

LAMONS

GASKET HANDBOOK

THE GASKET COMPANY

Edited for Lamons Gasket Co.

In cooperation with

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INTRODUCTION

The cost of leaky joints in industry today is staggering. Out-of-pocket costs run into billions of dollars annually in lost production, waste of energy, loss of product and, most recently, impact on the environment. These problems are increasing, not decreasing. It behooves all of us to consolidate our knowledge and experience to solve or at least minimize these problems. This publication is being produced because we, as gasket manufacturers and suppliers, are constantly called upon to solve sealing problems after the fact. Too often we find insufficient time and attention has been given to:

- proper design of flanged joint
- installation procedures and
- selection of the optimum gasket material required to solve a particular sealing problem.

We will endeavor to outline in this publication those areas we believe to be essential in a properly designed, installed and maintained gasketed joint.

We believe most people involved with the design, installation, and maintenance of gasketed joints realize that no such thing as "zero" leakage can be achieved. Whether or not a joint is "tight" depends on the sophistication of the methods used to measure leakage. In certain applications the degree of leakage may be perfectly acceptable if one drop of water per minute is noted at the gasketed joint. Other requirements are that no bubbles would be observed if the gasketed joint was subjected to an air or gas test underwater and a still more stringent inspection would require passing a mass spectrometer test. The rigidity of the test method would be determined by:

- the hazard of the material being confined
- loss of critical materials in a process flow
- impact on the environment should a particular fluid escape into the atmosphere
- danger of fire or of personal injury

All of these factors dictate proper attention must be given to:

- design of flange joints or closures
- proper selection of gasket type
- proper gasket material
- proper installation procedures

Care in these areas will ensure that the best technology goes into the total package and will minimize operating costs, pollution of the environment and hazards to employees and the general public.

SECTION I — DESIGN

WHY GASKETS ARE USED

Gaskets are used to create a static seal between two stationary members of a mechanical assembly and to maintain that seal under operating conditions which may vary dependent upon changes in pressures and temperatures. If it were possible to have perfectly mated flanges and if it were possible to maintain an intimate contact of these perfectly mated flanges throughout the extremes of operating conditions, a gasket would not be required. This is virtually an impossibility either because of

- the size of the vessel and/or the flanges
- the difficulty in maintaining such extremely smooth flange finishes during handling and assembly
- corrosion and erosion of the flange surfaces during operations.

As a consequence, relatively inexpensive gaskets are used to provide the sealing element in these mechanical assemblies. In most cases, the gasket provides a seal by external forces flowing the gasket material into the imperfections between the mating surfaces. It follows then that in a properly designed gasket closure, three major considerations must be taken into account in order for a satisfactory seal to be achieved.

- Sufficient force must be available to initially seat the gasket. Stating this another way, adequate means must be provided to flow the gasket into the imperfections in the gasket seating surfaces.
- Sufficient forces must be available to maintain a residual stress on the gasket under operating conditions to ensure that the gasket will be in intimate contact with the gasket seating surfaces to prevent blow-by or leakage.
- The selection of the gasket material must be such that it will withstand the pressures exerted against the gasket, satisfactorily resist the entire temperature range to which the closure will be exposed and withstand corrosive attack of the confined medium.

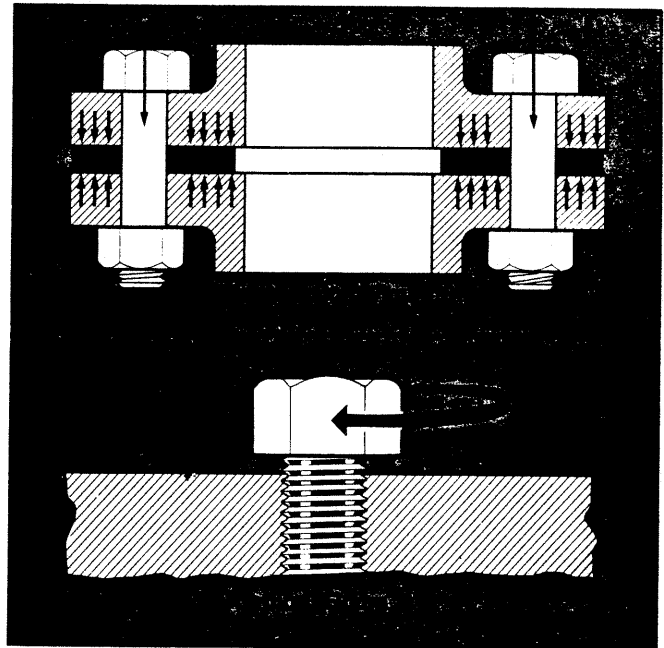
EFFECTING A SEAL

A seal is affected by compressing the gasket material and causing it to flow into the imperfections on the gasket seating surfaces so that intimate contact is made between the gasket and the gasket seating surfaces preventing the escape of the confined fluid. Basically there are four different methods that may be used either singly or in combination to achieve this unbroken barrier.

- Compression (Figure 1). This is by far the most common method of effecting a seal on a flange joint and the compression force is normally applied by bolting.
- Attrition (Figure 2). Attrition is a combination of a dragging action combined with compression such as in a spark plug gasket where the spark plug is turned down on a gasket that is both compressed and screwed into the flange.

- By heat, such as in the case of sealing a bell and spigot joint on cast iron pipe by means of molten lead. Note, however, that after the molten lead is poured, it is tamped into place using a tamping tool and a hammer.
- Gasket lip expansion. This is a phenomenon that would occur due to edge swelling when the gasket would be affected by confined fluid, as in the case of elastomeric compounds affected by the confined fluids, such as solvents, causing the gasket material to swell and increase the interaction of the gasket against the flange faces.

In general, it is considered that only the gasket surface provides the seal in the joint. It is assumed that the gasket material is impervious and will not permit any passage of fluid through the gasket seal.



GASKET SEATING

There are two major factors to be considered with regard to gasket seating.

The first is the gasket material itself. The ASME Unfired Pressure Vessel Code Section VIII, Division 1 defines minimum design seating stresses for a variety of gasket materials. These design seating stresses range from zero psi for so-called self-sealing gasket types such as low durometer elastomers and O-rings to 26,000 psi to properly seat solid flat metal gaskets. Between these two extremes there are a multitude of materials available to the designer enabling him to make a selection based upon the specific operating conditions under investigation. Table No. 1 indicates the more popular types of gaskets covered by ASME Unfired Pressure Vessel Code. (con't on page 6)

TABLE 1

TABLE UA-49.1
GASKET MATERIALS AND CONTACT FACINGS

Gasket Factors (<i>m</i>) for Operating Conditions and Minimum Design Seating Stress (<i>y</i>)					Refer to Table UA-49.2	
NOTE: This table gives a list of many commonly used gasket materials and contact facings with suggested design values of <i>m</i> and <i>y</i> that have generally proved satisfactory in actual service when using effective gasket seating width <i>b</i> given in Table UA-49.2. The design values and other details given in this table are suggested only and are not mandatory.						
Gasket material		Gasket factor <i>m</i>	Min. design seating stress <i>y</i> (psi)	Sketches and notes	Use facing sketch	Use column
Self-Energizing types O Rings. Metallic. Elastomer other gasket types considered as self-sealing		0	0	—	—	—
Elastomers without fabric or a high percentage of asbestos fiber: Below 75 Shore Durometer 75 or higher Shore Durometer		0.50 1.00	0 200		1 (a, b, c, d) 4, 5	II
Asbestos with a suitable binder for the operating conditions	1/8 thick	2.00	1600			
	1/16 thick	2.75	3700			
	1/32 thick	3.50	6500			
Elastomers with cotton fabric insertion		1.25	400			
Elastomers with asbestos fabric insertion, with or without wire reinforcement	3-ply	2.25	2200			
	2-ply	2.50	2900			
	1-ply	2.75	3700			
Vegetable fiber		1.75	1100			
Spiral-wound metal, asbestos filled		Carbon Stainless or Monel	3.00	10000		1 (a, b)
Corrugated metal, asbestos inserted or Corrugated metal, jacketed asbestos filled	Soft Aluminum	2.50	2900		1 (a, b, c, d)	
	Soft copper or brass	2.75	3700			
	Iron or soft steel	3.00	4500			
	Monel or 4-6% chrome	3.25	5500			
	Stainless steels	3.50	6500			
Corrugated metal	Soft aluminum	2.75	3700			
	Soft copper or brass	3.00	4500			
	Iron or soft steel	3.25	5500			
	Monel or 4-6% chrome	3.50	6500			
	Stainless steels	3.75	7600			
Flat metal jacketed asbestos filled	Soft aluminum	3.25	5500		1a, 1b, 1c*, 1d*, 2*	
	Soft copper or brass	3.50	6500			
	Iron or soft steel	3.75	7600			
	Monel	3.50	8000			
	4-6% chrome	3.75	9000			
Grooved metal	Soft aluminum	3.25	5500		1 (a, b, c, d) 2, 3	
	Soft copper or brass	3.50	6500			
	Iron or soft steel	3.75	7600			
	Monel or 4-6% chrome	3.75	9000			
	Stainless steels	4.25	10100			
Solid flat metal	Soft aluminum	4.00	8800		1 (a, b, c, d) 2, 3, 4, 5	I
	Soft copper or brass	4.75	13000			
	Iron or soft steel	5.50	18000			
	Monel or 4-6% chrome	6.00	21800			
	Stainless steels	6.50	26000			
Ring joint	Iron or soft steel	5.50	18000		6	
	Monel or 4-6% chrome	6.00	21800			
	Stainless steels	6.50	26000			

*The surface of a gasket having a lap should be against the smooth surface of the facing and not against the nubbin.

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TABLE 2

TABLE UA-49.2
EFFECTIVE GASKET WIDTH

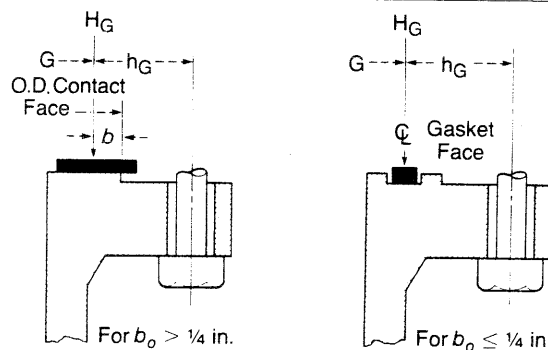
Facing Sketch Exaggerated		Basic Gasket Seating Width, b_o	
		Column I	Column II
1a			
1b*		$\frac{N}{2}$	$\frac{N}{2}$
1c	 $w \leq N$		
1d*	 $w \leq N$	$\frac{w + T}{2}, \left(\frac{w + N}{4} \text{ max} \right)$	$\frac{w + T}{2}, \left(\frac{w + N}{4} \text{ max} \right)$
2	 $w \leq \frac{N}{2}$	$\frac{w + N}{4}$	$\frac{w + 3N}{8}$
3	 $w \leq \frac{N}{2}$	$\frac{N}{4}$	$\frac{3N}{8}$
4*	 $w \leq \frac{N}{2}$	$\frac{3N}{8}$	$\frac{7N}{16}$
5*	 $w \leq \frac{N}{2}$	$\frac{N}{4}$	$\frac{3N}{8}$
6	 $w \leq \frac{N}{2}$	$\frac{w}{8}$	

Effective Gasket Seating Width, "b"

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

$$b = \sqrt{\frac{b_o}{2}}, \text{ when } b_o > \frac{1}{4} \text{ in.}$$

Location of Gasket Load Reaction



NOTE: The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.

*Where serrations do not exceed 1/64 in. depth and 1/32 in. width spacing, sketches 1b and 1d shall be used.

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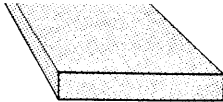
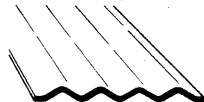
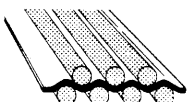
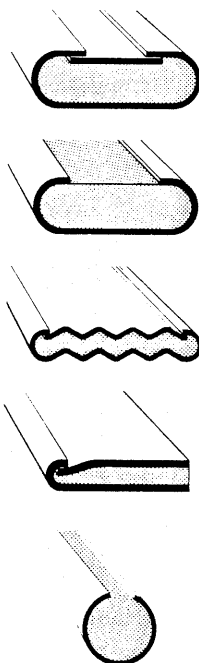
(con't from page 3)

The second major factor to take into consideration must be the surface finish of the gasket seating surface. As a general rule, it is necessary to have a relatively rough gasket seating surface for elastomeric, compressed asbestos and PTFE gaskets on the order of magnitude of 500 microinches to 750 microinches. Solid metal gaskets normally require a surface finish not rougher than 63 microinches. Semi-metallic gaskets such as spiral-wound fall between these two general types. The reason for the difference is that with non-metallic gaskets such as rubber and compressed asbestos, there must be sufficient roughness on the gasket seating surfaces to bite into the gasket thereby preventing excessive extrusion and increasing resistance to gasket blowout. In the case of solid metal gaskets, extremely high unit loads are required to flow the gasket into imperfections on the gasket seating sur-

faces. This requires that the gasket seating surfaces be as smooth as possible to ensure an effective seal. Spiral-wound gaskets, which have become extremely popular in the last fifteen to twenty years, do require some surface roughness to prevent excessive radial slippage of the gasket under compression. The characteristics of the type of gasket being used dictate the proper flange surface finish that must be taken into consideration by the flange designer and there is no such thing as a single optimum gasket surface finish for all types of gaskets. The problem of the proper finish for gasket seating surface is further complicated by the type of the flange design. For example a totally enclosed facing such as tongue and groove will permit the use of a much smoother gasket seating surface than can be tolerated with a raised face.

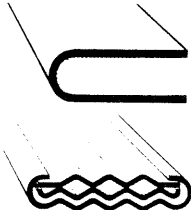
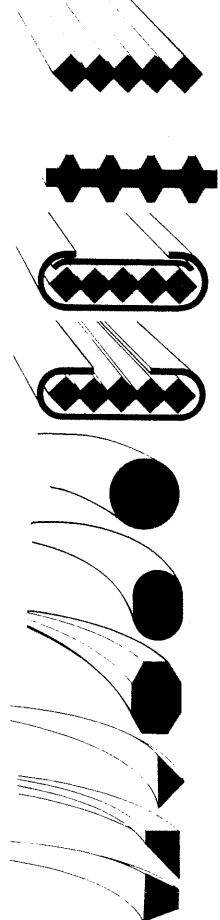
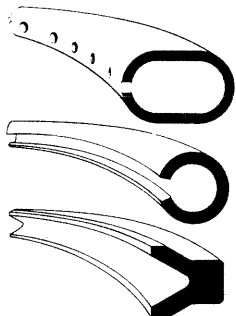
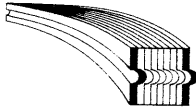
Table 3 includes recommendations for normal finishes for the various types of gaskets.

TABLE 3
GASKET SEATING SURFACE FINISHES

Gasket Description	Gasket Cross-Section	Flange Surface Finish μ'' — AARH
Flat — Non-Metallic (all materials)		500 — 750
Flat — Metallic * SEE NOTE 1		63
Corrugated metal		63
Corrugated metal with soft filler		125
Metal jacketed gaskets		63-80

NOTE: This table gives a list of suggested surface finishes that have generally proven satisfactory in actual service. They are suggested only and not mandatory; however, they are based upon the best cross-section of successful design experience currently available.

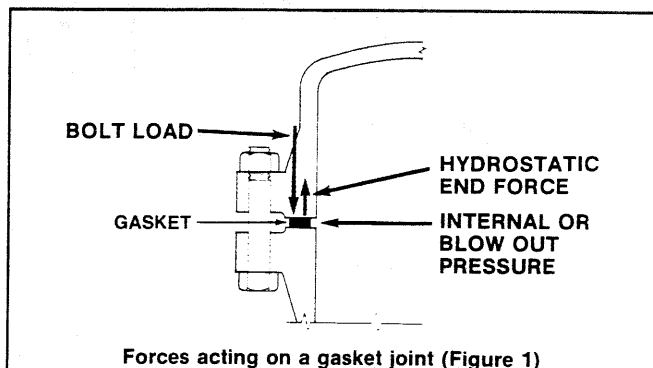
TABLE 3 — GASKET SEATING SURFACE FINISHES CONT.

Gasket Description	Gasket Cross-Section	Flange Surface Finish μ'' — AARH
Metal jacketed gaskets (cont.)		63-80
Solid metal		63
Hollow metal		32
Spiral wound ** SEE NOTE 2		80-500

Note ① — Solid metal washer type gaskets require extremely high seating stresses to seal. This usually necessitates a bolt area to gasket area greater than a ratio of 2:1. If this is not possible, it is preferred to use a profiled or serrated gasket to achieve the necessary seating load on the gasket.

Note ② — Refer to page 25 for more details on flange surface finishes for spiral wound gaskets.

FORCES ACTING ON A GASKETED JOINT



Forces acting on a gasket joint (Figure 1)

- **THE INTERNAL PRESSURE:** These are the forces continually trying to unseal a gasketed joint by exerting pressure against the gasket (blowout pressure) and against the flanges holding the gasket in place (hydrostatic end force). See Figure 1.
- **THE FLANGE LOAD:** The total force compressing the gasket to create a seal, i.e., the effective pressure resulting from the bolt loading.
- **TEMPERATURE:** Temperature creates thermo-mechanical effects, expanding or contracting the metals, affecting the gasket material by promoting "creep relaxation" which is a permanent strain or relaxation quality of many soft materials under stress. The effect of certain confined fluids may become increasingly degrading as temperature rises and attack upon organic gasket materials is substantially greater than at the ambient temperatures (about 75°F). As a rule, the higher the temperature, the more critical becomes the selection of the proper gasket.
- **MEDIUM:** The liquid or gas against which the gasket is to seal.
- **GENERAL CONDITIONS:** The type of flange, the flange surfaces, the type of bolt material, the spacing and tightness of the bolts, etc.

Each of these factors require consideration before an effective gasket material is finally chosen. However, the proper gasket may often be rejected because failure occurred due to a poorly cleaned flange face, or improper bolting-up practice. These details require careful attention, but if complied with will help eliminate gasket blow-out or failure.

There are three principal forces acting on any gasketed joint. They are:

- Bolt load and/or other means of applying the initial compressive load that flows the gasket material into surface imperfections to form a seal.
- The hydrostatic end force, that tends to separate flanges when the system is pressurized.
- Internal pressure acting on the portion of the gasket exposed to internal pressure, tending to blow the gasket out of the joint and/or to bypass the gasket under operating conditions.

There are other shock forces that may be created due to sudden changes in temperature and pressure. Creep relaxation is another factor that may come into the picture. Figure 1 indicates the three primary forces acting upon a gasketed joint which we will consider for this discussion. The initial compression force applied to a joint must serve several purposes.

- It must be sufficient to initially seat the gasket and flow the gasket into the imperfections on the

gasket seating surfaces regardless of operating conditions.

- Initial compression force must be great enough to compensate for the total hydrostatic end force that would be present during operating conditions.
- It must be sufficient to maintain a residual load on the gasket/flange interface.

From a practical standpoint, residual gasket load must be "X" times internal pressure if a tight joint is to be maintained. This unknown quantity "X" is what is known as the "m" factor in the ASME unfired pressure vessel code and will vary depending upon the type of gasket being used. Actually the "m" value is the ratio of residual unit stress (bolt load minus hydrostatic end force) on gasket (psi) to internal pressure of the system. The larger the number used for "m," the more conservative the flange design would be, and the more assurance the designer has of obtaining a tight joint.

BOLT LOAD FORMULAS

The ASME Unfired Pressure Vessel Code, Section VIII, Division 1 defines the initial bolt load required to seat a gasket sufficiently as:

$$W_{m2} = \pi b G y$$

The required operating bolt load must be at least sufficient, under the most severe operating conditions, to contain the hydrostatic end force and, in addition, to maintain a residual compression load on the gasket that is sufficient to assure a tight joint. ASME defines this bolt load as:

$$W_{m1} = \frac{\pi}{4} G^2 P + 2b\pi G m P$$

After W_{m1} and W_{m2} are calculated, then the minimum required bolt area A_m is determined:

$$A_{m1} = \frac{W_{m1}}{S_b}$$

$$A_{m2} = \frac{W_{m2}}{S_a}$$

$$A_m = A_{m1} \text{ if } A_{m1} \geq A_{m2}$$

OR

$$A_m = A_{m2} \text{ if } A_{m2} \geq A_{m1}$$

Bolts are then selected so that the actual bolt area A_b is equal to or greater than A_m

$$A_b = (\text{Number of Bolts}) \times (\text{Minimum Cross-Sectional Area of Bolt in Square Inches})$$

$$A_b \geq A_m$$

The maximum unit load $S_{g(max)}$ on the gasket bearing surface is equal to the total maximum bolt load in pounds divided by the actual sealing area of the gasket in square inches.

$$S_{g(max)} = \frac{A_b S_a}{\frac{\pi}{4} [(OD - 0.125)^2 - (ID)^2]} \quad \left\{ \begin{array}{l} \text{Spiral} \\ \text{Wound} \\ \text{Gaskets} \end{array} \right.$$

$$S_{g(max)} = \frac{A_b S_a}{\frac{\pi}{4} [(OD)^2 - (ID)^2]} \quad \left\{ \begin{array}{l} \text{All Other} \\ \text{Types of} \\ \text{Gaskets} \end{array} \right.$$

NOTATION SYMBOLS AND DEFINITIONS

Except as noted, the symbols and definitions below are those given in Appendix II of the 1977 AMSE Boiler and Pressure Vessel Code, Section VIII.

A_b	= actual total cross-sectional area of bolts at root of thread or section of least diameter under stress, square inches.	m	= gasket factor. Table 1.
A_m	= total required cross-sectional area of bolts, taken as the greater of A_{m1} or A_{m2} , square inches.	N	= width, in inches, used to determine the basic gasket seating width b_0 , based upon the possible contact width of the gasket. Table 2.
A_{m1}	= total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the operating conditions.	P	= design pressure, pounds per square inch.
A_{m2}	= total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket seating.	S_a	= allowable bolt stress at ambient temperature, pounds per square inch.
b	= effective gasket or joint-contact-surface seating width, inches. Table 2	S_b	= allowable bolt stress at operating temperature, pounds per square inch.
b_0	= basic gasket seating width, inches. Table 2.	S_g	= Actual unit load at the gasket bearing surface, pounds per square inch.
G	= diameter at location of gasket load reaction. Table 2.	W_{m1}	= required bolt load for operating conditions, pounds.
		W_{m2}	= minimum required bolt load for gasket seating, pounds.
		y	= gasket or joint-contact-surface unit seating load, minimum design seating stress, PSI Table 1 pounds per square inch.

When $b_0 \leq 1/4$ in., G = mean diameter of gasket contact face, inches.

When $b_0 > 1/4$ in., G = outside diameter of gasket contact face less $2b$, inches.

TABLE 4 MAXIMUM S_g VALUES FOR SPIRAL WOUND GASKETS

Canadian or Blue African Asbestos Filled	25,000 PSI
TFE Filled	13,000 PSI
GRAFOIL Filled	20,000 PSI

NOTE: Suggested Values Based on Successful Design Experience

SAMPLE GASKET CALCULATION PROBLEMS

EXAMPLE A:

An existing heat exchanger is to be utilized in a refinery in a new application more severe than the original design. The engineer requests suggestions for the optimum gasket to be used under the new conditions. His preference is for a spiral wound gasket.

CONDITIONS:

Flange is existing. Major remachining operations are to be avoided if possible. