		139 1.5	331 1.5	380 1.5	573 1.5	797 1.5	1694 1.5				
	 Welding Neck			159 1.5	355 1.5	430 1.5	652 1.5	1074 1.5	2069 1.5				
	 Lap Joint			165 1.5	355 1.5	415 1.5	566 1.5	820 1.5	1769 1.5				
FLANGED FITTINGS	 Blind	209 1.5	396 1.5	228 1.5	440 1.5	572 1.5	762 1.5	1030 1.5					
	 S.R. 90° Elbow	599 5.8	1060 6	711 5.8	1126 6	1340 6.2	1793 6.6	2817 7					
	 L.R. 90° Elbow		1350 7.4	941 7.4	1426 7.4								
	 45° Elbow	439 4.4	870 4.7	521 4.4	901 4.7	1040 4.8	1543 5	2252 5.2					
	 Tee	879 8.6	1625 9	1010 8.6	1602 9	1909 9.3	2690 9.9	4327 10.5					
VALVES	 Flanged Bonnet Gate	1629 8.2	2578 9.3	1510 6	2505 6.5	3765 7	4460 7.8	6675 8.5					
	 Flanged Bonnet Globe or Angle												
	 Flanged Bonnet Check	1371 10.5											
	 Pressure Seal Bonnet — Gate						3100 5.5	3400 5.6	4200 6				
	 Pressure Seal Bonnet — Globe												

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

20" O.D. 20" PIPE

PIPE	Schedule No.	5S	10S	10	20	30	40	60	80	100	120	140	160
	Wall Designation				Std.	XS							
	Thickness — In.	.188	.218	.250	.375	.500	.593	.812	1.031	1.281	1.500	1.750	1.968
	Pipe — Lbs/Ft	40.0	46.0	52.7	78.6	104.1	122.9	166.4	208.9	256.1	296.4	341.1	379.0
	Water — Lbs/Ft	131.0	130.2	129.5	126.0	122.8	120.4	115.0	109.4	103.4	98.3	92.6	87.9
WELDING FITTINGS		160.0 5	185.0 5		322 5	438 5							
		106.0 3.4	122.0 3.4		238 3.4	278 3.4							
		80.3 2.1	92.5 2.1		160 2.1	228 2.1							
		112.0 4	130.0 4		378 4	490 4							
		228.0 8.3	265.0 8.3		396 8.3	625 8.3							
		71.6 1.7	87.6 1.7		142 1.7	186 1.7							
		17.7 2.3	20.5 2.3		71 2.3	93 2.3							
INSULATION	Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200
		Nom. Thick., in.	1½	1½	2	2½	3	3	3½	4	4	4½	4½
		Lbs./Ft	8.45	8.45	11.6	14.6	17.7	17.7	21.1	24.6	24.6	28.1	28.1
		Nom. Thick., in.						3	3½	4	4	4½	4½
		Lbs./Ft						24.7	30.7	37.0	37.0	43.1	43.1
FLANGES		Pressure Rating psi	Cast Iron		Steel								
			125	250	150	300	400	600	900	1500	2500		
			153 1.5		180 1.5	378 1.5	468 1.5	733 1.5	972 1.5	2114 1.5			
					195 1.5	431 1.5	535 1.5	811 1.5	1344 1.5	2614 1.5			
			275 1.5	487 1.5	297 1.5	545 1.5	711 1.5	976 1.5	1287 1.5				
FLANGED FITTINGS			792 6	1315 6.3	922 6	1375 6.3	1680 6.5	2314 6.9	3610 7.3				
			1132 7.8	1725 7.8	1352 7.8	1705 7.8							
			592 4.6	1055 4.8	652 4.6	1105 4.8	1330 4.9	1917 5.2	2848 5.4				
			1178 9	2022 9.5	1378 9	1908 9.5	2370 9.7	3463 10.1	5520 11				
VALVES			1934 8.3	3823 9.5	1855 6	3370 7	5700 8	5755 8					
													
			1772 11										
													
													

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

24" O.D. 24" PIPE

PIPE		Schedule No.	5S	10	20		30	40	60	80	120	140	160	
		Wall Designation			Std.	XS								
		Thickness — In.	218	250	.375	.500	.562	.687	.968	1.218	1.812	2.062	2.343	
		Pipe — Lbs/Ft	55.0	63.4	94.6	125.5	140.8	171.2	238.1	296.4	429.4	483.1	541.9	
		Water — Lbs/Ft	188.9	188.0	183.8	180.1	178.1	174.3	165.8	158.3	141.4	134.5	127.0	
      	L.R. 90° Elbow	260.0 6		500 6	578 6									
	S.R. 90° Elbow	178.0 3.7		305 3.7	404 3.7									
	L.R. 45° Elbow	130.0 2.5		252 2.5	292 2.5									
	Tee	174.0 4.9		544 4.9	607 4.9									
	Lateral	361.0 10		544 10	875 10									
	Reducer	107.0 1.7		167 1.7	220 1.7									
	Cap	28.6 2.8		102 2.8	134 2.8									
INSULATION	Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
		Nom. Thick., In.	1½	1½	2	2½	3	3	3½	4	4	4½	4½	
		Lbs./Ft	10.0	10.0	13.4	17.0	21.0	21.0	24.8	28.7	28.7	32.9	32.9	
	Combina- tion	Nom. Thick., In.						3	3½	4	4	4½	4½	
		Lbs./Ft						29.2	36.0	43.1	43.1	50.6	50.6	
   	Pressure Rating psi	Cast Iron		Steel								Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation. Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot. Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes. To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe. Valve weights are approximate. When possible, obtain weights from the manufacturer. Cast iron valve weights are for flanged end valves; steel weights for welding end valves. All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.		
		125	250	150	300	400	600	900	1500	2500				
	Screwed or Slip-On	236 1.5		245 1.5	577 1.5	676 1.5	1056 1.5	1823 1.5	3378 1.5					
	Welding Neck			295 1.5	632 1.5	777 1.5	1157 1.5	2450 1.5	4153 1.5					
	Lap Joint			295 1.5	617 1.5	752 1.5	1046 1.5	2002 1.5	3478 1.5					
	Blind	404 1.5	757 1.5	446 1.5	841 1.5	1073 1.5	1355 1.5	2442 1.5						
   	S.R. 90° Elbow	1231 6.7	2014 6.8	1871 6.7	2174 6.8	2474 7.1	3506 7.6	6155 8.1						
	L.R. 90° Elbow	1711 8.7	2644 8.7	1821 8.7	2874 8.7									
	45° Elbow	871 4.8	1604 5	1121 4.8	1634 5	1974 5.1	2831 5.5	5124 6						
	Tee	1836 10	3061 10.2	2276 10	3161 10.2	3811 10.6	5184 11.4	9387 12.1						
    	Flanged Bonnet Gate	3062 8.5	6484 9.8	2500 5	4675 7	6995 8.7	8020 9.5							
	Flanged Bonnet Globe or Angle													
	Flanged Bonnet Check	2956 12												
	Pressure Seal Bonnet — Gate													
	Pressure Seal Bonnet — Globe													

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

26" O.D. 26" PIPE

PIPE		Schedule No.			10		20																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																							
------	--	--------------	--	--	----	--	----	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

28" O.D. 28" PIPE

PIPE	Schedule No.		10		20	30							
	Wall Designation			Std.	XS								
	Thickness — In.	250	.312	.375	.500	.625	.750	.875	1.000	1.125			
	Pipe — Lbs./Ft	74.0	92.4	110.6	146.9	182.7	218.0	253.0	288.0	323.0			
	Water — Lbs./Ft	257.3	255.0	252.7	248.1	243.6	238.9	234.4	230.0	225.6			
WELDING FITTINGS		L.R. 90° Elbow			626 9	829 9							
		S.R. 90° Elbow			415 5.4	551 5.4							
		L.R. 45° Elbow			312 3.6	413 3.6							
		Tee			729 7	910 7							
		Lateral											
		Reducer			210 2.7	290 2.7							
		Cap			120 4.5	160 4.5							
INSULATION	Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200
	85% Magnesia Calcium Silicate	Nom. Thick., in.	1½	1½	2	2½	3	3½	4	4½	5	5	6
		Lbs./Ft	11.2	11.2	15.1	19.2	23.4	27.8	32.3	36.9	41.6	41.6	51.4
	Combina- tion	Nom. Thick., in.						3½	4½	5½	6	6½	7
		Lbs./Ft						39.5	55.4	72.2	80.9	89.8	99.0
FLANGES	Pressure Rating psi	Cast Iron		Steel									
		125	250	150	300	400	600	900	1500	2500			
				334 1.5	853 1.5	780 1.5	1075 1.5	1800 1.5					
				364 1.5	943 1.5	880 1.5	1175 1.5	1850 1.5					
													
				669 1.5	1408 1.5	1425 1.5	1750 1.5	2575 1.5					
FLANGED FITTINGS		S.R. 90° Elbow											
		L.R. 90° Elbow											
		45° Elbow											
		Tee											
VALVES		Flanged Bonnet Gate											
		Flanged Bonnet Globe or Angle											
		Flanged Bonnet Check											
		Pressure Seal Bonnet — Gate											
		Pressure Seal Bonnet — Globe											

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

30" O.D. 30" PIPE

PIPE	Schedule No.	5S	10 & 10S		20	30							
	Wall Designation			Std.	XS								
	Thickness — In.	.250	.312	.375	.500	.625	.750	.875	1.000	1.125			
	Pipe — Lbs./Ft	79.0	98.9	118.7	157.6	196.1	234.0	272.0	310.0	347.0			
	Water — Lbs./Ft	296.3	293.5	291.0	286.0	281.1	276.6	271.8	267.0	262.2			
WELDING FITTINGS	 L.R. 90° Elbow	478.0 10		775 10	953 10		596.0 10						
	 S.R. 90° Elbow	319.0 5.9		470 5.9	644 5.9		388.0 5.9						
	 L.R. 45° Elbow	239.0 3.9		358 3.9	475 3.9		298.0 3.9						
	 Tee			855 7.8	1065 7.8								
	 Lateral												
	 Reducer			220 3.9	315 3.9								
	 Cap			125 4.8	175 4.8								
Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
INSULATION	85% Magnesia Calcium Silicate	Nom. Thick., In.	1½	1½	2	2½	3	3½	4	4½	5	5	6
		Lbs./Ft	11.9	11.9	16.1	20.5	25.0	29.5	34.3	39.1	44.1	44.1	54.4
	Combina-tion	Nom. Thick., In.						3½	4½	5½	6	6½	7
		Lbs./Ft						42.1	58.9	76.5	85.7	95.1	104.7
FLANGES	Pressure Rating psi	Cast Iron		Steel									
		125	250	150	300	400	600	900	1500	2500			
	 Screwed or Slip-On			365 1.5	975 1.5	900 1.5	1175 1.5	2075 1.5					
	 Welding Neck			410 1.5	1095 1.5	1000 1.5	1300 1.5	2150 1.5					
	 Lap Joint												
FLANGED FITTINGS	 Blind			770 1.5	1665 1.5	1675 1.5	2000 1.5	3025 1.5					
	 S.R. 90° Elbow												
	 L.R. 90° Elbow												
	 45° Elbow												
VALVES	 Tee												
	 Flanged Bonnet Gate												
	 Flanged Bonnet Globe or Angle												
	 Flanged Bonnet Check												
	 Pressure Seal Bonnet — Gate												
	 Pressure Seal Bonnet — Globe												

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

32" O.D. 32" PIPE

PIPE	Schedule No.		10		20	30	40						
	Wall Designation			Std.	XS								
	Thickness — In.	250	.312	.375	.500	.625	.688	.750	.875	1.000	1.125		
	Pipe — Lbs/Ft	85.0	105.8	128.7	168.2	209.4	229.9	250.0	291.0	331.0	371.0		
	Water — Lbs/Ft	337.8	335.0	323.3	327.0	321.8	319.2	316.7	311.6	306.4	301.3		
WELDING FITTINGS		LR 90° Elbow			818 10.5	1090 10.5							
		SR 90° Elbow			546 6.3	722 6.3							
		LR 45° Elbow			408 4.2	541 4.2							
		Tee			991 8.4	1230 8.4							
		Lateral											
		Reducer			255 3.1	335 3.1							
		Cap			145 5.2	190 5.2							
Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
INSULATION	85% Magnesia Calcium Silicate	Nom. Thick., In.	1½	1½	2	2½	3	3½	4	4½	5	5	6
		Lbs./Ft	12.7	12.7	17.1	21.7	26.5	31.3	36.3	41.4	46.6	46.6	57.5
	Combination	Nom. Thick., In.						3½	4½	5½	6	6½	7
		Lbs./Ft						44.7	62.3	80.9	90.5	100.4	110.5
FLANGES	Pressure Rating psi	Cast Iron		Steel									
		125	250	150	300	400	600	900	1500	2500			
				476 1.5	1093 1.5	1025 1.5	1375 1.5	2500 1.5					
				516 1.5	1228 1.5	1150 1.5	1500 1.5	2575 1.5					
													
FLANGED FITTINGS		Blind			951 1.5	1978 1.5	1975 1.5	2300 1.5	3650 1.5				
		SR 90° Elbow											
		LR 90° Elbow											
		45° Elbow											
VALVES		Tee											
		Flanged Bonnet Gate											
		Flanged Bonnet Globe or Angle											
		Flanged Bonnet Check											
		Pressure Seal Bonnet — Gate											
VALVES		Pressure Seal Bonnet — Globe											

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

34" O.D. 34" PIPE

PIPE	Schedule No.		10		20	30	40						
	Wall Designation			Std.	XS								
	Thickness — In.	.250	.312	.375	.500	.625	.688	.750	.875	1.000	1.125		
	Pipe — Lbs./Ft	90.0	112.4	134.7	178.9	222.8	244.6	266.0	310.0	353.0	395.0		
	Water — Lbs./Ft	382.0	379.1	376.0	370.3	365.0	362.2	359.5	354.1	348.6	343.2		
WELDING FITTINGS	 L.R. 90° Elbow			926 11	1230 11								
	 S.R. 90° Elbow			617 5.5	817 5.5								
	 L.R. 45° Elbow			463 4.4	615 4.4								
	 Tee			1136 8.9	1420 8.9								
	 Lateral												
	 Reducer			270 3.3	355 3.3								
	 Cap			160 5.6	210 5.6								
INSULATION	Temperature Range °F	100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
	85% Magnesia												
	Calcium Silicate												
	Combina- tion												
FLANGES	Pressure Rating psi	Cast Iron		Steel									
		125	250	150	300	400	600	900	1500	2500			
	 Screwed or Slip-On			515 1.5	1281 1.5	1150 1.5	1500 1.5	2950 1.5					
	 Welding Neck			560 1.5	1406 1.5	1300 1.5	1650 1.5	3025 1.5					
	 Lap Joint												
FLANGED FITTINGS	 Blind			1085 1.5	2231 1.5	2250 1.5	2575 1.5	4275 1.5					
	 S.R. 90° Elbow												
	 L.R. 90° Elbow												
	 45° Elbow												
VALVES	 Tee												
	 Flanged Bonnet Gate												
	 Flanged Bonnet Globe or Angle												
	 Flanged Bonnet Check												
	 Pressure Seal Bonnet — Gate												
	 Pressure Seal Bonnet — Globe												

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

36" O.D. 36" PIPE

PIPE	Schedule No.			10	20	30	40												
	Wall Designation				Std.	XS													
	Thickness — In.		250	.312	.375	.500	.625	.750	.875	1.000	1.125								
	Pipe — Lbs./Ft		96.0	119.1	142.7	189.6	236.1	282.4	328.0	374.0	419.0								
	Water — Lbs./Ft		429.1	425.9	422.6	416.6	411.0	405.1	399.4	393.6	387.9								
WELDING FITTINGS		L.R. 90° Elbow			1040 12	1380 12													
		S.R. 90° Elbow			692 5	913 5													
		L.R. 45° Elbow			518 4.8	686 4.8													
		Tee			1294 9.5	1610 9.5													
		Lateral																	
		Reducer			340 3.6	360 3.6													
		Cap			175 6	235 6													
INSULATION	Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200						
		Nom. Thick., In.	1½	1½	2	2½	3	3½	4	4½	5	5	6						
		Lbs./Ft	14.2	14.2	19.2	24.2	29.5	34.8	40.3	45.9	51.7	51.7	63.5						
		Nom. Thick., In.							3½	4½	5½	6	6½	7					
Lbs./Ft								49.8	69.3	89.7	100.2	111.0	122.0						
FLANGES		Pressure Rating psi	Cast Iron		Steel								Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation. Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs./cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs./cubic foot and the outer layer at 11 lbs./cubic foot. Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes. To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe. Valve weights are approximate. When possible, obtain weights from the manufacturer. Cast iron valve weights are for flanged end valves; steel weights for welding end valves. All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.						
			125	250	150	300	400	600	900	1500	2500								
		Screwed or Slip-On			588 1.5	1485 1.5	1325 1.5	1600 1.5	3350 1.5										
		Welding Neck			628 1.5	1585 1.5	1475 1.5	1750 1.5	3450 1.5										
		Lap Joint																	
FLANGED FITTINGS		Blind			1233 1.5	2560 1.5	2525 1.5	2950 1.5	4900 1.5										
		S.R. 90° Elbow																	
		L.R. 90° Elbow																	
		45° Elbow																	
		Tee																	
VALVES		Flanged Bonnet Gate																	
		Flanged Bonnet Globe or Angle																	
		Flanged Bonnet Check																	
		Pressure Seal Bonnet — Gate																	
		Pressure Seal Bonnet — Globe																	

WEIGHT OF PIPING MATERIALS (Courtesy of Anvil International)

42" O.D. 42" PIPE

PIPE	Schedule No.			20	30	40							
	Wall Designation		Std.	XS									
	Thickness — In.	250	.375	.500	.625	.750	1.000	1.250	1.500				
	Pipe — Lbs/Ft	112.0	166.7	221.6	276.0	330.0	438.0	544.0	649.0				
	Water — Lbs/Ft	586.4	578.7	571.7	565.4	558.4	544.8	531.2	517.9				
WELDING FITTINGS	 L.R. 90° Elbow		1420 15	1880 15									
	 S.R. 90° Elbow		1079 9	1430 9									
	 L.R. 45° Elbow		707 6	937 6									
	 Tee		1870	2415									
	 Lateral												
	 Reducer		310 4.5	410 4.5									
	 Cap		230 7.5	300 7.5									
Temperature Range °F		100-199	200-299	300-399	400-499	500-599	600-699	700-799	800-899	900-999	1000-1099	1100-1200	
INSULATION	85% Magnesia Calcium Silicate	Nom. Thick., In.	1½	1½	2	2½	3	3½	4	4½	5	5	6
		Lbs./Ft	16.5	16.5	22.2	28.0	34.0	40.1	46.4	52.7	59.2	59.2	72.6
	Combina-	Nom. Thick., In.						3½	4½	5½	6	6½	7
	tion	Lbs./Ft						57.4	79.7	102.8	114.8	126.9	139.3
FLANGES	Pressure Rating psi		Cast Iron		Steel								
			125	250	150	300	400	600	900	1500	2500		
	 Screwed or Slip-On				792 1.5	1895 1.5	1759 1.5	2320 1.5					
	 Welding Neck				862 1.5	2024 1.5	1879 1.5	2414 1.5					
	 Lap Joint												
FLANGED FITTINGS	 Blind				1733 1.5	3449 1.5	3576 1.5	4419 1.5					
	 S.R. 90° Elbow												
	 L.R. 90° Elbow												
	 45° Elbow												
VALVES	 Tee												
	 Flanged Bonnet Gate												
	 Flanged Bonnet Globe or Angle												
	 Flanged Bonnet Check												
	 Pressure Seal Bonnet — Gate												
	 Pressure Seal Bonnet — Globe												

Boldface type is weight in pounds. Lightface type beneath weight is weight factor for insulation.

Insulation thicknesses and weights are based on average conditions and do not constitute a recommendation for specific thicknesses of materials. Insulation weights are based on 85% magnesia and hydrous calcium silicate at 11 lbs/cubic foot. The listed thicknesses and weights of combination covering are the sums of the inner layer of diatomaceous earth at 21 lbs/cubic foot and the outer layer at 11 lbs/cubic foot.

Insulation weights include allowances for wire, cement, canvas, bands and paint, but not special surface finishes.

To find the weight of covering on flanges, valves or fittings, multiply the weight factor by the weight per foot of covering used on straight pipe.

Valve weights are approximate. When possible, obtain weights from the manufacturer.

Cast iron valve weights are for flanged end valves; steel weights for welding end valves.

All flanged fitting, flanged valve and flange weights include the proportional weight of bolts or studs to make up all joints.

