CHAPTER A7
BOLTED JOINTS

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INTRODUCTION

While other chapters of the *Piping Handbook* deal with the pressure integrity of the piping system, this chapter deals with managing the leak integrity of bolted flanged systems. It covers the main elements of a bolted joint system to provide an understanding of the bolted joint connection and the science of joint sealing.

This chapter focuses exclusively on bolted joints subjected to internal pressures. While integrity of mechanical (structural) joints are also critical, they are not covered in this book.

Oil, gas, and power plants and other process industries are under constant pressure to work their plants at maximum design limitations and for longer periods. The bolted joint is often regarded as the weak link in the plant's pressure envelope. Whether a pipe flange, heat exchanger, reactor manway, or valve bonnet, the joint integrity relies not only on the mechanical design of the flange and its components, but also on its condition, maintenance, and assembly. Plant personnel are looking for equipment to achieve leak-free joints with reduced shutdown periods while increasing the time between shutdowns. Similarly, flanged joints in other piping and distribution systems found throughout industrial, commercial, and residential facilities are required to maintain their structural integrity and leak tightness.

Several standards have been written to enable designers to design bolted joints. Compliance to the requirements of these standards ensures mechanical integrity of bolted joints. However, these standards do not provide adequate and effective requirements or guidelines to assure leak integrity of flanged joints.

To achieve leak integrity, a broader view of the bolted flange joint as a system must be adopted. Ideally, a process is to be followed that manages the key elements of the bolted system, which allows the design potential of the bolted joint to be realized and helps in achieving continued leak-free operation.

This chapter reviews the process required to achieve flange-joint integrity.

COST OF A LEAK

Some believe leaking flanges are normal and leaks cannot be prevented. Some also hold similar views about health and safety. Safety professionals now know that accidents can be prevented and that the goal of zero accidents is achievable. The goal of zero leaks is also achievable. Leaks are still very commonplace. A thorough survey throughout North American industry, performed by Pressure Vessel Research Council (PVRC), concluded that the average plant experiences 180 leaks per year. A breakdown of the severity of these leaks is shown in Fig. A7.1.

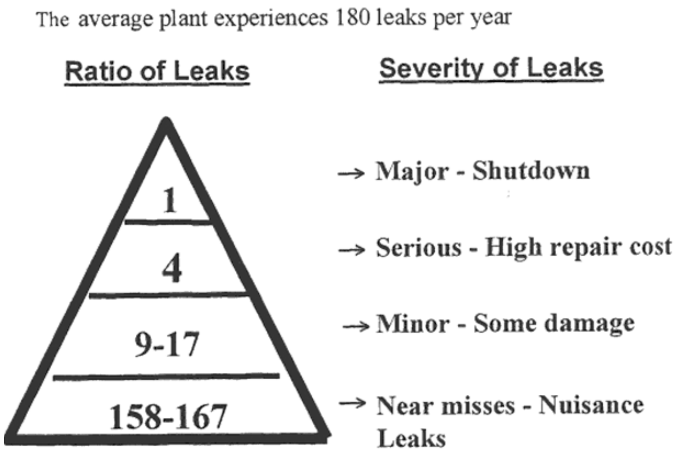


FIGURE A7.1 Industry leak study (PVRC Study, July 1985).

In a manner similar to accident ratio statistics, there is a relationship between minor, serious, and other dangerous events. All events represent failure in control. Failures in control that result in leaks cost industry millions of dollars yearly due to:

- Emission
- Pollution, spills
- Rework
- Leak sealing
- Fires
- Lost product
- Late schedules
- Forced shut downs—production losses

Control is the issue. Leaks are controllable. Control is achieved by implementation of a Flange Joint Integrity program. Joint Integrity is a control program that becomes an integral part of a plant's safety and reliability.

THE PROCESS OF JOINT INTEGRITY

To assist in managing a process, ask yourself the following questions: why, what, who, and how?

Why do we need a Flange Joint Integrity program? This was addressed in the previous section, "Cost of a Leak." The stakes are enormous. A Flange Joint Integrity program will help improve plant safety and reliability while reducing its environmental impact.

What do we need to control? The operating environment, the components, and assembly all need to be controlled.

Who do we need to control? The designers, field operatives, and supervisors.

How do we control? Train personnel to required competency. Design components using latest engineering standards. Develop best practices for assembly and maintenance. Implement a quality assurance program that provides traceability and ensures compliance to specifications.

There are over 120 variables that affect flange joint integrity. These can be controlled through the following categories:

- Environment (internal and external)
- Components
- Assembly

The internal environment outlines the design and operating conditions of temperature, pressure, and fluid. With the external environment, consideration is given to location of the flange, whether it is operating in air or sub-sea, and externally applied piping loads. An understanding of the environment is crucial to the design and selection of the appropriate components with the correct assembly methods.

The components include the most appropriately designed and selected flange, gasket, and bolting, commensurate with the risk dictated by the environment.

Assembly includes checking the condition of the components and proceeding according to established procedures. Proper assembly requires that

- Flange faces meet the standards
- Gasket-seating stress is achieved
- Bolts, nuts, and gaskets are free of defects
- Appropriate lubrication is used

Execution requires trained, competent people using the correct tools and following procedures.

The steps in the joint integrity process are shown in Fig. A7.2.

FLANGE JOINT COMPONENTS

Flanges

There are numerous types of flanges available. The type and material of flanges is dependent on the service environment. The service environment is specified in the Piping and Instrumentation Drawing (P&ID) and other design documents. Refer

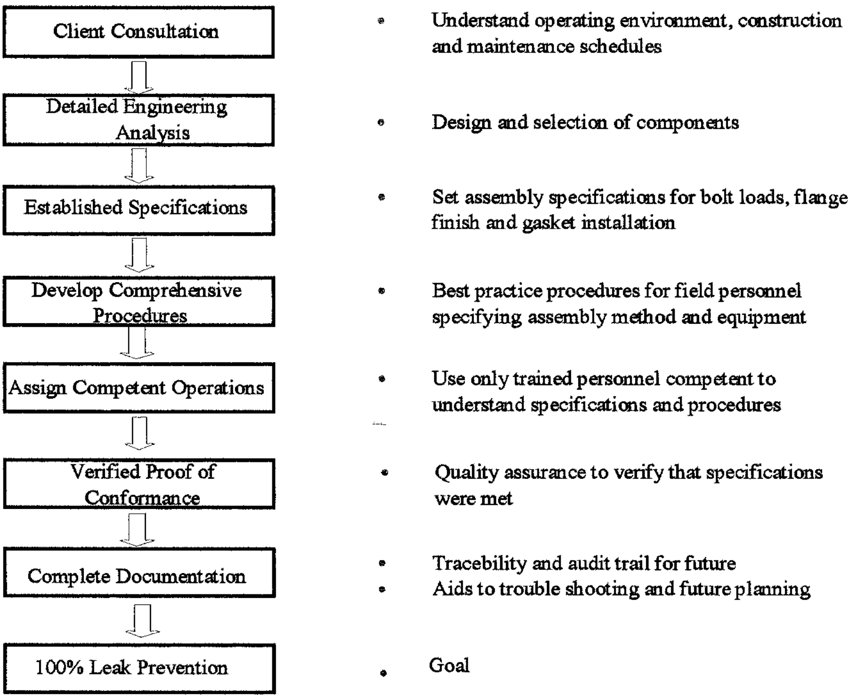


FIGURE A7.2 Steps in joint integrity process.

to Chap. B1 of this handbook. Selection of flange materials is done in conjunction with piping specification.

Flange Standards

There are a variety of standards used in the design and selection of flanges. The following codes and standards relate to pipe flanges:

ASME Codes and Standards:

B16.1	Cast Iron Flanges and Flanged Fittings
B16.5	Pipe Flanges and Flanged Fittings
B16.24	Bronze Flanges and Fittings–150 and 300 Classes
B16.42	Ductile Iron Pipe Flanges and Flanged Fittings–150 and 300 Classes
B16.47	Large Diameter Steel Flanges

Section VIII

Division 1	Pressure Vessels
Appendix 3	Mandatory Rules for Bolted Flange Connections

ANSI/AWWA Standards

C-111/A21.15	Flanged C.I. Pipe with Threaded Flanges
C-207	Steel Pipe Flanges

API Specifications

Spec 6A-96	Specification for Wellhead and Christmas Tree Equipment

The two most commonly used flange standards for process and utilities pipework are ASME B16.5 and BS 1560 (British Standards). API 6A (American Petroleum Institute) specifies flanges for wellhead and Christmas tree equipment. Less common flange standards which may be encountered are flanges for metric or DIN standards.

Refer to Chap. A4 for other codes and standards.

Flange-End Connection

The flange-end connection defines the way in which it is attached to the pipe. The following are commonly available standard flange end types:

Weld-Neck (WN) Flange. Weld-neck flanges are distinguished from other types by their long, tapered hub and gentle transition to the region where the WN flange is butt-welded to the pipe. The long, tapered hub provides an important reinforcement of the flange, increasing its strength and resistance to dishing. WN flanges are typically used on arduous duties involving high pressures or hazardous fluids.

The butt-weld may be examined by radiography or ultrasonic inspection. Usually, the butt-welds are subject to visual, surface, or volumetric examinations, or a combination thereof, depending on the requirements of the code of construction for piping or a component. There is, therefore, a high degree of reliability in the integrity of the weld. A butt-weld also has good fatigue performance, and its presence does not induce high local stresses in the pipework.

Socket-Weld (SW) Flange. Socket-weld flanges are often used on hazardous duties involving high pressure but are limited to a nominal pipe size NPS 2 (DN 50) and smaller. The pipe is fillet-welded to the hub of the SW flange. Radiography is not practical on the fillet weld; therefore correct fitting and welding is crucial. The fillet weld may be inspected by surface examination, magnetic particle (MP), or liquid penetrant (PT) examination methods.

Slip-on Flanges. Slip-on flanges are preferred to weld-neck flanges by many users because of their initial low cost and ease of installation. Their calculated strength under internal pressure is about two-thirds of that of weld-neck flanges. They are typically used on low-pressure, low-hazard services such as fire water, cooling water, and other services. The pipe is ''double-welded'' to both the hub and the bore of the flange, but, again, radiography is not practical. MP, PT, or visual examination is used to check the integrity of the weld. When specified, the slip-on flanges are used on pipe sizes greater than NPS 2½ (DN 65).

Composite Lap-Joint Flange. This type of flanged joint is typically found on high alloy pipe work. It is composed of a hub, or ''stub end,'' welded to the pipe and a backing flange, or lapped flange, which is used to bolt the joint together. An alloy hub with a galvanized steel backing flange is cheaper than a complete alloy flange. The flange has a raised face, and sealing is achieved with a flat ring gasket.

Swivel-Ring Flange. As with the composite lap-joint flange, a hub will be butt-welded to the pipe. A swivel ring sits over the hub and allows the joint to be bolted together. Swivel-ring flanges are normally found on sub–sea services where the swivel ring facilitates flange alignment. The flange is then sealed using a ring-type joint (RTJ) metal gasket.

Blind Flange. Blind flanges are used to blank off the ends of piping, valves, and pressure vessel openings. From the standpoint of internal pressure and bolt loading, blind flanges, particularly in the larger sizes, are the most highly stressed of all the standard flanges. However, since the maximum stresses in a blind flange are bending stresses at the center, they can be safely permitted to be stressed more than other types of flanges.

These common flange types are shown in Fig. A7.3.

Flange Faces

There are five types of flange faces commonly found. The surface finish of the faces are specified in the flange standards quoted above.

Raised Face (RF). The raised face is the most common facing employed with bronze, ductile iron, and steel flanges. The RF is 1⁄16-in high for Class 150 and Class 300 flanges and 1⁄4-in high for all pressure classes, higher than Class 300. The facing on a RF flange has a concentric or phonographic groove with a controlled surface finish. Sealing is achieved by compressing a flat, soft, or semimetallic gasket between mating flanges in contact with the raised face portion of the flange.

Ring-Type Joint (RTJ). This type is typically used in the most severe duties, for example, in high-pressure-gas pipe work. Ring-type metal gaskets must be used on this type of flange facing.

RTJ for API 6A Type 6B, BS 1560 and ASME B16.5 Flanges
The seal is made by plastic deformation of the RTJ gasket into the groove in the flange, resulting in intimate metal-to-metal contact between the gasket and the flange groove. The faces of the two opposing flange faces do not come into contact because a gap is maintained by the presence of the gasket. Such RTJ flanges will normally have raised faces, but flat faces may also be used or specified.

RTJ for API 6A Type 6BX Flanges
API 6A Type 6BX flanges have raised faces. These flanges incorporate special metal ring joint gaskets. The pitch diameter of the ring is slightly greater than the pitch diameter of the flange groove. This factor preloads the gasket and creates a pressure-energized seal. A Type 6BX flange joint that does not achieve face-to-face contact will not seal and, therefore, must not be put into service.

Flat Face (FF). Flat-face flanges are a variant of raised face flanges. Sealing is achieved by compression of a flat nonmetallic gasket (very rarely a flat metallic

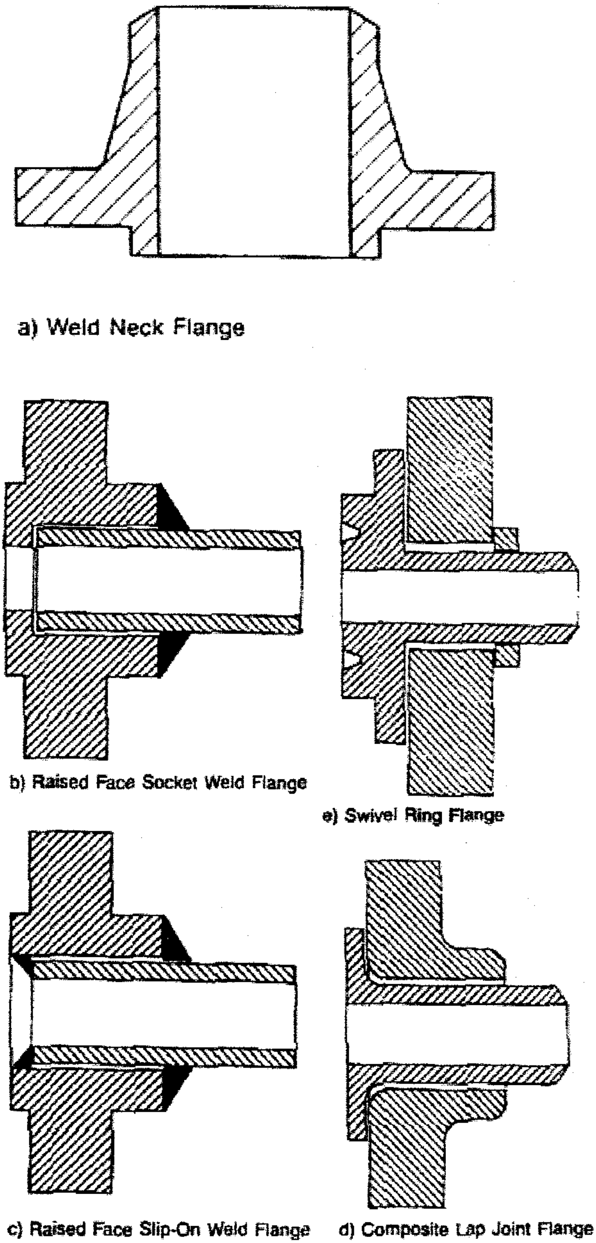


FIGURE A7.3 Common flange types.

gasket) between the grooved surfaces of the mating FF flanges. The gasket fits over the entire face of the flange. FF flanges are normally used on the least arduous of duties, such as low pressure water piping having Class 125 and Class 250 flanges and flanged valves and fittings. In this case the large gasket contact area spreads the flange loading and reduces flange stresses.

Note: Both ASME B16.5 and BS 1560 specify flat face flanges and raised face flanges as well as RTJ flanges. API 6A is specific to RTJ flanges only.

Male and Female Facings. The female face is $\frac{3}{16}$-in deep, the male face is $\frac{1}{4}$-in high, and both are smooth finished. The outer diameter of the female face acts to locate and retain the gasket. Custom male and female facings are commonly found on the heat exchanger shell to channel and cover flanges.

Tongue-and-Groove Facings. Tongue-and-groove facings are standardized in both large and small types. They differ from male-and-female in that the inside diameters of the tongue-and-groove do not extend into the flange base, thus retaining the gasket on its inner and outer diameter. These are commonly found on pump covers and valve bonnets.

Flange Specification and Identification

A flange is specified by the following information:

Type and Facing. The flange is specified according to whether it is, for example, "weld-neck RTJ" or "socket-weld RF." Ring joint facing and RTJ gasket dimensions for ASME B16.5 are shown in Table A7.1.

Nominal Pipe Size (NPS). This is a dimensionless designation to define the nominal pipe size (NPS) of the connecting pipe, fitting, or nozzle. Examples include NPS 4 and NPS 6.

Flange Pressure Class. This designates the pressure temperature rating of the flange, which is required for all flanges. Examples include Classes 150, 300, 900, and 1500.

Standard. Basic flange dimensions for ASME B16.5 are shown in Table A7.2. Examples include ASME B16.5, BS 1560, DIN or API 6A.

Material. A material specification for flanges must be specified and be compatible to the piping material specifications.

Pipe Schedule. This is only for WN, composite lap-joint and swivel-ring flanges where the flange bore must match that of the pipe, such as schedule 40, 80, 120, and 160.

Gaskets

A gasket is a material or combination of materials designed to clamp between the mating faces of a flange joint. The primary function of gaskets is to seal the irregularities of each face of the flange, preventing leakage of the service fluid from inside the flange to the outside. The gasket must be capable of maintaining a seal during the operating life of the flange, provide resistance to the fluid being sealed, and meet the temperatures and pressure requirements.

Gasket Standards

There are a variety of standards that govern dimensions, tolerances, and fabrication of gaskets. The more common international standards are

ASME B16.20-1997	Metallic Gaskets for Pipe Flanges, Ring-Joint, Spiral Wound and Jacketed
ASME B16.21-1990	Nonmetallic Flat Gaskets for Pipe Flanges
BS 4865 Part 1	Flat Ring Gaskets to Suit BS4504 and DIN Flange
BS 3381	Spiral Wound Gaskets to Suit BS 1560 Flanges
API 6A	Specification for Wellhead and Christmas Tree Equipment

Types of Gaskets

Gaskets can be defined into three main categories: nonmetallic, semimetallic, and metallic types.

Nonmetallic Gaskets. Usually composite sheet materials are used with flat-face flanges and low pressure class applications. Nonmetallic gaskets are manufactured with nonasbestos material or compressed asbestos fiber (CAF). Nonasbestos types include arimid fiber, glass fiber, elastomer, Teflon (PTFE), and flexible graphite gaskets. Full-face gasket types are suitable for use with flat-face (FF) flanges. Flat-ring gasket types are suitable for use with raised faced (RF) flanges.

Gasket dimensions for ASME B16.5 flanges are shown in Table A7.3. Gasket dimensions for ASME B16.47 Series A large diameter steel flanges are shown in Table A7.4a. Gasket dimensions for ASME B16.47 Series B large diameter steel flanges are shown in Table A7.4b.

Semimetallic Gaskets. Semimetallic gaskets are composites of metal and nonmetallic materials. The metal is intended to offer strength and resiliency, while the nonmetallic portion of a gasket provides conformability and sealability. Commonly used semimetallic gaskets are spiral wound, metal jacketed, camprofile, and a variety of metal-reinforced graphite gaskets. Semimetallic gaskets are designed for the widest range of operating conditions of temperature and pressure. Semimetallic gaskets are used on raised face, male-and-female, and tongue-and-groove flanges.

Spiral Wound Gaskets. Spiral wound gaskets are the most common gaskets used on raised face flanges. They are used in all pressure classes from Class 150 to Class 2500. The part of the gasket that creates the seal between the flange faces is the spiral wound section. It is manufactured by winding a preformed metal strip and a soft filler material around a metal mandrel. The inside and outside diameters are reinforced by several additional metal windings with no filler.

TABLE A7.1 Ring Joint Facing and RTJ Gasket Dimensions

a) ASME B16.5 Class 150

Nominal pipe size		½	¾	1	1½	2	3	4	6	8	10	12	14	16	18	20	24
Diameter of raised section	I			2½	3¼	4	5¼	6¾	8⅝	10¾	13	16	16¾	19	21½	23½	28
Groove pitch diameter	J	CLASS 150		1⅞	2 9/16	3¼	4½	5⅞	7⅝	9¾	12	15	15⅝	17⅞	20⅜	22	26½
Depth of groove	K	FLANGES		¼	¼	¼	¼	¼	¼	¼	¼	¼	¼	¼	¼	¼	¼
Width	L	NOT		11/32	11/32	11/32	11/32	11/32	11/32	11/32	11/32	11/32	11/32	11/32	11/32	11/32	11/32
Outside diameter	M	SPECIFIED		2 3/16	2⅞	3 9/16	4 13/16	6 3/16	7 15/16	10 1/16	12 5/16	15 5/16	15 15/16	18 3/16	20 11/16	22 5/16	26 13/15
Inside diameter	N	IN THESE		1 9/16	2¼	2 15/16	4 3/16	5 9/16	7 5/16	9 7/16	11 11/16	14 11/16	15 5/16	17 9/16	20 1/16	21 11/16	26 1/16
Width	O	SIZES		5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16
Thickness	P			½	½	½	½	½	½	½	½	½	½	½	½	½	½
R number				15	19	22	29	36	43	48	52	56	59	64	68	72	76

Notes:
1. All dimensions in inches.
2. Ring dimensions are per ANSI B16.20.

TABLE A7.1 Ring Joint Facing and RTJ Gasket Dimensions

b) ASME B16.5 Class 300

Nominal pipe size	1/2	3/4	1	1 1/2	2	3	4	6	8	10	12	14	16	18	20	24
Diameter of raised section I	2	2 1/2	2 3/4	3 9/16	4 1/4	5 3/4	6 7/8	9 1/2	11 7/8	14	16 1/4	18	20	22 5/8	25	19 1/2
Groove pitch diameter J	1 11/32	1 11/16	2	2 11/16	3 1/4	4 7/8	5 7/8	8 5/16	10 5/8	12 3/4	15	16 1/2	18 1/2	21	23	27 1/4
Depth of groove K	7/32	1/4	1/4	1/4	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16	3/8	5/16
Width L	9/32	11/32	11/32	11/32	15/32	15/32	15/32	15/32	15/32	15/32	15/32	15/32	15/32	15/32	17/32	21/32
Outside diameter M	1 18/32	2	2 5/16	3	3 11/16	5 5/16	6 5/16	8 3/4	11 1/16	13 3/16	15 7/16	16 15/16	18 15/16	21 7/16	23 1/2	27 7/8
Inside diameter N	1 3/32	1 3/8	1 11/16	2 3/8	2 13/16	4 7/16	5 7/16	7 1/8	10 3/16	12 5/16	14 9/16	16 1/16	18 1/16	20 9/16	22 1/2	26 5/8
Width O	1/4	5/16	5/16	5/16	7/16	7/16	7/16	7/16	7/16	7/16	7/16	7/16	7/16	7/16	1/2	5/8
Thickness P	3/8	1/2	1/2	1/2	5/8	5/8	5/8	5/8	5/8	5/8	5/8	5/8	5/8	5/8	11/16	11/16
R number	11	13	16	20	23	31	37	45	49	53	57	61	65	69	73	77

Notes:
1. All dimensions in inches.
2. Ring dimensions are per ANSI B16.20.

TABLE A7.1 Ring Joint Facing and RTJ Gasket Dimensions

c) ASME B16.5 Class 600

Nominal pipe size		1/2	3/4	1	1 1/2	2	3	4	6	8	19	12	14	16	18	20	24
Diameter of raised section	I	2	2 1/4	2 3/4	3 9/16	4 1/4	5 3/4	6 7/8	9 1/2	11 7/8	14	16 1/4	18	20	22 5/8	25	29 1/2
Groove pitch diameter	J	1 11/32	1 11/16	2	2 11/16	3 1/4	4 7/8	5 7/8	8 5/16	10 5/8	12 3/4	15	16 1/2	18 1/2	21	23	27 1/4
Depth of groove	K	7/32	1/4	1/4	1/4	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/16	5/17	5/16	3/8	5/16
Width	L	9/32	11/32	11/32	11/32	15/32	15/32	15/32	15/32	15/32	15/32	15/32	15/32	15/32	15/32	17/32	21/32
Outside diameter	M	1 19/32	2 5/16	3	3 11/16	5 5/16	6 5/16	8 3/4	11 1/16	13 3/16	15 7/16	16 13/16	18 15/16	21 7/16	23 1/2	27 7/8	
Inside diameter	N	1 3/32	1 3/8	1 11/16	2 3/8	2 13/16	4 7/16	5 7/16	7 1/8	10 3/16	12 5/16	14 9/16	16 1/16	18 1/16	20 9/16	22 1/2	26 5/8
Width	O	1/4	5/16	5/16	5/16	7/16	7/16	7/16	7/16	7/16	7/16	7/16	7/16	7/16	7/16	1/2	5/8
Thickness	P	3/8	1/2	1/2	1/2	5/8	5/8	5/8	5/8	5/8	5/8	5/8	5/8	5/8	5/8	11/16	11/16
R number		11	13	16	20	23	31	37	45	49	53	57	61	65	69	73	77

Notes:
1. All dimensions in inches.
2. Ring dimensions are per ANSI B16.20.

TABLE A7.1 Ring Joint Facing and RTJ Gasket Dimensions

d) ASME B16.5 Class 900

Nominal pipe size	1/2	3/4	1	1 1/2	2	3	4	6	8	10	12	14	16	18	20	24
Diameter of raised section I						6 1/8	7 1/8	9 1/2	12 1/8	14 1/4	16 1/2	18 3/8	20 5/8	23 3/8	25 1/2	30 3/8
Groove pitch diameter J						4 7/8	5 7/8	8 5/16	10 5/8	12 3/4	15	16 1/2	18 1/2	21	23	27 1/4
Depth of groove K	USE CLASS 1500					5/16	5/16	5/16	5/16	5/16	5/16	7/16	7/16	1/2	1/2	5/8
Width L	DIMENSIONS					15/32	15/32	15/32	15/32	15/32	15/32	21/32	21/32	25/32	25/32	1 1/16
Outside diameter M	IN					5 5/16	6 5/16	8 3/4	11 1/16	13 3/16	15 7/16	17 1/8	19 1/8	21 3/4	23 3/4	28 1/4
Inside diameter N	THESE SIZES					4 7/16	5 7/16	7 7/8	10 3/16	12 5/16	14 9/16	15 7/8	17 7/8	20 1/8	22 1/4	26 1/4
Width O						7/16	7/16	7/16	7/16	7/16	7/16	5/8	5/8	3/4	3/4	1
Thickness P						5/8	5/8	5/8	5/8	5/8	5/8	13/16	13/16	15/16	15/16	1 1/4
R number						31	37	45	49	53	57	62	66	70	74	78

Notes:
1. All dimensions in inches.
2. Ring dimensions are per ANSI B16.20.

TABLE A7.1 Ring Joint Facing and RTJ Gasket Dimensions

e) ASME B16.5 Class 1500

Nominal pipe size		1/2	3/4	1	1 1/2	2	3	4	6	8	10	12	14	16	18	20	24
Diameter of raised section	I	2 3/8	2 5/8	2 13/16	3 5/8	4 7/8	6 3/8	7 5/8	9 3/4	12 1/2	14 3/8	17 1/4	19 1/4	21 1/2	24 1/8	26 1/2	31 1/4
Groove pitch diameter	J	1 9/16	1 3/4	2	2 11/16	3 3/4	5 3/8	6 3/8	8 5/16	10 5/8	12 3/4	15	16 1/2	18 1/2	21	23	27 1/4
Depth of groove	K	1/4	1/4	1/4	1/4	5/16	5/16	5/16	3/8	5/16	7/16	9/16	5/8	11/16	11/16	11/16	13/16
Width	L	11/32	11/32	11/32	11/32	15/32	15/32	15/32	17/32	21/32	21/32	29/32	1 1/16	1 3/16	1 3/16	1 5/16	1 7/16
Outside diameter	M	1 7/8	2 1/16	2 5/16	3	4 3/16	5 13/16	6 13/16	8 13/16	11 1/4	13 3/8	15 7/8	17 1/2	19 5/8	22 1/8	24 1/4	28 5/8
Inside diameter	N	1 1/4	1 7/16	1 11/16	2 3/8	3 5/16	4 15/16	5 15/16	7 14/16	10	12 1/8	14 1/8	15 1/2	17 3/8	19 7/8	21 3/4	25 7/8
Width	O	5/16	5/16	5/16	5/16	7/16	7/16	7/16	1/2	5/8	5/8	7/8	1	1 1/8	1 1/8	1 1/4	1 3/8
Thickness	P	1/2	1/2	1/2	1/2	5/8	5/8	5/8	11/16	13/16	13/16	11/16	1 1/4	1 3/8	1 3/8	1 1/2	1 5/8
R number		12	14	16	20	24	35	39	46	50	54	58	63	67	71	75	79

Notes:
1. All dimensions in inches.
2. Ring dimensions are per ANSI B16.20.

TABLE A7.1 Ring Joint Facing and RTJ Gasket Dimensions

f) ASME B16.5 Class 2500

Nominal pipe size		½	¾	1	1½	2	3	4	6	8	10	12	14	16	18	20	24
Diameter of raised section	I	2⁹⁄16	2⅞	3¼	4½	5¼	6⅜	8	11	13⅜	16¾	19½					
Groove pitch diameter	J	1¹¹⁄16	2	2⅜	3¼	4	5	6³⁄16	9	11	13½	16					
Depth of groove	K	¼	¼	¼	⁵⁄16	⁵⁄16	⅜	⁷⁄16	½	⁹⁄16	¹¹⁄16	¹¹⁄16					
Width	L	¹¹⁄32	¹¹⁄32	¹¹⁄32	¹⁵⁄32	¹⁵⁄32	¹⁷⁄32	²¹⁄32	²⁵⁄32	²⁹⁄32	1³⁄16	1⁵⁄16					
Outside diameter	M	2	2⁵⁄16	2¹¹⁄16	3¹¹⁄16	4⁷⁄16	5½	6¹³⁄16	9¾	11⅞	14⅝	17¼					
Inside diameter	N	1⅜	1¹¹⁄16	2¹⁄16	2¹³⁄16	3⁹⁄16	4½	5⁹⁄16	8¼	10⅛	12⅜	14¾					
Width	O	⁵⁄16	⁵⁄16	⁵⁄16	⁷⁄16	⁷⁄16	½	⅝	¾	⅞	1⅛	1¼					
Thickness	P	½	½	½	⅝	⅝	¹¹⁄16	¹³⁄16	¹³⁄16	1¹⁄16	1⅜	1½					
R number		13	16	18	23	26	32	38	47	51	55 60						

Notes:
1. All dimensions in inches.
2. Ring dimensions are per ANSI B16.20.

TABLE A7.2 Basic Flange Dimensions

a) ASME B16.5 Class 150

Nominal pipe size	1/2	3/4	1	1 1/2	2	3	4	6	8	10	12	14	16	18	20	24
Outside diameter	27/32	1 3/64	1 5/16	1 29/32	2 3/8	3 1/2	4 1/2	6 5/8	8 5/8	10 3/4	12 3/4	14	16	18	20	24
Thickness A1	7/16	1/2	9/16	11/16	3/4	15/16	15/16	1	1 1/8	1 3/16	1 1/4	1 3/8	1 7/16	1 9/16	1 11/16	1 7/8
Outside diameter B	3 1/2	3 7/8	4 1/4	5	6	7 1/2	9	11	13 1/2	16	19	21	23 1/2	25	27 1/2	32
Hub diameter C	1 3/16	1 1/2	1 15/16	2 9/16	3 1/16	4 1/4	5 5/16	7 9/16	9 11/16	12	14 3/8	15 3/4	18	19 7/8	22	26 1/8
Slip on	5/8	5/8	11/16	7/8	1	1 3/16	1 5/16	1 9/16	1 3/4	1 15/16	2 3/16	2 1/4	2 1/2	2 11/16	2 7/8	3 1/4
Lapped	5/8	5/8	11/16	7/8	1	1 3/16	1 5/16	1 9/16	1 3/4	1 15/16	2 3/16	3 1/8	3 7/16	3 13/16	4 1/16	4 3/8
Weld neck	1 7/8	2 1/16	2 3/16	2 7/16	2 1/2	2 3/4	3	3 1/2	4	4	4 1/2	5	5	5 1/2	5 11/16	6

TABLE A7.2 Basic Flange Dimensions

b) ASME B16.5 Class 300

Nominal pipe size	1/2	3/4	1	1 1/2	2	3	4	6	8	10	12	14	16	18	20	24
Outside diameter	27/32	1 3/64	1 5/16	1 29/32	2 3/8	3 1/2	4 1/2	6 5/8	8 5/8	10 3/4	12 3/4	14	16	18	20	24
Thickness A1	9/16	5/8	11/16	13/16	7/8	1 1/8	1 1/4	1 7/16	1 5/8	1 7/8	2	2 1/8	2 1/4	2 3/8	2 1/2	2 3/4
Outside diameter B	3 3/4	4 5/8	4 7/8	6 1/8	6 1/2	8 1/4	10	12 1/2	15	17 1/2	20 1/2	23	25 1/2	28	30 1/2	36
Hub diameter C	1 1/2	1 7/8	2 1/8	2 3/4	3 5/16	4 5/8	5 3/4	8 1/8	10 1/4	12 5/8	14 3/4	16 3/4	19	21	23 1/8	27 5/8
Slip on	7/8	1	1 1/16	1 3/16	1 5/16	1 11/16	1 7/8	2 1/16	2 7/16	2 5/8	2 7/8	3	3 1/4	3 1/2	3 3/4	4 3/16
Lapped	7/8	1	1 1/16	1 3/16	1 5/16	1 11/16	1 7/8	2 1/16	2 7/16	3 3/4	4	4 3/8	4 3/4	5 1/8	5 1/2	6
Weld neck	2 1/16	2 1/4	2 7/16	2 11/16	2 3/4	3 1/8	3 3/8	3 7/8	4 3/8	4 5/8	5 1/8	5 5/8	5 3/4	6 1/4	6 3/8	6 5/8

TABLE A7.2 Basic Flange Dimensions

c) ASME B16.5 Class 600

Nominal pipe size	1/2	3/4	1	1 1/2	2	3	4	6	8	10	12	14	16	18	20	24
Outside diameter	27/32	1 3/64	1 5/16	1 29/32	2 3/8	3 1/2	4 1/2	6 5/8	8 5/8	10 3/4	12 3/4	14	16	18	20	24
Thickness A2	9/16	5/8	11/16	7/8	1	1 1/4	1 1/2	1 7/8	2 3/16	2 1/2	2 5/8	2 3/4	3	3 1/4	3 1/2	4
Outside diameter B	3 3/4	4 5/8	4 7/8	6 1/8	6 1/2	8 1/4	10 3/4	14	16 1/2	20	22	23 3/4	27	29 1/4	32	37
Hub diameter C	1 1/2	1 7/8	2 1/8	2 3/4	3 5/16	4 5/8	6	8 3/4	10 3/4	13 1/2	15 3/4	17	19 1/2	21 1/2	24	28 1/4
Slip on	7/8	1	1 1/16	1 1/4	1 1/16	1 13/16	2 1/8	2 5/8	3	3 3/8	3 5/8	3 11/16	4 3/16	4 5/8	5	5 1/2
Lapped	7/8	1	1 1/16	1 1/4	1 67/16	1 13/16	2 1/8	2 5/8	3	4 3/8	4 5/8	5	5 1/2	6	6 1/2	7 1/4
Weld neck	2 1/16	2 1/4	2 7/16	2 3/4	2 7/8	3 1/4	4	4 5/8	5 1/4	6	6 1/8	6 1/2	7	7 1/4	7 1/2	8

TABLE A7.2 Basic Flange Dimensions

d) ASME B16.5 Class 900

		½	¾	1	1½	2	3	4	6	8	10	12	14	16	18	20	24
Pipe	Nominal pipe size	½	¾	1	1½	2	3	4	6	8	10	12	14	16	18	20	24
Pipe	Outside diameter	27/32	1 3/64	1 4/16	1 29/32	2⅜	3½	4½	6⅝	8⅝	10¾	12¾	14	16	18	20	24
Flange	Thickness A2	Use Class 1500 dimensions in these sizes					1½	1¾	2 3/16	2½	2¾	3⅛	3⅜	3½	5	4¼	5½
Flange	Outside diameter B						9½	11½	15	18½	21½	24	25¼	27¾	31	33¾	41
Flange	Hub diameter C						5	6¼	9¼	11¾	14½	16½	17¾	20	22¼	24½	29½
Flange: Length through hub D2	Slip on						2⅛	2¾	3⅜	4	4½	4⅝	5⅛	5¼	6	6¼	8
Flange: Length through hub D2	Lapped						2⅛	2¾	3⅜	4½	5	5⅝	6⅛	6½	7½	8¼	10½
Flange: Length through hub D2	Weld neck						4	4½	5½	6⅜	7¼	7⅞	8⅜	8½	9	9¾	11½

TABLE A7.2 Basic Flange Dimensions

e) ASME B16.5 Class 1500

Nominal pipe size	½	¾	1	1½	1	3	4	6	8	10	12	14	16	18	20	24
Outside diameter	27⁄32	1 3⁄64	1 5⁄16	1 29⁄32	2⅜	3½	4½	6⅝	8⅝	10¾	12¾	14	16	18	20	24
Thickness A2	⅞	1	1⅛	1¼	1½	1⅞	2⅛	3¼	3⅝	4¼	4⅞	5¼	5¾	6⅜	7	8
Outside diameter B	4¾	5⅛	5⅞	7	8⁄12	10½	12¼	15½	19	23	26½	29½	32½	36	38¾	46
Hub diameter C	1½	1¾	2 1⁄16	2¾	4⅛	5¼	6⅜	9	11½	14½	17¾	19½	21¾	23½	25¼	30
Slip on	1¼	1⅜	1⅝	1¾	2¼	NOT SPECIFIED FOR CLASS 1500										
Lapped	1¼	1⅜	1⅝	1¾	2¼	2⅞	3 9⁄16	4 11⁄16	5⅝	7	8⅝	9½	10¼	10⅞	11½	13
Weld neck	2⅜	2¾	2⅞	3¼	.4	4⅝	4⅞	6¾	8⅜	10	11⅛	11¾	12¼	12⅞	14	16

TABLE A7.2 Basic Flange Dimensions

f) ASME B16.5 Class 2500

Nominal pipe size	½	¾	1	1½	2	3	4	6	8	10	12	14	16	18	20	24
Outside diameter	27/32	1 3/64	1 5/16	1 29/32	2⅜	3½	4½	6⅝	8⅝	10¾	12¾	Class 2500 flanges not specified in these sizes				
Thickness A2	1 3/16	1¼	1⅜	1¾	2	2⅝	3	4¼	5	6½	7¼					
Outside diameter B	5¼	5½	6¼	8	9¼	12	14	19	21¾	26½	30					
Hub diameter C	1 11/16	2	2¼	3⅛	3¾	5¼	6½	9¼	12	14¾	17⅜					
Slip on	NOT SPECIFIED FOR CLASS 2500															
Lapper	1 9/16	1 11/16	1⅞	2⅜	2¾	3⅝	4¼	6	7	9	10					
Weld neck	2⅞	3⅛	3½	4⅜	5	6⅝	7½	10¾	12½	16½	18¼					

TABLE A7.3 Gasket Dimensions for ASME B16.5 Pipe Flanges and Flange Fittings

a) Class 150

		Class 150 gaskets				Class 300 gaskets			
Nominal pipe size	Gasket ID	OD	Number of holes	Hole diameter	Bolt circle diameter	OD	Number of holes	Hole diameter	Bolt circle diameter
½	0.84	3.50	4	0.62	2.38	3.75	4	0.62	2.62
¾	1.06	3.88	4	0.62	2.75	4.62	4	0.75	3.25
1	1.31	4.25	4	0.62	3.12	4.88	4	0.75	3.50
1¼	1.66	4.62	4	0.62	3.50	5.25	4	0.75	3.88
1½	1.91	5.00	4	0.62	3.88	6.12	4	0.88	4.50
2	2.38	6.00	4	0.75	4.75	6.50	8	0.75	5.00
2½	2.88	7.00	4	0.75	5.50	7.50	8	0.88	5.88
3	3.50	7.50	4	0.75	6.00	8.25	8	0.88	6.62
3½	4.00	8.50	8	0.75	7.00	9.00	8	0.88	7.25
4	4.50	9.00	8	0.75	7.50	10.00	8	0.88	7.88
5	5.56	10.00	8	0.88	8.50	11.00	8	0.88	9.25
6	6.62	11.00	8	0.88	9.50	12.50	12	0.88	10.63
8	8.62	13.50	8	0.88	11.75	15.00	12	1.00	13.00
10	10.75	16.00	12	1.00	14.25	...	...	...	...
12	12.75	19.00	12	1.00	17.00	...	...	...	...

General note: Dimensions are in inches.

TABLE A7.3 Gasket Dimensions for ASME B16.5 Pipe Flanges and Flange Fittings
b) Class 300, 400, 600 and 900

Nomimal pipe size	Gasket ID	Gasket OD			
		Glass 300	Class 400	Class 600	Class 900
½	0.84	2.12	2.12	2.12	2.50
¾	1.06	2.62	2.62	2.62	2.75
1	1.31	2.88	2.88	2.88	3.12
1¼	1.66	3.25	3.25	3.25	3.50
1½	1.91	3.75	3.75	3.75	3.88
2	2.38	4.38	4.38	4.38	5.62
2½	2.88	5.12	5.12	5.12	6.50
3	3.50	5.88	5.88	5.88	6.62
3½	4.00	6.50	6.38	6.38	. . .
4	4.50	7.12	7.00	7.62	8.12
5	5.56	8.50	8.38	9.50	9.75
6	6.62	9.88	9.75	10.50	11.38
8	8.62	12.12	12.00	12.62	14.12
10	10.75	14.25	14.12	15.75	17.12
12	12.75	16.62	16.50	18.00	19.62
14	14.00	19.12	19.00	19.38	20.50
16	16.00	21.25	21.12	22.25	22.62
18	18.00	23.50	23.38	24.12	25.12
20	20.00	25.75	25.50	26.88	27.50
24	24.00	30.50	30.25	31.12	33.00

General note: Dimensions are in inches.

For applications involving raised face flanges, the spiral wound gasket is supplied with an outer ring; for critical applications it is supplied with both outer and inner rings. The outer ring provides the centering capability of the gasket as well as the blow-out resistance of the windings and acts as a compression stop. The inner ring provides additional load-bearing capability from high-bolt loading. This is particularly advantageous in high-pressure applications. The inner ring also acts as a barrier to the internal fluids and provides resistance against buckling of the windings.

Spiral wound–ring gaskets are also used in tongue-and-groove flanges. Inner rings should be used with spiral wound gaskets on male-and-female flanges, such as those found in heat-exchanger, shell, channel, and cover-flange joints.

Spiral wound gaskets are designed to suit ASME B16.5 and DIN flanges. See Table A7.5 for dimensions for spiral wound gaskets used with ASME B16.5 flanges.

See Table A7.6a and A7.6b for dimensions for spiral wound gaskets used with ASME B16.47 large diameter steel flanges.

See Table A7.7 for inner-ring inner diameters for spiral wound gaskets.

Camprofile Gaskets. Camprofile gaskets are made from a solid serrated metal core faced on each side with a soft nonmetallic material.

The term camprofile (or kammprofile) comes from the groove profile found on each face of the metal core. Two profiles are commonly used: the DIN 2697 profile and the shallow profile. The shallow profile is similar to the DIN 2697 profile except

TABLE A7.4a Flat Ring Gasket Dimensions for ASME B16.47 Series A (MSSSP44) Large Diameter Steel Flanges, Classes 150, 300, 400 and 600

Nominal pipe size	ID	OD			
		Class 150	Class 300	Class 400	Class 600
22	22.00	26.00	27.75	27.63	28.88
26	26.00	30.50	32.88	32.75	34.12
28	28.00	32.75	35.38	35.12	36.00
30	30.00	34.75	37.50	37.25	38.25
32	32.00	37.00	39.62	39.50	40.25
34	34.00	39.00	41.62	41.50	42.25
36	36.00	41.25	44.00	44.00	44.50
38	38.00	43.75	41.50	42.26	43.50
40	40.00	45.75	43.88	44.58	45.50
42	42.00	48.00	45.88	46.38	48.00
44	44.00	50.25	48.00	48.50	50.00
46	46.00	52.25	50.12	50.75	52.26
48	48.00	54.50	52.12	53.00	54.75
50	50.00	56.50	54.25	55.25	57.00
52	52.00	58.75	56.25	57.26	59.00
54	54.00	61.00	58.75	59.75	61.25
56	56.00	63.25	60.75	61.75	63.50
58	58.00	65.50	62.75	63.75	65.50
60	60.00	67.50	64.75	66.25	67.75

General note: Dimensions are in inches.

TABLE A7.4b Flat Ring Gasket Dimensions for ASME B16.47 Series B (API 605) Large Diameter Steel Flanges, Classes 75, 150, 300, 400 and 600

Nominal pipe size	Gasket ID	Gasket OD				
		Class 75	Class 150	Class 300	Class 400	Class 600
26	26.00	27.88	28.56	30.38	29.38	30.12
28	28.00	29.88	30.56	32.50	31.50	32.25
30	30.00	31.88	32.56	34.88	33.75	34.62
32	32.00	33.88	34.69	37.00	35.88	36.75
34	34.00	35.88	36.81	39.12	37.88	39.25
36	36.00	38.31	38.88	41.25	40.25	41.25
38	38.00	40.31	41.12	43.25	...	...
40	40.00	42.31	43.12	45.25	...	...
42	42.00	44.31	45.12	47.25	...	...
44	44.00	46.50	47.12	49.25	...	...
46	46.00	48.50	49.44	51.88	...	...
48	48.00	50.50	51.44	53.88	...	...
50	50.00	52.50	53.44	55.88	...	...
52	52.00	54.62	55.44	57.88	...	...
54	54.00	56.62	57.62	61.25	...	...
56	56.00	58.88	59.62	62.75	...	...
58	58.00	60.88	62.19	65.19	...	...
60	60.00	62.88	64.19	67.12	...	...

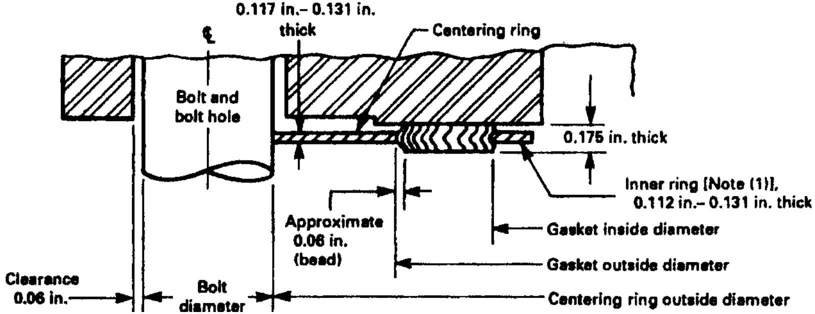
TABLE A7.5 Dimensions for Spiral Wound Gaskets Used with ASME B16.5 Flanges

Flange size (NPS)	Outside diameter of gaskets: Classes 150, 300, 400, 600	Outside diameter of gaskets: Classes 900, 1500, 2500	Inside diameter of gasket by class: 150	300	400	600	900	1500	2500	Outside diameter of centering ring by class: 150	300	400	600	900	1500	2500
½	1.25	1.25	0.75	0.75	(5)	0.75	(5)	0.75	0.75	1.88	2.13	(5)	2.13	(5)	2.50	2.75
¾	1.56	1.56	1.00	1.00	(5)	1.00	(5)	1.00	1.00	2.25	2.63	(5)	2.63	(5)	2.75	3.00
1	1.88	1.88	1.25	1.25	(5)	1.25	(5)	1.25	1.25	2.63	2.88	(5)	2.88	(5)	3.13	3.38
1¼	2.38	2.38	1.88	1.88	(5)	1.88	(5)	1.56	1.56	3.00	3.25	(5)	3.25	(5)	3.50	4.13
1½	2.75	2.75	2.13	2.13	(5)	2.13	(5)	1.88	1.88	3.38	3.75	(5)	3.75	(5)	3.88	4.63
2	3.38	3.38	2.75	2.75	(5)	2.75	(5)	2.31	2.31	4.13	4.38	(5)	4.38	(5)	5.63	5.75
2½	3.88	3.88	3.25	3.25	(5)	3.25	(5)	2.75	2.75	4.88	5.13	(5)	5.13	(5)	6.50	6.63
3	4.75	4.75	4.00	4.00	(5)	4.00	3.75	3.63	3.63	5.38	5.88	(5)	5.88	6.63	6.88	7.75
4	5.88	5.88	5.00	5.00	4.75	4.75	4.75	4.63	4.63	6.88	7.13	7.00	7.63	8.13	8.25	9.25
5	7.00	7.00	6.13	6.13	5.81	5.81	5.81	5.63	5.63	7.75	8.50	8.38	9.50	9.75	10.00	11.00
6	8.25	8.25	7.19	7.19	6.88	6.88	6.88	6.75	6.75	8.75	9.88	9.75	10.50	11.38	11.13	12.50
8	10.38	10.13	9.19	9.19	8.88	8.88	8.75	8.50	8.50	11.00	12.13	12.00	12.63	14.13	13.88	15.25
10	12.50	12.25	11.31	11.31	10.81	10.81	10.88	10.50	10.63	13.38	14.25	14.13	15.75	17.13	17.13	18.75
12	14.75	14.50	13.38	13.38	12.88	12.88	12.75	12.75	12.50	16.13	16.63	16.50	18.00	19.63	20.50	21.63
14	16.00	15.75	14.63	14.63	14.25	14.25	14.00	14.25	(5)	17.75	19.13	19.00	19.38	20.50	22.75	
16	18.25	18.00	16.63	16.63	16.25	16.25	16.25	16.00	(5)	20.25	21.25	21.13	22.25	22.63	25.25	
18	20.75	20.50	18.69	18.69	18.50	18.50	18.25	18.25	(5)	21.63	23.50	23.38	24.13	25.13	27.75	
20	22.75	22.50	20.69	20.69	20.50	20.50	20.50	20.25	(5)	23.88	25.75	25.50	26.88	27.50	29.75	
24	27.00	26.75	24.75	24.75	24.75	24.75	24.75	24.25	(5)	28.25	30.50	30.25	31.13	33.00	35.50	

TABLE A7.6a Dimensions for Spiral Wound Gaskets Used with ASME B16.47 Series A (MSS SP44) Flanges

Flange size (NPS)	Class 150			Class 300			Class 400			Class 600			Class 900		
	Gasket		Centering ring outside diameter	Gasket		Centering ring outside diameter	Gasket		Centering ring outside diameter	Gasket		Centering ring outside diameter	Gasket		Centering ring outside diameter
	Inside diameter	Outside diameter		Inside diameter	Outside diameter		Inside diameter	Outside diameter		Inside diameter	Outside diameter		Inside diameter	Outside diameter	
26	26.50	27.75	30.50	27.00	29.00	32.88	27.00	29.00	32.75	27.00	29.00	34.13	27.00	29.00	34.75
28	28.50	29.75	32.75	29.00	31.00	35.38	29.00	31.00	35.13	29.00	31.00	36.00	29.00	31.00	37.25
30	30.50	31.75	34.75	31.25	33.25	37.50	31.25	33.25	37.25	31.25	33.25	38.25	31.25	33.25	39.75
32	32.50	33.88	37.00	33.50	35.50	39.63	33.50	35.50	39.50	33.50	35.50	40.25	33.50	35.50	42.25
34	34.50	35.88	39.00	35.50	37.50	41.63	35.50	37.50	41.50	35.50	37.50	42.25	35.50	37.50	44.75
36	36.50	38.13	41.25	37.63	39.63	44.00	37.63	39.63	44.00	37.63	39.63	44.50	37.75	39.75	47.25
38	38.50	40.13	43.75	38.50	40.00	41.50	38.25	40.25	42.25	39.00	41.00	43.50	40.75	42.75	47.25
40	40.50	42.13	45.75	40.25	42.13	43.88	40.38	42.38	44.38	41.25	43.25	45.50	43.25	45.25	49.25
42	42.50	44.25	48.00	42.25	44.13	45.88	42.38	44.38	46.38	43.50	45.50	48.00	45.25	47.25	51.25
44	44.50	46.38	50.25	44.50	46.50	48.00	44.50	46.50	48.50	45.75	47.75	50.00	47.50	49.50	53.88
46	46.50	48.38	52.25	46.38	48.38	50.13	47.00	49.00	50.75	47.75	49.75	52.25	50.00	52.00	56.50
48	48.50	50.38	54.50	48.63	50.63	52.13	49.00	51.00	53.00	50.00	52.00	54.75	52.00	54.00	58.50
50	50.50	52.50	56.50	51.00	53.00	54.25	51.00	53.00	55.25	52.00	54.00	57.00			
52	52.50	54.50	58.75	53.00	55.00	56.25	53.00	55.00	57.25	54.00	56.00	59.00			
54	54.50	56.50	61.00	55.25	57.25	58.75	55.25	57.25	59.75	56.25	58.25	61.25			
56	56.50	58.50	63.25	57.25	59.25	60.75	57.25	59.25	61.75	58.25	60.25	63.50			
58	58.50	60.50	65.50	59.50	61.50	62.75	59.25	61.25	63.75	60.50	62.50	65.50			
60	60.50	62.50	67.50	61.50	63.50	64.75	61.75	63.75	66.25	62.75	64.75	68.25			

TABLE A7.6b Dimensions for Spiral Wound Gaskets Used with ASME B16.47 Series B (API 605) Flanges



Flange size (NPS)	Class 150			Class 300			Class 400			Class 500			Class 900		
	Gasket		Centering ring outside diameter	Gasket		Centering ring outside diameter	Gasket		Centering ring outside diameter	Gasket		Centering ring outside diameter	Gasket		Centering ring outside diameter
	Inside diameter	Outside diameter		Inside diameter	Outside diameter		Inside diameter	Outside diameter		Inside diameter	Outside diameter		Inside diameter	Outside diameter	
26	26.50	27.50	28.56	26.50	28.00	30.38	26.25	27.50	29.38	26.13	28.13	30.13	27.25	29.50	33.00
28	28.50	29.50	30.56	28.50	30.00	32.50	28.13	29.50	31.50	27.75	29.75	32.25	29.25	31.50	35.50
30	30.50	31.50	32.56	30.50	32.00	34.88	30.13	31.75	33.75	30.63	32.63	34.63	31.75	33.75	37.75
32	32.50	33.50	34.69	32.50	34.00	37.00	32.00	33.88	35.88	32.75	34.75	36.75	34.00	36.00	40.00
34	34.50	35.75	36.81	34.50	36.00	39.13	34.13	35.88	37.88	35.00	37.00	39.25	36.25	38.25	42.25
36	36.50	37.75	38.88	36.50	38.00	41.25	36.13	38.00	40.25	37.00	39.00	41.25	37.25	39.25	44.25
38	38.37	39.75	41.13	39.75	41.25	43.25	38.25	40.25	42.25	39.00	41.00	43.50	40.75	42.75	47.25
40	40.25	41.88	43.13	41.75	43.25	45.25	40.38	42.38	44.38	41.25	43.25	45.50	43.25	45.25	49.25
42	42.50	43.88	45.13	43.75	45.25	47.25	42.38	44.38	46.38	43.50	45.50	48.00	45.25	47.25	51.25
44	44.25	45.88	47.13	45.75	47.25	49.25	44.50	46.50	48.50	45.75	47.75	50.00	47.50	49.50	53.88
46	46.50	48.19	49.44	47.88	49.38	51.88	47.00	49.00	50.75	47.75	49.75	52.25	50.00	52.00	56.50
48	48.50	50.00	51.44	49.75	51.63	53.88	49.00	51.00	53.00	50.00	52.00	54.75	52.00	54.00	58.50
50	50.50	52.19	53.44	51.88	53.38	55.88	51.00	53.00	55.25	52.00	54.00	57.00			
52	52.50	54.19	55.44	53.88	55.38	57.88	53.00	55.00	57.25	54.00	56.00	59.00			
54	54.50	56.00	57.63	55.25	57.25	60.25	55.25	57.25	59.75	56.25	58.25	61.25			
56	56.88	58.18	59.63	58.25	60.00	62.75	57.25	59.25	61.75	58.25	60.25	63.50			
58	59.07	60.19	62.19	60.44	61.94	65.19	59.25	61.25	63.75	60.50	62.50	65.50			
60	61.31	62.44	64.19	62.56	64.19	67.19	61.75	63.75	66.25	62.75	64.75	68.25			

TABLE A7.7 Inner Ring Inside Dimensions for Spiral Wound Gaskets

Flange size (NPS)	Pressure class						
	150	300	400	600	900	1500	2500
½	0.56	0.56		0.56		0.56	0.56
¾	0.81	0.81		0.81		0.81	0.81
1	1.06	1.06		1.06		1.06	1.06
1¼	1.50	1.50		1.50		1.31	1.31
1½	1.75	1.75		1.75		1.63	1.63
2	2.19	2.19		2.19		2.06	2.06
2½	2.62	2.62		2.62		2.50	2.50
3	3.19	3.19		3.19	3.19	3.19	3.19
4	4.19	4.19	4.19	4.19	4.19	4.19	4.19
5	5.19	5.19	5.19	5.19	5.19	5.19	5.19
6	6.19	6.19	6.19	6.19	6.19	6.19	6.19
8	8.50	8.50	8.25	8.25	8.25	8.12	7.88
10	10.56	10.56	10.25	10.25	10.25	10.15	9.75
12	12.50	12.50	12.50	12.50	12.38	12.38	11.50
14	13.75	13.75	13.75	13.75	13.50	13.38	
16	15.75	15.75	15.75	15.75	15.50	15.25	
18	17.69	17.69	17.69	17.69	17.50	17.25	
20	19.69	19.69	19.69	19.69	19.50	19.25	
24	23.75	23.75	23.75	23.75	23.75	22.75	

that the groove depth is 0.5 mm (versus 0.75 mm for DIN 2697). This allows for a cost advantage for the shallow profile. The profile can be made from sheet metal or strip with a thickness of 3 mm instead of a thickness of 4 mm for DIN profile. For the original German Standard see Fig. A7.4, DIN 2697, Profile for Camprofile Gasket.

The most common facing for camprofile gaskets is flexible graphite. Other facings such as expanded or sintered PTFE and CAF are also used. The camprofile gasket combines the strength, blowout, and creep resistance of a metal core with a soft sealing material that conforms to the flange faces providing a seal. Standard camprofile gaskets are available to suit ASME B16.5, BS1560, and DIN 2697. These dimensions are shown in Table A7.8, Camprofile Dimensions to Suit Standard Flanges.

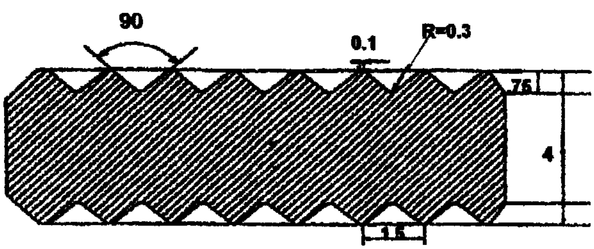


FIGURE A7.4 DIN 2697 Profile for camprofile gasket.

TABLE A7.8 Camprofile Dimensions to Suit Standard Flanges

a) Suit ASME B16.5 and BS 1560 Flanges Class 150 to 2500

Style PN, ZG & ZA to suit ASME B16.5 and BS 1560 flanges class 150 up to 2500									
Dimensions in inches			150	300	400	600	900	1500	2500
NPS	d1	d2	d3						
½	29⁄32	15⁄16	1⅞	2⅛	2⅛	2⅛	2½	2½	2¾
¾	1⅛	1 9⁄16	2¼	2⅝	2⅝	2⅝	2¾	2¾	3
1	1 7⁄16	1⅞	2⅝	2⅞	2⅞	2⅞	3⅛	3⅛	3⅜
1¼	1¾	2⅜	3	3¼	3¼	3¼	3½	3½	4⅛
1½	2 1⁄16	2¾	3⅜	3¾	3¾	3¾	3⅞	3⅞	4⅝
2	2¾	3½	4⅛	4⅜	4⅜	4⅜	5⅝	5⅝	5¾
2½	3¼	4	4⅞	5⅛	5⅛	5⅛	6½	6½	6⅝
3	3⅞	4⅞	5⅜	5⅞	5⅞	5⅞	6⅝	6⅞	7¾
3½	4⅜	5⅜	6⅜	6½	6⅜	6⅜	7½	7⅜	—
4	4⅞	6 1⁄16	6⅞	7⅛	7	7⅝	8⅛	8¼	9¼
5	5 15⁄16	7 3⁄16	7¾	8½	8⅜	9½	9¾	10	11
6	7	8⅜	8¾	9⅞	9¾	10½	11⅜	11⅛	12½
8	9	10½	11	12⅛	12	12⅝	14⅛	13⅞	15¼
10	11⅛	12⅝	13⅜	14¼	14⅛	15¾	17⅛	17⅛	18¾
12	13⅜	14⅞	16⅛	16⅝	16½	18	19⅝	20½	21⅝
14	14⅝	16⅛	17¾	19⅛	19	19⅜	20½	22¾	—
16	16⅝	18⅜	20¼	21¼	21⅛	22¼	22⅝	25¼	—
18	18⅞	20⅞	21⅝	23½	23⅜	24⅛	25⅛	27¾	—
20	20⅞	22⅞	23⅞	25¾	25½	26⅞	27½	29¾	—
22	22⅞	24⅞	26	27¾	27⅝	28⅞	—	—	—
24	24⅞	26⅞	28¼	30½	30¼	31⅛	33	35½	—

TABLE A7.8 Camprofile Dimensions to Suit Standard Flanges

b) Suit DIN 2697 PN 64 to PN 400

Style PN, ZG & ZA in accordance with DIN 2697, PN64 to PN400								
Dimensions in mm			64	100	160	250	320	400
DN	d1	d2	d3					
10	22	40	56	56	56	67	67	67
15	25	45	61	61	61	72	72	77
25	36	68	82	82	82	82	92	103
40	50	88	102	102	102	108	118	135
50	62	102	112	118	118	123	133	150
65	74	122	137	143	143	153	170	192
80	90	138	147	153	153	170	190	207
100	115	162	173	180	180	202	229	256
125	142	188	210	217	217	242	274	301
150	165	218	247	257	257	284	311	348
(175)	190	260	277	287	284	316	358	—
200	214	285	309	324	324	358	398	442
250	264	345	364	391	388	442	488	—
300	310	410	424	458	458	—	—	—
350	340	465	486	512	—	—	—	—
400	386	535	543	—	—	—	—	—

Camprofile gaskets are used on all pressure classes from Class 150 to Class 2500 in a wide variety of service fluids and operating temperatures.

Jacketed Gaskets. Jacketed gaskets are made from a nonmetallic gasket material enveloped in a metallic sheath. This inexpensive gasket arrangement is used occasionally on standard flange assemblies, valves, and pumps. Jacketed gaskets are easily fabricated in a variety of sizes and shapes and are an inexpensive gasket for heat exchangers, shell, channel, and cover flange joints. Their metal seal makes them unforgiving to irregular flange finishes and cyclic operating conditions.

Jacketed gaskets come in a variety of metal envelope styles. The most common style is double jacketed, shown in Fig. A7.5.

Metallic Gaskets. Metallic gaskets are fabricated from one or a combination of metals to the desired shape and size. Common metallic gaskets are ring-joint gaskets and lens rings. They are suitable for high-temperature and pressure applications and require high-bolt loads to seal.

Ring-Joint Gaskets. Standard ring-joint gaskets can be categorized into three groups: Style R, RX, and BX.

They are manufactured to API 6A and ASME B16.20 standards.

Dimensions of Style R gaskets are shown in Table A7.1.

Style R gaskets are either oval or octagonal. Style RX is a pressure-energized adaptation of the standard Style R ring-joint gasket. The RX is designed to fit the same groove design as the Standard Style R. These gasket styles are shown in Fig. A7.6. Dimensions of RX gaskets are shown in Table A7.9.

Style BX pressure-energized ring joints are designed for use on pressurized systems up to 20,000 psi (138 MPa). Flange faces using BX-style gaskets will come in contact with each other when the gasket is correctly fitted and bolted up. The BX gasket incorporates a pressure-balance hole to ensure equalization of pressure which may be trapped in the grooves. Dimensions of BX gaskets are shown in Table A7.10.

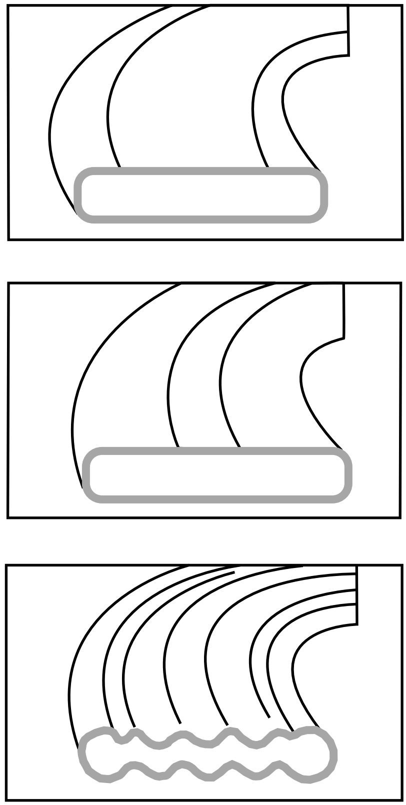


FIGURE A7.5 Double-jacketed gaskets.

Lens Rings Gaskets. Lens rings gaskets have a spherical surface and are suited for use with conical flange faces manufactured to DIN 2696. They are used in specialized high-pressure and high-temperature applications. Standard lens rings gaskets in accordance with DIN-2696 are shown in Table A7.11.

Other specialty metallic seals are available, including welded-membrane gaskets and weld-ring gaskets. These gaskets come in pairs and are seal-welded to their mating flanges and to each other to provide a zero-leakage high-integrity seal.

Bolts and Nuts

Bolts and nuts provide for clamping of the flange and gasket components. Bolting is a term that includes studbolts, nuts, and washers.

Bolting Standards. The following are international standards that pertain to bolting:

ASME B1.1	Unified Inch Screw Threads
ASME B18.2.1	Square and Hex Bolts and Screws

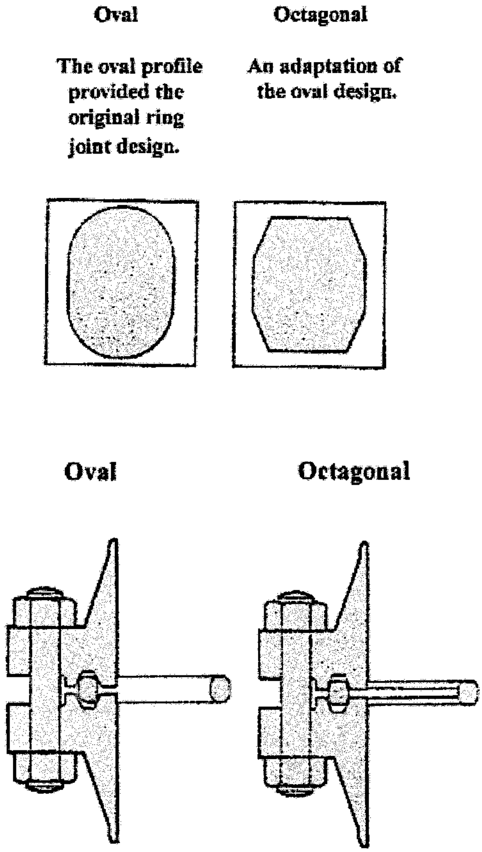


FIGURE A7.6 Style R (oval and octagonal) and RX gaskets.

ASME B18.2.2	Square and Hex Nuts
ASME B18.21.1	Lock Washers
ASME B18.22.1	Plain Washers
ASTM F436	Mechanical Properties of Plain Washers
BS 4882	Bolting for Flanges and Pressure Containing Purposes

Bolts. A bolt is a fastener with a head integral with the shank and threaded at the opposite end. Bolting for flanges and pressure-containing purposes are usually studbolts. Studbolts are fasteners that are threaded at both ends or for the whole length. The general forms of studbolts are shown in Fig. A7.7. Screw threads for studbolts for all materials are shown in Table A7.12. The nominal length of an

TABLE A7.9 Type RX Ring Gasket Dimensions

Ring number	Outside diameter of ring *OD*	Width of ring *A*	Width of flat *C*	Height of outside bevel *D*	Height of ring *H*	Radius in ring R_1	Hole size *E*
RX-20	3.000	0.344	0.182	0.125	0.750	0.06	N/A
RX-23	3.672	0.469	0.254	0.167	1.000	0.06	N/A
RX-24	4.172	0.469	0.254	0.167	1.000	0.06	N/A
RX-25	4.313	0.344	0.182	0.125	0.750	0.06	N/A
RX-26	4.406	0.469	0.254	0.167	1.000	0.06	N/A
RX-27	4.656	0.469	0.254	0.167	1.000	0.06	N/A
RX-31	5.297	0.469	0.254	0.167	1.000	0.06	N/A
RX-35	5.797	0.469	0.254	0.167	1.000	0.06	N/A
RX-37	6.297	0.469	0.254	0.167	1.000	0.06	N/A
RX-39	6.797	0.469	0.254	0.167	1.000	0.06	N/A
RX-41	7.547	0.469	0.254	0.167	1.000	0.06	N/A
RX-44	8.047	0.469	0.254	0.167	1.000	0.06	N/A
RX-45	8.734	0.469	0.254	0.167	1.000	0.06	N/A
RX-46	8.750	0.531	0.263	0.188	1.125	0.06	N/A
RX-47	9.656	0.781	0.407	0.271	1.625	0.09	N/A
RX-49	11.047	0.469	0.254	0.167	1.000	0.06	N/A
RX-50	11.156	0.656	0.335	0.208	1.250	0.06	N/A
RX-53	13.172	0.469	0.254	0.167	1.000	0.06	N/A
RX-54	13.281	0.656	0.335	0.208	1.250	0.06	N/A
RX-57	15.422	0.469	0.254	0.167	1.000	0.06	N/A
RX-63	17.391	1.063	0.582	0.333	2.000	0.09	N/A
RX-65	18.922	0.469	0.254	0.167	1.000	0.06	N/A
RX-66	18.031	0.656	0.335	0.208	1.250	0.06	N/A
RX-69	21.422	0.469	0.254	0.167	1.000	0.06	N/A
RX-70	21.656	0.781	0.407	0.271	1.625	0.09	N/A
RX-73	23.469	0.531	0.263	0.208	1.250	0.06	N/A
RX-74	23.656	0.781	0.407	0.271	1.625	0.09	N/A
RX-82	2.672	0.469	0.254	0.167	1.000	0.06	0.06
RX-84	2.922	0.469	0.254	0.167	1.000	0.06	0.06
RX-85	3.547	0.531	0.263	0.167	1.000	0.06	0.06
RX-86	4.078	0.594	0.335	0.188	1.125	0.06	0.09
RX-87	4.453	0.594	0.335	0.188	1.125	0.06	0.09
RX-88	5.484	0.688	0.407	0.208	1.250	0.06	0.12
RX-89	5.109	0.719	0.407	0.208	1.250	0.06	0.12
RX-90	6.875	0.781	0.479	0.292	1.750	0.09	0.12
RX-91	11.297	1.188	0.780	0.297	1.781	0.09	0.12
RX-99	9.672	0.469	0.254	0.167	1.000	0.06	N/A
RX-201	2.026	0.226	0.126	0.057	0.445	0.02	N/A
RX-205	2.453	0.219	0.120	0.072	0.437	0.02	N/A
RX-210	3.844	0.375	0.213	0.125	0.750	0.03	N/A
RX-215	5.547	0.469	0.210	0.167	1.000	0.06	N/A

TABLE A7.10 Type BX Ring Gasket Dimensions

Ring number	Nominal size (in)	Outside diameter of ring *OD*	Height of ring *H*	Width of ring *A*	Diameter of flat *ODT*	Width of flat *C*	Hole size *D*
BX-150	1 11/16	2.842	0.366	0.366	2.790	0.314	0.06
BX-151	1 13/16	3.008	0.379	0.379	2.954	0.325	0.06
BX-152	2 1/16	3.334	0.403	0.403	3.277	0.346	0.06
BX-153	2 9/16	3.974	0.448	0.448	3.910	0.385	0.06
BX-154	3 1/16	4.600	0.488	0.488	4.531	0.419	0.06
BX-155	4 1/16	5.825	0.560	0.560	5.746	0.481	0.06
BX-156	7 1/16	9.367	0.733	0.733	9.263	0.629	0.12
BX-157	9	11.593	0.826	0.826	11.476	0.709	0.12
BX-158	11	13.860	0.911	0.911	13.731	0.782	0.12
BX-159	13 5/8	16.800	1.012	1.012	16.657	0.869	0.12
BX-160	13 5/8	15.850	0.938	0.541	15.717	0.408	0.12
BX-161	16 5/8	19.347	1.105	0.638	19.191	0.482	0.12
BX-162	16 5/8	18.720	0.560	0.560	18.641	0.481	0.06
BX-163	18 3/4	21.896	1.185	0.684	21.728	0.516	0.12
BX-164	18 3/4	22.463	1.185	0.968	22.295	0.800	0.12
BX-165	21 1/4	24.595	1.261	0.728	24.417	0.550	0.12
BX-166	21 1/4	25.198	1.261	1.029	25.020	0.851	0.12
BX-167	26 3/4	29.896	1.412	0.516	29.696	0.316	0.06
BX-168	26 3/4	30.128	0.142	0.632	29.928	0.432	0.06
BX-169	5 1/8	6.831	0.624	0.509	6.743	0.421	0.06
BX-170	6 5/8	8.584	0.560	0.560	8.505	0.481	0.06
BX-171	8 9/16	10.529	0.560	0.560	10.450	0.481	0.06
BX-172	11 5/32	13.113	0.560	0.560	13.034	0.481	0.06
BX-303	30	33.573	1.494	0.668	33.361	0.457	0.06

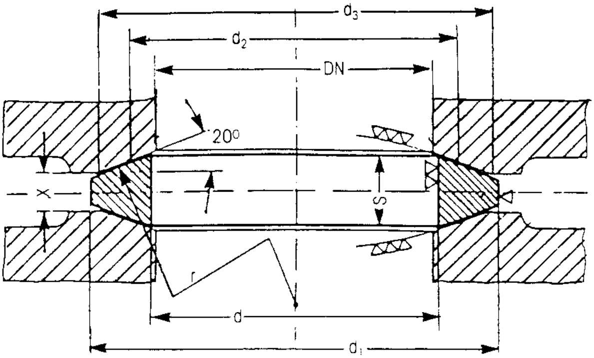
inch-series studbolt is the overall length, excluding the point at each end. The ends of the studbolt are finished with a point having an included angle of approximately 90 degrees to a depth slightly exceeding the depth of the thread. Markings indicating the grade of studbolt are applied to one end of the studbolt.

The minimum length of the studbolt should ensure full engagement of the nut such that the point protrudes above the face of the nut. For applications that utilize hydraulic stud-tensioning tools for tightening, one bolt-diameter is added to this minimum length. Hydraulic stud tensioning and other tightening methods are covered later in this chapter. While there is no maximum length of thread, unnecessarily long studs are avoided due to cost and to prevent corrosion and other damage to exposed threads, which would make subsequent removal difficult.

Nuts

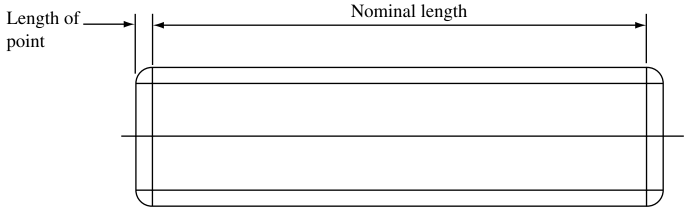
Heavy Series. Heavy-series nuts are generally used with studs on pressure piping. The nonbearing face of a nut has a 30-degree chamfer, while its bearing face is finished with a washer face. Dimensions of heavy-series nuts are shown in Table A7.13.

TABLE A7.11 Lens Rings Gasket Dimensions in Accordance with DIN 2696

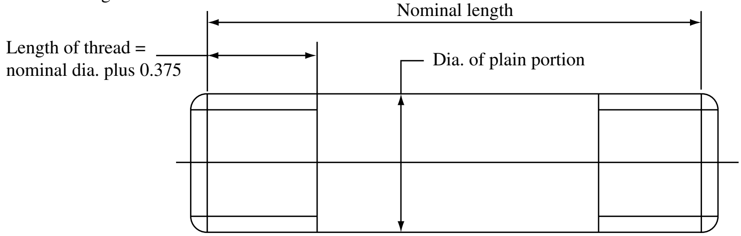


Nominal pipe size DN	d min.	d max.	d_1	S for d max	d_2 middle contact diameter	r	d_3	X
Nominal pressure PN64–400								
10	10	14	21	7	17.1	25	18	5.7
15	14	18	28	8.5	22	32	27	6
25	20	29	43	11	34	50	39	6
40	34	43	62	14	48	70	55	8
50	46	55	78	16	60	88	68	9
65	62	70	102	20	76.6	112	85	13
80	72	82	116	22	88.2	129	97	13
100	94	108	143	26	116	170	127	15
125	116	135	180	29	149	218	157	22
150	139	158	210	33	171	250	183	26
Nominal pressure PN64 and 100								
(175)	176	183	243	41	202.5	296	218	28
200	198	206	276	35	225	329	243	27
250	246	257	332	37	277.7	406	298	25
300	295	305	385	40	323.5	473	345	26
350	330	348	425	41	368	538	394	23
400	385	395	475	42	417.2	610	445	24
Nominal pressure PN160–400								
(175)	162	177	243	37	202.5	296	218	21
200	183	200	276	40	225	329	243	25
250	230	246	332	46	277.7	406	298	25
300	278	285	385	50	323.5	473	345	30

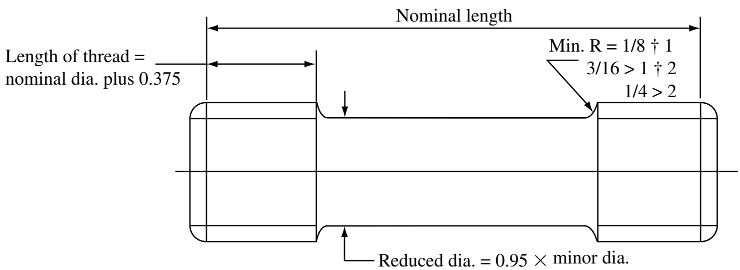
Avoid nominal pipe sizes in brackets.



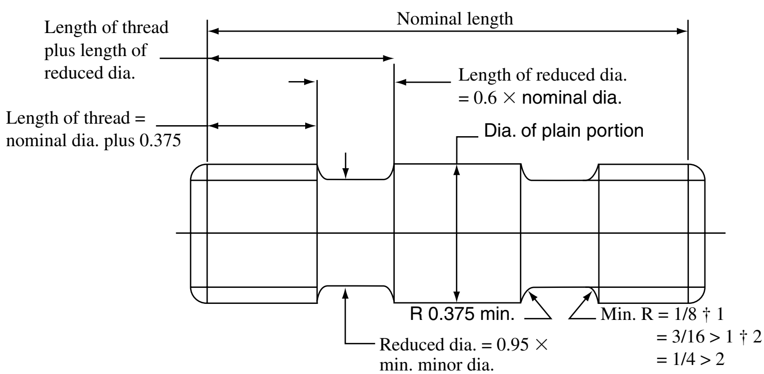
(a) Studbolt threaded full length



(b) Studbolt threaded each end with nominal diameter portion at center



(c) Studbolt threaded each end with reduced diameter portion at center



(d) Studbolt threaded each end with two reduced diameter portions and nominal diameter portion at center

Dimensions are in inches.

NOTE 1. Exclusion of length to points from nominal length is in agreement with USA oil industry practice.
NOTE 2. Dimensions at each end are the same for all designs. Centering holes are permitted in types (c) and (d).

FIGURE A7.7 Dimensions of studbolts—inch series.

TABLE A7.12 Pitch of Screw Threads for Studbolts

a) Metric sizes

Nominal diameter	Pitch
<M27	ISO Metric coarse (see BS 3643)
≥M30 ≤ M43	3 mm
≥M45 ≤ M100	4 mm

b) Inch sizes

Nominal diameter	Pitch
≤1 inch	ISO Unified inch coarse (UNC)
≥ 1⅛ inch	8 threads/in UN Series

Lock Nuts. The primary purpose of a self-locking nut is to resist loosening under service conditions experiencing vibration and shock. The self-locking nut produces an interference fit between the bolt threads and the nut threads. Most common self-locking nuts contain a nylon insert. The degree of interference is controlled during manufacture of the nylon-insert minor diameter. The elastic nature of the nylon provides uniform reaction from nut to nut. Generally, in pressure-piping systems, the primary concern is obtaining and maintaining proper stud preload to affect the gasket seal. Vibration is not normally a concern in these applications, and in situations where vibration is prevalent, adequate preload control will prevent nut rotation and loosening.

Washers

Flat Washers. Flat washers are used principally to minimize embedment of the nut and to aid torquing. Plain washers are manufactured in accordance with standard ANSI/ASME B18.22.1. Hardened washers are utilized in high-torque applications. Suitable mechanical properties for hardened, stamped, plain washers are covered by ASTM F436. Applicable properties for plain washers rolled from wire shall be AISI 1060 steel or equivalent, heat treated to a hardness of Rockwell C 45–53.

Dimensions of preferred sizes of Type A plain washers are shown in Table A7.14.

Live Loading. Live loading using belleville springs improves the elasticity of the flange joint. A belleville spring is a washer that is dished in the center to give it a cone shape. The cone shape provides for a very stiff spring, in comparison to coil springs. The cone will deflect and flatten at a specified spring rate (ratio of load to deflection) when subjected to the axial load (Fp) generated in a stud. Figure A7.8 shows a section of a belleville spring. Belleville springs are described by the following dimensions:

OD = outside diameter
ID = inside diameter
t = material thickness
h = deflection to flat
H = overall height

TABLE A7.13 Dimensions of Heavy Series Nut—Metric Series

Nominal size and pitch	Width across flats *s*		Width across corners *e*	Thickness *m*		Tolerance on squareness
	max	min	min	max	min	
	mm	mm	mm	mm	mm	mm
M10 × 1.5	16.00	15.57	17.59	8	7.42	0.29
M12 × 1.75	18.00	17.57	19.85	10	9.42	0.32
(M14 × 2)	21.00	20.16	22.78	11	10.30	0.37
M16 × 2	24.00	23.16	26.17	13	12.30	0.41
M20 × 2.5	30.00	29.16	32.95	16	15.30	0.51
(M22 × 2.5)	34.00	33.00	37.29	18	17.30	0.54
M24 × 3	36.00	35.00	39.55	19	18.16	0.61
M27 × 3	41.00	40.00	45.20	22	21.16	0.70
M30 × 3	46.00	45.00	50.85	24	23.16	0.78
M33 × 3	50.00	49.00	55.37	26	25.16	0.85
M36 × 3	55.00	53.80	60.79	29	28.16	0.94
M39 × 3	60.00	58.80	66.44	31	30.00	1.03
M42 × 3	65.00	63.80	72.09	34	33.00	1.11
M45 × 4	70.00	68.80	77.74	36	35.00	1.20
M48 × 4	75.00	73.80	83.39	38	37.00	1.29
M52 × 4	80.00	78.80	89.04	42	41.00	1.37
M56 × 4	85.00	83.60	94.47	45	44.00	1.46
M64 × 4	95.00	93.60	105.77	51	49.80	1.63
M70 × 4	100.00	98.60	114.42	56	54.80	1.76
M72 × 4	105.00	103.60	117.07	58	56.80	1.81
M76 × 4	110.00	108.60	122.72	61	59.80	1.89
M82 × 4	120.00	118.60	134.01	66	64.80	2.02
M90 × 4	130.00	128.60	145.32	72	70.80	2.20
M95 × 4	135.00	133.60	150.97	76	74.80	2.31
M100 × 4	145.00	143.60	162.27	80	78.80	2.42

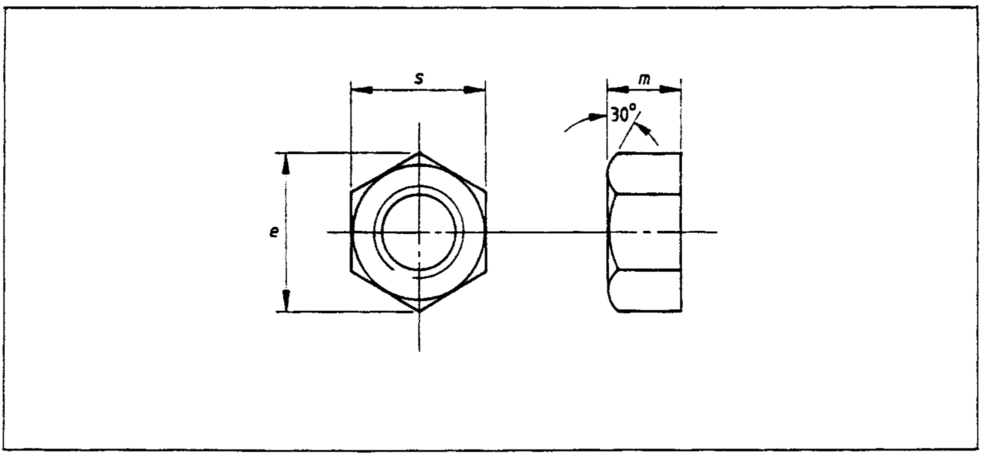
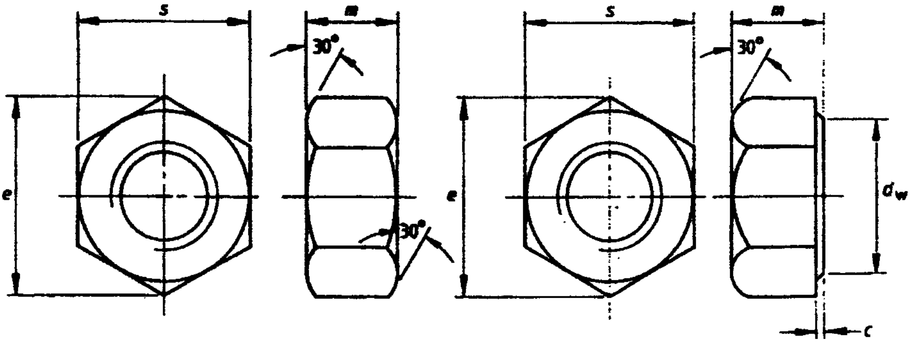


TABLE A7.13 Dimensions of Heavy Series Nut—Inch Series

Nominal size	Threads per inch	Width across flat *s*		Width across corners *e*	Washerface			Thickness *m*		Tolerance on squareness
					Diameter d_w		Thickness			
		max	min	max	max	min	*c*	max	min	
in	threads/in	in	in	in	in	in	in	in	in	in
½	13	0.875	0.85	1.01	0.836	0.808	1/64	0.504	0.464	0.015
⅝	11	1.062	1.031	1.23	1.013	0.979	1/64	0.631	0.587	0.016
¾	10	1.250	1.212	1.44	1.189	1.150	1/64	0.758	0.710	0.019
⅞	9	1.438	1.394	1.66	1.366	1.324	1/64	0.885	0.833	0.023
1	8	1.625	1.575	1.88	1.539	1.496	1/64	1.012	0.956	0.023
1⅛	8	1.812	1.756	2.09	1.710	1.668	1/64	1.139	1.079	0.027
1¼	8	2.000	1.938	2.31	1.892	1.841	1/64	1.251	1.187	0.027
1⅜	8	2.188	2.119	2.53	2.070	2.013	1/64	1.378	1.310	0.030
1½	8	2.375	2.3	2.74	2.251	2.185	1/64	1.505	1.433	0.030
1⅝	8	2.562	2.481	2.96	2.433	2.857	1/64	1.632	1.566	0.030
1¾	8	2.750	2.662	3.18	2.605	2.529	1/64	1.759	1.679	0.030
1⅞	8	2.938	2.844	3.39	2.779	2.702	1/64	1.886	1.802	0.035
2	8	3.125	3.025	3.61	2.949	2.874	1/64	2.013	1.925	0.035
2¼	8	3.500	3.388	4.04	3.296	3.219	1/32	2.251	2.155	0.040
2½	8	3.875	3.75	4.47	3.65	3.563	1/32	2.505	2.401	0.045
2¾	8	4.250	4.112	4.91	4.012	3.906	1/32	2.759	2.647	0.050
3	8	4.625	4.475	5.34	4.373	4.251	1/32	3.013	3.893	0.055
3½	8	5.375	5.2	6.21	5.061	4.94	1/32	3.506	3.370	0.060
3¾	8	5.750	5.563	6.64	5.42	5.27	1/32	3.760	3.616	0.065
4	8	6.125	5.925	7.07	5.78	5.62	1/32	4.014	3.862	0.070

Note: The dimensions are illustrated in figure 6.

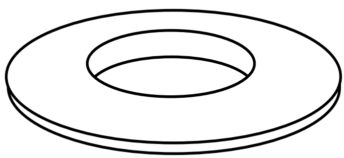


Flange assemblies always tend to relax in time, particularly at elevated temperatures. The rate of relaxation is dependent on many factors, including embedment relaxation of studs and nuts, flange rotation, bolt creep, and gasket creep. The relaxation phenomenon is covered more fully in the section "Behavior of the Flanged Joint System." The high load-deflection or spring rate, characteristics of belleville springs, aid in maintaining bolt preload, compensating for some of the joint relaxation.

The spring rate of a belleville spring depends on geometry, material, and loading conditions. The load-deflection characteristics can be varied by stacking springs in combinations of series and parallel stacks. Figure A7.9 shows load-deflection curves

TABLE A7.14 Dimensions of Preferred Sizes of Type A Plain Washers

Normal washer size	A Inside diameter tolerance			B Outside diameter tolerance			C Thickness		
	Basic	Plus	Minus	Basic	Plus	Minus	Basic	Max	Min
No. ⅝ 0.625 N	0.656	0.030	0.007	1.312	0.030	0.007	0.095	0.121	0.074
⅝ 0.625 W	0.688	0.030	0.007	1.750	0.030	0.007	0.134	0.160	0.108
¾ 0.750 N	0.812	0.030	0.007	1.469	0.030	0.007	0.134	0.160	0.108
¾ 0.750 W	0.812	0.030	0.007	2.000	0.030	0.007	0.148	0.177	0.122
⅞ 0.875 N	0.938	0.007	0.030	1.750	0.030	0.007	0.134	0.160	0.108
⅞ 0.875 W	0.938	0.007	0.030	2.250	0.030	0.007	0.165	0.192	0.136
1 1.000 N	1.062	0.007	0.030	2.000	0.030	0.007	0.134	0.160	0.108
1 1.000 W	1.062	0.007	0.030	2.500	0.030	0.007	0.165	0.192	0.136
1⅛ 1.125 N	1.250	0.030	0.007	2.250	0.030	0.007	0.134	0.160	0.108
1⅛ 1.125 W	1.250	0.030	0.007	2.750	0.030	0.007	0.165	0.192	0.136
1¼ 1.250 N	1.375	0.030	0.007	2.500	0.030	0.007	0.165	0.192	0.136
1¼ 1.250 W	1.375	0.030	0.007	3.000	0.030	0.007	0.105	0.192	0.136
1⅜ 1.375 N	1.500	0.030	0.007	2.750	0.030	0.007	0.165	0.192	0.136
1⅜ 1.375 W	1.500	0.045	0.010	3.250	0.045	0.010	0.180	0.213	0.153
1½ 1.500 N	1.625	0.030	0.007	3.000	0.030	0.007	0.165	0.192	0.136
1½ 1.500 W	1.625	0.045	0.010	3.500	0.045	0.010	0.180	0.213	0.153
1⅝ 1.625	1.750	0.045	0.010	3.750	0.045	0.010	0.180	0.213	0.153
1¾ 1.750	1.875	0.045	0.010	4.000	0.045	0.010	0.180	0.213	0.153
1⅞ 1.875	2.000	0.045	0.010	4.250	0.045	0.010	0.180	0.213	0.153
2 2.000	2.125	0.045	0.010	4.500	0.045	0.010	0.180	0.213	0.153
2¼ 2.250	2.375	0.045	0.010	4.750	0.045	0.010	0.220	0.248	0.193
2½ 2.500	2.625	0.045	0.010	5.000	0.045	0.010	0.238	0.280	0.210
2¾ 2.750	2.875	0.065	0.010	5.250	0.065	0.010	0.259	0.310	0.228
3 3.000	3.125	0.065	0.010	5.500	0.065	0.010	0.284	0.327	0.249



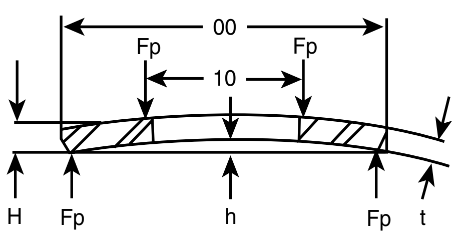


FIGURE A7.8 Section of Belleville spring.

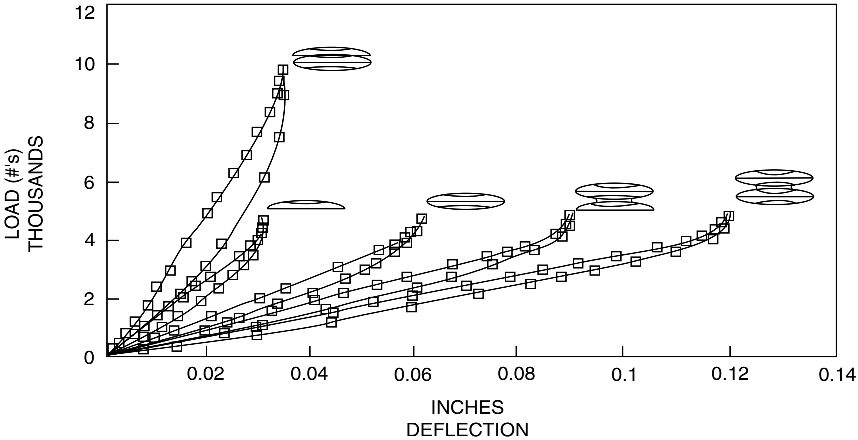


FIGURE A7.9 Load deflection in curves of several Belleville arrangements.

of several different belleville arrangements. The hysteresis between increasing and decreasing load (the upper and lower curves, respectively) is caused by the friction between the spring and the loading surfaces. Two springs stacked in parallel doubles the load to flatten the pair with no further increase in deflection. Two springs stacked in series will produce twice the deflection at the same load.

FUNCTION OF GASKETS

The function of a gasket is to conform to the irregularities of the flange faces to affect a seal, preventing the inside fluid from leaking out. See Fig. A7.10 for a typical flange-gasket arrangement. The leak performance of the gasket is dependent on the stress on the gasket during operation. Each different type of gasket has its own inherent leak-tightness capabilities. The higher the gasket stress, the higher the leak-tightness capability.

The ideal gasket is comprised of a body with good load-bearing and recovery characteristics, with a soft conformable surface layer. Gaskets have a combination of elastic and plastic characteristics. Ideal gaskets should have the following properties:

1. Compressibility—Gaskets that have sufficient compressibility to suit the style and surface finish of the flange, ensuring that all the imperfections will be filled with the gasket material.
2. Resilience—Gaskets that have high resilience will enable the gasket to move with the dynamic loadings of the flange to maintain its seating stress.
3. No change in thickness—Gaskets that will not continue to deform under varying load cycles of temperature and pressure or under a constant load at elevated temperatures (creep).

Unfortunately, most gaskets available on the market are not ideal gaskets. Most gaskets usually just have one or sometimes two of the above properties. For critical

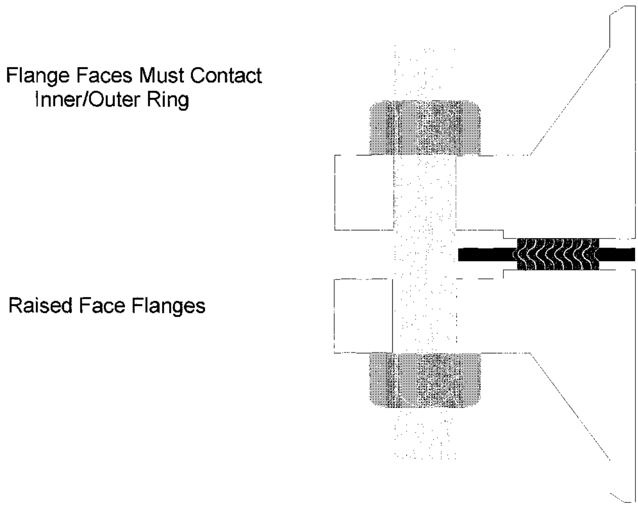


FIGURE A7.10 Typical flange gasket arrangement.

applications, designers are always on the lookout for gaskets that have all three properties.

The most difficult, often critical, property required of a gasket is its ability to resist creep during operation. In high-temperature services, the flanges will heat up at a faster rate than the bolts and under steady-state conditions will continue to be hotter than the bolts as a result of the thermal gradient. This results in a higher thermal expansion of the flanges with respect to the bolts, increasing the boat load and concurrently the gasket stress. The gasket will then deform under the higher applied load during this cycle. Most gaskets will deform permanently and will not rebound when the load cycle goes away with varying conditions. The permanent set or plastic deformation that occurred during operation will cause loss of bolt load and concurrently loss of gasket stress. As gasket stress decreases leak rate increases.

FUNCTION OF BOLTS

The function of a bolt is to provide a clamp load or preload (Fp) to sufficiently compress and stress the gasket and resist the parting forces exerted by the hydrostatic end force and other external loads. The hydrostatic end force is created by the pressure of the internal fluid across the internal area of flange. The internal area is generally the inside diameter of the sealing element.

All bolts behave like a heavy spring. As you turn down the nut against the flange, the bolt stretches and the flange and gasket compress. All bolt-tightening methods result in stretching the bolt. The torque-tightening method uses the thread helix of turning the nut against the reactive forces of the flange to stretch the bolt. The hydraulic bolt-tensioning method utilizes an annular piston threaded on the end of the bolt to provide an axial stretch. Torque tightening and hydraulic bolt tensioning are discussed in the section "Methods of Bolt Tightening." In its elastic region, bolts stretch according to Hooke's law:

$$\Delta L_b = \frac{Fp \times L_b}{E \times A_s} \tag{A7.1}$$

ΔL_b = Change in length of bolt, in (mm)

F_p = Applied tensile load (Preload), lb (kN)

L_b = Effective length of bolt Length of bolt in which tensile stress is applied, in (mm)

E = Young s Modulus of Elasticity, psi (N/mm^2)

A_s = Tensile stress area of bolt, in^2(mm^2)

$= 0.7854(D - 0.9743/n)^2$

where D = Nominal diameter of bolt
n = is the number of threads per in

Preload is the applied bolt load generated during tightening.

BEHAVIOR OF THE FLANGED JOINT SYSTEM

It is important to recognize that the individual components of flanges, gaskets, nuts, and bolts operate together as a system. Gasket companies are continually fielding questions from concerned users about their "gasket" leakage. Gasket leakage is symptomatic of a broader problem. To focus exclusively on the gasket as the cause of the leakage fails to recognize that the flange joint operates as a system, and a systems approach should be used to design flange joints and trouble shoot flange problems.

Under actual operating conditions, the confined fluid, under pressure, creates a hydrostatic end force trying to separate the flange faces. The preload developed in the bolts keeps the flanges together while maintaining a residual gasket-seating stress. The internal pressure of the fluid tries to move, go through, or bypass the gasket. This is illustrated in Fig. A7.11, "What Happens under Actual Operating Conditions."

Joint Stiffness

The flange joint consists of a series of springs in a combination of tension and compression. The bolts are springs in tension, while the flange is a spring in compression. The interaction of the two depends on their respective stiffness. The interaction between the stiffness of the bolt and flange can be represented by a joint diagram. See Fig. A7.12, "Joint Diagram of Simple Elastic Joints."

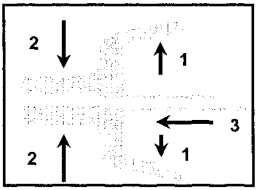


FIGURE A7.11 What happens under actual operating conditions.

The stiffness of the bolt is:

$$K_b = \frac{F_p}{\Delta L_b} \qquad \frac{\text{lb}}{\text{in}} \frac{(\text{kN})}{(\text{mm})} \tag{A7.2}$$

The stiffness of the joint is:

$$K_i = \frac{F_p}{\Delta L_j} \qquad \frac{\text{lb}}{\text{in}} \frac{(\text{kN})}{(\text{mm})} \tag{A7.3}$$

where F_p = Preload lb(kN)
ΔL_b = Change in length of bolt, in (mm)
ΔL_j = Change in compression of joint, in (mm)

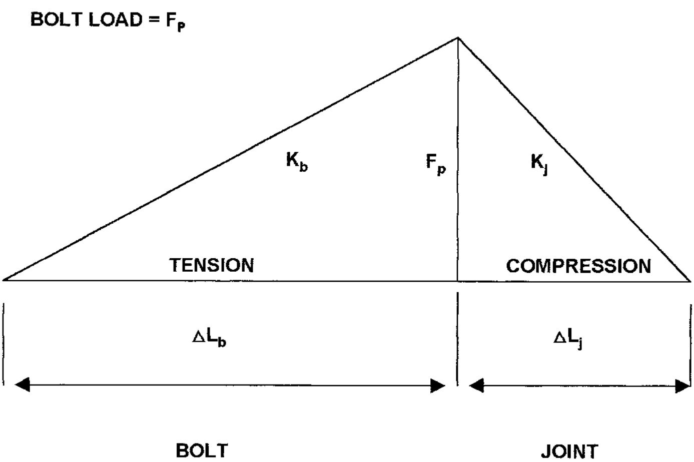


FIGURE A7.12 Joint diagram of simple elastic joints.

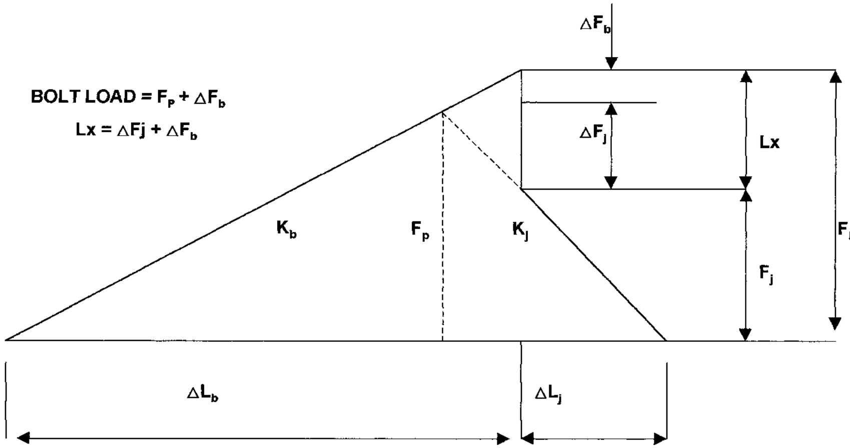


FIGURE A7.13 Joint diagram with external tensile load (Lx).

At the mating surfaces, the bolt sees the preload (F_p) in tension while the joint sees the same preload in compression. Their deflection under this preload is proportional to their respective stiffness.

If an external tensile load (L_x) is applied (i.e., pressure end force), the bolt load increases and the bolt lengthens, while the joint unloads. The change in deformation of the bolt equals its change in deformation of the joint such that they maintain contact with each other. The external load is shared between the bolts and the joint in proportion to their stiffness. This is illustrated in Fig. A7.13.

In a flange joint containing a gasket, behavior is governed to a great degree by the gasket. Unfortunately, the gasket stiffness is nonlinear and very difficult to predict. Gaskets unload quickly following a steep curve, as shown in Fig. A7.14.

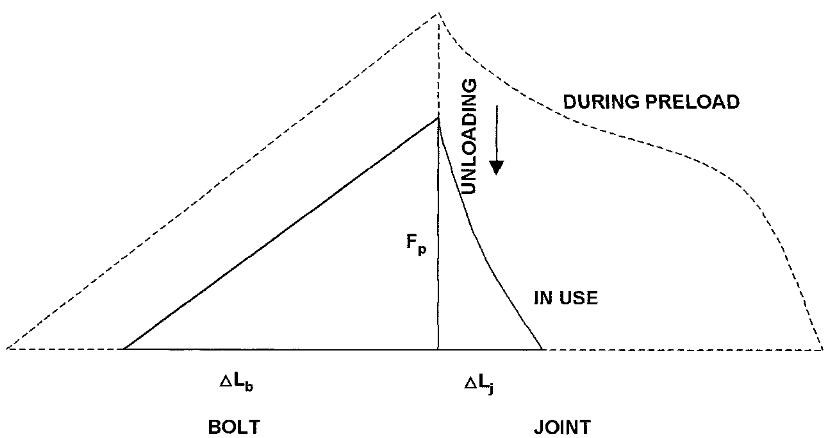


FIGURE A7.14 Joint stiffness diagram for a flanged connection with spiral wound gasket.

Therefore, externally applied loads have a significant effect on reducing the stress of the gasket. A sufficiently high enough bolt preload is required to compensate for gasket unloading in order to maintain sufficient stresses to seal during operation. If the bolt preload is lost due to bolt creep, gasket creep, or flange rotation, the gasket stress drops dramatically and leakage follows.

Elastic Interaction

As one bolt is tightened, the flange and gasket partially compress in relation to their relative stiffness. As subsequent bolts are tightened, the joint compresses further. As each additional bolt is tightened, the compression on the joint will tend to reduce the preload in adjacent bolts. Figure A7.15 shows the elastic behavior of a simplified four-bolt flange. After tightening bolts 1 and 2, bolts 3 and 4 are tightened by compressing the joint further and relaxing the previously tightened bolts 1 and 2. The effect of tightening bolts separately and affecting the loads in adjacent bolts is referred to as elastic interaction, or cross talk. Elastic interaction is one reason why wide scatter in bolt preloads are found in flanged joints. Figure A7.16 shows a typical load scatter of a 28-bolt heat-exchanger channel to shell flange. The top line is the preload for each stud as it was originally tightened with torquing. Notice the wide variation in bolt load with this method of tightening.

Relaxation of the Flange Joint

Flange joint relaxation is one of the most important areas to consider when designing or troubleshooting flange systems. Over and over again, flanges are hydrostatically tested to verify conformance to leak tightness requirements. After successful hydrostatic testing, some flanged joints are found to be leaking during startup, shutdown, or at some time during their operating life. Verification of actual bolt load (using ultrasonic measurement) has revealed that the residual load in the studs after the hydrostatic test is usually lower than the original bolt preload achieved during tightening.

Relaxation of the bolt load observed is due to permanent deformation of the

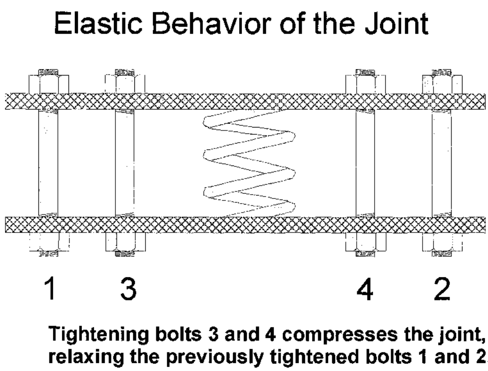


FIGURE A7.15 Elastic behavior of simplified 4-bolt flange.

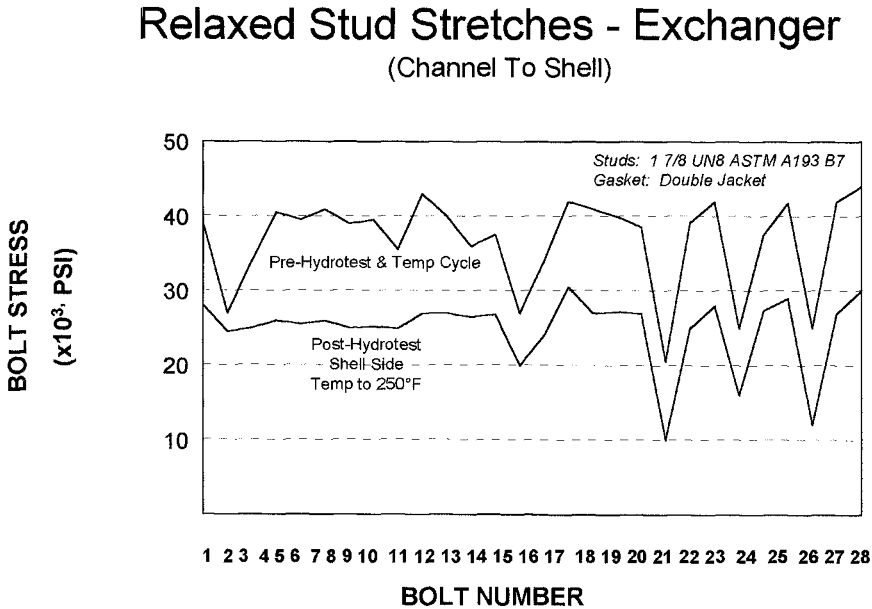


FIGURE A7.16 Typical load scatter of 28-bolt heat exchanger.

gasket element experienced as a result of the pressure test loads. During the hydrostatic test, high external compressive loads are added to the gasket. The gasket will continue to compress (deform) as a result of the additional hydrostatic end load. Since most gaskets have poor elastic properties, the hydrostatic end force will result in permanent deformation of the gasket. On conclusion of the hydrostatic test, the permanent deformation of the gasket will be seen as loss of bolt load and overall joint relaxation. This is illustrated in Fig. A7.16. The lower curve is the residual bolt load measured after the hydrostatic test.

The wide scatter shown is consistent with uncontrolled tightening techniques. The relaxation effects are typical of gaskets that continue to deform under varying load cycles of temperature and pressure or poor creep resistance at elevated temperatures. This flange would likely leak at any number of points in its operating life. The wide scatter of the bolt loads illustrated in Fig. A7.16 may lead to failure after the hydrostatic test.

With the wide load scatter in combination with relaxation of the joint after hydrotest, the joint may leak during startup. In addition, operating temperatures and pressure cycles will continue to relax the joint until there is insufficient bolt load and gasket stress at a particular position around the flange to maintain a seal, and leakage results.

Operating relaxation of the flanged joint is affected by the creep-resistant material properties of the flange, studbolts, and gaskets. Materials that continue to creep (deform) during operation will lead to leakage.

The solution to relaxation-affected leak problems is to

1. Control the initial bolt preload to eliminate the wide scatter around the flange and to ensure the bolt loads are sufficient to maintain a seal throughout the operating life. Controlled bolting is described more fully later in this chapter.

2. Design and install components that are resistant to creep by ensuring that they are suitable for the operating temperatures and pressures.

GASKET SELECTION

The proper selection of gasket is critical to the success of achieving long-term leak tightness of flanged joints. Due to their widespread usage, gaskets are often taken for granted. Industry demands for reduced flange leakage in environments of increasing process temperatures and pressures have led gasket manufacturers to develop a wide variety of gasket types and materials, with new gaskets being introduced on an ongoing basis. This rapidly changing environment makes, and will continue to make, gasket selection difficult.

It is highly recommended that the gasket manufacturer be consulted on the proper selection of gaskets for each application. Gasket manufacturers are familiar with the industry codes and standards and conduct extensive testing of their products to ascertain performance under a variety of operating conditions.

Flange design details, service environment, and operating performance guide the gasket selection process. Start with the flange design. Identify the appropriate flange standard, outlining size, type, facing, pressure rating, and materials (i.e., ASME B16.5, NPS 4, Class 1500, RF, carbon steel). Identify the service environment of temperature, pressure, and process fluid. It is useful to highlight gasket-operating performance.

Gasket-Operating Performance

New flange and gasket designs are incorporating tightness factors in their calculations to reduce leak rates. Traditional ASME Section VIII code utilizes m and y gasket factors in the design calculations of flanges. These factors are useful to establish the flange design required to help ensure the overall pressure integrity of the system; however, they are not useful parameters to predict flange leak rates.

All flanges leak to a certain degree. Industry requirements are demanding reduction in leak rates along with predictable performance. This has lead to a more rigorous approach to establishing gasket factors and the associated methods for gasketed flanged-joint design.

Significant progress has been made in the last six years in Europe by CEN and in North America by ASME's Pressure Vessel Research Council (PVRC) to establish gasket test procedures and the development of design constants that greatly improve the gasketed flanged-joint design. Maximum allowable leak rates have been established for various classes of equipment. EPA Fugitive Emissions basic limits are shown below.

Component	Allowable leakage level
Flange	500 ppmv
Pump	1,000 ppmv
Valve	500 ppmv
Agitator	10,000 ppmv

PVRC has established a new set of gasket factors, G_b, a, and G_s and a related tightness parameter, T_p, which can be used in place of traditional m and y factors in determining required bolt load.

G_b and a (Part A testing) represent the initial gasket-compression characteristics. G_b is the gasket stress at a tightness parameter (T_p) of 1; a is the slope of the line of gasket stress versus tightness parameter plotted on a log-log curve. This line shows that the tightness parameter (or leak tightness) increases with increasing gasket stress. That is, the higher the gasket stress, the lower the expected leakage. G_s is the unloading (Part B) gasket stress at a $T_p = 1$.

A low value of G_b indicates that the gasket requires low levels of gasket stress for initial seating. Low values of G_s indicate that the gasket requires lower stresses to maintain tightness during operation and can tolerate higher levels of unloading, which maintain sealability. An idealized tightness curve showing the basis for gasket constants G_b, a, and G_s is shown in Fig. A7.17.

The data for many gasket styles and materials have been published in various PVRC-sponsored publications. Typical PVRC gasket factors for a variety of gasket types are shown in Table A7.15.

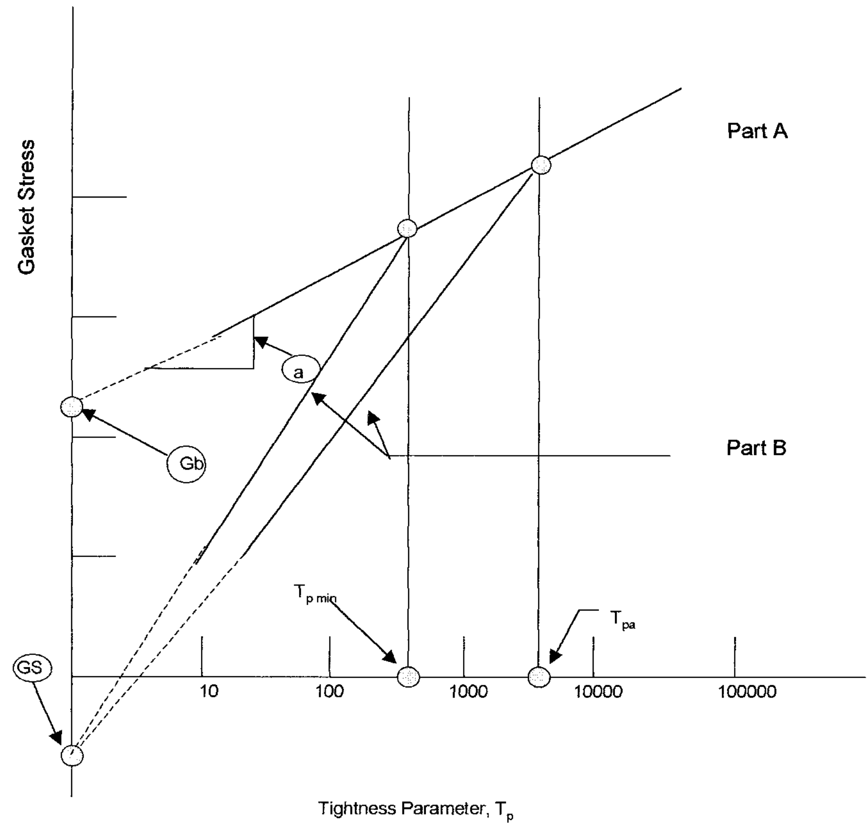


FIGURE A7.17 PVRC idealized tightness curve.

TABLE A7.15 Typical PVRC Gasket Factors

Type	Material	Gb(psi)	a	Gs(psi)
Spiral wound (Class 150 to 2500)	SS/Graphite	2300	0.237	13
	SS/Graphite with inner-ring	2530	0.241	4
	SS/Asbestos	3400	0.300	7
Metal-reinforced graphite	SS/Graphite	1665	0.293	0.02
Sheet gaskets	Graphite	1047	0.35	0.07
	Expanded PTFE	310	0.352	3.21
	Filled PTFE	444	0.332	.013
	CAF	2500	0.15	117
Corrugated gaskets	Soft iron	3000	0.160	115
	Stainless steel	4700	0.150	130
	Soft copper	1500	0.240	430
Metal jacketed	Soft iron	2900	0.230	15
	Stainless steel	2900	0.230	15
	Soft copper	1800	0.350	15
Metal-jacketed corr.	Soft iron	8500	0.134	230
Camprofile	SS/Graphite	387	0.33	14

Note: All data presented in this table is based on currently available published information. The PVRC continues to refine data-reduction techniques, and values are therefore subject to further review and alteration.

PVRC Convenient Method. The PVRC Convenient Method provides an easy conservative method for determining bolt load (W_{mo}) used in flange and gasket design as an alternate to using m and y values.

Gasket operating stress

$$S_{m1} = G_s[G_b/G_s \times T_{pa}{}^a]^{1/Tr} \tag{A7.4}$$

Seating stress

$$S_{m2} = G_b[e \times 1.5]\, T_{pa}{}^a = Pd[Ai/Ag] \tag{A7.5}$$

Design factor

$$M_o = \text{the greater of } \frac{S_{m1}}{Pd} \text{ or } \frac{S_{m2}}{Pd} \text{ or } 2 \tag{A7.6}$$

Design bolt load

$$W_{mo} = Pd(AgMo + Ai) \tag{A7.7}$$

where a—The slope associated with Part A tightness data

A_g—Area of gasket-seating surface, in^2 (mm^2) = .7854(OD^2 − ID^2)

A_i—Hydrostatic area; the area against which the internal pressure is acting, in^2 (mm^2) = $.7854G^2$

b_o—Basic gasket seating width, in (mm)
b_o = (OD − ID)/4

b—Effective gasket seating width

$b = b_o$, when $b_o \leq \frac{1}{4}$ in

$b = \sqrt{b_o/2}$, when $b_o > \frac{1}{4}$ in, in (mm)

C—Tightness constant
C = 0.1 for tightness class T1 (economy)
C = 1.0 for tightness class T2 (standard)
C = 10.0 for tightness class T3 (tight)

e—Joint assembly efficiency; recognizes that gasket-operating stress is improved depending on the actual gasket stress achieved during boltup; also recognizes the reliability of more sophisticated bolting methods and equipment in actually achieving desired bolt loads
e = 0.75 for manual boltup
e = 1.0 for ''ideal'' boltup, e.g., hydraulic stud tensioners, ultrasonics

G—Diameter of location of gasket load reaction, in (mm), from ASME Section 8

$$G = \frac{(\mathrm{OD} + \mathrm{ID})}{2} \text{ if } b_o \leq \tfrac{1}{4} \text{ in, in (mm)}$$
$$= \mathrm{OD} - 2b, \text{ if } b_o > \tfrac{1}{4} \text{ in, in (mm)}$$

G_b—The stress intercept at T_p = 1, associated with Part A tightness data psi (MPa)

G_s—The stress intercept at T_p = 1, associated with Part B tightness data psi (MPa)

P_d—Design pressure, psi (MPa)

P_t—Test pressure (generally 1.5 × P_d), psi (MPa)

S_{m1}—Operating gasket stress, psi (MPa)

S_{m2}—Seating gasket stress, psi (MPa)

M_o—Design factor

TC—Tightness class that is acceptable for the application, depending on the severity of the consequences of a leaker
T1 (economy) represents a mass leak rate per unit diameter of 0.2 mg/sec-mm
T2 (standard) represents a mass leak rate per unit diameter of 0.002 mg/sec-mm
T3 (tight) represents a mass leak rate per unit diameter of 0.0002 mg/sec-mm

T_p—Tightness parameter. T_p is a dimensionless parameter used to relate the performance of gaskets with various fluids, based on mass leak rate. Recognizes that leakage is proportional to gasket diameter (leak rate per unit diameter). T_p is the pressure (in atmospheres) required to cause a helium leak rate of 1 mg/sec for a 150 mm OD gasket in a joint. PVRC researchers have related T_p to other fluids through actual testing as well as use of laminar flow theory.

T_{pa}—Assembly tightness; the tightness actually achieved at assembly = .1243 × C × P_t

$T_{p\min}$—Minimum tightness; the minimum acceptable tightness for a particular application = .1243 × C × P_d

T_r—Tightness ratio; = log (T_{pa})/log ($T_{p\min}$)

W_{mo}—Design bolt load, lb (kN)

Example A7.1 Example of PVRC Convenient Method

Input data

Application: Heat exchanger. Reboiler channel

Design pressure (P_d) = 400 psi

Test pressure (P_t) = 600 psi

Operating temperature

Gasket type: camprofile

Gasket dimensions ID = 37.63 in

OD = 38.88 in

Tightness class required T3, therefore $C = 10.0$

Assembly efficiency (e) = 1.0, using hydraulic stud tensioners

From Table A7.15 $G_b = 387$ psi

$a = .33$

$G_s = 14.0$ psi

Calculations:

$G = 38.32$ in

$A_g = 75.10\ \text{in}^2$

$A_i = 1{,}153.8\ \text{in}^2$

$T_{p\min} = 497.20$

$T_{pa} = 745.80$

$T_p = 1.07$

$S_{m1} = 2{,}450$ psi

$S_{m2} = (3{,}852)$ psi

$$M_o = \frac{2{,}450}{400} = 6.13$$

Design total bolt load to achieve T3 leak tightness

$$W_{mo} = 400 \times (75.10 \times 6.13 + 1{,}153)$$
$$= 645{,}345 \text{ lb}$$

To illustrate the usefullness of PVRC calculations in gasket selection, the following example shows the same calculations using a double-jacketed gasket typically found in the above application instead of the camprofile.

Input data (as above except)

Gasket type: Double jacketed

$$G_b = 2900 \text{ psi}$$
$$a = .23$$
$$G_s = 15$$

This changes the calculation of S_{m1} and S_{m2} to

$$S_{m1} = 8{,}759 \text{ psi}$$
$$S_{m2} = 2{,}709 \text{ psi}$$
$$M_o = 21.90$$
$$W_{mo} = 1{,}119{,}255 \text{ lb}$$

TABLE A7.16 Application of Types of Gaskets

Gasket type	Pressure class: Low Class 150–300	Medium Class 600–900	High Class 1500–2500	Maximum temperature of materials (°F)
Nonmetallic				
–CAF	x	—	—	650–1000
–Nonasbestos fibre	x	—	—	550
–PTFE	x	—	—	390–550
–Graphite	x	—	—	750
Semimetallic				
–Metal jacketed	x	x	—	750+*
–Metal reinforced graphite	x	x	—	750+*
–Spiral wound	x	x	x	750+*
–Camprofile	x	x	x	750+*
Metallic				
–Ring-joint gaskets	—	x	x	650+*
–Lens ring	—	x	x	650+*
–Machined ring	—	x	x	650+*

x applicable
– not applicable
* depends on material

The total bolt load to achieve the same leak tightness of T3 is 1,119,255 lb.

These examples would indicate that higher leak tightness can be achieved using the camprofile gasket versus the double-jacketed gasket under the design conditions outlined.

Types of Gaskets

As discussed earlier, gaskets can be defined into three main categories: nonmetallic, semimetallic, and metallic. The general applications for each gasket type are shown in Table A7.16.

High Temperature Selection

In high temperature applications, above 650°F (343°C), gasket selection becomes even more critical. Many gaskets may perform well at low temperatures but fail to meet leak-tightness requirements at elevated temperatures.

Many gaskets lose their resiliency at elevated temperatures, with changes in their elastic behavior. The gasket's inherent stiffness will also tend to diminish, resulting in the gasket continuing to deform under the applied flange loads. This deformation (or creep) will result in loss of gasket stress, bolt load, and leak tightness. In elevated temperature applications, search out materials that retain their resiliency and gasket designs that will not change in thickness (retain its stiffness).

Considerable technical information on gasket selection is available from gasket manufacturers and from other technical sources such as the Pressure Vessel Research Council and industry trade associations such as the Fluid Sealing Association (FSA).

BOLT SELECTION

Bolts and nuts should be selected to conform to the design specifications set out with the flange design. Care is taken to ensure that the correct grade of material is selected to suit the recommended bolting temperature and stress ranges. Material specifications for bolts are outlined in BS 4882 and ASME Section VIII.

Common material specifications for bolts and nuts are shown in Table A7.17.

The following information should be specified when ordering bolts and nuts:

1. Quantity
2. Grade of material, identifying symbol of bolt or nut
3. Form
   - Bolts or studbolts
   - Nuts, regular or heavy series
4. Dimensions
   - Nominal diameter, length
   - Diameter of plain and reduced portion, length of thread (if applicable)
5. Identification of tests in addition to those stated in the standard
6. Manufacturer's test certificate (if required). Fully threaded studbolts and heavy series nuts are most common in industrial applications.

TABLE A7.17 Material Specifications for Bolts and Nuts and Recommended Bolting Temperature Range

Material specifications	Alloy types	Mechanical properties: Tensile strength		Mechanical properties: Yield strength .2% proof stress min		Mechanical properties: Recommended bolting temperature range(1) °C		Recommended corresponding nut grades
		N/mm^2	psi	N/mm^2	psi	min	max	
B7, L7 BS 1506–621A	1% Chromium molybdenum steel	860	123,000	730	103,000	−100	400	2H, 4, 7, or 8 (L4, 7 or 8 with L7 bolts)
B16 BS 1506–661	1% Chromium molybdenum vanadium steel	860	123,000	730	103,000	0	520	4, 7, or 8
B8, L8 BS 1506–801B	Austenitic chromium nickel 18/8 type steel	540	77,000	210	30,000	−250	575	8, 8F
B8, CX BS 1506–821T:	Stabilized austenitic chromium nickel 18/8 type steel, cold worked after solution treatment	860	123,000	700	99,000	−250	575	8 CX
B17B	Precipitation hardening austenitic nickel chromium steel	900	128,000	590	84,000	−250	650	17B
B80A BS 3076 NA20	Precipitation hardening nickel chromium titanium aluminum alloy	1000	143,000	620	88,000	−250	750	80A

Note: (1) Temperature of bolting refers to actual metal temperatures.

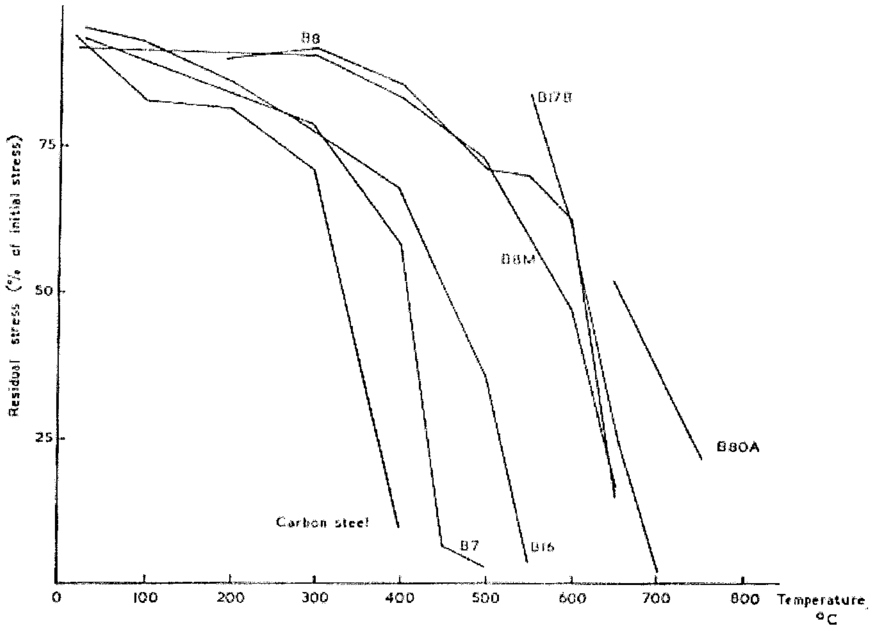


FIGURE A7.18 Stress relaxation behavior of various bolting materials showing percentage of initial stress retained at 1000 hours.

High Temperature Bolting Applications

The relaxation of bolt stress under constant strain conditions is widely recognized and has been measured in research on studbolt assemblies. At temperatures in excess of 300°C, special steels and alloys are required to improve upon the stress relaxation performance of low alloy steels. The relaxation behaviors of different bolting materials are shown in Fig. A7.18.

Different nut materials influence the stress-relaxation behavior of the stud, nut assembly.

The recommended nut for each grade of stud is shown in Table A7.17.

High temperature relaxation is a combined effect of gasket creep, bolt creep, and flange rotation. All three or any combination may occur. The symptoms show up as loose bolts that reduce gasket stress, resulting in increased leakage.

FLANGE STRESS ANALYSIS

The most common design standard for flanges is in ASME Section VIII, Appendix 3—"Mandatory Rules for Bolted Flange Connections." This standard applies in the design of flanges subject to hydrostatic end loads and to establish gasket seating.

The maximum allowable stress values for bolting outlined in the ASME code are design values to be used in determining the *minimum* amount of bolting required under the code. A distinction is made in the code between the design value and the bolt stress that may actually exist in the field. The ASME code Appendix S further acknowledges that an initial bolt stress higher than design value may (and, in some cases, must) be developed in the tightening operation. This practice to increase bolt stress higher than the design values is permitted by the code, provided that regard is given to ensure against excessive bolt loads, flange distortion, and gross crushing of the gasket.

General Requirements

Bolt Loads. In the design of the bolted flange connection, the bolt loads are calculated based on two design conditions of operating and gasket seating.

Operating Condition. The operating condition determines the minimum load according to

$$W_{m1} = \frac{3.14}{4} G^2 P_t + 2b\, 3.14\, G m P_t \tag{A7.8}$$

where b, G and P_t are defined previously and m is gasket factor expressed as a multiple of internal pressure

The equation is the sum of the hydrostatic end force plus a residual gasket load equaling a multiple of internal pressure.

Gasket Seating. The second design condition requires a minimum bolt load determined to seat the gasket regardless of internal pressure according to

$$W_{m2} = 3.14\, bGy \tag{A7.9}$$

where y is the minimum seating stress for the gasket selected

PVRC Method. As discussed earlier the PVRC method can be used as an alternate to W_{m1} or W_{m2} in calculating the bolt loads used in the design of the flange.

Total Required Bolt Areas. These design values on bolt loads are used to establish minimum total cross-sectional areas of the bolts A_m. A_m is determined as follows:

$$A_{m1} = \frac{W_{m1}}{Sb}, \quad \text{where } Sb \text{ is allowable bolt stress at operating temperature}$$

$$A_{m2} = \frac{W_{m2}}{Sa}, \quad \text{where } Sa \text{ is allowable bolt stress at atmospheric temperature}$$

Using PVRC bolt loads:

$$A_{mo} = \frac{W_{mo}}{Sa}$$

A_m is greater of A_{m1} or A_{m2} or A_{mo}. Bolts are then selected so that the actual bolt area, A_b, is equal to or greater than A_m.

Example Calculation. Using the same application outlined in the "Gasket Selection" section, the following shows the calculation of bolt loads using m and y factors.

Input Data

Gasket type: Camprofile

$m = 2$

$y = 2500$

ID = 37.63 in

OD = 38.88 in

Design Pressure (P) = 400 psi

Operating Conditions

$$W_{m1} = \frac{3.14}{4} \times 38.32^2 \times 400 + 2 \times .28 \times 3.14 \times 38.32 \times 2 \times 400 = 515{,}225 \text{ lb}$$

Gasket Seating

$$W_{m2} = 3.14 \times .28 \times 38.32 \times 2500 = 84{,}227 \text{ lb}$$

$W_{m1} > W_{m2}$, therefore W_{m1} would govern in the flange design. Note that using the PVRC method, the design bolt load was 645,345 lb, higher than both W_{m1} and W_{m2}. This will be a common occurrence, revealing that higher bolt loads than assumed using m and y factors are required to achieve required leak tightness.

Flange Design. The bolt loads used in the flange design by the code is

$$W = \frac{(A_m + A_b)Sa}{2} \tag{A7.10}$$

Alternately, where additional safety is desired, the code recommends that the bolt load for flange design is actual bolt area (A_b) times the allowable bolt stress (Sa).

For critical flanges, it is suggested that a more conservative approach to flange design be adopted, calculating the design bolt load as actual bolt area (A_b) times expected field bolt stress (Se). The expected field-bolt stress (Se) achieved is often $1.5 \times Sa$. By using this approach a higher bolt load is determined. This will increase the flange thickness. The benefits to increased flange thickness are

1. Thicker flanges will rotate less and distribute the applied bolt load more uniformly to the gasket.
2. Thicker flanges require longer bolts. Longer bolts have more strain energy and are more forgiving to joint relaxation.

Finite Element Analysis

Finite Element Analysis (FEA) is being used more frequently to review designs of critical flanges. FEA costs are dropping dramatically while the procedure's effectiveness to model complex structure is increasing.

FEA can be used to predict the behavior of the flange structure subjected to its operating conditions. It is possible to predict the behavior of the flange structure mathematically because the behavior of the materials can be described mathematically. Hooke's law describes the mechanical behavior of the metal materials and their elastic response. Other types of stress-strain relationships have been developed to model the nonlinear, plastic behavior of the gasket.

The key is to determine the actual operating stress on the gasket to predict its leak-tightness performance subjected to thermal effects, pressure, bolt stress, relaxation, and flange rotation.

ASSEMBLY CONDITIONS

The flange components consisting of flange, gaskets, and bolts may have been adequately designed but their performance to specifications will be affected by assembly conditions.

Flange Surface Finish

Flange surface finish is critical to achieve the design-sealing potential of the gasket. Again, gasket-leak tightness is dependent upon its operating gasket stress. Flanges that are warped, pitted, rotated, and have incorrect flange gasket-surface finish will impair the leak tightness of the gasket.

Flanges out of parallelism and flatness should be held within ASME B 16.5 specifications. This will ensure that the uniform bolt loads translate to uniform gasket stress.

The resiliency and compressibility of the gasket are affected by flange surface finish. Recommended flange surface finishes for various gasket types are shown in Table A7.18.

TABLE A7.18 Recommended Flange Surface Finish for Various Gasket Types

Gasket type	Flange surface finish microinch CLA	Flange surface finish micrometer Ra
Soft cut sheet gaskets	Material <1.5 mm thick 125–250	Material <1.5 mm thick 3.2–6.3
	Material ≥1.5 mm thick 125–500	Material ≥1.5 mm thick 3.2–12.5
Camprofile	125–250	3.2–6.3
Metal reinforced graphite	125–250	3.2–6.3
Spiral wound	125–250	3.2–6.3
Metal-jacketed gaskets	100 max	2.5 max
Solid metal gaskets	63 max	1.6 max

Gasket Condition

Never reuse a gasket. A gasket's compressibility and resiliency are severely reduced once it has been used.

Check the gasket for any surface defects along the contact faces that may impair sealing.

Keep the gasket on its storage board until immediately prior to assembly.

Do not use any gasket compounds to install the gasket to the flange, as it affects the compressibility, resiliency, and creep behavior of the gasket. Consult the gasket manufacturer when installing large diameter gaskets for a recommendation on how to secure them to the flange during installation.

Bolt Condition

Bolts and nuts may be reused providing they are in new condition. Ensure bolts and nuts are clean, free of rust, and that the nut runs freely on the bolt threads. Install bolts and nuts well lubricated by using a high quality anti-seize lubricant to the stud threads and the nut face.

Methods of Bolt Tightening

Once the total bolt loads (W) are calculated for the flanges, specifications, and procedures should be adopted outlining how to achieve the design bolt load.

The total bolt load (W) for the flange is divided by the number of bolts to determine the individual bolt preload (F_p).

To achieve improved leak tightness sufficient and uniform gasket stress must be realized in the field. This obviously requires uniform and correct applied bolt load. The higher the requirement to reduce leakage, the more controlled the method bolt tightening.

The common methods of bolt tightening are:

- hammer, impact wrenches
- torque wrenches
- hydraulic tensioning systems

Each method has its own assembly efficiency. Bolt tightening methods and their assembly efficiencies are shown in Table A7.19.

Hammer, Impact Wrenches Method

This method remains the most common form of bolt tightening. The advantages are speed and ease of use. Disadvantages include a lack of preload control and the inability to generate sufficient preload on large bolts.

Torque Method

Torque wrenches are often regarded as a means to improve control over bolt preload in comparison with hammer-tightening methods. However, as indicated in Table A7.19, significant variation in stud-to-stud load control is still evident.

TABLE A7.19 Tightening Methods and Assembly Efficiencies

Method to control bolt preload	Tightening method	Stud-to-stud load variation from the mean (%)	Assembly efficiency (e)
No torque/stretch control	Power impact, lever or hammer wrench	$> \pm 50\%$	0.75
Torque control	Calibrated torque wrench or hydraulic wrench	± 30 to $\pm 50\%$	0.85
Tensioner load control	Multiple stud tensioners	± 10 to $\pm 15\%$	0.95
Direct measurement of stress or strain	Ultrasonic extensometer, calipers, strain gages	$\pm 10\%$ or less	1

Much attention is given to the level of torque that should be applied to a specific application. However, it is not the torque that is important but the end result of the torque-bolt preload. Control over bolt preload is the factor for ensuring proper gasket-seating stresses are achieved.

Torque is the measure of the torsion required to turn a nut up the inclined plane of a thread. The efficiency of the nut's turn along the bolt thread to generate preload is dependent upon many factors, including thread pitch, friction between the threads, and friction between the nut face and the flange face.

In general, only about 10 percent of the applied torque goes toward providing bolt preload. The rest is lost in overcoming friction: 50 percent in overcoming the friction between the nut and flange faces, and 40 percent in overcoming friction between the threads of the nut and the bolt.

Another variable to overcome is the elastic behavior of the joint as illustrated in Fig. A7.15.

As the bolts are tightened creating the desired preload, the flange will partially compress. As additional bolts are tightened, the flange joint will compress a little further. The continuous deflection of the flanged joint reduces the stretch (or preload) of previously tightened joints. This phenomenon is referred to as cross talk and is a result of tightening a multistud flange one bolt at a time.

A typical wide variation in bolt and bolt preload is experienced using torquing because of the uncontrolled effects of friction and cross talk, as illustrated in Fig. A7.16, "Typical Load Scatter of 28 Bolt Heat Exchanger Flange."

Torque Calculations. The amount of torque that is required to generate a specific bolt preload is calculated by

$$T = \frac{K D F_p}{12} \text{ ft-lb} \tag{A7.11}$$

where K = nut factor, experimentally determined (see Table A7.20)
D = nominal diameter of stud, in
F_p = desired bolt preload, calculated by dividing total design bolt load (W) by number of bolts

TABLE A7.20 Torque Nut Factors (K)

Bolt and lubricant	Nut factor (K) reported range
As received alloy bolt	0.158–0.267
As received stainless studbolt	0.3
Copper-based antiseize	0.08–0.23
Nickel-based antiseize	0.13–0.27
Moly paste or grease	0.10–0.18

Note: It is important to remember that the K value is an experimentally derived constant. The K value should be verified in the field for each new application.

Example: Torque Calculation. Application: Heat exchanger–Reboiler channel (see Section "Gasket Selection")

Input data:

Type of stud bolt: ASTM A193, Gr. B7

Number of bolts: 56

Size of bolt: $(d) = .875$ in

Design bolt load: $(W_{mo}) = 645{,}345$ lb

Nut factor $(K) = .18$ copper based antiseize lubricant

Calculations:

$$F_p = 645{,}345/56 = 11{,}524 \text{ lb/bolt}$$

$$T = .18 \times .875 \times 11{,}524/12 = 150 \text{ ft-lb}$$

Torque Procedure. Torquing should be applied in multipasses following a cross pattern to reduce warping of flange, crushing the gasket, and to minimize cross talk in achieving bolt preload.

Pass	**Torque**
1	⅓ of final torque (T). Start at bolt no. 1 and follow cross pattern
2	⅔ of final torque (T) following cross pattern
3	At final torque (T) following cross pattern
4	At final torque (T), start at highest bolt number and tighten in a counterclockwise sequence

The cross pattern is easily followed once the bolts are numbered in the flange. Randomly select a bolt and designate it as bolt number 1. Proceed in a clockwise motion to the next bolt and add four to the previous bolt number. Moving clockwise, the next bolt number would be 1, 5, 9, and so on.

This system of adding four to the previous bolt number continues until adding four to the previous number exceeds the total number of bolts in a flange. At this point, start again at bolt number 3. Continue in the same clockwise direction, numbering bolts 3, 5, 11, and so on until again this number is larger than the total number of bolts in a flange. At this point the next number is 2; continue as previously described: 2, 6, 10, and so on. The last series of bolt numbers start with bolt number 4 and continue 8, 12, 16, and so on.

A sample 16-bolt flange showing a typical cross pattern is shown in Fig. A7.19.

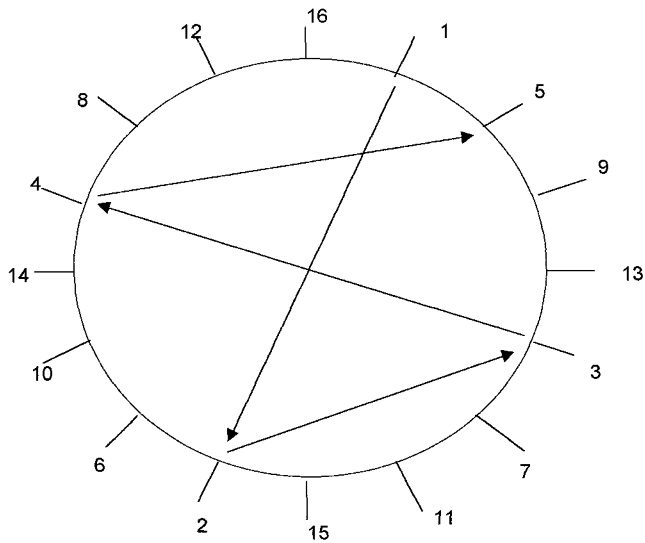


FIGURE A7.19 Typical torquing cross pattern of a 16-bolt flange.

Tensioning Systems

Many of the variables that reduce the control of bolt preload using the torque process are eliminated using hydraulic tensioning systems.

Hydraulic tensioners are hollow hydraulic compact cylinders that are threaded onto a protruding section of the studbolt generally using a pulling device. A bridge supports the hydraulic head straddling the nut and reacting against the flange while hydraulic pressure is applied to the hydraulic head. Under the applied hydraulic load, the bolt stretches at the same time as it compresses the flange and gasket. Residual bolt load equivalent to the desired preload (F_p) is achieved by manually turning down the nut under the tensioner bridge during the applied hydraulic load. Applied bolt load is directly proportional to the hydraulic pressure and the area of the hydraulic cylinder. There are no frictional losses associated with tensioning, as compared to torquing. A cross section of a hydraulic stud tensioner is illustrated in Fig. A7.20. The residual load (preload F_p) = Applied Load − Load Loss Factor.

The load loss factor is dependent upon the stud stress realized, bolt diameter, and effective length of the bolt. For each application its load loss factor can be precisely calculated to determine the necessary applied load to generate the residual preload. Development of thorough procedures is essential to maintain the accuracy of hydraulic stud-tensioning process.

Cross talk is significantly reduced by utilizing multistud tensioning. Generally 50 percent of the studs in a flange are tensioned simultaneously by using multiple

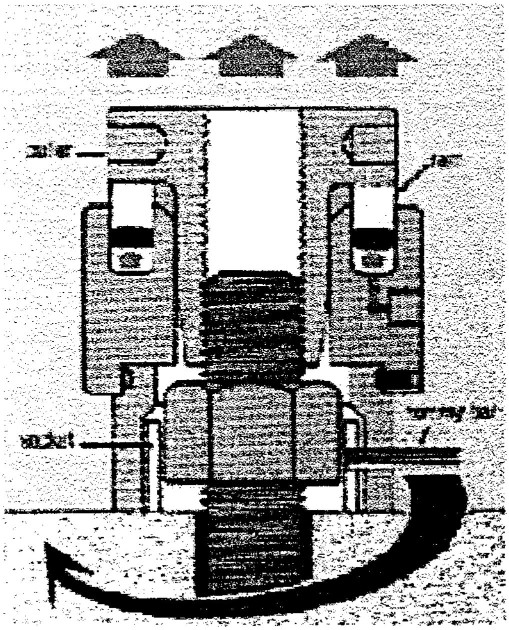


FIGURE A7.20 Hydraulic stud tensioning.

tools interconnected with a high-pressure hose tied into a common pump source. Many flange configurations allow for 100 percent of the studs to be tensioned simultaneously. This completely eliminates cross talk.

Hydraulic tensioning provides the most controlled tightening method for achieving specified bolt preload.

Controlled Bolting

Controlled bolting is the method where the loading-stress of the flange bolts is measured using ultrasonic equipment to ensure that the correctly specified bolt preload is achieved.

The application of torque alone to the flange is not controlled bolting, as there remain many uncertainties about the actual bolt load.

Torquing in combination with ultrasonic measure provides necessary controls to achieve the required bolt preload.

Multistud tensioning following established procedures provides a high degree of control over bolt preload. In critical application, multistud tensioning should also be combined with ultrasonic measurement to verify that all specifications are met.

BOLT LOAD MONITORING

Monitoring of the actual residual bolt load after tightening is essential to ensure that leak-tightness goals are achieved and becomes an important part of the quality assurance process of achieving flange joint integrity.

All tightening methods provide a degree of stud-preload scatter as a function of their process capability. The only way to be sure that specified stud preload is achieved is to measure it.

There are several methods for performing stud-stretch measuring, including strain gauges, bow micrometers, mechanical extensometers, and ultrasonic extensometers. The most common and versatile is the ultrasonic extensometer.

Theory of Operation

The ultrasonic extensometer operates by placing a high-frequency transducer at one end of the stud. Frequencies used for stud measurement range from 1 to 20 megahertz. At these frequencies a liquid couplant (gel) is used to couple the ultrasound from the transducer to the stud.

An ultrasound wave is generated by the transducer and travels down the body length of the stud. The wave reflects off the opposite end of the stud and travels back to the transducer.

The ultrasonic instrument measures the time of flight of the ultrasound in the stud. Many factors, including material density, stud length, temperatures, and stress are used to convert the time-of-flight measurement into an ultrasonic reference length.

Hooke's Law and Stud-Stretch Measurement

All studs elongate in their elastic region following Hooke's law, as outlined in the section "Function of Bolts".

In the relaxed state, a reference length is measured using the ultrasonic extensometer. After the stud has been tightened, an additional reading is made to measure stud stretch. Given the known parameters of effective bolt length (L_b), tensile area of bolt (A_s), Young's modulus of elasticity (E), the preload (F_p) can now be directly correlated to stretch. Rearranging equation A7.1 allows the calculation of bolt preload (F_p):

$$F_p = \frac{\Delta L_b}{L_b} \times E \times A_s$$

The measured residual preload can then be compared to design preload to ensure it falls within an acceptable tolerance. Alternately the stretch reading ΔL_b actual is compared to ΔL_b design.

Example A7.2 Example Calculation: Application: Heat exchange–Reboiler channel

Input data:

$$A_s = .7854\left(.875 - \frac{.9743}{9}\right)^2 = .462 \text{ in}^2$$

Stud size $(D) = 0.875$ in

Thread pitch$(n) = 9$

Material = A193, Gr. B7

Young's modulus $(E) = 30 \times 10^6$ psi

Effective length (L_b) = Grip length $+ 1 \times D$, where grip length equals measured distance between nut faces

Design bolt load $(F_p) = 11{,}524$ lb

Calculation:

$$\Delta L_b = \frac{11{,}524 \times 9.5}{30 \times 10^6 \times .462} = 0.008 \text{ in}$$

Expected tolerance on critical application is ± 10%; therefore, actual ΔL_b should fall between 0.0072 in and .0088 in.

MANAGING FLANGE JOINT INTEGRITY

Leaks are a threat to profits, safety, and the environment. Problems resulting from leaks in flanges can range from local in severity to plant-wide catastrophe. Although the range of negative results can vary widely, leaks have one thing in common: all leaks are preventable.

Leaks don't happen by accident, they happen by design. Rather than being symptoms of product failure, leaks are generally evidence of failure in process control. When you fix the process control, you fix the leaks before they happen.

This chapter has reviewed the key elements to achieving flange joint integrity to assure leak-free integrity of the bolted flange joint. An integrated approach must be adopted to ensure success of the process of joint integrity.

This process begins with an understanding of the operating environment, continues with design and selection of the flange components, setting of assembly specifications, establishment of best-practices procedures, assignment of competent personnel, quality assurance, traceability through complete documentation, and finishes with meeting the goal of leak prevention.

The goal of leak prevention is achievable and starts with a mind-set of doing things right the first time.