WEIGHT OF FLANGED GATE VALVES (lb)

Size (in.)	150#	300#	600#
2	46	74	84
3	76	108	160
4	110	165	300
5	155	235	
6	175	320	640
8	310	500	1080
10	455	760	1550
12	650	1020	2100
14	860	1380	
16	1120	1960	
18	1400	2450	
20	2125	3890	
24	3120	5955	

*Courtesy of Crane Valves***WEIGHT OF FLANGED CHECK (SWING) VALVES (lb)**

Size (in.)	150#	300#	600#
2	33	46	62
3	59	86	115
4	93	154	192
5	152	255	
6	165	276	495
8	275	420	780
10	440	640	1400
12	680	1000	1750
14	950	1550	
16	1225	1700	
18	1700	2200	
20	1850	2800	
24	2900	3650	

*Courtesy of Crane Valves***WEIGHT OF FLANGED GLOBE VALVES (lb)**

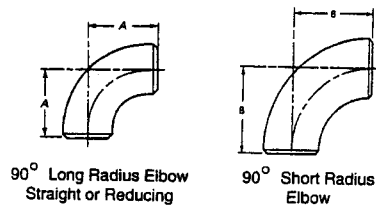
Size (in.)	150#	300#	600#
2	47	60	88
3	82	117	160
4	134	176	270
5	199	290	
6	240	340	550
8	370	530	1000
10	525	750	
12	900	1100	
14	1000		

Courtesy of Crane Valves

Bolt Weights (lb) ⁽¹⁾

Flange Size (in.)	Flange Schedule								
	Cast Iron		Steel						
	125	250	150	300	400	600	900	1500	2500
1/2	1.0		1.0	1.0		1.0		3.2	3.4
3/4	1.0		1.0	2.0		2.0		3.3	3.6
1	1.0	2.0	1.0	2.0		2.0		6.0	6.0
1 1/4	1.0	2.0	1.0	2.0		2.0		6.0	9.0
1 1/2	1.0	2.5	1.0	3.5		3.5		9.0	12.0
2	1.5	3.5	1.5	4.0		4.5		12.5	21.0
2 1/2	1.5	6.0	1.5	7.0		8.0	19.0	19.0	27.0
3	1.5	6.0	1.5	7.5		8.0	12.5	25.0	37.0
3 1/2	3.5	6.5	3.5	7.5		12.0			
4	4.0	6.5	4.0	7.5	12.0	12.5	25.0	34.0	61.0
5	6.0	6.5	6.0	8.0	12.5	19.5	33.0	60.0	98.0
6	6.0	10.0	6.0	11.5	19.0	30.0	40.0	76.0	145.0
8	6.5	16.0	6.5	18.0	30.0	40.0	69.0	121.0	232.0
10	15.0	33.0	15.0	38.0	52.0	72.0	95.0	184.0	445.0
12	15.0	44.0	15.0	49.0	69.0	91.0	124.0	306.0	622.0
14	22.0	57.0	22.0	62.0	88.0	118.0	159.0	425.0	
16	31.0	76.0	31.0	83.0	114.0	152.0	199.0	570.0	
18	41.0	93.0	41.0	101.0	139.0	193.0	299.0	770.0	
20	52.0	95.0	52.0	105.0	180.0	242.0	361.0	1010.0	
24	71.0	174.0	71.0	174.0	274.0	360.0	687.0	1560.0	

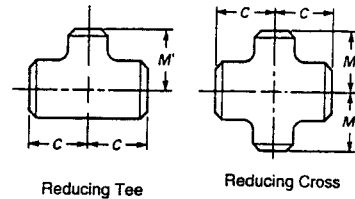
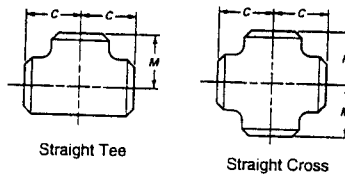
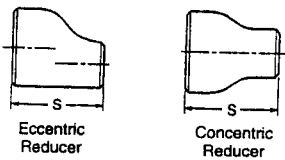
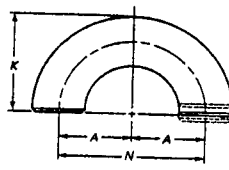
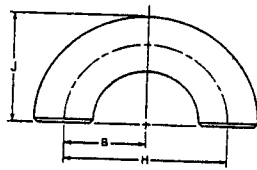
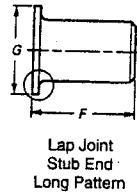
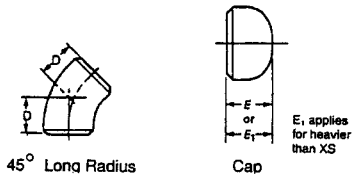
⁽¹⁾ Weights are for one complete flanged joint and include bolts and nuts.



Steel Butt-Welding Fitting Dimensions

These dimensions are per
ASME B16.9 – 2001 edition.

(all dimensions in inches)



Dimension "M" is shown in table below; refer to large table for dimension "C"

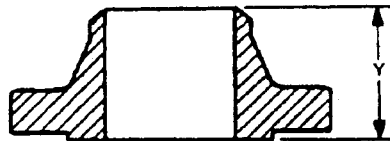
Size			Size		
a	b	M'	a	b	M'
1	x 3/4	1.50	6	x 3 1/2	5.00
1 1/4	x 1	1.88		x 3	4.88
	x 3/4	1.88		x 2 1/2	4.75
1 1/2	x 1 1/4	2.25	8	x 6	6.62
	x 1	2.25		x 5	6.38
	x 3/4	2.25		x 4	6.12
2	x 1 1/2	2.38		x 3 1/2	6.00
	x 1 1/4	2.25	10	x 8	8.00
	x 1	2.00		x 6	7.62
	x 3/4	1.75		x 5	7.50
2 1/2	x 2	2.75		x 4	7.25
	x 1 1/2	2.62	12	x 10	9.50
	x 1 1/4	2.50		x 8	9.00
	x 1	2.25		x 6	8.62
3	x 2 1/2	3.25		x 5	8.50
	x 2	3.00	14	x 12	10.62
	x 1 1/2	2.88		x 10	10.12
	x 1 1/4	2.75		x 8	9.75
3 1/2	x 3	3.62		x 6	9.38
	x 2 1/2	3.50	16	x 14	12.00
	x 2	3.25		x 12	11.62
	x 1 1/2	3.12		x 10	11.12
4	x 3 1/2	4.00		x 8	10.75
	x 3	3.88	18	x 16	13.00
	x 2 1/2	3.75		x 14	13.00
	x 2	3.50		x 12	12.62
	x 1 1/2	3.38		x 10	12.12
5	x 4	4.62	20	x 18	14.50
	x 3 1/2	4.50		x 16	14.00
	x 3	4.38		x 14	14.00
	x 2 1/2	4.25		x 12	13.62
	x 2	4.12	24	x 20	17.00
6	x 5	5.38		x 18	16.50
	x 4	5.12		x 16	16.00

Size	A	B	D	E	E'	F	G	H	J	K	M	N	S*	Pipe Schedule Numbers for:	
														Std. Ftg.	Extra Strong
1/2	1.50		0.62	1.00	1.00	3.00	1.38			1.88	1.00	3.00		40	80
3/4	1.50		0.75	1.00	1.00	3.00	1.38			2.00	1.12	3.00	1.50	40	80
1	1.50	1.00	0.88	1.50	1.50	4.00	2.00	2.00	1.62	2.19	1.50	3.00	2.00	40	80
1 1/4	1.88	1.25	1.00	1.50	1.50	4.00	2.50	2.50	2.06	2.75	1.88	3.75	2.00	40	80
1 1/2	2.25	1.50	1.12	1.50	1.50	4.00	2.88	3.00	2.44	3.25	2.25	4.50	2.50	40	80
2	3.00	2.00	1.38	1.50	1.75	6.00	3.62	4.00	3.19	4.19	2.50	6.00	3.00	40	80
2 1/2	3.75	2.50	1.75	1.50	2.00	6.00	4.12	5.00	3.94	5.19	3.00	7.50	3.50	40	80
3	4.50	3.00	2.00	2.00	2.50	6.00	5.00	6.00	4.75	6.25	3.38	9.00	3.50	40	80
3 1/2	5.25	3.40	2.25	2.50	3.00	6.00	5.50	7.00	5.50	7.25	3.75	10.50	4.00	40	80
4	6.00	4.00	2.50	2.50	3.00	6.00	5.19	8.00	6.25	8.25	4.12	12.00	4.00	40	80
5	7.50	5.00	3.12	3.00	3.50	8.00	7.31	10.00	7.75	10.31	4.88	15.00	5.00	40	80
6	9.00	6.00	3.75	3.50	4.00	8.00	8.50	12.00	9.31	12.31	5.62	18.00	5.50	40	80
8	12.00	8.00	5.00	4.00	5.00	8.00	10.62	16.00	12.31	16.31	7.00	24.00	6.00	40	80
10	15.00	10.00	6.25	5.00	6.00	10.00	12.75	20.00	15.38	20.38	8.50	30.00	7.00	40	80
12	18.00	12.00	7.50	6.00	7.00	10.00	15.00	24.00	18.38	24.38	10.00	36.00	8.00	40	80
14	21.00	14.00	8.75	6.50	7.50	12.00	16.25	28.00	21.00	28.00	11.00	42.00	13.00	30	
16	24.00	16.00	10.00	7.00	8.00	12.00	18.50	32.00	24.00	32.00	12.00	48.00	14.00	30	
18	27.00	18.00	11.25	8.00	9.00	12.00	21.00	36.00	27.00	36.00	13.50	54.00	15.00		40
20	30.00	20.00	12.50	9.00	10.00	12.00	23.00	40.00	30.00	40.00	15.00	60.00	20.00	20	30
22	33.00	22.00	13.50	10.00	10.00	12.00	25.25	44.00	33.00	44.00	16.50	66.00	20.00		
24	36.00	24.00	15.00	10.50	12.00	12.00	27.25	48.00	36.00	48.00	17.00	72.00	20.00	20	

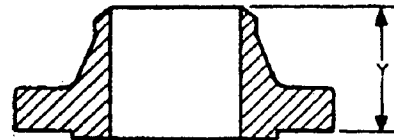
* Size is that of large end.

**Welding Neck Flange Overall Length, Y (in.)
(ASME B16.5, 1998 Edition)**

NPS	Flange Class						
	150	300	400	600	900	1500	2500
½	1.88	2.06		2.06		2.38	2.88
¾	2.06	2.25		2.25		2.75	3.12
1	2.19	2.44		2.44		2.88	3.50
1 ¼	2.25	2.56		2.62		2.88	3.75
1 ½	2.44	2.69		2.75		3.25	4.38
2	2.50	2.75		2.88		4.00	5.00
2 ½	2.75	3.00		3.12		4.12	5.62
3	2.75	3.12		3.25	4.00	4.62	6.62
3 ½	2.81	3.19		3.38			
4	3.00	3.38	3.50	4.00	4.50	4.88	7.50
5	3.50	3.88	4.00	4.50	5.00	6.12	9.00
6	3.50	3.88	4.06	4.62	5.50	6.75	10.75
8	4.00	4.38	4.62	5.25	6.38	8.38	12.50
10	4.00	4.62	4.88	6.00	7.25	10.00	16.50
12	4.50	5.12	5.38	6.12	7.88	11.12	18.25
14	5.00	5.62	5.88	6.50	8.38	11.75	
16	5.00	5.75	6.00	7.00	8.50	12.25	
18	5.50	6.25	6.50	7.25	9.00	12.88	
20	5.69	6.38	6.62	7.50	9.75	14.00	
24	6.00	6.62	6.88	8.00	11.50	16.00	



Class 150 and 300



Class 400 - 2500

Gate Valves ⁽¹⁾
Solid Wedge, Double Disc, and Conduit
End-to-End Dimensions "A" Per ASME B16.10 (1992 Rev)
Flanged and Buttwelding Ends

Nominal Size		125/150		250/300 ⁽²⁾	600		900		1500		2500	
NPS	DN	Flanged End ⁽³⁾	Butt Weld ⁽⁵⁾		Long Pattern ⁽³⁾	Short Pattern ⁽⁴⁾	Long Pattern ⁽³⁾	Short Pattern ⁽⁴⁾	Long Pattern ⁽³⁾	Short Pattern ⁽⁴⁾	Long Pattern ⁽³⁾	Short Pattern ⁽⁴⁾
1/4	8	4.00	4.00									
3/8	10	4.00	4.00									
1/2	15	4.25	4.25	5.50	6.50						10.38	
3/4	20	4.62	4.62	6.00	7.50						10.75	
1	25	5.00	5.00	6.50	8.50	5.25	10.00	5.50	10.00	5.50	12.12	7.31
1 1/4	32	5.50	5.50	7.00	9.00	5.75	11.00	6.50	11.00	6.50	13.75	9.12
1 1/2	40	6.50	6.50	7.50	9.50	6.00	12.00	7.00	12.00	7.00	15.12	9.12
2	50	7.00	8.50	8.50	11.50	7.00	14.50	8.50	14.50	8.50	17.75	11.00
2 1/2	65	7.50	9.50	9.50	13.00	8.50	16.50	10.00	16.50	10.00	20.00	13.00
3	80	8.00	11.12	11.12	14.00	10.00	15.00	12.00	18.50	12.00	22.75	14.50
4	100	9.00	12.00	12.00	17.00	12.00	18.00	14.00	21.50	16.00	26.50	18.00
5	125	10.00	15.00	15.00	20.00	15.00	22.00	17.00	26.50	19.00	31.25	21.00
6	150	10.50	15.88	15.88	22.00	18.00	24.00	20.00	27.75	22.00	36.00	24.00
8	200	11.50	16.50	16.50	26.00	23.00	29.00	26.00	32.75	28.00	40.25	30.00
10	250	13.00	18.00	18.00	31.00	28.00	33.00	31.00	39.00	34.00	50.00	36.00
12	300	14.00	19.75	19.75	33.00	32.00	38.00	36.00	44.50	39.00	56.00	41.00
14	350	15.00	22.50	22.50 (30.00)	35.00	35.00	40.50	39.00	49.50	42.00		44.00
16	400	16.00	24.00	24.00 (33.00)	39.00	39.00	44.50	43.00	54.50	47.00		49.00
18	450	17.00	26.00	26.00 (36.00)	43.00	43.00	48.00		60.50	53.00		55.00
20	500	18.00	28.00	28.00 (39.00)	47.00	47.00	52.00		65.50	58.00		
22	550	[20.00]	30.00	43.00	51.00							
24	600	20.00	32.00	31.00 (45.00)	55.00	55.00	61.00		76.50			
26	650	22.00	34.00	49.00	57.00							
28	700	24.00	36.00	53.00	61.00							
30	750	24.00 [26.00]	36.00	55.00	65.00							
32	800	[28.00]	38.00	60.00	70.00							
34	850	[30.00]	40.00	64.00	76.00							
36	900	28.00 [32.00]	40.00	68.00	82.00							

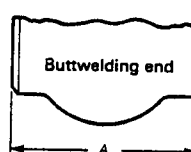
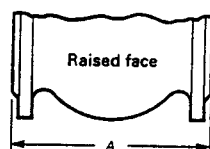
⁽¹⁾ Not all valve types are available in all sizes, refer to valve manufacturer information and/or ASME B16.10.

⁽²⁾ Steel Valves in parenthesis, when different. Conduit gate valves in brackets "[]" when different.

⁽³⁾ Flanged and Buttwelding, Long Pattern.

⁽⁴⁾ Buttwelding, Short Pattern.

⁽⁵⁾ Steel, Class 150 only.



Globe, Lift Check and Swing Check Valves ⁽¹⁾
End-to-End Dimensions "A" Per ASME B16.10 (1992 Rev)
Flanged and Buttwelding Ends

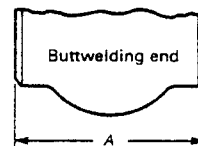
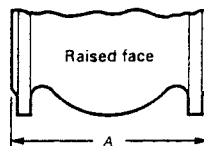
Nominal Size		125/150 ⁽²⁾	250/300 ⁽²⁾	600		900		1500		2500	
NPS	DN			Long Pattern ⁽³⁾	Short Pattern ⁽⁴⁾	Long Pattern ⁽³⁾	Short Pattern ⁽⁴⁾	Long Pattern ⁽³⁾	Short Pattern ⁽⁴⁾	Long Pattern ⁽³⁾	Short Pattern ⁽⁴⁾
1/4	8	4.00									
3/8	10	4.00									
1/2	15	4.25	6.00	6.50				8.50		10.38	
3/4	20	4.62	7.00	7.50		9.00		9.00		10.75	
1	25	5.00	8.00	8.50	5.25	10.00		10.00		12.12	
1 1/4	32	5.50	8.50	9.00	5.75	11.00		11.00		13.75	
1 1/2	40	6.50	9.00	9.50	6.00	12.00		12.00		15.12	
2	50	8.00	10.50	11.50	7.00	14.50		14.50	8.50	17.75	11.00
2 1/2	65	8.50	11.50	13.00	8.50	16.50	10.00	16.50	10.00	20.00	13.00
3	80	9.50	12.50	14.00	10.00	15.00	12.00	18.50	12.00	22.75	14.50
4	100	11.50	14.00	17.00	12.00	18.00	14.00	21.50	16.00	26.50	18.00
5	125	13.00 (14.00)	15.75	20.00	15.00	22.00	17.00	26.50	19.00	31.25	21.00
6	150	14.00 (16.00)	17.50	22.00	18.00	24.00	20.00	27.75	22.00	36.00	24.00
8	200	19.50	21.00 (22.00)	26.00	23.00	29.00	26.00	32.75	28.00	40.25	30.00
10	250	24.50	24.50	31.00	28.00	33.00	31.00	39.00	34.00	50.00	36.00
12	300	27.50	28.00	33.00	32.00	38.00	36.00	44.50	39.00	56.00	41.00
14	350	31.00 (21.00)		35.00		40.50	39.00	49.50	42.00		
16	400	36.00 [34.00]		39.00		44.50	43.00	54.50	47.00		
18	450	38.50		43.00		48.00		60.50			
20	500	38.50		47.00		52.00		65.50			
22	550	42.00		51.00							
24	600	51.00		55.00		61.00		76.50			
26	650	51.00		57.00							
28	700	57.00		63.00							
30	750	60.00		65.00							
36	900	77.00		82.00							

⁽¹⁾ Not all valve types are available in all sizes, refer to valve manufacturer information and/or ASME B16.10.

⁽²⁾ Steel Valves in parenthesis, when different. Swing check steel valve in brackets "[]" when different.

⁽³⁾ Flanged and Buttwelding, Long Pattern.

⁽⁴⁾ Buttwelding, Short Pattern.



Angle and Lift Check ⁽¹⁾
End-to-End Dimensions "D" Per ASME B16.10 (1992 Rev)
Flanged and Buttwelding Ends

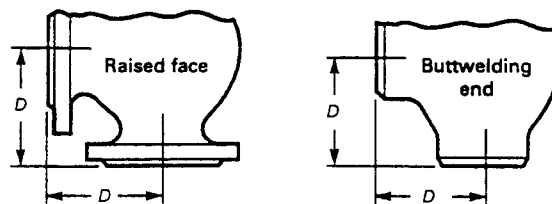
Nominal Size		125/150 ⁽²⁾	250/300 ⁽²⁾	600		900		1500 Long Pattern ⁽³⁾	2500 Long Pattern ⁽³⁾
NPS	DN			Long Pattern ⁽³⁾	Short Pattern ⁽⁴⁾	Long Pattern ⁽³⁾	Short Pattern ⁽⁴⁾		
1/4	8	2.00							
3/8	10	2.00							
1/2	15	2.25	3.00	3.25				4.25	5.19
3/4	20	2.50	3.50	3.75		4.50		4.50	5.38
1	25	2.75	4.00	4.25		5.00		5.00	6.06
1 1/4	32	3.00	4.25	4.50		5.50		5.50	6.88
1 1/2	40	3.25	4.50	4.75		6.00		6.00	7.56
2	50	4.00	5.25	5.75	4.25	7.25		7.25	8.88
2 1/2	65	4.25	5.75	6.50	5.00	8.25		8.25	10.00
3	80	4.75	6.25	7.00	6.00	7.50	6.00	9.25	11.38
4	100	5.75	7.00	8.50	7.00	9.00	7.00	10.75	13.25
5	125	6.50 (7.00)	7.88	10.00	8.50	11.00	8.50	13.25	15.62
6	150	7.00 (8.00)	8.75	11.00	10.00	12.00	10.00	13.88	18.00
8	200	9.75	10.50 (11.00)	13.00		14.50	13.00	16.38	20.12
10	250	12.25	12.25	15.50		16.50	15.50	19.50	25.00
12	300	13.75	14.00	16.50		19.00	18.00	22.25	28.00
14	350	15.50				20.25	19.50	24.75	
16	400	18.00				26.00			
18	450					29.00			
20	500					32.50			
22	550								
24	600					39.00			

⁽¹⁾Not all valve types are available in all sizes, refer to valve manufacturer information and/or ASME B16.10.

⁽²⁾Steel Valves in parenthesis, when different.

⁽³⁾Flanged and Buttwelding, Long Pattern.

⁽⁴⁾Buttwelding, Short Pattern. Dimensions apply to pressure seal or flangeless bonnet valves. They may be applied at the manufacturer's option to valves with flanged bonnets.



Total Thermal Expansion, U.S. Units, for Metals
Total Linear Thermal Expansion Between 70°F and Indicated Temperature, in./100 ft
Per ASME B31.3 – 1999 Edition

Temp., ° F	Material					Temp., ° F	Material				
	Carbon Steel Carbon-Moly- Low-Chrome (Through 3Cr-Mo)	5Cr-Mo Through 9Cr-Mo	Austenitic Stainless Steels 18 Cr-8Ni	UNS No8XXX Series Ni-Fe-Cr	UNS No6XXX Series Ni-Cr-Fe		Carbon Steel Carbon-Moly- Low-Chrome (Through 3Cr-Mo)	5Cr-Mo Through 9Cr-Mo	Austenitic Stainless Steels 18 Cr-8Ni	UNS No8XXX Series Ni-Fe-Cr	UNS No6XXX Series Ni-Cr-Fe
-325	-2.37	-2.22	-3.85			575	4.35	4.02	5.93	5.44	4.77
-300	-2.24	-2.10	-3.63			600	4.60	4.24	6.24	5.72	5.02
-275	-2.11	-1.98	-3.41			625	4.86	4.47	6.55	6.01	5.27
-250	-1.98	-1.86	-3.19			650	5.11	4.69	6.87	6.30	5.53
-225	-1.85	-1.74	-2.96			675	5.37	4.92	7.18	6.58	5.79
-200	-1.71	-1.62	-2.73			700	5.63	5.14	7.50	6.88	6.05
-175	-1.58	-1.50	-2.50			725	5.90	5.38	7.82	7.17	6.31
-150	-1.45	-1.37	-2.27			750	6.16	5.62	8.15	7.47	6.57
-125	-1.30	-1.23	-2.01			775	6.43	5.86	8.47	7.76	6.84
-100	-1.15	-1.08	-1.75			800	6.70	6.10	8.80	8.06	7.10
-75	-1.00	-0.94	-1.50			825	6.97	6.34	9.13	8.35	
-50	-0.84	-0.79	-1.24			850	7.25	6.59	9.46	8.66	
-25	-0.68	-0.63	-0.98			875	7.53	6.83	9.79	8.95	
0	-0.49	-0.46	-0.72			900	7.81	7.07	10.12	9.26	
25	-0.32	-0.30	-0.46			925	8.08	7.31	10.46	9.56	
50	-0.14	-0.13	-0.21			950	8.35	7.56	10.80	9.87	
70	0	0	0	0	0	975	8.62	7.81	11.14	10.18	
100	0.23	0.22	0.34	0.28	0.26	1000	8.89	8.06	11.48	10.49	
125	0.42	0.40	0.62	0.52	0.48	1025	9.17	8.30	11.82	10.80	
150	0.61	0.58	0.90	0.76	0.70	1050	9.46	8.55	12.16	11.11	
175	0.80	0.76	1.18	0.99	0.92	1075	9.75	8.80	12.50	11.42	
200	0.99	0.94	1.46	1.23	1.15	1100	10.04	9.05	12.84	11.74	
225	1.21	1.13	1.75	1.49	1.38	1125	10.31	9.28	13.18	12.05	
250	1.40	1.33	2.03	1.76	1.61	1150	10.57	9.52	13.52	12.38	
275	1.61	1.52	2.32	2.03	1.85	1175	10.83	9.76	13.86	12.69	
300	1.82	1.71	2.61	2.30	2.09	1200	11.10	10.00	14.20	13.02	
325	2.04	1.90	2.90	2.59	2.32	1225	11.38	10.26	14.54	13.36	
350	2.26	2.10	3.20	2.88	2.56	1250	11.66	10.53	14.88	13.71	
375	2.48	2.30	3.50	3.18	2.80	1275	11.94	10.79	15.22	14.04	
400	2.70	2.50	3.80	3.48	3.05	1300	12.22	11.06	15.56	14.39	
425	2.93	2.72	4.10	3.76	3.29	1325	12.50	11.30	15.90	14.74	
450	3.16	2.93	4.41	4.04	3.53	1350	12.78	11.55	16.24	15.10	
475	3.39	3.14	4.71	4.31	3.78	1375	13.06	11.80	16.58	15.44	
500	3.62	3.35	5.01	4.59	4.02	1400	13.34	12.05	16.92	15.80	
525	3.86	3.58	5.31	4.87	4.27	1425			17.30	16.16	
550	4.11	3.80	5.62	5.16	4.52	1450			17.69	16.53	
						1475			18.08	16.88	
						1500			18.47	17.25	

APPENDIX IV

A PRACTICAL GUIDE TO EXPANSION JOINTS

This appendix is largely a copy of the publication, "A Practical Guide to Expansion Joints" by the Expansion Joint Manufacturers Association, Inc. 25 North Broadway, Tarrytown, NY 10591. It is reproduced, courtesy of EJMA.

IV-1 WHAT ARE EXPANSION JOINTS

IV-1.1 Definition of an Expansion Joint

For purposes of this publication, the definition of an expansion joint is "any device containing one or more metal bellows used to absorb dimensional changes such as those caused by thermal expansion or contraction of a pipe-line, duct, or vessel."

IV-1.2 Expansion Devices (Fig. IV-1.1)

Bellows expansion joints provide an alternative to pipe expansion loops and slip-type joints.

IV-1.3 Manufacturing Methods

There are several methods of manufacturing bellows, the two most common bellows types are:

(1) **FORMED BELLOWS** – (Fig. IV-1.2.1) Whether formed hydraulically or mechanically, this is by far the most common type available today. Formed from a thin-walled tube, a formed bellows contains only longitudinal welds, and exhibits significant flexibility. Formed bellows of varying shapes are made from single or multiple plies of suitable material (usually stainless steel) ranging in thickness from 0.004" to 0.125" and greater, and in diameters of $\frac{3}{4}$ " to over 12 ft. These bellows are usually categorized according to convolution shape (Fig. IV-1.2.1 and IV-1.4).

(2) **FABRICATED BELLOWS** – Made by welding together a series of thin gage diaphragms or discs (Fig. IV-1.2.2 and IV-1.4). Fabricated bellows are usually made of heavier gage material than formed bellows and are therefore able to withstand higher pressures.

Due to the variations in manufacturing and design configurations, the number, height, and pitch of the convolutions should never be specified in the purchase of an expansion joint as a condition of design, but should be left to the decision of the manufacturer. These dimensions will vary from one manufacturer to another and in particular will depend on the specific design conditions and the manufacturer's own rating procedure which has been established on the basis of laboratory tests and field experience.

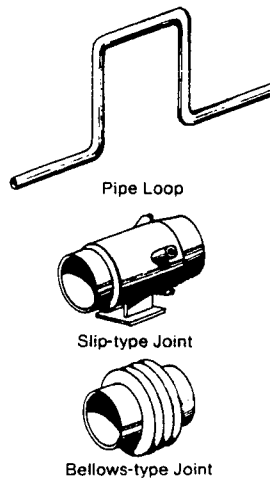


FIG. IV-1.1
EXPANSION DEVICES

IV-1.4 Design Variables

If an expansion joint is to fulfill its intended function safely and reliably, it must be kept in mind that it is a highly specialized product. Interchangeability is rare in expansion joints, and in a real sense, each unit is custom-made for an intended application. It becomes necessary then, to supply the expansion joint manufacturer with accurate information regarding the conditions of design that the expansion joint will be subjected to in service. Because many of these design conditions interact with each other, designing a bellows becomes much like assembling a jigsaw puzzle; one needs all of the pieces (design conditions) before a clear picture of an expansion joint design can emerge. The following is a listing of the basic design conditions that should be supplied to the manufacturer when specifying an expansion joint.

IV-1.4.1 Size. Size refers to the diameter of the pipeline (or dimensions of the duct in the case of rectangular joints) into which the expansion joint is to be installed. The size of an expansion joint affects its pressure-retaining capabilities, as well as its ability to absorb certain types of movements.

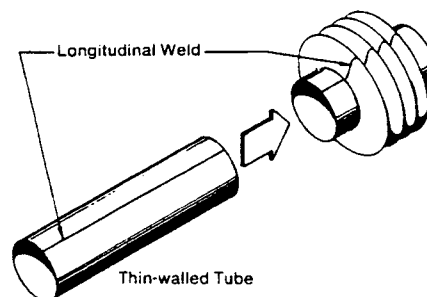


FIG. IV-1.2.1
FORMED BELLOWS

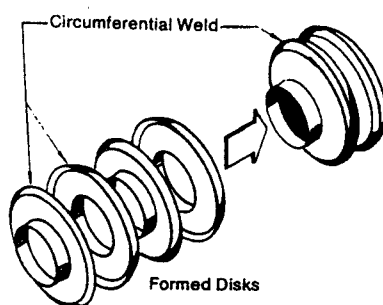


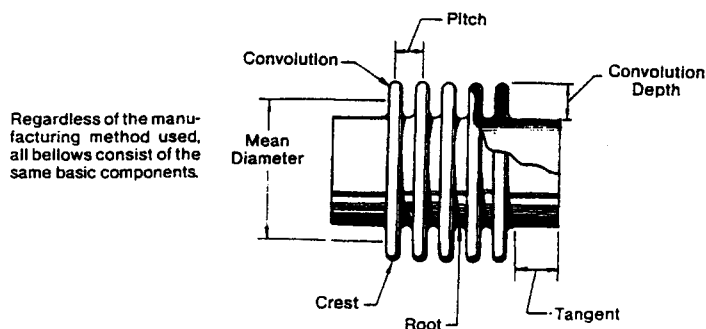
FIG. IV-1.2.2
FABRICATED BELLOWS

Note: When specifying pipe sizes, misunderstandings often result due to the confusing array of nominal pipe sizes (NPS) and different pipe schedules. To eliminate any misconceptions, pipe sizes should be supplied to the expansion joint manufacturer in terms of the actual outside diameter and the actual wall thickness of the pipe. The outside diameters, schedules, and wall thickness of pipe sizes up thru 36 inches can be found in the American Society of Mechanical Engineers (ASME B36.10).

IV-1.4.2 Flowing Medium. The substances that will come in contact with the expansion joint should be specified. In some cases, due to excessive erosion, or corrosion potential, or in cases of high viscosity, special materials and accessories should be specified. When piping systems containing expansion joints are cleaned periodically, the cleaning solution must be compatible with the bellows materials.

IV-1.4.3 Pressure. Pressure is possibly the most important factor determining expansion joint design. Minimum and maximum anticipated pressure should be accurately determined. If a pressure test is to be performed, this pressure should be specified as well. While the determination of pressure requirements is important, care should be exercised to insure that these specified pressures are not increased by unreasonable safety factors as this could result in a design which may not adequately satisfy other performance characteristics.

IV-1.4.4 Temperature. The operating temperature of the expansion joint will affect its pressure capacity, allowable stresses, cycle life, and material requirements. All possible temperature sources should be investigated when determining minimum and maximum temperature requirements. In so



Regardless of the manufacturing method used, all bellows consist of the same basic components.

FIG. IV-1.3
BELLOWS COMPONENTS

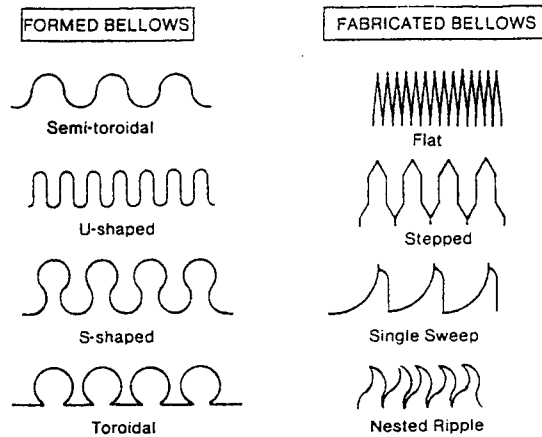


FIG. IV-1.4
CONVOLUTION SHAPES

doing, however, it is important that only those temperatures occurring at the expansion joint location itself be specified. Specifying temperatures remote from the expansion joint may unnecessarily result in the need for special materials at additional expense.

IV-1.4.5 Motion. Movements due to temperature changes or mechanical motion to which the expansion joint will be subjected must be specified. (Methods for determining movements are presented in Section IV-2.) In addition, other extraneous movements, such as wind loading or installation misalignment must be considered. The various dimensional changes which an expansion joint is required to absorb, such as those resulting from thermal changes in a piping system, are as follows (Fig. IV-1.5):

Axial Motion is motion occurring parallel to the center line of the bellows and can be either extension or compression.

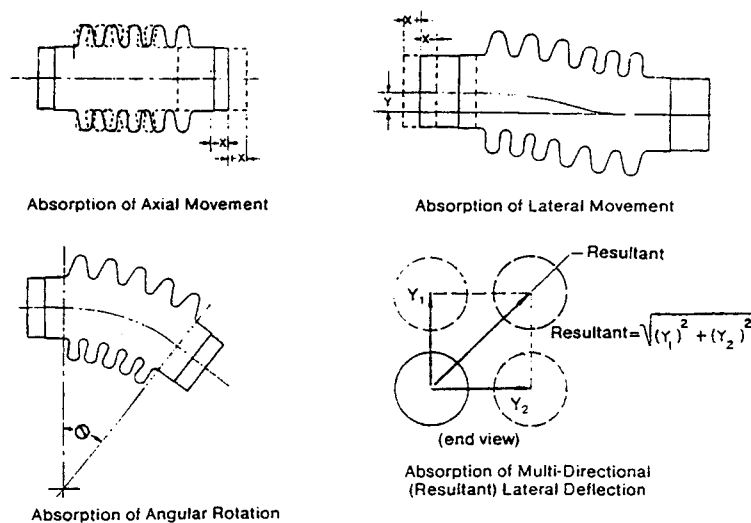


FIG. IV-1.5
TYPES OF EXPANSION JOINT MOVEMENT

In addition to Axial, Lateral, and Angular Movements, an expansion joint may be subjected to Torsional motion, or twisting. Torsion imposes severe stresses on the expansion joint and all such cases should be referred to the manufacturer.

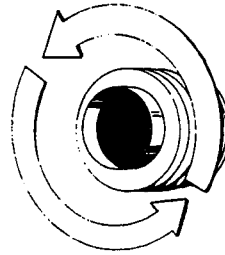


FIG. IV-1.6
TORSIONAL MOVEMENT

Lateral Deflection, or offset, is motion which occurs perpendicular, or at right angles to the centerline of the bellows. Lateral deflection can occur along one or more axis simultaneously.

Angular Rotation is the bending of an expansion joint along its centerline.

IV-1.5 Expansion Joint Accessories

The basic unit of every expansion joint is the bellows. By adding additional components, expansion joints of increasing complexity and capability are created which are suitable for a wide range of applications.

Refer to Figure IV-1.7 and the following list for a description of the basic expansion joint accessories.

1. **BELLOWS** – The flexible element of an expansion joint, consisting of one or more convolutions, formed from thin material.
2. **LINER** – A device which minimizes the effect on the inner surface of the bellows by the fluid flowing through it. Liners are primarily used in high velocity applications to prevent erosion of the inner surface of the bellows and to minimize the likelihood of flow induced vibration. Liners come in single, tapered, or telescoping configurations according to the application requirements. An expansion joint, if provided with liners, must be installed in the proper orientation with respect to flow direction. Liners are sometimes referred to as internal sleeves.
3. **COVER** – A device used to provide external protection to the bellows from foreign objects, mechanical damage, and/or external flow. The use of a cover is strongly recommended to all applications. A cover is sometimes referred to as a shroud.
4. **WELD END** – The ends of an expansion joint equipped with pipe for weld attachment to adjacent equipment or piping. Weld ends are commonly supplied beveled for butt welding.

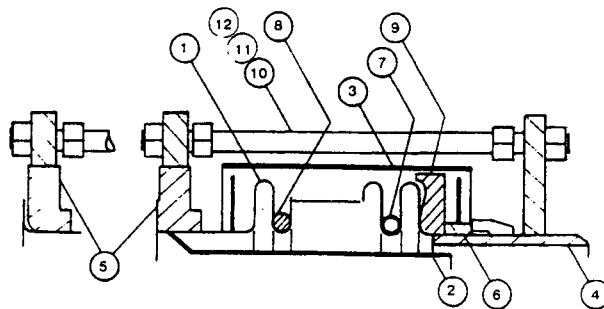


FIG. IV-1.7
EXPANSION JOINT ACCESSORIES

5. **FLANGED END** – The ends of an expansion joint equipped with flanges for the purpose of bolting the expansion joint to the mating flanges of adjacent equipment or piping.
6. **COLLAR** – A ring of suitable thickness which is used to reinforce the bellows tangent, or cuff, from bulging due to pressure.
7. **HOLLOW REINFORCING RINGS** – Devices used on some expansion joints, fitting snugly in the roots of the convolutions. The primary purpose of these devices is to reinforce the bellows against internal pressure. Hollow rings are usually formed from a suitable pipe or tubing section.
8. **SOLID ROOT RINGS** – Identical in function to hollow root rings, but formed from solid bar stock for greater strength.
9. **EQUALIZING RINGS** – “T” Shaped in cross-section, these rings are made of cast iron, steel, stainless steel or other suitable alloys. In addition to resisting internal pressure, equalizing rings limit the amount of compression movement per convolution.
10. **CONTROL RODS** – Devices, usually in the form of rods or bars, attached to the expansion joint assembly whose primary function is to distribute the applied movement between the two bellows of a universal expansion joint. Control rods are NOT designed to restrain bellows pressure thrust.
11. **LIMIT RODS** – Devices, usually in the form of rods or bars, attached to the expansion joint assembly whose primary function is to restrict the bellows movement range (axial, lateral and angular) during normal operation. In the event of a main anchor failure, limit rods are designed to prevent bellows over-extension or over-compression while restraining the full pressure loading and dynamic forces generated by the anchor failure.
12. **TIE RODS** – Devices, usually in the form of rods or bars, attached to the expansion joint assembly whose primary function is to continuously restrain the full bellows pressure thrust during normal operation while permitting only lateral deflection. Angular rotation can be accommodated only if two tie rods are used and located 90° from the direction of rotation.
13. **PANTOGRAPHIC LINKAGES** – A scissors-like device. A special form of control rod attached to the expansion joint assembly whose primary function is to positively distribute the movement equally between the two bellows of the universal joint throughout its full range of movement. Pantographic linkages, like control rods, are NOT designed to restrain pressure thrust.

IV-1.6 Types of Expansion Joints

There are several different types of expansion joints. Each is designed to operate under a specific set of design conditions. The following is a listing of the most basic types of expansion joint designs, along with a brief description of their features and application requirements. A more in-depth discussion of specific expansion joint applications appears in Section IV-2.2.

All of the following expansion joint types are available in both circular and rectangular models

- (1) **SINGLE EXPANSION JOINT.** The simplest form of expansion joint; of single bellows construction, it absorbs all of the movement of the pipe section into which it is installed.
- (2) **DOUBLE EXPANSION JOINT.** A double expansion joint consists of two bellows jointed by a common connector which is anchored to some rigid part of the installation by means of an anchor base. The anchor base may be attached to the common connector either at installation or time of manufacturer. Each bellows of a double expansion joint functions independently as a single unit. Double bellows expansion joints should not be confused with universal expansion joints.
- (3) **UNIVERSAL EXPANSION JOINT.** A universal expansion joint is one containing two bellows joined by a common connector for the purpose of absorbing any combination of the three (3)

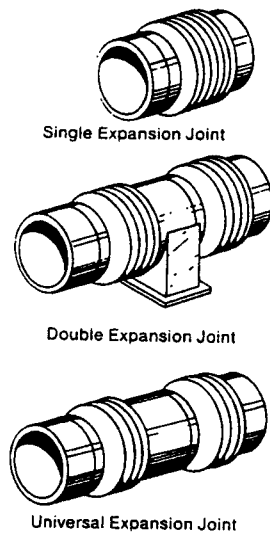


FIG. IV-1.8
EXPANSION JOINT TYPES

basic movements. (Section IV-1.4.5) A universal expansion joint is used in cases to accommodate greater amounts of lateral movement than can be absorbed by a single expansion joint.

- (4) **UNIVERSAL TIED EXPANSION JOINTS.** Tied universal expansion joints are used when it is necessary for the assembly to eliminate pressure thrust forces from the piping system. In this case the expansion joint will absorb lateral movement and will not absorb any axial movement external to the tied length.

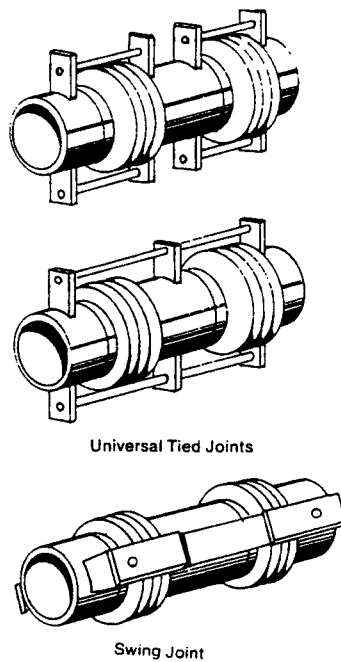
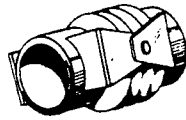
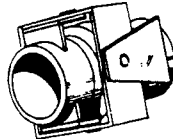


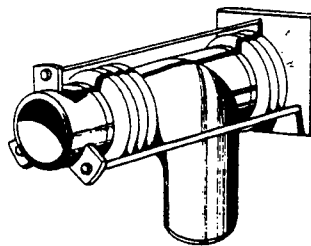
FIG. IV-1.8.1
EXPANSION JOINT TYPES (CONT.)



Hinged Expansion Joint



Gimbal Expansion Joint



Pressure-Balanced Expansion Joint

FIG. IV-1.8.2
EXPANSION JOINT TYPES (CONT.)

- (5) **SWING EXPANSION JOINT.** A swing expansion joint is designed to absorb lateral deflection and/or angular rotation in one plane only by the use of swing bars, each of which is pinned at or near the ends of the unit.
- (6) **HINGED EXPANSION JOINT.** A hinged expansion joint contains one bellows and is designed to permit angular rotation in one plane only by the use of a pair of pins running through plates attached to the expansion joint ends. Hinged expansion joints should be used in sets of 2 or 3 to function properly.
- (7) **GIMBAL EXPANSION JOINT.** A gimbal expansion joint is designed to permit angular rotation in any plane by the use of 2 pairs of hinges affixed to a common floating gimbal ring.
 NOTE: Expansion joints Types 4, 5, 6, and 7 are normally used for restraint of pressure thrust forces. (Section IV-2.1.3.2)
- (8) **PRESSURE-BALANCED EXPANSION JOINT.** A pressure-balanced expansion joint is designed to absorb axial movement and/or lateral deflection while restraining the bellows pressure thrust force (Section IV-2.1.3.2) by means of the devices interconnecting the flow bellows with an opposed bellows also subjected to line pressure. This type of joint is installed where a change of direction occurs in a run of pipe.

IV-2 USING EXPANSION JOINTS

IV-2.1 System Preparation

When thermal growth of a piping system has been determined to be a problem, and the use of an expansion joint is indicated, the most effective expansion joint should be selected as a solution. When using expansion joints it should be kept in mind that movements are not eliminated, but are only

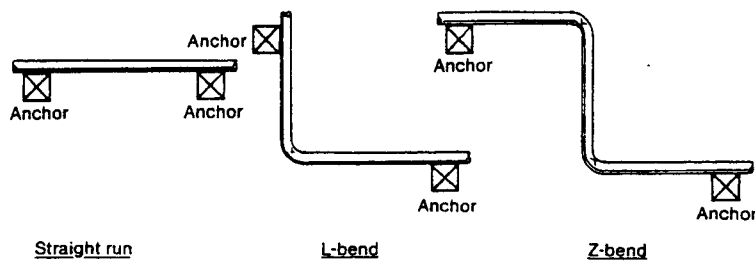


FIG. IV-2.1
STANDARD PIPE SECTIONS

directed to certain points where they can best be absorbed by the expansion joint. The methods used and the steps that must be taken in preparing a piping system for the introduction of an expansion joint are covered in this section. As will be seen, many factors must be considered to insure the proper functioning of an expansion joint.

IV-2.1.1 Simplify the System. The first step in the selection of expansion joints is to choose the tentative location of the pipe anchors. The purpose of these anchors is to divide a piping system into simple, individual expanding sections. Since thermal growth cannot be restrained, it becomes the function of the pipe anchors to limit and control the amount of movement that the expansion joints, installed between them, must absorb. It is generally advisable to begin with the assumption that using single or double expansion joints in straight axial movement will provide the most simple and economical layout. *Never install more than one 'single' type joint between any two anchors in a straight run of pipe.*

In general, piping systems should be anchored so that a complex system is divided into several sections which conform to one of the configurations shown in Fig. IV-2.1.

Major pieces of equipment such as turbines, pumps, compressors, etc. may function as anchors provided they are designed for this use. Fig. IV-2.2 illustrates a hypothetical piping system in which expansion joints are to be installed. The following sections will describe expansion joint application methods as applied to this system.

IV-2.1.2 Calculating Thermal Growth. Step 1 in determining thermal movement is to choose tentative anchor positions. The goal here is to divide a system into several straight-line sections exhibiting only axial movement. Referring to Fig. IV-2.3 demonstrates this method.

The pump and two tanks are logical anchor positions. The addition of anchors at pts. B and C result in the desired series of straight-line piping legs. The next step is to determine the actual change in length of each of the piping legs due to changes in temperature.

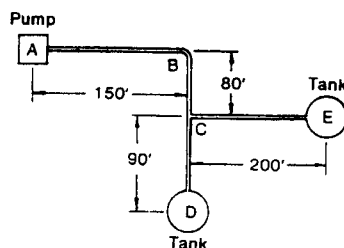


FIG. IV-2.2
TYPICAL PIPING LAYOUT

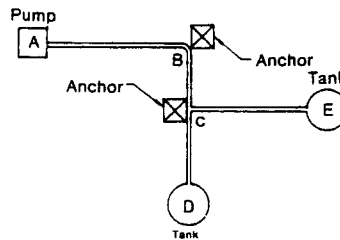


FIG. IV-2.3
ANCHOR POSITIONS

In determining thermal movement, all potential sources of temperature must be considered. Usually, the temperature of the flow medium is the major source of dimensional changes, but in certain extreme cases, ambient temperature outside the system can contribute to thermal movement. Movements caused by other sources such as mechanical movements or wind loading, must also be considered.

IV-2.1.3 Pipe Anchors and Forces. Pipe anchors, their attachments, and the structures to which they are attached, must be designed to withstand all of the forces acting upon them. In addition to the normal pipe system forces, anchors in systems which contain expansion joints are subjected to additional forces. Two significant forces which are unique to expansion joint systems are spring force and pressure thrust force.

IV-2.1.3.1 Spring Force

Spring force is the force required to deflect an expansion joint a specific amount. It is based on operating conditions and manufacturing methods and materials. Spring force imparts a resisting force throughout the system much as a spring would when compressed or otherwise deflected. (Fig. IV-2.4) In order for an expansion joint to operate properly, this spring force must be restrained by anchors.

The magnitude of spring force is determined by the expansion joint's spring rate, and by the amount of movement to which the expansion joint is subjected. The spring rate varies for each expansion joint and is based upon the physical dimensions and material of a specific expansion joint.

A simple example will illustrate the magnitude of spring force for an expansion joint installed in a 24 inch diameter pipe system. The axial spring rate is given as: 1568 lbs./in. If the expansion joint deflects $\frac{1}{2}$ in. axially, the spring force is:

$$\text{Spring Force} = 1568 \times .5 = 784 \text{ lbs.}$$

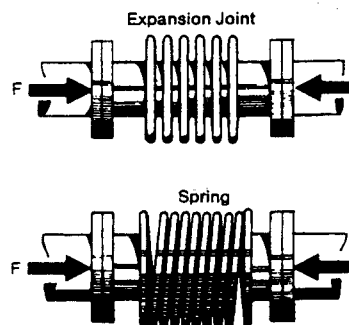


FIG. IV-2.4
SPRING FORCE

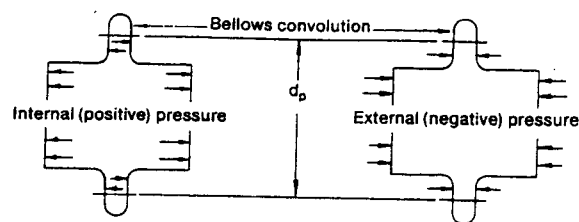


FIG. IV-2.5
PRESSURE THRUST

IV-2.1.3.2 Pressure Thrust Force

Pressure thrust force is often misunderstood. It is a condition created by the installation of a flexible unit, such as an expansion joint, into a rigid piping system which is under pressure. Pressure thrust force is a function of the system pressure and mean diameter (Fig. IV-2.5) of the bellows. Mean diameter (d_p) is usually greater than the pipe diameter. Fig IV-2.5 illustrates the effect of pressure on a bellows convolution. In cases of internal, or positive pressure, the convolutions are pushed apart, causing the bellows to extend or increase in length while the opposite is observed in cases of external or negative pressure. The force required to maintain the bellows at it's proper length is equal to this pressure thrust and can be significantly higher than all other system forces combined. See Fig. (IV-2.6) for further details of pressure thrust effects.

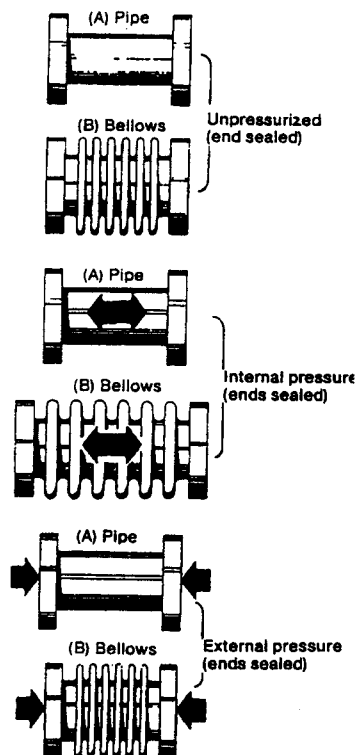


FIG. IV-2.6
PRESSURE THRUST EFFECTS

IV-2.1.3.3 Determining Pressure Thrust Force

The magnitude of pressure thrust force (F_s) in lbs., is determined by the following equation:

$$F_s = Pa$$

where

(P) is the pressure (psig) and

(a) is the effective area of the expansion joint (in^2)

The following example will illustrate the effect of pressure thrust. Suppose that an expansion joint is to be installed in a 24 in. diameter pipe system which operates a 150 psig pressure. From the manufacturer of the expansion joint the effective area is found to be 560 in^2 .

$$a = \frac{\pi (d_p)^2}{4}$$

The pressure thrust force (F_s) is found to be:

$$F_s = Pa \text{ or } F_s = (150)(560) = 84,000 \text{ lbs.}$$

Note that when determining pressure thrust forces, the pressure value used in $F_s = Pa$ should equal the maximum anticipated pressure that the system is likely to experience. For this reason, any pressure test must be considered. Be careful not to specify unrealistic safety factors which could lead to over-design of the anchors at additional cost.

The force is transmitted from the ends of the expansion joint along the pipe. Referring to Fig. IV-2.6, a comparison of the effect of pressure thrust is shown between a section of rigid pipe (A) and a flexible bellows (B).

In an unpressurized state, there is no pressure thrust force transmitted by A or B and their lengths remain the same.

Under internal pressure, a force is acting upon both A and B that is equal to the pressure times the effective areas of A & B. The force of this thrust is totally absorbed by the rigid walls of the pipe (A) and its length remains unchanged. The length of (B) will tend to increase under internal pressure since the flexible expansion joint is unable to resist the force. It therefore yields, or stretches.

In cases of external pressure or vacuum conditions, the effect of internal pressure is reversed and the expansion joint will tend to compress.

To stop the expansion joint from extending or compressing due to pressure thrust, main anchors must be located at some point on each side of the joint to withstand the forces of pressure thrust and keep the expansion joint at its proper length. Tie rods, hinges and/or gimbals may also be used to restrain pressure thrust force by attaching the ends of the expansion joint to each other. In this case, these restraints prevent the bellows from absorbing any axial movement external to the expansion joint. Pressure-balanced expansion joints restrain pressure thrust and absorb axial movement external to the joint.

As can be seen, the forces exerted on a piping system due to the presence of expansion joints can be quite significant. In the preceding example the spring force was found to be 784 lbs., while the pressure thrust force was equal to 84,000 lbs.—a combined force of 84,784 lbs! One can see that proper anchoring of a piping system containing expansion joints is vital.

IV-2.1.3.4 Main Anchors

A main anchor must be designed to withstand the forces and moments imposed upon it by each of the pipe sections to which it is attached. In the case of a pipe section containing one or more unrestrained expansion joints, these will consist of the full bellows thrust due to pressure, media flow, the forces and/or moments required to deflect the expansion joint or joints their full rated movement, and the frictional forces caused by the pipe guides, directional anchors and supports. In certain applications it may be necessary to consider the weight of the pipe and its contents and any other forces and/or moments resulting from wind loading etc.

In systems containing expansion joints, main anchors are installed at any of the following locations:

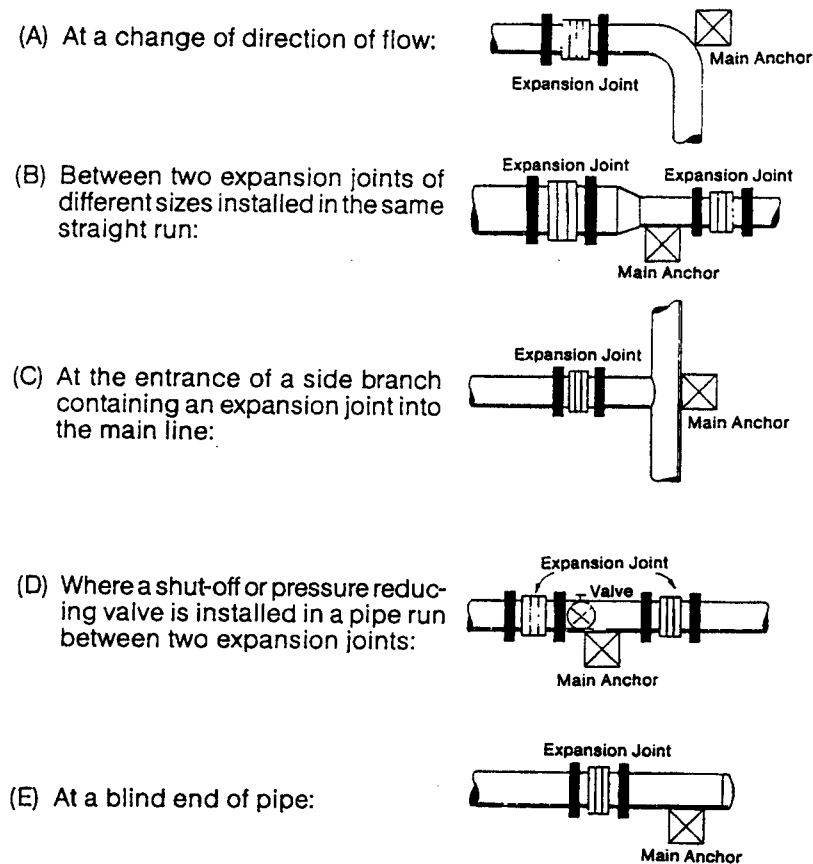


FIG. IV-2.7

IV-2.1.3.5 Intermediate Anchors

Intermediate anchors are not intended to withstand pressure thrust force. This force is absorbed by main anchors, by devices on the expansion joint such as tie rods, swing bars, hinges, gimbals, etc., or, as in the case of a pressure-balanced or double expansion joint, is balanced by an equal pressure force acting in the opposite direction. An intermediate anchor must withstand all of the non-pressure forces acting upon it by each of the pipe sections to which it is attached. In the case of a pipe section containing an expansion joint, these forces will consist of the force required to move the expansion joint and the frictional forces caused by the pipe guides.

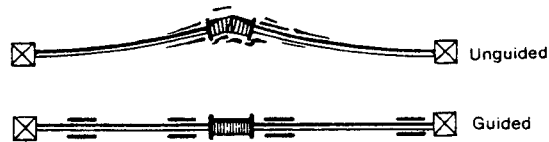


FIG. IV-2.8.1
PIPE GUIDES AND STABILITY

IV-2.1.4 Pipe Guides and Supports. Correct alignment of the pipe adjoining an expansion joint is of vital importance to its proper function. Although expansion joints are designed and built for a long and satisfactory life, maximum service will be obtained only when the pipeline has the recommended number of guides and is anchored and supported in accordance with good piping practice. Proper supporting of the pipeline is required not only to support the pipe itself, but also to provide support at each end of the expansion joint. Pipe guides are necessary to insure proper alignment of movement to the expansion joint and to prevent buckling of the line. Buckling is caused by a combination of the expansion joint flexibility and the internal pressure loading on the pipe which causes it to act like a column loaded by the pressure thrust of the expansion joint. (Fig. IV-2.8.1).

IV-2.1.4.1 Pipe Guide Application

When locating pipe guides for applications involving axial movement only, it is generally recommended that the expansion joint be located near an anchor, and that the first guide be located a maximum of 4 pipe diameters away from the expansion joint. This arrangement will provide proper movement guiding as well as proper support for each end of the expansion joint. The distance between the first and second guide should not exceed 14 pipe diameters, and the distance between the remaining anchors should be determined from paragraph IV-2.2.4.2. of the EJMA Standards.

Common types of pipe guides are shown in Fig. IV-2.9. For systems subjected to lateral motion or angular rotation, directional anchors or planar guides may be required. Examples of these guides are shown in Section IV-2.2.

The recommendations given for pipe anchors and guides represent the minimum requirements for controlling pipelines which contain expansion joints and are intended to protect the expansion joint and pipe system from abuse and failure. However, additional pipe supports are often required between guides in accordance with standard piping practice.

IV-2.1.4.2 Pipe Support Application

A pipe support is any device which permits free movement of the piping while carrying the dead weight of the pipe and any valves or attachments. Pipe supports must also be capable of carrying the live weight. Pipe supports cannot be substituted for pipe alignment or planar guides. Pipe rings, U-bolts, roller supports, spring hanger etc., are examples of conventional pipe support devices.

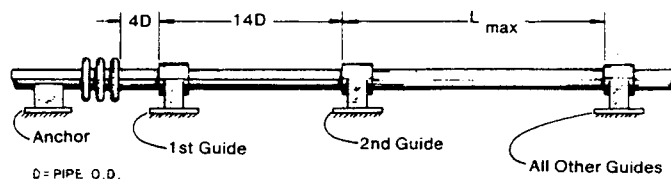
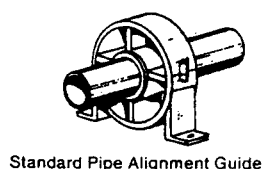
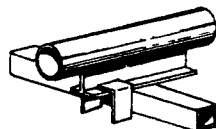


FIG. IV-2.8.2
PROPER PIPE GUIDE LOCATION



Standard Pipe Alignment Guide



Tee Guide



Strap Guide

FIG. IV-2.9
COMMON PIPE GUIDES

IV-2.2 Expansion Joint Applications

The following examples demonstrate common types of expansion joint applications, some dealing with simple axial movement and others with more complex and specialized applications.

IV-2.2.1 Axial Movement. Figure IV-2.10 typifies good practice in the use of a single expansion joint to absorb axial pipeline expansion. Note the use of just one expansion joint between anchors, the nearness of the joint to an anchor and the spacing of the pipe guides.

Fig. IV-2.11 typifies good practice in the use of a double expansion joint to absorb axial pipeline expansion. Note the addition of the intermediate anchor which, in conjunction with the two main anchors effectively divides this section into two distinct expanding segments, so that there is only one expansion joint between anchors.

Fig. IV-2.12 typifies good practice in the use of a pressure-balanced expansion joint (Section IV-1.6, Paragraph 8) to absorb axial movement. Note that the expansion joint is located at a change of direction of the piping and that the elbow and the end of the pipe are secured by intermediate anchors. Since the pressure thrust is absorbed by the expansion joint itself, and only the force required to deflect the expansion joint is imposed on the piping, a minimum of guiding is required. Guiding near the expansion joint as shown will frequently suffice. In long, small-diameter pipelines, additional guiding may be necessary.

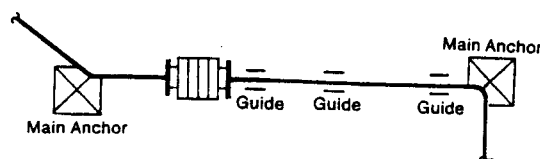


FIG. IV-2.10
SINGLE EXPANSION JOINT APPLICATION



FIG. IV-2.11
DOUBLE EXPANSION JOINT APPLICATION

IV-2.2.2 Combined Movements, Lateral Deflection, and Angular Rotation. Because it offers the lowest expansion joint cost, the single expansion joint is usually considered first for any application. Fig. IV-2.13 shows a typical application of a single expansion joint used to absorb lateral deflection, as well as axial compression. The joint is located near the end of the long pipe run with main anchors at each end and guides properly spaced for both movement control and protection of the piping against buckling. In this case, as opposed to the similar configuration in Fig. IV-2.10, the anchor at left is a directional main anchor which, while absorbing the main anchor loading in the direction of the expansion joint axis, permits the thermal expansion of the short leg to act upon the expansion joint as lateral deflection. Because the main anchor loading (pressure thrust) exists only in the leg containing the expansion joint, the anchor at the end of the short piping leg is an intermediate anchor.

Figure IV-2.14 represents modifications of Figure IV-2.10 in which the main anchors at either end of the expansion joint are replaced by tie rods. Where the piping configuration permits, the use of the tie rods adjusted to prevent axial movement frequently simplifies and reduces the cost of the installation. Because of these tie rods, the expansion joint is not capable of absorbing any axial movement other than its own thermal expansion (See Section IV-2.1.3.3). The thermal expansion in the short leg is imposed as deflection on the long piping leg.

Fig. IV-2.15 shows a tied universal expansion joint used to absorb lateral deflection in a single plane 'Z' bend. The thermal movement of the horizontal lines is absorbed as lateral deflection by the expansion joint. Both anchors need only to be intermediate anchors since the pressure thrust is restrained by the tie rods. Where possible, the expansion joint should fill the entire offset leg so that its expansion is absorbed within the tie rods as axial movement, and bending in the horizontal lines is minimized.

Figure IV-2.16 shows a typical application of a tied universal expansion joint in a 3-plane 'Z' bend. Since the universal expansion joint can absorb lateral deflection in any direction, the two horizontal legs may lie at any angle in the horizontal plane.

Figure IV-2.17 shows a case of pure angular rotation using a pair of hinge joints in a single-plane 'Z' bend. Since the pressure thrust is restrained by the hinges, only intermediate anchors are required. The axial expansion of the pipe leg between the expansion joints is imposed on the two horizontal legs if there is sufficient flexibility.

If not, a 3 hinge system such as in Fig. IV-2.18 may be required. The thermal expansion of the offset leg is absorbed by joints (A), (B), & (C). Expansion joint (B) must be capable of absorbing the total of the rotation of (A) & (C). Hence, it is frequently necessary that the center expansion joint contain a greater number of convolutions than those at either end.

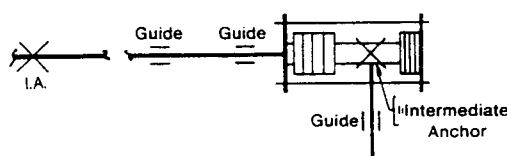


FIG. IV-2.12
PRESSURE-BALANCED APPLICATION

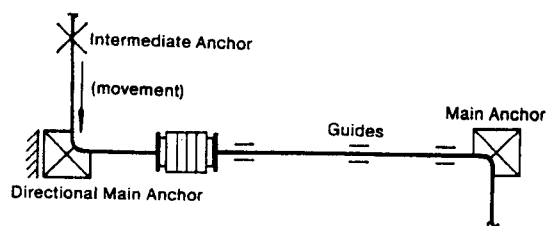


FIG. IV-2.13
COMBINED AXIAL AND LATERAL MOVEMENT

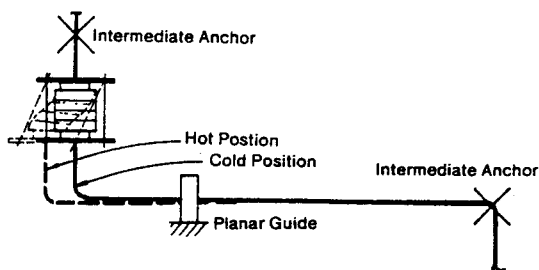


FIG. IV-2.14
SINGLE-TIED APPLICATION

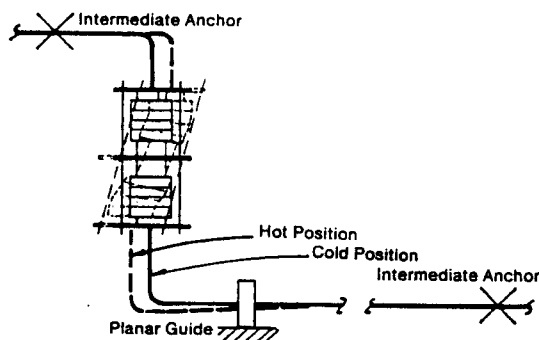


FIG. IV-2.15
UNIVERSAL-TIED APPLICATION

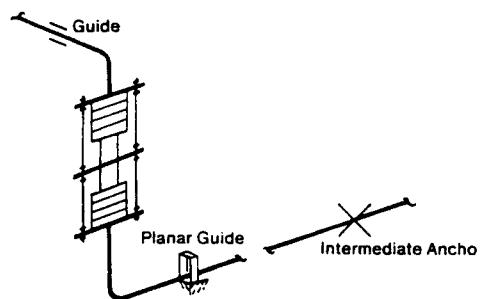


FIG. IV-2.16
UNIVERSAL-TIED APPLICATION

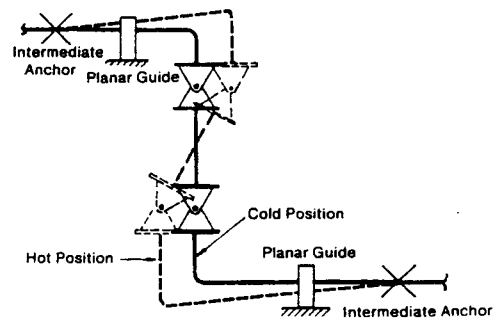


FIG. IV-2.17
TWO-HINGE APPLICATION

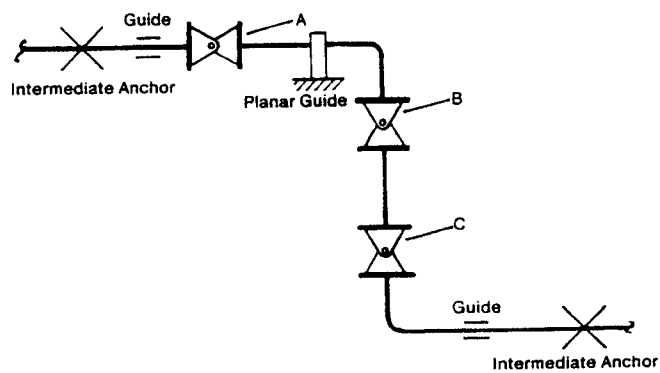


FIG. IV-2.18
THREE-HINGE APPLICATION

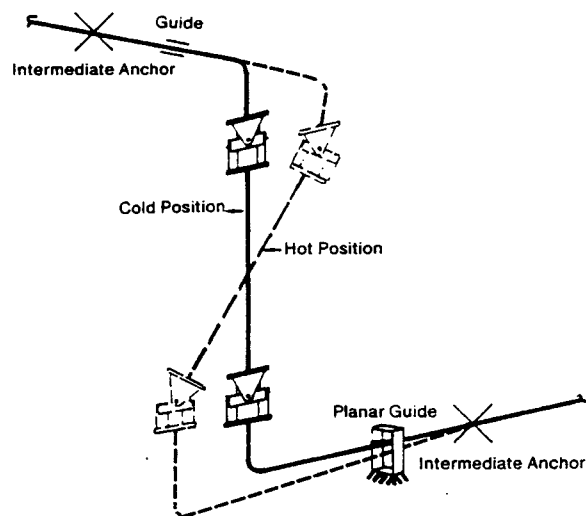


FIG. IV-2.19
TWO-GIMBAL APPLICATION

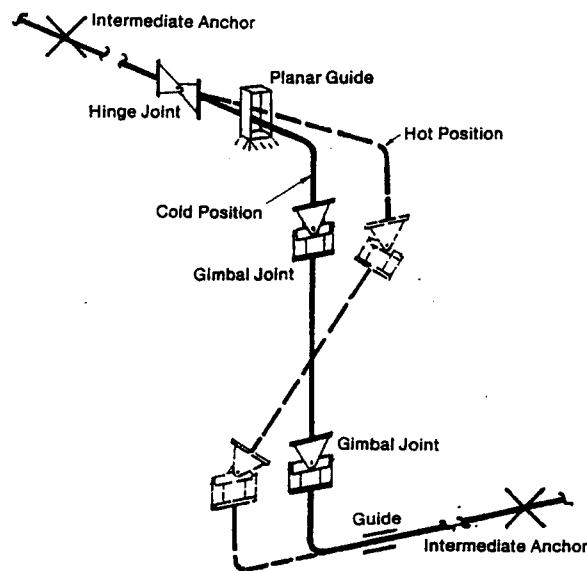


FIG. IV-2.20
ONE HINGE/TWO GIMBAL APPLICATION

Just as hinged expansion joints may offer great advantages in single-plane applications, gimbal expansion joints are designed to offer similar advantages in multi-plane systems. The ability of gimbal expansion joints to absorb angular rotation in any plane is frequently applied by utilizing two such units to absorb lateral deflection.

Figure IV-2.19 shows a case of pure angular rotation using a pair of gimbal expansion joints in a multi-plane 'Z' bend. Since the pressure thrust is restrained by the gimbal assemblies, only intermediate anchors are required.

In applications where the flexibility of the two horizontal piping legs is not sufficient to absorb the axial growth of the pipe section between the two gimbal joints in Figure IV-2.19, the addition of a hinge joint, as shown in Figure IV-2.20, is indicated. Since the expansion of the offset leg takes place in only one plane, the use of the simpler hinge expansion joint is justified.

IV-3 EXPANSION JOINT HANDLING, INSTALLATION, AND SAFETY RECOMMENDATIONS

IV-3.1 Shipping and Handling

Responsible manufacturers of expansion joints take every design and manufacturing precaution to assure the user of a reliable product. The installer and the user have the responsibility to handle, store, install, and apply the expansion joints in a way which will not impair the quality built into them.

IV-3.1.1 Shipping Devices. All manufacturers should provide some way of maintaining the face-to-face dimension of an expansion joint during shipment and installation. These usually consist of overall bars or angles welded to the flanges or weld ends at the extremities of the expansion joint. Washers or wooden blocks between equalizing rings are also used for this purpose. (Fig. IV-3.1) Do not remove these shipping devices until all expansion joints, anchors, and guides in the system have been installed. Shipping devices manufactured by members of the Expansion Joint Manufacturers Association, Inc., are usually painted yellow or otherwise distinctively marked as an aid to the installer. The shipping devices must be removed prior to start-up or testing of the system. (Ref. Section IV-3.4)

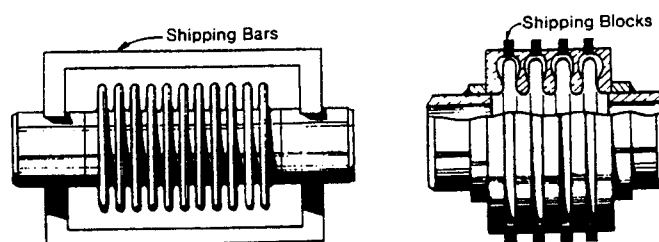


FIG. IV-3.1
SHIPPING DEVICES MAINTAIN OVERALL LENGTH DURING SHIPMENT
AND INSTALLATION

IV-3.1.2 Storage. Some conditions of outdoor storage may be detrimental, and where possible, should be avoided; preferably, the storage should be in a cool, dry area. Where this cannot be accomplished, the expansion joint manufacturer should be so advised either through the specifications or purchase contract. Care must be taken to avoid mechanical damage such as caused by stacking, bumping or dropping. For this reason, it is strongly suggested that covers be specified on all expansion joints to protect the bellows. Certain industrial and natural atmospheres can be detrimental to some bellows materials. If expansion joints are to be stored or installed in such atmospheric environments, materials should be specified which are compatible with these environments.

IV-3.1.3 Installation Instructions. Expansion joints are shipped with documents which furnish the installer with instructions covering the installation of the particular expansion joint. These documents should be left with the expansion joint until installation is completed.

IV-3.2 Installation

Metal bellows-type expansion joints have been designed to absorb a specified amount of movement by flexing of the thin-gage bellows. If proper care is not exercised in the installation of the expansion joint, cycle life and pressure capacity could be reduced, leading to premature failure and damage to the piping system.

It is important that the expansion joint be installed at the length specified by the manufacturer. They should never be extended or compressed in order to make up for deficiencies in length, nor should they be offset to accommodate misaligned pipe.

Remember that a bellows is designed to absorb motion by flexing. The bellows, therefore, must be sufficiently thick to withstand the design pressure, while being thin enough to absorb the required flexing. Optimum design will always require a bellows to be of thinner material than virtually any other component in the piping system in which it is installed. The installer must recognize this relative fragility of the bellows and take every possible measure to protect it during installation. Avoid denting, weld spatter, arc strikes or the possibility of allowing foreign matter to interfere with the proper flexing of the bellows. It is highly recommended that a cover be specified for every expansion joint. The small cost of a cover is easily justified when compared to the cost of replacing a damaged bellows element. With reasonable care during storage, handling, and installation, the user will be assured of the reliability designed and built into the expansion joint.

IV-3.3 Do's and Don't's – Installation and Handling

The following recommendations are included to avoid the most common errors that occur during installation. When in doubt about an installation procedure, contact the manufacturer for clarification before attempting to install the expansion joint.

DO'S...

- Do ... Inspect for damage during shipment such as: dents, broken hardware, water marks on carton, etc.
- Do ... Store in a clean, dry area where it will not be exposed to heavy traffic or damaging environment.
- Do ... Use only designated lifting lugs when provided.
- Do ... Make the piping system fit the expansion joint. By stretching, compressing, or offsetting the joint to fit the piping, the expansion joint may be overstressed when the system is in service.
- Do ... Leave one flange loose on the adjacent piping when possible, until the expansion joint has been fitted into position. Make necessary adjustments of this loose flange before welding.
- Do ... Install the joint with the arrow pointing in the direction of the flow.
- Do ... Install single vanstone liners pointing in the direction of flow. Be sure also to install a gasket between a vanstone liner and flange.
- Do ... In case of telescoping liner, install the smallest I.D. liner pointing in the direction of flow.
- Do ... Remove all shipping devices after the installation is complete and before any pressure test of the fully installed system.
- Do ... Remove any foreign material that may have become lodged between the convolutions.
- Do ... Refer to the proper guide spacing and anchoring recommendations in Section IV-2.1.3 & IV-2.1.4.

DON'T'S...

- Don't ... Drop or strike expansion joint.
- Don't ... Remove the shipping bars until the installation is complete.
- Don't ... Remove any moisture-absorbing desiccant bags or protective coatings until ready for installation.
- Don't ... Use hanger lugs or shipping bars as lifting lugs.
- Don't ... Use chains or any lifting device directly on the bellows or bellows cover.
- Don't ... Allow weld spatter to hit unprotected bellows.
- Don't ... Use cleaning agents which contain chlorides.
- Don't ... Use steel wool or wire brushes on bellows.
- Don't ... Force or rotate one end of an expansion joint for alignment of bolt holes. Bellows are not ordinarily capable of absorbing torsion.
- Don't ... Hydrostatic pressure test or evacuate the system before proper installation of all guides and anchors.
- Don't ... Use shipping bars to restrain the pressure thrust during testing.
- Don't ... Use pipe hangers as guides.
- Don't ... Exceed the manufacturers rated test pressure of the expansion joint.

Caution: The manufacturers warranty may be void if improper installation procedures have been used.

IV-3.4 Safety Recommendations

This section was prepared in order to better inform the user of those factors which many years of experience have shown to be essential for the successful installation and performance of piping systems containing bellows expansion joints.

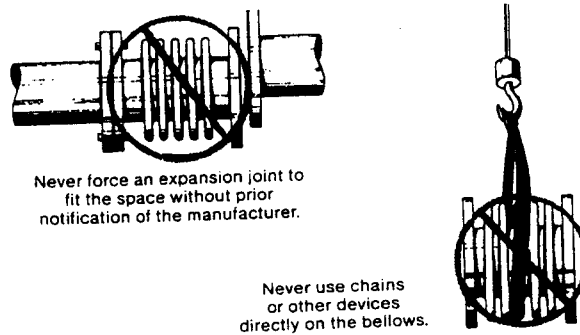


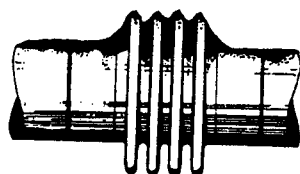
FIG. IV-3.2

IV-3.4.1 Inspection Prior to Start-up or Pressure Test. Expansion joints are usually considered to be non-repairable items and generally do not fall into the category for which maintenance procedures are required. However, immediately after the installation is complete a careful visual inspection should be made of the entire piping system to insure that there is no evidence of damage, with particular emphasis on the following:

1. Are anchors, guides, and supports installed in accordance with the system drawings?
2. Is the proper expansion joint in the proper location?
3. Are the expansion joint's flow direction and pre-positioning correct?
4. Have all of the expansion joint shipping devices been removed?
5. If the system has been designed for a gas, and is to be tested with water, has provision been made for proper support of the additional dead weight load on the piping and expansion joint? Some water may remain in the bellows convolutions after the test. If this is detrimental to the bellows or system operation, means should be provided to remove this water.
6. Are all guides, pipe supports and the expansion joints free to permit pipe movement?
7. Has any expansion joint been damaged during handling and installation?
8. Is any expansion joint misaligned? This can be determined by measuring the joint overall length, inspection of the convolution geometry, and checking clearances at critical points on the expansion joint and at other points in the system.
9. Are the bellows and other movable portions of the expansion joint free of foreign material?

IV-3.4.2 Inspection During and Immediately after Pressure Test – Warning: Extreme care must be exercised while inspecting any pressurized system or component. A visual inspection of the system should include the following:

1. Evidence of leaking or loss of pressure.
2. Distortion or yielding of anchors, expansion joint hardware, the bellows and other piping components.
3. Any unanticipated movement of the piping due to pressure.
4. Evidence of instability in the bellows.
5. The guides, expansion joints, and other movable parts of the system should be inspected for evidence of binding.
6. Any evidence of abnormality or damage should be reviewed and evaluated by competent design authority.



Inspect periodically for a build-up of debris between the bellows convolutions, or any other circumstance which may restrict the free-flexing of the bellows.

FIG. IV-3.3

IV-3.4.3 Periodic Inservice Inspection – Warning: Extreme care must be exercised while inspection any pressurized system or component.

1. Immediately after placing the system in operation, a visual inspection should be conducted to insure that the thermal expansion is being absorbed by the expansion joints in the manner for which they were designed.
2. The bellows should be inspected for evidence of unanticipated vibration.
3. A program of periodic inspection should be planned and conducted throughout the operating life of the system. The frequency of these inspections should be determined by the service and environmental conditions involved. Such inspections can spot the more obvious potential problems such as external corrosion, loosening of threaded fasteners, and deterioration of anchors, guides and other hardware.

IT MUST BE UNDERSTOOD THAT THIS INSPECTION PROGRAM, WITHOUT ANY OTHER BACKUP INFORMATION, CANNOT GIVE EVIDENCE OF DAMAGE DUE TO FATIGUE, STRESS CORROSION OR GENERAL INTERNAL CORROSION. THESE CAN BE THE CAUSE OF SUDDEN FAILURES AND GENERALLY OCCUR WITHOUT ANY VISIBLE OR AUDIBLE WARNING.

4. When any inspection reveals evidence of malfunction, damage, or deterioration, this should be reviewed by competent design authority for resolution. Additionally, any changes in the system operating conditions such as pressure, temperature, movement, flow, velocity, etc. that may adversely affect the expansion joint should be reported to and evaluated by a competent design authority.

IV-3.5 Key to Symbols Used

IV-3.6 Specification and Ordering Guide

This guide has emphasized that the most reliable and safe bellows expansion joint installations are the result of a high degree of understanding between the user and the manufacturer. The prospect for successful long-life performance of piping systems containing expansion joints begins with the submittal of accurate data by the system designer.

When preparing specifications for expansion joints, it is imperative that the system designer completely review the piping system layout and provide the necessary data relating to the selection of an expansion joint: flow medium, pressure, temperature, movements, etc.

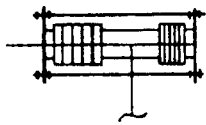
To aid the user in the specification and ordering of expansion joints, the Expansion Joint Manufacturers Association has provided the following expansion joint specification sheets. Complete data, when submitted in accordance with these specifications, will help in obtaining an optimum design. Other data that must be considered but which is not covered, may be included in the "remarks" section of the specification sheets.



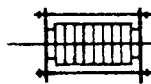
SINGLE EXPANSION JOINT



DOUBLE EXPANSION JOINT
WITH INTERMEDIATE ANCHOR



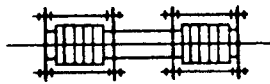
PRESSURE BALANCED
EXPANSION JOINT



SINGLE EXPANSION JOINT
WITH TIE RODS



UNIVERSAL EXPANSION JOINT
WITH OVERALL TIE RODS



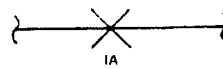
UNIVERSAL EXPANSION JOINT
WITH SHORT TIE RODS



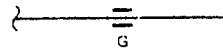
MAIN ANCHOR



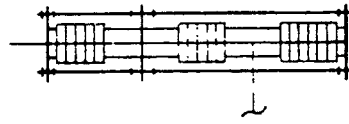
DIRECTIONAL
MAIN ANCHOR



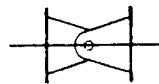
INTERMEDIATE
ANCHOR



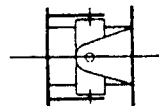
PIPE ALIGNMENT
GUIDE



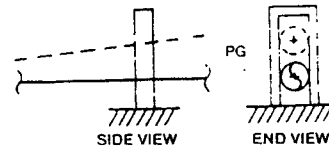
UNIVERSAL PRESSURE BALANCED
EXPANSION JOINT



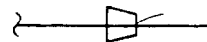
HINGED EXPANSION JOINT



GIMBAL EXPANSION JOINT



PLANAR PIPE ALIGNMENT GUIDE



PIPE REDUCER

APPENDIX A-1					
STANDARD EXPANSION JOINT SPECIFICATION SHEET					
COMPANY:				DATE / /	
PROJECT:				SHEET OF	
				INQUIRY NO.	
				JOB NO.	
ITEM NO./EJ TAG NO.					
1	QUANTITY			EJMA PAGE REFERENCE	
2	NOMINAL SIZE/I.D./O.D. (IN.)				
3	EXPANSION JOINT TYPE			1	
4a	FLUID INFORMATION	MEDIUM GAS/LIQUID		5, 6, 147 77	
4b		VELOCITY (FT./SEC)			
4c		FLOW DIRECTION			
5	DESIGN PRESSURE, PSIG			6, 19,	
6	TEST PRESSURE, PSIG			83, 135	
7a	TEMPERATURE	DESIGN (°F)		6, 13	
7b		MAXIMUM/MINIMUM (°F)			
7c		INSTALLATION (°F)			
8a	MAXIMUM INSTALLATION MOVEMENT	AXIAL COMPRESSION (IN.)		6, 7, 8, 141	
8b		AXIAL EXTENSION (IN.)			
8c		LATERAL (IN.)			
8d		ANGULAR (DEG.)			
9a	MAXIMUM DESIGN MOVEMENTS	AXIAL COMPRESSION (IN.)		6, 7, 13, 47	
9b		AXIAL EXTENSION (IN.)			
9c		LATERAL (IN.)			
9d		ANGULAR (DEG.)			
9e		NO. OF CYCLES			
10a	OPERATING FLUCTUATIONS	AXIAL COMPRESSION (IN.)		6, 7, 13, 47	
10b		AXIAL EXTENSION (IN.)			
10c		LATERAL (IN.)			
10d		ANGULAR (DEG.)			
10e		NO. OF CYCLES			
11a	MATERIALS OF CONSTRUCTION	BELLOWS		5, 6, 45 77, 78 3, 7, 72 3, 43	
11b		LINERS			
11c		COVER			
11d		PIPE SPECIFICATION			
11e		FLANGE SPECIFICATION			
12	RODS (TIE/LIMIT/CONTROL)			3, 4, 41	
13	PANTOGRAPHIC LINKAGE			4	
14	ANCHOR BASE (MAIN/INTERMEDIATE)			1, 2, 17	
15a	DIMENSIONAL LIMITATIONS	OVERALL LENGTH (IN.)		54	
15b		OUTSIDE DIAMETER (IN.)			
15c		INSIDE DIAMETER (IN.)			
16a	SPRING RATE LIMITATIONS	AXIAL (LBS./IN.)		54	
16b		LATERAL (LBS./IN.)			
16c		ANGULAR (IN.-LBS./DEG.)			
17	INSTALLATION POSITION HORIZ./VERT.			8, 141	
18a	QUALITY ASSURANCE REQUIREMENTS	BELLOWS WELD NDE	LONG SEAM	133	
18b			ATTACH.		
18c		PIPING NDE			
18d		DESIGN CODE REQ'D.			
18e		PARTIAL DATA REQ'D.			
18f					
18g					
19		VIBRATION AMPLITUDE/FREQUENCY			



APPENDIX A-2

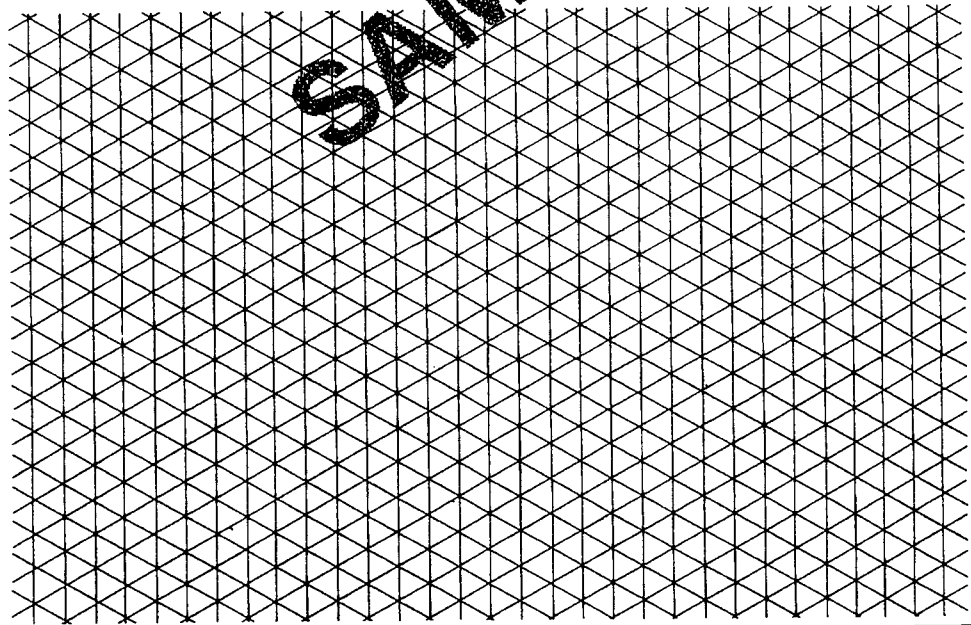


SUPPLEMENTAL SPECIFICATION SHEET
To be used with Standard Expansion Joint Specification Sheet

Company _____ Date _____
 _____ Proposal No. _____
 Project _____ Inquiry/Job No. _____
 _____ Sheet _____ of _____

ITEM NO.				
20. PURGE INSTRUMENTATION CONNECTION				
21a.	SPECIAL FLANGE DESIGN	FACING		
21b.		O.D. (IN.)		
21c.		I.D. (IN.)		
21d.		THICKNESS (IN.)		
21e.		B.C. DIAMETER (IN.)		
21f.		NO. HOLES		
21g.		SIZE HOLES		
21h.		HOLE ORIENTATION		

ISOMETRIC PIPING SKETCH:



APPENDIX V

CONVERSION FACTORS

CONVERSION FACTORS
(Courtesy of Anvil International)

Multiply	by	To Obtain	Multiply	by	To Obtain
Absolute viscosity (poise)	1	Gram/second centimeter	BTU/minute	17.57	Watts
Absolute viscosity (centipoise)	0.01	Poise	BTU/pound	0.556	Calories (Kg)/Kilogram
Acceleration due to gravity (g)	32.174	feet/second ²	Bushels	2150.4	Cubic inches
	980.6	Centimeters/second ²		35.24	Liters
Acres	0.4047	Hectares		4	Pecks
	10	Square Chains		32	Quarts (dry)
	43,560	Square Feet	Cables	120	Fathoms
	4047	Square Meters	Calories (gm)	0.003968	BTU
	0.001562	Square Miles		0.001	Calories (Kg)
	4840	Square Yards		3.088	Foot pounds
	160	Square Rods		1.558×10^{-6}	Horse power hours
Acre-feet	43,560	Cubic Feet		4.185	Joules
	325,851	Gallons (US)		0.4265	Kilogram meters
	1233.49	Cubic Meters		1.1628×10^{-6}	Kilowatt hours
	1,233,490	Liters		0.0011628	Watt hours
Acre-feet/hour	726	Cubic feet/Minute	Cal (gm)/sec/cm ² /°C/cm	242.13	BTU/Hr/ft ² /°F/ft
	5430.86	Gallons Minute	Calories (Kg)	3.968	BTU
Angstroms	10^{-10}	Meters		1000	Calories (gm)
Ares	0.01	Hectares		3088	Foot pounds
	1076.39	Square Feet		0.001558	Horse power hours
	0.02471	Acres		4185	Joules
Atmospheres	76.0	Cms of Hg at 32°F		426.5	Kilograms meters
	29.921	Inches of Hg at 32° F		0.0011628	Kilowatt hours
	33.94	Feet of Water at 62° F		1.1628	Watt hours
	10.333	Kgs Square meter	Calories (Kg) Cu meter	0.1124	BTU/Cu foot at 0° C
	14.6963	Pounds Square inch	Cal (Kg)/Hr/M ² /°C/M	0.671	BTU Hr/ft ² /°F/foot
	1.058	Tons/Square foot	Calories (Kg)/Kg	1.8	BTU pound
	1013.15	Millibars	Calories (Kg) minute	51.43	Foot pounds/second
	235.1408	Ounces/Square inch		0.09351	Horse power
Bags of cement	94	Pounds of cement		0.06972	Kilowatts
Barrels of oil	42	Gallons of oil (US)	Carats (diamond)	200	Milligram
Barrels of cement	376	Pounds of cement	Centares (Centiares)	1	Square meters
Barrels (not legal) or	31	Gallons (US)	Centigram	0.01	Grams
	31.5	Gallons (US)	Centiliters	0.01	Liters
Board feet	144×1 in.*	Cubic inches	Centimeters	0.3937	Inches
Boiler horse power	33,479	BTU/hour		0.032808	Feet
	9.803	Kilowatts		0.01	Meters
	34.5	pounds of water evaporated/ hour at 212° F		10	Millimeters
BTU	252.016	Calories (gm)	Centimeters of Hg at 32° F	0.01316	Atmospheres
	0.252	Calories (Kg)		0.4461	Feet of water at 62° F
	778.26	Foot pounds		136	Kgs/Square meter
	0.0003927	Horse power hours		27.85	Pounds/Square foot
	1055.1	Joules		0.1934	Pounds/Square inch
	107.5	Kilogram meters	Centimeters/second	1.969	Feet/minute
	0.0002928	Kilowatt hours		0.03281	Feet/second
BTU/Cu foot	8.89	Calories (Kg)/Cu meter at 32° F		0.036	Kilometers/hour
BTU/Hr/ft ² /°F/ft	0.00413	Cal (gm)/Sec/cm ² /°C/cm		0.6	Meters/minute
	1.49	Cal (Kg)/Hr/M ² /°C/Meter		0.02237	Miles/hour
BTU/minute	12.96	Foot pounds/second		0.0003728	Miles/minute
	0.02356	Horse Power	Centimeters/second ²	0.03281	Feet/second ²
	0.01757	Kilowatts	Centipoise	0.000672	Pounds/sec foot
				2.42	Pounds/hour foot
				0.01	Poise
			Chains (Gunter's)	4	Rods
				66	Feet
				100	Links

* For thickness less than 1 in. use actual thickness in decimals of an inch.

CONVERSION FACTORS (CONTINUED)
(Courtesy of Anvil International)

Multiply	by	To Obtain	Multiply	by	To Obtain
Cheval-vapeur	1	Metric horse power		807.9	Quarts (liq. US)
	75	Kilogram meters/second	Cubic yards/minute	0.45	Cubic feet/second
	0.98632	Horse power		3.367	Gallons (US)/second
Circular inches	10 ⁶	Circular mils		12.74	Liters/second
	0.7854	Square inches	Cubit	18	Inches
	785,400	Square mils	Days (mean)	1440	Minutes
Circular mils	0.7854	Square mils		24	Hours
	10 ⁻⁶	Circular inches		86,400	Seconds
	7.854 × 10 ⁻⁵	Square inches	Days (sidereal)	86,164.1	Solar seconds
Cubic centimeters	3.531 × 10 ⁻⁵	Cubic feet	Decigrams	0.1	Grams
	0.06102	Cubic inches	Deciliters	0.1	Liters
	10 ⁻⁶	Cubic meters	Decimeters	0.1	Meters
	1.308 × 10 ⁻⁶	Cubic yards	Degrees (angle)	60	Minutes
	0.0002642	Gallons (US)		0.01745	Radians
	0.001	Liters		3600	Seconds
	0.002113	Pints (liq. US)	Degrees F [less 32]	0.5556	Degrees C
	0.001057	Quarts (liq. US)	Degrees F	1 [plus 460]	Degrees F above absolute 0
Cubic feet	0.0391	Ounces (fluid)	Degrees C	1.8 [plus 32]	Degrees F
	28.320	Cubic centimeters		1 [plus 273]	Degrees C above absolute 0
	1728	Cubic inches	Degrees/second	0.01745	Radians/second
	0.02832	Cubic meters		0.1667	Revolutions/minute
	0.03704	Cubic yards		0.002778	Revolutions/second
	7.48052	Gallons (US)	Dekagrams	10	Grams
	28.32	Liters	Dekaliters	10	Liters
	59.84	Pints (liq. US)	Dekameters	10	Meters
	29.92	Quarts (liq. US)	Diameter (circle)	3.14159265359	Circumference
	2.296 × 10 ⁻⁵	Acre feet	(approx)	3.1416	
	0.803564	Bushels	(approx)	3.14	
Cubic feet of water	62.4266	Pounds at 39.2° F	(approx)	$\frac{22}{7}$	
	62.3352	Pounds at 62° F	Diameter (circle)	0.88623	Side of equal square
Cubic feet/minute	472	Cubic centimeters/sec		0.7071	Side of inscribed square
	0.1247	Gallons (US)/second	Diameter ³ (sphere)	0.5236	Volume (sphere)
	0.472	Liters/second	Diam (major) × diam (minor)	0.7854	Area of ellipse
	62.34	Pounds water/min at 62° F	Diameter ² (circle)	0.7854	Area (circle)
	7.4805	Gallons (US)/minute	Diameter ² (sphere)	3.1416	Surface (sphere)
	10.772	Gallons/24 hours	Diam (inches) × RPM	0.262	Belt speed ft/minute
	0.033058	Acre feet/24 hours	Digits	0.75	Inches
Cubic feet/second	646.317	Gallons (US)/24 hours	Drams (avoirdupois)	27.34375	Grains
	448.831	Gallons/minute		0.0625	Ounces (avoir.)
	1.98347	Acre feet/24 hours		1.771845	Grams
Cubic inches	16.387	Cubic centimeters	Fathoms	0.16667	Feet
	0.0005787	Cubic feet	Feet	30.48	Centimeters
	1.639 × 10 ⁻⁵	Cubic meters		12	Inches
	2.143 × 10 ⁻⁵	Cubic yards		0.3048	Meters
	0.004329	Gallons (US)		1/3	Yards
	0.01639	Liters		0.06061	Rods
	0.03463	Pints (liq. US)	Feet of water at 62	0.029465	Atmospheres
	0.01732	Quarts (liq. US)		0.88162	Inches of Hg at 32° F
Cubic meters	10 ⁶	Cubic centimeters		62.3554	Pounds/square foot
	35.31	Cubic feet		0.43302	Pounds/square inch
	61.023	Cubic inches		304.44	Kilogram/sq meter
	1.308	Cubic yards	Feet/minute	0.5080	Centimeters/second
	264.2	Gallons (US)		0.01667	Feet/second
	1000	Liters		0.01829	Kilometers/hour
	2113	Pints (liq. US)			
	1057	Quarts (liq. US)			
Cubic yards	764,600	Cubic centimeters			
	27	Cubic feet			
	46,656	Cubic inches			
	0.7646	Cubic meters			
	202	Gallons (US)			
	764.6	Liters			
	1616	Pints (liq. US)			

CONVERSION FACTORS (CONTINUED)
(Courtesy of Anvil International)

Multiply	by	To Obtain	Multiply	by	To Obtain
Feet/minute	0.3048	Meters/minute		0.0022857	Ounces (avoir.)
	0.01136	Miles/hour	Grains/gallon (US)	17.118	Parts/million
Feet/Second	30.48	Centimeters/second		142.86	Pounds/million gallons (US)
	1.097	Kilometers/hour	Grams	980.7	Dynes
	0.5921	Knots		15.43	Grains
	18.29	Meters/minute		0.001	Kilograms
	0.6818	Miles/hour		1000	Milligrams
	0.01136	Miles/minute		0.03527	Ounces (avoir.)
Feet/second ²	30.48	Centimeters/second ²		0.03215	Ounces (troy)
	0.3048	Meters/second ²		0.002205	Pounds
Flat of a hexagon	1.155	Distance across corners	Grams/centimeters	0.00521	Pounds/inch
Flat of a square	1.414	Distance across corners	Grams/cubic centimeter		62.45 Pounds/cubic foot
Foot pounds	0.00128492	BTU		0.03613	Pounds/cubic inch
	0.32383	Calories (gm)		4.37	Grains/100 cubic ft
	0.0003238	Calories (Kg)	Grams/liter	58.405	Grains/gallon (US)
	5.05×10^{-7}	Horse power hours		8.345	Pounds/100 gallons (US)
	1.3558	Joules		0.062427	Pounds/cubic foot
	0.13826	Kilogram meters		1000	Parts/million
	3.766×10^{-7}	Kilowatt hours	Gravity (g)	32.174	Feet/second ²
	0.0003766	Watt hours		980.6	Centimeters/second ²
Foot pounds/minute	0.001286	BTU/minute	Hand	4	Inches
	0.01667	Foot pounds/second		10.16	Centimeters
	3.03×10^{-5}	Horse power	Hectares	2.471	Acres
	0.0003241	Calories (Kg)/minute		107,639	Square feet
	2.26×10^{-5}	Kilowatts		100	Ares
Foot pounds/second	0.07717	BTU/minute	Hectograms	100	Grams
	0.001818	Horse power	Hectoliters	100	Liters
	0.01945	Calories (Kg)/minute	Hectometers	100	Meters
	0.001356	Kilowatts	Hectowatts	100	Watts
Furlong	40	Rods	Hogshead	63	Gallons (US)
	220	Yards		238.4759	Liters
	660	Feet	Horse power	42.44	BTU/minute
	0.125	Miles		33,000	Foot pounds/minute
	0.2042	Kilometers		550	Foot pounds/second
Gallons (Imperial)	277.42	Cubic inches		1.014	Metric horse power (Cheval vapeur)
	4.543	Liters		10.7	Calories (Kg)/min
	1.20095	Gallons (US)		0.7457	Kilowatts
Gallons (US)	3785	Cubic centimeters		745.7	Watts
	0.13368	Cubic feet	Horse power (boiler)	33,479	BTU/hour
	231	Cubic inches		9.803	Kilowatts
	0.003785	Cubic meters		34.5	Pounds of water evaporated/hour at 212° F
	0.004951	Cubic yards	Horse power hours	2546.5	BTU
	3.785	Liters		641,700	Calories (gm)
	8	Pints (liq. US)		641.7	Calories (Kg)
	4	Quarts (liq. US)		1980198	Foot pounds
	0.83267	Gallons (Imperial)		2688172	Joules
	3.069×10^{-6}	Acre feet		273,740	Kilogram meters
Gallons (US) of water at 62° F	8.333	Pounds of water		0.7455	Kilowatt hours
Gallons (US) of water/minute	6.0086	Tons of water/24 hours		745.5	Watt hours
Gallons(US)/minute	0.002228	Cubic feet/second	Inches	2.54	Centimeters
	0.13368	Cubic feet/minute		0.08333	Feet
	8.0208	Cubic feet/hour		1000	Mils
	0.06309	Liters/second		12	Lines
	3.78533	Liters/minute		72	Points
	0.0044192	Acre feet/24 hours	Inches of Hg at 32° F	0.03342	Atmospheres
Grains	1	Grains (avoirdupois)		345.3	Kilograms/square meter
	1	Grains (apothecary)		70.73	Pounds/square foot
	1	Grains (troy)		0.49117	Pounds/square inch
	0.0648	Grams		1.1343	Feet of water at 62° F
	0.0020833	Ounces (troy)			

CONVERSION FACTORS (CONTINUED)
(Courtesy of Anvil International)

Multiply	by	To Obtain	Multiply	by	To Obtain
Inches of Hg at 32° F	13.6114	Inches of water at 62° F	Kilowatt hours	859,999	Calories (gm)
	7.85872	Ounces/square inch		858.99	Calories (Kg)
Inches of water at 62° F	0.002455	Atmospheres		2,655,200	Foot pounds
	25.37	Kilograms/square meter		1.341	Horse power hours
	0.5771	Ounces/square inch		3,600,000	Joules
	5.1963	Pounds/square foot		367,100	Kilogram meters
	0.03609	Pounds/square inch		1000	Watt hours
	0.07347	Inches of Hg at 32° F	Knots	1	Nautical miles/hour
Joules	0.00094869	BTU		1.1516	Miles/hour
	0.239	Calories (gm)		1.8532	Kilometers/hours
	0.000239	Calories (Kg)	Leagues	3	Miles
	0.73756	Foot pounds	Lines	0.08333	Inches
	3.72×10^{-7}	Horse power hours	Links	7.92	Inches
	0.10197	Kilogram meters	Liters	1000	Cubic Centimeters
	2.778×10^{-7}	Kilowatt hours		0.03531	Cubic feet
	0.0002778	Watt hours		61.02	Cubic inches
	1	Watt second		0.001	Cubic meters
Kilograms	980,665	Dynes		0.001308	Cubic yards
	2.205	Pounds		0.2642	Gallons (US)
	0.001102	Tons (short)		0.22	Gallons (Imp)
	1000	Grams		2.113	Pints (liq. US)
	35.274	Ounces (avoir.)		1.057	Quarts (liq. US)
	32.1507	Ounces (troy)		8.107×10^{-7}	Acre Feet
Kilogram meters	0.009302	BTU		2.2018	Pounds of water at 62° F
	2.344	Calories (gm)	Liter/minute	0.0005886	Cubic feet/second
	0.002344	Calories (Kg)		0.004403	Gallons (US)/second
	7.233	Foot pounds		0.26418	Gallons (US)/minute
	3.653×10^{-6}	Horse power hours	Meters	100	Centimeters
	9.806	Joules		3.281	Feet
	2.724×10^{-6}	Kilowatt hours		39.37	Inches
	0.002724	Watt hours		1.094	Yards
Kilograms/ cubic meter	0.06243	Pounds/ cubic foot		0.001	Kilometers
Kilograms/meter	0.6720	Pounds/foot		1000	Millimeters
Kilograms/sq centimeter	14.223	Pounds/sq inch	Meters/minute	1.667	Centimeters/second
	1	Metric atmosphere		3.281	Feet/minute
Kilogram/sq meter	9.678×10^{-5}	Atmospheres		0.05468	Feet/second
	0.003285	Feet of water at 62° F		0.06	Kilometers/hour
	0.002896	Inches of Hg at 32° F		0.03728	Miles/hour
	0.2048	Pounds/square foot	Meters/second	196.8	Feet/minute
	0.001422	Pounds/square inch		3.281	Feet/second
	0.007356	Centimeters of Hg at 32° F		3.6	Kilometers/hour
Kiloliters	1000	Liters		0.06	Kilometers/minute
Kilometers	100,000	Centimeters		2.237	Miles/hour
	1000	Meters		0.03728	Miles/minute
	3281	Feet	Microns	10^{-6}	Meters
	0.6214	Miles		0.001	Millimeters
	1094	Yards		0.03937	Mils
Kilometers/hour	27.78	Centimeters/second	Mils	0.001	Inches
	54.68	Feet/minute		0.0254	Millimeters
	0.9113	Feet/second		25.4	Microns
	16.67	Meters/minute	Miles	160,934	Centimeters
	0.6214	Miles/hour		5280	Feet
	0.5396	Knots		63,360	Inches
Kilometers/hr/sec	27.78	Centimeters/sec/sec		1.609	Kilometers
	0.9113	Feet/sec/sec		1760	Yards
	0.2778	Meters/sec/sec		80	Chains
Kilowatts	56.92	BTU/minute		320	Rods
	44,250	Foot pounds/minute		0.8684	Nautical miles
	737.6	Foot pounds/second	Miles/hour	44.70	Centimeters/second
	1.341	Horse power		88	Feet/minute
	14.34	Calories (Kg)/min		1.467	Feet/second
	1000 Watts			1.609	Kilometers/hours
Kilowatt hours	3413	BTU		0.8684	Knots
				26.82	Meters/minute

CONVERSION FACTORS (CONTINUED)
(Courtesy of Anvil International)

Multiply	by	To Obtain	Multiply	by	To Obtain
Miles/minute	2682	Centimeters/second		1.315	Horse power
	88	Feet/second	Pounds (avoirdupois)	16	Ounces (avoir.)
	1,609	Kilometers/minute		256	Drams (avoir.)
	60	Miles/hour		7000	Grains
Millibars	0.000987	Atmosphere		0.0005	Tons (short)
Milliers	1000	Kilograms		453.5924	Grams
Milligrams	0.01	Grams		1.21528	Pounds (troy)
	0.01543	Grains		14.5833	Ounces (troy)
Milligrams/liter	1	Parts/million	Pounds (troy)	5760	Grains
Milliliters	0.001	Liters		240	Pennyweights (troy)
Million gals/24 hours	1.54723	Cubic feet/second		12	Ounces (troy)
Millimeters	0.1	Centimeters		373.24177	Grams
	0.03937	Inches		0.822857	Pounds (avoir.)
	39.37	Mils		13.1657	Ounces (avoir.)
	1000	Microns		0.00036735	Tons (long)
Miner's inches	1.5	Cubic feet/minute		0.00041143	Tons (short)
Minutes (angle)	0.0002909	Radians		0.00037324	Tons (metric)
Nautical miles	6080.2	Feet	Pounds of water at 62° F	27.72	0.01604 Cubic feet
	1.516	Miles		0.120	Cubic inches
Ounces (avoirdupois)	16	Drams (avoir.)	Pounds of water/min. at 62° F	0.0002673	Gallons (US)
	437.5	Grains			
	0.0625	Pounds (avoir.)	Pounds/cubic foot	0.01602	Cubic feet/second
	28.349527	Grams		16.02	Grams/cubic centimeter
	0.9115	Ounces (troy)		0.0005787	Kilograms/cubic meter
Ounces (fluid)	1.805	Cubic inches			Pounds/cubic inch
	0.02957	Liters	Pounds/cubic inch	27.68	Grams/cubic centimeter
	29.57	Cubic centimeters		27.680	Kilograms/cubic meter
	0.25	Gills		1728	Pounds/cubic foot
Ounces (troy)	480	Grains	Pounds/foot	1.488	Kilograms/meter
	20	Pennyweights (troy)	Pounds/inch	178.6	Grams/centimeter
	0.08333	Pounds (troy)	Pounds/hour foot	0.4132	Centipoise
	31.103481	Grams		0.004132	Poise grams/sec cm
	1.09714	Ounces (avoir.)	Pounds/sec foot	14.881	Poise grams/sec cm
Ounces/square inch	0.0625	Pounds/square inch		1488.1	Centipoise
	1.732	Inches of water at 62° F	Pounds/square foot	0.016037	Feet of water at 62° F
	4.39	Centimeters of water at 62° F		4.882	Kilograms/square meter
	0.12725	Inches of Hg at 32° F		0.006944	Pounds/square inch
	0.004253	Atmospheres		0.014139	Inches of Hg at 32° F
Palms	3	Inches		0.0004725	Atmospheres
Parts/million	0.0584	Grains/gallon (US)	Pounds/square inch	0.068044	Atmospheres
	0.07016	Grains/gallon (Imp)		2.30934	Feet of water at 62° F
	8.345	Pounds/million gal (US)		2.0360	Inches of Hg at 32° F
Pennyweights (troy)	24	Grains		703.067	Kilograms/square meter
	1.55517	Grams		27.912	Inches of water at 62° F
	0.05	Ounces (troy)	Quadrants (angular)	90	Degrees
	0.0041667	Pounds (troy)		5400	Minutes
Pints (liq. US)	4	Gills		324,000	Seconds
	16	Ounces (fluid)		1.751	Radians
	0.5	Quarts (liq. US)	Quarts (dry)	67.20	Cubic inches
	28.875	Cubic inches	Quarts (liq. US)	2	Pints (liq. US)
	473.1	Cubic centimeters		0.9463	Liters
Pipe	126	Gallons (US)		32	Ounces (fluid)
Points	0.01389	Inches		57.75	Cubic inches
Poise	0.0672	Pounds/sec foot		946.3	Cubic centimeters
	242	Pounds/hour foot	Quintal, Argentine	101.28	Pounds
	100	Centipoise	Brazil	129.54	Pounds
Poncelots	100	Kilogram meters/second	Castile, Peru	101.43	Pounds
			Chile	101.41	Pounds
			Metric	220.46	Pounds
			Mexico	101.47	Pounds

CONVERSION FACTORS (CONTINUED)
(Courtesy of Anvil International)

Multiply	by	To Obtain	Multiply	by	To Obtain
Quires	25	Sheets	Square Miles	2.590	Square kilometers
Radians	57.30	Degrees		259	Hectares
	3438	Minutes		3,097,600	Square yards
	206.186	Seconds		102,400	Square rods
	0.637	Quadrants		1	Sections
Radians/second	57.30	Degrees/second	Square millimeters	0.01	Square centimeters
	0.1592	Revolutions/second		0.00155	Square inches
	9.549	Revolutions/minute		1550	Square mils
Radians/second ²	573.0	Revolutions/minute ²		1973	Circular mils
	0.1592	Revolutions/second ²	Square mils	1.27324	Circular mils
Reams	500	Sheets		0.0006452	Square millimeters
Revolutions	360	Degrees		10 ⁻⁶	Square inches
	4	Quadrants	Square yards	0.0002066	Acres
	6.283	Radians		9	Square feet
Revolutions/minute	6	Degrees/second		0.8361	Square meters
	0.1047	Radians/second		3.228 × 10 ⁻⁷	Square miles
	0.01667	Revolutions/second	Stere	1	Cubic meters
Revolutions/minute ²	0.001745	Radians/second ²	Stone	14	Pounds
	0.0002778	Revolutions/second ²		6.35029	Kilograms
Revolutions/second	360	Degrees/second	Tons (long)	1016	Kilograms
	6.283	Radians/second		2240	Pounds
	60	Revolutions/minute		1.12	Tons (short)
Revolutions/second ²	6.283	Radians/second ²	Tons (metric)	1000	Kilograms
	3600	Revolutions/minute ²		2205	Pounds
Rods	16.5	Feet		1.1023	Tons (short)
	5.5	Yards	Tons (short)	2000	Pounds
Seconds (angle)	4.848 × 10 ⁻⁶	Radians		32,000	Ounces
Sections	1	Square miles		907.185	Kilograms
Side of a square	1.4142	Diameter of inscribed circle		0.90718	Tons (metric)
	1.1284	Diameter of circle with equal area		0.89286	Tons (long)
Span	9	Inches	Tons of refrigeration	12,000	BTU/hour
Square centimeters	0.001076	Square feet		288,000	BTU/24 hours
	0.1550	Square inches	Tons of water/24 hours at 62° F	83.33	Pounds of water/hour
	0.0001	Square meters		0.16510	Gallons (US)/minute
	100	Square millimeters		1.3263	Cubic feet/hour
Square feet	2.296 × 10 ⁻⁵	Acres	Watts	0.05692	BTU/minute
	929.0	Square centimeters		44.26	Foot pounds/minute
	144	Square inches		0.7376	Foot pounds/second
	0.0929	Square meters		0.001341	Horse power
	3.587 × 10 ⁻⁸	Square miles		0.01434	Calories (Kg)/minute
	0.1111	Square yards		0.001	Kilowatts
Square inches	6.452	Square centimeters		1	Joule/second
	0.006944	Square feet	Watt hours	3.413	BTU
	645.2	Square millimeters		860.5	Calories (gm)
	1.27324	Circular inches		0.8605	Calories (Kg)
	1,273,239	Circular mils		2655	Foot pounds
	1,000,000	Square mils		0.001341	Horse power hours
Square kilometers	247.1	Acres		3600	Joules
	10,760,000	Square feet		367.1	Kilogram meters
	1,000,000	Square meters		0.001	Kilowatt hours
	0.3861	Square miles	Watts/square inch	8.2	BTU/square foot/minute
	1,196,000	Square yards		6373	Foot pounds/sq ft/minute
Square meters	0.0002471	Acres		0.1931	Horse power/ square foot
	10.764	Square feet	Yards	91.44	Centimeters
	1.196	Square yards		3	Feet
	1	Centares		36	Inches
Square miles	640	Acres		0.9144	Meters
	27,878,400	Square feet		0.1818	Rods
			Year (365 days)	8760	Hours

REFERENCES

- ANSI B16.1, *Cast Iron Pipe Flanges and Flanged Fittings*, American National Standards Institute.
- API 526, *Flanged Steel Pressure Relief Valves*, American Petroleum Institute.
- API 570, *Piping Inspection Code: Inspection Repair, Alteration, and Rerating of In-Service Piping Systems*, American Petroleum Institute.
- API 594, *Wafer and Wafer-Lug Check Valves*, American Petroleum Institute.
- API 599, *Metal Plug Valves—Flanged and Welding Ends*, American Petroleum Institute.
- API 600, *Steel Gate Valves—Flanged and Butt-Welding Ends, Bolted, and Pressure Seal Bonnets*, American Petroleum Institute.
- API 602, *Compact Steel Gate Valves—Flanged, Threaded, Welding and Extended Body Ends*, American Petroleum Institute.
- API 603, *Class 150, Cast, Corrosion-Resistant, Flanged-End Gate Valves*, American Petroleum Institute.
- API 608, *Metal Ball Valves—Flanged, Threaded, and Butt-Welding Ends*, American Petroleum Institute.
- API 609, *Butterfly Valves: Double Flanged, Lug- and Water-Type*, American Petroleum Institute.
- API 610, *Centrifugal Pumps for General Refinery Services*, American Petroleum Institute.
- API 611, *General Purpose Steam Turbines for Petroleum, Chemical, and Gas Industry Services*, American Petroleum Institute.
- API 612, *Special Purpose Steam Turbines for Petroleum, Chemical, and Gas Industry Services*, American Petroleum Institute.
- API 617, *Centrifugal Compressors for Petroleum, Chemical, and Gas Service Industries*, American Petroleum Institute.
- API 619, *Rotary-Type Positive Displacement Compressors for Petroleum, Chemical, and Gas Industry Services*, American Petroleum Institute.
- API 661, *Air-Cooled Heat Exchangers for General Refinery Services*, American Petroleum Institute.
- ASME B1.20.1, *Pipe Threads, General Purpose (Inch)*, American Society of Mechanical Engineers.
- ASME B16.3, *Malleable Iron Threaded Fittings*, American Society of Mechanical Engineers.
- ASME B16.4, *Gray Iron Threaded Fittings*, American Society of Mechanical Engineers.
- ASME B16.5, *Pipe Flanges and Flanged Fittings*, American Society of Mechanical Engineers.
- ASME B16.9, *Factory-Made Wrought Steel Buttwelding Fittings*, American Society of Mechanical Engineers.

- ASME B16.10, *Face-to-Face and End-to-End Dimensions of Valves*, American Society of Mechanical Engineers.
- ASME B16.11, *Forged Steel Fittings, Socket-Welding and Threaded*, American Society of Mechanical Engineers.
- ASME B16.14, *Ferrous Pipe Plugs, Bushings, and Locknuts with Pipe Threads*, American Society of Mechanical Engineers.
- ASME B16.15, *Cast Bronze Threaded Fittings, Classes 125 and 250*, American Society of Mechanical Engineers.
- ASME B16.18, *Cast Copper Alloy Solder Joint Pressure Fittings*, American Society of Mechanical Engineers.
- ASME B16.22, *Wrought Copper and Copper Alloy Solder Joint Pressure Fittings*, American Society of Mechanical Engineers.
- ASME B16.24, *Bronze Pipe Flanges and Flanged Fittings Classes 150, 300, 400, 600, 900, 1500, and 2500 and Flanged Fittings, Class 150 and 300*, American Society of Mechanical Engineers.
- ASME B16.26, *Cast Copper Alloy Fittings for Flared Copper Tubes*, American Society of Mechanical Engineers.
- ASME B16.28, *Wrought Steel Buttwelding Short Radius Elbows and Returns*, American Society of Mechanical Engineers.
- ASME B16.34, *Valves—Flanged, Threaded, and Welding End*, American Society of Mechanical Engineers.
- ASME B16.36, *Orifice Flanges, Classes 300, 600, 900, 1500, and 2500*, American Society of Mechanical Engineers.
- ASME B16.39, *Malleable Iron Threaded Pipe Unions, Classes 150, 250, and 300*, American Society of Mechanical Engineers.
- ASME B16.42, *Ductile Iron Pipe Flanges and Flanged Fittings, Classes 150 and 300*, American Society of Mechanical Engineers.
- ASME B16.47, *Large Diameter Steel Flanges, NPS 26 Through NPS 60*, American Society of Mechanical Engineers.
- ASME B31.1, *Pressure Piping*, American Society of Mechanical Engineers.
- ASME B31.3, *Process Piping*, American Society of Mechanical Engineers.
- ASME B31.4, *Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids*, American Society of Mechanical Engineers.
- ASME B31.5, *Refrigeration Piping*, American Society of Mechanical Engineers.
- ASME B31.8, *Gas Transmission and Distribution Piping Systems*, American Society of Mechanical Engineers.
- ASME B31.9, *Building Services Piping*, American Society of Mechanical Engineers.
- ASME B31.11, *Slurry Piping*, American Society of Mechanical Engineers.
- ASME B31H, *Standard Method to Establish Maximum Allowable Design Pressures for Piping Components*, American Society of Mechanical Engineers.
- ASME B31J, *Standard Method to Develop Stress Intensification and Flexibility Factors for Piping Components*, American Society of Mechanical Engineers.

ASME Boiler and Pressure Vessel Code, Section I, *Power Boilers*, American Society of Mechanical Engineers.

ASME Boiler and Pressure Vessel Code, Section II, *Materials*, Part A, *Ferrous Material Specifications*, American Society of Mechanical Engineers.

ASME Boiler and Pressure Vessel Code, Section II, *Materials*, Part B, *Nonferrous Material Specifications*, American Society of Mechanical Engineers.

ASME Boiler and Pressure Vessel Code, Section II, *Materials*, Part D, *Properties*, American Society of Mechanical Engineers.

ASME Boiler and Pressure Vessel Code, Section III, *Rules for Construction of Nuclear Power Plant Components*, American Society of Mechanical Engineers.

ASME Boiler and Pressure Vessel Code, Section V, *Nondestructive Examination*, American Society of Mechanical Engineers.

ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, *Pressure Vessels*, American Society of Mechanical Engineers.

ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, *Pressure Vessels, Alternative Rules*, American Society of Mechanical Engineers.

ASME Boiler and Pressure Vessel Code, Section VIII, Division 3, *Pressure Vessels, Alternative Rules for Construction of High Pressure Vessels*, American Society of Mechanical Engineers.

ASME Boiler and Pressure Vessel Code, Section VIII, Divisions 1 and 2, Code Case 2286, *Alternative Rules for Determining Allowable Compressive Stresses for Cylinders, Cones, Spheres, and Formed Heads*, American Society of Mechanical Engineers.

ASME Boiler and Pressure Vessel Code, Section IX, *Welding and Brazing Qualifications*, American Society of Mechanical Engineers.

ASME PCC-1, *Guidelines for Pressure Boundary Bolted Flange Joint Assembly*, American Society of Mechanical Engineers.

ASME Standard OM-S/G, *Standards and Guides for Operation and Maintenance of Nuclear Power Plants*, American Society of Mechanical Engineers.

ASTM A53, *Standard Specification for Pipe, Steel, Black and Hot-Dipped, Zinc-Coated, Welded and Seamless*, American Society for Testing and Materials.

ASTM A106, *Standard Specification for Seamless Carbon Steel Pipe for High-Temperature Service*, American Society for Testing and Materials.

ASTM C582, *Standard Specification for Contact-Molded Reinforced Thermosetting Plastic (RTP) Laminates for Corrosion Resistant Equipment*, American Society for Testing and Materials.

ASTM D1599, *Standard Test Method for Short-Time Hydraulic Failure Pressure of Plastic Pipe, Tubing, and Fittings*, American Society for Testing and Materials.

ASTM D2837, *Standard Test Method for Obtaining Hydrostatic Design Basis for Thermoplastic Pipe Materials*, American Society for Testing and Materials.

ASTM D2992, *Standard Practice for Obtaining Hydrostatic or Pressure Design Basis for "Fiberglass" (Glass-Fiber-Reinforced Thermosetting-Resin) Pipe and Fittings*, American Society for Testing and Materials.

AWS D10.10, *Recommended Practices for Local Heating of Welds in Piping and Tubing*, American Welding Society.

AWWA C110, *Ductile-Iron and Gray-Iron Fittings, 3 Inch Through 48 Inch (75 mm Through 1200 mm), for Water and Other Liquids*, American Water Works Association.

AWWA C115, *Flanged Ductile-Iron with Ductile-Iron or Gray-Iron Threaded Flanges*, American Water Works Association.

AWWA C207, *Steel Pipe Flanges for Water Works Service, Sizes 4 Inch Through 144 Inch (100 mm Through 3,600 mm)*, American Water Works Association.

AWWA C208, *Dimensions for Fabricated Steel Water Pipe Fittings*, American Water Works Association.

AWWA C500, *Metal-Seated Gate Valves for Water Supply Service*, American Water Works Association.

AWWA C504, *Rubber-Seated Butterfly Valves*, American Water Works Association.

Becht IV, C., 1980, "Root Bulge of Bellows," DOE Research and Development Report, ESG-DOE-13320, Rockwell International.

Becht IV, C., and Skopp, G., 1981a, "Stress Analysis of Bellows," *Metallic Bellows and Expansion Joints*, PVP-Vol. 51, American Society of Mechanical Engineers.

Becht IV, C., and Skopp, G., 1981b, "Root Bulge of Bellows," *Metallic Bellows and Expansion Joints*, PVP-Vol. 51, American Society of Mechanical Engineers.

Becht IV, C., Horton, P., and Skopp, G., 1981c, "Verification of Theoretical Plastic Ratchet Boundaries for Bellows," *Metallic Bellows and Expansion Joints*, PVP-Vol. 51, American Society of Mechanical Engineers.

Becht IV, C., 1985, "Predicting Bellows Response by Numerical and Theoretical Methods," *A Decade of Progress in Pressure Vessels and Piping Technology—1985*, American Society of Mechanical Engineers.

Becht IV, C., 1989a, "Fatigue and Elevated Temperature Design of Bellows," *Metallic Bellows and Expansion Joints—1989*, PVP-Vol. 168, American Society of Mechanical Engineers.

Becht IV, C., 1989b, "Considerations in Bellows Thrust Forces and Proof Testing," *Metallic Bellows and Expansion Joints—1989*, PVP-Vol. 168, American Society of Mechanical Engineers.

Becht IV, C., Chen, Y., and Benteftifa, C., 1992, "Effect of Pipe Insertion on Slip-On Flange Performance," *Design and Analysis of Pressure Vessels, Piping, and Components—1992*, PVP-Vol. 235, American Society of Mechanical Engineers.

Becht IV, C., 1999, "Evaluation of Plastic Strain Concentration in Bellows," *Fracture, Design, Analysis of Pressure Vessels, Heat Exchangers, Piping Components, and Fitness for Service—1999*, PVP-Vol. 388, American Society of Mechanical Engineers.

Becht IV, C., 2000a, "The Effect of Bellows Convolution Profile on Stress Distribution and Plastic Strain Concentration," *Fitness for Service, Stress Classification and Expansion Joint Design*, PVP-Vol. 401, American Society of Mechanical Engineers.

Becht IV, C., 2000b, "An Evaluation of EJMA Stress Calculations for Unreinforced Bellows," *Fitness for Service, Stress Classification and Expansion Joint Design*, PVP-Vol. 401, American Society of Mechanical Engineers.

Becht IV, C., 2000c, "Fatigue of Bellows, A New Design Approach," *International Journal of Pressure Vessels and Piping*, 77(13), pp. 843–850.

Becht IV, C., 2001, *Behavior of Bellows*, WRC Bulletin 466, Welding Research Council, Nov 2001.

- Bednar, H., 1986, *Pressure Vessel Design Handbook*, Van Nostrand Reinhold, New York.
- Bergman, E. O., 1960, "The New-Type Code Chart for the Design of Vessels Under External Pressure," *Pressure Vessel and Piping Design, Collected Papers 1927-1959*, American Society of Mechanical Engineers, pp. 647-654.
- Biersteker, M., Dietemann, C., Sureshwala, S., and Haupt, R. W., 1991, "Qualification of Non-standard Piping Product Form for ASME Code for Pressure Piping, B31 Applications," *Codes and Standards and Applications for Design and Analysis of Pressure Vessels and Piping Components*, PVP-Vol. 210-1, American Society of Mechanical Engineers.
- Bijlaard, P. P., "Stresses from Local Loadings in Cylindrical Pressure Vessels," *Trans. A.S.M.E.*, 77, 802-816 (1955).
- Bijlaard, P. P., "Stresses from Radial Loads in Cylindrical Pressure Vessels," *Welding Jnl.*, 33 (12), Research supplement, 615-s to 623-s (1954).
- Bijlaard, P. P., "Stresses from Radial Loads and External Moments in Cylindrical Pressure Vessel," *Ibid.*, 34 (12), Research supplement, 608-s to 617-s (1955).
- Bijlaard, P. P., "Computation of the Stresses from Local Loads in Spherical Pressure Vessels or Pressure Vessel Heads," *Welding Research Council Bulletin* No. 34, (March 1957).
- Bijlaard, P. P., "Local Stresses in Spherical Shells from Radial or Moment Loadings," *Welding Jnl.*, 36 (5), Research Supplement, 240-s to 243-s (1957).
- Bijlaard, P. P., "Stresses in a Spherical Vessel from Radial Loads Acting on a Pipe," *Welding Research Council Bulletin* No. 49, 1-30 (April 1959).
- Bijlaard, P. P., "Stresses in a Spherical Vessel from External Moments Acting on a Pipe," *Ibid.*, No. 49, 31-62 (April 1959).
- Bijlaard, P. P., "Influence of a Reinforcing Pad on the Stresses in a Spherical Vessel Under Local Loading," *Ibid.*, No. 49, 63-73 (April 1959).
- Bijlaard, P. P., "Stresses in Spherical Vessels from Local Loads Transferred by a Pipe," *Ibid.*, No. 50, 1-9 (May 1959).
- Bijlaard, P. P., "Additional Data on Stresses in Cylindrical Shells Under Local Loading," *Ibid.*, No. 50, 10-50 (May 1959).
- Boardman, H. C., 1943, "Formulas for the Design of Cylindrical and Spherical Shells to Withstand Uniform Internal Pressure," *Water Tower*, 30, pp. 14-15.
- Broyles, R., 1989, "Bellows Instability," *Metallic Bellows and Expansion Joints—1989*, American Society of Mechanical Engineers.
- Broyles, R., 1994, "EJMA Design Equations," *Developments in a Progressing Technology—1994*, PVP-Vol. 279, American Society of Mechanical Engineers.
- Copper Development Association, 1995, *The Copper Tube Handbook*, Copper Development Association, New York.
- EJMA, *Standards of the Expansion Joint Manufacturer's Association*, EJMA, Tarrytown, New York.
- Holt, M., 1960, "A Procedure for Determining the Allowable Out-of-Roundness for Vessels Under External Pressure," *Pressure Vessel and Piping Design, Collected Papers 1927-1959*, American Society of Mechanical Engineers, pp. 655-660.
- ISO 15649, 2001, Petroleum and natural gas industries-piping.

- Kobatake, K., et al, 1986, "Simplified Fatigue Life Evaluation Method of Bellows Expansion Joints at Elevated Temperature," *Fatigue and Fracture Assessment by Analysis and Testing*, PVP-Vol. 103, American Society of Mechanical Engineers.
- Langer, B. F., 1961, "Design of Pressure Vessels for Low Cycle Fatigue," *ASME Paper No. 61-WA-18*, American Society of Mechanical Engineers.
- Markl, A., 1960a, "Fatigue Tests of Piping Components," *Pressure Vessel and Piping Design, Collected Papers, 1927-1959*, American Society of Mechanical Engineers, pp. 402-418.
- Markl, A., 1960b, "Fatigue Tests of Welding Elbows and Comparable Double-Mitre Bends," *Pressure Vessel and Piping Design, Collected Papers, 1927-1959*, American Society of Mechanical Engineers, pp. 371-393.
- Markl, A., 1960c, "Fatigue Tests on Flanged Assemblies," *Pressure Vessel and Piping Design, Collected Papers, 1927-1959*, American Society of Mechanical Engineers, pp. 91-101.
- Markl, A., 1960d, Piping-Flexibility Analysis, *Pressure Vessel and Piping Design, Collected Papers, 1927-1959*, American Society of Mechanical Engineers, pp. 419-441.
- Mershon, J., Mokhtarian, K., Ranjan G., and Rodabaugh, E., 1984, *Local Stresses in Cylindrical Shells Due To External Loadings on Nozzles—Supplement to WRC Bulletin No. 107*, Bulletin 297, Welding Research Council, New York.
- MSS SP-42, *Class 150 Corrosion-Resistant Gate, Globe, Angle and Check Valves with Flanged and Buttweld Ends*, Manufacturers Standardization Society of the Valve and Fittings Industry.
- MSS SP-43, *Wrought Stainless Steel Buttwelding Fittings*, Manufacturers Standardization Society of the Valve and Fittings Industry.
- MSS SP-44, *Steel Pipe Line Flanges*, Manufacturers Standardization Society of the Valve and Fittings Industry.
- MSS SP-51, *Class 150LW Corrosion Resistant Cast Flanges and Flanged Fittings*, Manufacturers Standardization Society of the Valve and Fittings Industry.
- MSS SP-58, *Pipe Hangers and Supports—Materials, Design, and Manufacture*, Manufacturers Standardization Society of the Valve and Fittings Industry.
- MSS SP-65, *High Pressure Chemical Industry Flanges and Threaded Stubs for Use with Lens Gaskets*, Manufacturers Standardization Society of the Valve and Fittings Industry.
- MSS SP-70, *Cast Iron Gate Valves, Flanged and Threaded Ends*, Manufacturers Standardization Society of the Valve and Fittings Industry.
- MSS SP-71, *Cast Iron Swing Check Valves, Flanged and Threaded Ends*, Manufacturers Standardization Society of the Valve and Fittings Industry.
- MSS SP-72, *Ball Valves with Flanged or Buttwelding Ends for General Service*, Manufacturers Standardization Society of the Valve and Fittings Industry.
- MSS SP-75, *Specifications for High Test Wrought Buttwelding Fittings*, Manufacturers Standardization Society of the Valve and Fittings Industry.
- MSS SP-79, *Socket-Welding Reducer Inserts*, Manufacturers Standardization Society of the Valve and Fittings Industry.
- MSS SP-80, *Bronze Gate, Globe, Angle and Check Valves*, Manufacturers Standardization Society of the Valve and Fittings Industry.

MSS SP-81, *Stainless Steel, Bonnetless, Flanged, Knife Gate Valves*, Manufacturers Standardization Society of the Valve and Fittings Industry.

MSS SP-83, *Class 3000 Steel Pipe Unions, Socket-Welding and Threaded*, Manufacturers Standardization Society of the Valve and Fittings Industry.

MSS SP-85, *Cast Iron Globe and Angle Valves, Flanged and Threaded Ends*, Manufacturers Standardization Society of the Valve and Fittings Industry.

MSS SP-88, *Diaphragm Type Valves*, Manufacturers Standardization Society of the Valve and Fittings Industry.

MSS SP-95, *Swage(d) Nipples and Bull Plugs*, Manufacturers Standardization Society of the Valve and Fittings Industry.

MSS SP-97, *Integrally Reinforced Forged Branch Outlet Fittings—Socket Welding, Threaded, and Buttwelding Ends*, Manufacturers Standardization Society of the Valve and Fittings Industry.

MSS SP-105, *Instrument Valves for Code Applications*, Manufacturers Standardization Society of the Valve and Fittings Industry.

NEMA SM 23, *Steam Turbines for Mechanical Drive Service*, National Electrical Manufacturers Association.

Osweiler, F., 1989, "Design of an Expansion Joint by a Finite Element Program—Comparison with the EJMA Standards," *Metallic Bellows and Expansion Joints—1989*, PVP-Vol. 168, American Society of Mechanical Engineers.

Robinson, E., 1960, "Steam-Piping Design to Minimize Creep Concentrations," *Pressure Vessel and Piping Design, Collected Papers 1927–1959*, American Society of Mechanical Engineers, pp. 451–466.

Rodabaugh, E., and George, H., 1960, "Effect of Internal Pressure on Flexibility and Stress-Intensification Factors of Curved Pipe or Welding Elbows," *Pressure Vessel and Piping Design, Collected Papers 1927–1959*, American Society of Mechanical Engineers, pp. 467–477.

SAE J513, *Refrigeration Tube Fittings—General Specifications*, Society of Automotive Engineers.

SAE J514, *Hydraulic Tube Fittings*, Society of Automotive Engineers.

SAE J518, *Hydraulic Flange Tube, Pipe, and Hose Connections, Four-Bolt Split Flanged Type*, Society of Automotive Engineers.

Saunders, H. E., and Windenburg, D., 1960, "Strength of Thin Cylindrical Shells Under External Pressure," *Pressure Vessel and Piping Design, Collected Papers 1927–1959*, American Society of Mechanical Engineers, pp. 600–611.

Short II, W. E., 1989, "Overview of Chapter VII, Nonmetallic Piping and Piping Lined with Nonmetals in the ASME B31.3 Chemical Plant & Petroleum Refinery Piping Code," *Codes and Standards and Applications for Design and Analysis of Pressure Vessel and Piping Components—1989*, PVP-Vol 161, American Society of Mechanical Engineers.

Short II, W. E., 1992, "Coverage of Non-Metals in the ASME B31.3 Chemical Plant and Petroleum Refinery Piping Code," *Journal of Process Mechanical Engineers*, 206, pp. 67–72.

Short II, W. E., Leon, G. F., Widera, G. E. O., and Zui, C. G., 1996, *Literature Survey and Interpretive Study on Thermoplastic and Reinforced-Thermosetting-Resin Piping and Component Standards*, WRC 415, Welding Research Council.

Sims, J., 1986, "Development of Design Criteria for a High Pressure Piping Code," *High Pressure Technology—Design, Analysis, and Safety of High Pressure Equipment*, PVP-Vol. 110, D. P. Kendall, ed., American Society of Mechanical Engineers.

SNT-TC-1A, *Recommended Practice for Nondestructive Testing Personnel Qualification and Certification*, Society for Nondestructive Testing.

Takezono, S., 1971, "Fatigue Strength of Expansion Joints of Pressure Vessels," *Bulletin of the Japanese Society of Mechanical Engineers*, 14(76).

Tsukimori, K., et al, 1989, "Fatigue and Creep-Fatigue Life Prediction of Bellows," *Metallic Bellows and Expansion Joints—1989*, PVP-Vol. 168, American Society of Mechanical Engineers.

Wichman, K., Hopper A., and Mershon J., 1979, *Local Stresses in Spherical and Cylindrical Shells due to External Loadings*, Bulletin 107, Welding Research Council.

Windenburg, D., 1960, "Vessels Under External Pressure: Theoretical and Empirical Equations Represented in Rules for the Construction of Unfired Pressure Vessels Subjected to External Pressure," *Pressure Vessel and Piping Design, Collected Papers 1927–1959*, American Society of Mechanical Engineers, pp. 625–632.

Windenburg, D., and Trilling, C., 1960, "Collapse by Instability of Thin Cylindrical Shells Under External Pressure," *Pressure Vessel and Piping Design, Collected Papers 1927–1959*, American Society of Mechanical Engineers, pp. 612–624.

Yamamoto, S., et al, 1986, "Fatigue and Creep-Fatigue Testing of Bellows at Elevated Temperature," *Fatigue and Fracture Assessment by Analysis and Testing*, PVP-Vol. 103, American Society of Mechanical Engineers.

Yamashita, T., et al, 1989, "Ratchetting Mechanism of Bellows and Evaluation Method," *Metallic Bellows and Expansion Joints—1989*, PVP-Vol. 168, American Society of Mechanical Engineers.

INDEX

A

- (parameter), 30
 - chart for determining, 31–32
- Allowable stress, 18–19
 - limits of, due to occasional loads, 54–55
 - materials used above highest temperature for provided, 88
 - for nonmetallic piping, 128
 - for supports, 75–76
 - for thermal expansion, 57–61
- Allowable stress tables, using, 95–100
- Alternative leak test, 131
- American National Standards Institute (ANSI), 2
- American Society of Mechanical Engineers (ASME), 2
- American Standards Association (ASA), 2
- ANSI (American National Standards Institute), 2
- Arc welding processes, 101–106
- ASA (American Standards Association), 2
- ASME (American Society of Mechanical Engineers), 2
- ASME B31.3, 1
 - addenda to, 6
 - answering questions about, 7
 - changing, 7
 - development and maintenance of, 5–6
 - editions of, 6
 - history of, 1–2
 - intent of, 3–4
 - organization of, 9–13
 - organizational features of, 10
 - responsibilities within, 4–5
 - scope of, 2–3
- Assembly
 - of bolted joints, 114–115
 - miscellaneous requirements for, 117
 - rules for, 101–117
- Attached equipment, load limits for, 85–86

B

- (parameter), 30
 - chart for determining, 33
- Barlow equation, 26
- Bellows expansion joints, 153–163
 - correlation testing, 158–160
 - elevated-temperature design, 160

- fatigue design for other alloys, 157–158
- fatigue design of, 155
- future developments in, 162–163
- pressure design of, 153–155
- pressure thrust considerations in, 162–163

Bends

- in-plane and out-plane bending moments in, 67
- limitations on, 47–48
- pressure design of, 37–39

Blanks, pressure design of, 41

Blind flanges, pressure design of, 41

Board on Pressure Technology Codes and Standards, 6

Boardman equation, 25–27

Boardman representation, empirical, 26

Bolted joints, assembly of, 114–115

Bolting, limitations on, 48–49

Bonding processes, 134–140

Branch connections

- under external pressure, additional considerations for, 36
- in-plane and out-plane bending moments in, 67
- limitations on, 47–48
- presumed to be acceptable, 36
- welded, pressure design of, 33–36

Brazing procedures, 113–114

Buckling check, 30

Butt-and-wrap joints, 135–136

Butt-welded steel pipe, Markl fatigue curve for, 60

C

Carbon steel materials, minimum temperature without impact testing for, 91

Category M piping, 9–10, 141–143

Charpy V-notch impact testing, 93–94

Closures, pressure design of, 40–41

Coincident conditions, 15

Cold spring, 69–70

Components

- iron, pressure ratings of, 172–174
- limitations on, 45–49
- listed, *see* Listed components
- standard, pressure ratings of, 172–184
- steel, pressure ratings of, 175–181
- unlisted, pressure design of, 42–43

- Computer Flexibility Analysis, 185–188
- Computer stress analysis, 65
- Constant-effort-spring hanger table, 80
- Conversion factors, 250–256
- Copper and copper alloys, pressure ratings of, 182–199
- Cylinders
 - longitudinal stress in, 29
 - under pressure, basic stress calculations for, 28–29
- D**
- Deflection limits, 53
- Deformation-controlled behavior, load-controlled behavior versus, 58
- Design conditions, 15–18
- Design criteria for thermal stress, 57–61
- Design minimum temperature, 18
- Design pressure, 15
- Design temperature, 15–18
- Designer, 4
- Displacement cycle conditions, combining different, 61
- E**
- Elastic buckling, 30
- Elastic follow-up, 71–73
- Elastic modulus variations due to temperature, 73
- Elbows, pressure design of, 37–39
- Engineered safeguards, 12–13
- Equipment
 - attached, load limits for, 85–86
 - reducing loads on, 86
- Erection, rules for, 101–117
- Erector, 5
- Examination, 119–123
 - acceptance criteria for, 122
 - of high-pressure piping, 150–151
 - of nonmetallic piping, 140
 - of nonmetallics, 122
 - progressive, 120–121
 - required, *see* Required examination
 - types of, 121–122
- Examination requirements, overview of, 119–120
- Expansion joints
 - bellows, *see* Bellows expansion joints
 - piping systems with, field test of, 161–162
 - practical guide, 224–249
 - shop test of, 160–161
- External pressure
 - branch connections under, additional considerations for, 36
 - for high-pressure piping, 147–148
 - pressure design of straight pipe under, 30–33
- Extruded outlet header, pressure design of, 36, 37–38
- F**
- Fabrication, rules for, 101–117
- Fabricator, 5
- Fatigue, high-pressure piping design for, 148–149
- Fatigue design of bellows expansion joints, 155
- FCAW (flux-cored arc welding), 104–105
- Field test of piping systems with expansion joints, 161–162
- Film coefficient, effective outside, 17
- Fittings, limitations on, 47–48
- Flanged joints, limitations on, 49–50
- Flanges
 - limitations on, 46–47
 - pressure design of, 41
- Flexibility analysis, 63–73, 189–223
 - equations for, 68–69
 - formal, exemptions from, 64
 - fundamental principles of, 63
 - required, 63–64
- Fluid service categories, 9–10
- Fluid service Category M, 9–10, 141–143
- Fluid service limitations, 95
- Fluid temperature, 16
- Flux-cored arc welding (FCAW), 104–105
- G**
- Gas metal arc welding (GMAW), 103–104
- Gas tungsten arc welding (GTAW), 102–103
- GMAW (gas metal arc welding), 103–104
- Groove depths, 165
- GTAW (gas tungsten arc welding), 102–103
- H**
- Hanger selection tables, 78, 80
- Hardness testing, 122
- Heat fusion, 138–139
- Heat treatment, 108, 112
 - governing thickness for, 112–113
 - requirements for, 110–111
- High-pressure piping, 145–152
 - design for sustained and occasional loads, 148
 - design for thermal expansion and fatigue, 148–149
 - examination of, 150–151
 - external pressure for, 147–148
 - fabrication of, 150
 - materials for, 149–150
 - pressure design of, 146–147
 - records for, 152
 - testing, 152
- Hoop stress, 26
- Hot gas welding, 137–138
- Hydropneumatic test, 127
- Hydrostatic test, 126

INDEX

A

- (parameter), 30
 - chart for determining, 31-32
- Allowable stress, 18-19
 - limits of, due to occasional loads, 54-55
 - materials used above highest temperature for provided, 88
 - for nonmetallic piping, 128
 - for supports, 75-76
 - for thermal expansion, 57-61
- Allowable stress tables, using, 95-100
- Alternative leak test, 131
- American National Standards Institute (ANSI), 2
- American Society of Mechanical Engineers (ASME), 2
- American Standards Association (ASA), 2
- ANSI (American National Standards Institute), 2
- Arc welding processes, 101-106
- ASA (American Standards Association), 2
- ASME (American Society of Mechanical Engineers), 2
- ASME B31.3, 1
 - addenda to, 6
 - answering questions about, 7
 - changing, 7
 - development and maintenance of, 5-6
 - editions of, 6
 - history of, 1-2
 - intent of, 3-4
 - organization of, 9-13
 - organizational features of, 10
 - responsibilities within, 4-5
 - scope of, 2-3
- Assembly
 - of bolted joints, 114-115
 - miscellaneous requirements for, 117
 - rules for, 101-117
- Attached equipment, load limits for, 85-86

B

- (parameter), 30
 - chart for determining, 33
- Barlow equation, 26
- Bellows expansion joints, 153-163
 - correlation testing, 158-160
 - elevated-temperature design, 160

- fatigue design for other alloys, 157-158
- fatigue design of, 155
- future developments in, 162-163
- pressure design of, 153-155
- pressure thrust considerations in, 162-163

Bends

- in-plane and out-plane bending moments in, 67
- limitations on, 47-48
- pressure design of, 37-39
- Blanks, pressure design of, 41
- Blind flanges, pressure design of, 41
- Board on Pressure Technology Codes and Standards, 6
- Boardman equation, 25-27
- Boardman representation, empirical, 26
- Bolted joints, assembly of, 114-115
- Bolting, limitations on, 48-49
- Bonding processes, 134-140
- Branch connections
 - under external pressure, additional considerations for, 36
 - in-plane and out-plane bending moments in, 67
 - limitations on, 47-48
 - presumed to be acceptable, 36
 - welded, pressure design of, 33-36
- Brazing procedures, 113-114
- Buckling check, 30
- Butt-and-wrap joints, 135-136
- Butt-welded steel pipe, Markl fatigue curve for, 60

C

- Carbon steel materials, minimum temperature without impact testing for, 91
- Category M piping, 9-10, 141-143
- Charpy V-notch impact testing, 93-94
- Closures, pressure design of, 40-41
- Coincident conditions, 15
- Cold spring, 69-70
- Components
 - iron, pressure ratings of, 172-174
 - limitations on, 45-49
 - listed, *see* Listed components
 - standard, pressure ratings of, 172-184
 - steel, pressure ratings of, 175-181
 - unlisted, pressure design of, 42-43

- Pipe (*Cont.*)
 - straight, pressure design of, *see* Straight pipe, pressure design of
 - Pipe bends, 113
 - Pipe support elements, *see* Supports
 - Piping
 - Category M, 9–10, 141–143
 - examination of, *see* Examination
 - high-pressure, *see* High-pressure piping
 - jacketed, 127
 - metallic, required examination for, 123
 - new, 3
 - nonmetallic, *see* Nonmetallic piping
 - plastic, thermosetting, 134–135
 - vacuum, 127
 - Piping components, *see* Components
 - Piping designer, 4
 - Piping flexibility, increasing, 70–71
 - Piping systems with expansion joints, field test of, 161–162
 - Plastic piping, thermosetting, 134–135
 - Plastics, bonding of, 133–140
 - Pneumatic test, 126
 - Preheat temperatures, 109
 - Preheating requirements, 108
 - Pressure design, 23–43
 - of bellows expansion joints, 153–155
 - of bends, 37–39
 - of blanks, 41
 - of blind flanges, 41
 - of closures, 40–41
 - of elbows, 37–39
 - of extruded outlet header, 36, 37–38
 - of flanges, 41
 - of high-pressure piping, 146–147
 - internal, methods for, 23–24
 - of miter bends, 39–40
 - of nonmetallic piping, 130–131
 - of reducers, 42
 - of straight pipe
 - under external pressure, 30–33
 - for internal pressure, 24–29
 - of unlisted components, 42–43
 - of welded branch connections, 33–36
 - Pressure design thickness, 24
 - Pressure Equipment Directive (PED), European, 2
 - Pressure ratings
 - of copper and copper alloys, 181–184
 - of iron components, 172–174
 - of listed components, 165–184
 - of standard components, 172–184
 - Pressure Technology Codes and Standards, Board on, 6
 - Pressure test requirements, overview of, 125
 - Pressure testing, 125–128
 - Pressure thrust considerations in bellows expansion joints, 162–163
 - Pressure variations, allowances for, 19–20
 - Pressure vessels, load limits for, 85–86
 - Primary longitudinal stress, 53
 - Process Piping Code*, *see* ASME B31.3
 - Progressive examination, 120–121
 - PT (liquid-penetrant examination), 121
- ## R
- Radiographic examination (RT), 121
 - Reducers, pressure design of, 42
 - Refractory-lined pipe, wall temperature for, 16–18
 - Reinforced-bellows fatigue curve development, 157
 - Required examination, 122
 - for metallic piping, 123
 - RT (radiographic examination), 121
- ## S
- Safeguarding, 11–13
 - SAW (submerged arc welding), 105–106
 - Sensitive leak test, 127
 - Set pressure, 21
 - Shakedown, stress-strain behavior illustrating, 59
 - Shielded metal arc welding (SMAW), 101–102
 - Shop test of expansion joints, 160–161
 - Slip-on flanges, 46–47
 - SMAW (shielded metal arc welding), 101–102
 - Soldering, 113
 - Solvent-cement joining, 136–137
 - Spring design, 77–79
 - Standard components, pressure ratings of, 172–184
 - Standards Committee, 6
 - Steel components, pressure ratings of, 175–181
 - Straight pipe, pressure design of
 - under external pressure, 30–33
 - for internal pressure, 24–29
 - Strain concentration, 71–73
 - Strength reduction factor, 54–55
 - Stress
 - allowable, *see* Allowable stress
 - hoop, 26
 - longitudinal, *see* Longitudinal stress
 - peak, 68
 - thermal, *see* Thermal stress
 - Stress analysis, computer, 65
 - Stress calculations, basic, for cylinders under pressure, 28–29
 - Stress classification, 79
 - Stress concentration factor, 66
 - Stress intensification factors, 65–68
 - Stress-strain behavior illustrating shakedown, 59
 - Submerged arc welding (SAW), 105–106
 - Supports, 75–81
 - design of, 76

fabrication of, 84
materials and allowable stress for, 75–76
overview of, 75
Sustained loads, high-pressure piping design for, 148
Sustained longitudinal stress, 53–54

T

Task Groups, 5–6

Temperature

design, 15–18
design minimum, 18
elastic modulus variations due to, 73
fluid, 16
highest, materials used above, for provided allowable stress, 88
metal, 16
minimum, without impact testing, for carbon steel materials, 91
preheat, 109
reduction in minimum design metal, without impact testing, 92
wall, for refractory-lined pipe, 16–18
Temperature variations, allowances for, 19–20
Thermal expansion
allowable stress for, 57–61
high-pressure piping design for, 148–149
Thermal stress, 57
design criteria for, 57–61
Thermosetting plastic piping, 134–135
Thickness, governing, for heat treatment, 112–113
Thread depths, 165–166

Threaded joints, limitations on, 50
Toughness tests, low-temperature, requirements for, 90
Tubing joints, limitations on, 50–51

U

Ultrasonic examination (UT), 121
Unlisted components, pressure design of, 42–43
Unlisted materials, qualification of, 85–86
Unreinforced-bellows fatigue curve development, 155–156
UT (ultrasonic examination), 121

V

Vacuum piping, 131
Valves, limitations on, 45–46
Variable-spring hanger table, 78
Vibration, 62
Visual examination (VT), 121

W

Wall temperature for refractory-lined pipe, 16–18
Wall thickness, 25
Wall thickness calculation, sample, 28
Weight of piping materials, 190–223
Welded branch connections, pressure design of, 33–36
Welded joint details, 115–117
Welded joints, limitations on, 49
Welder performance qualification, 107–108
Welding procedure qualification record, 106–107
Welding procedure specification, 106
Welding requirements, general, 101

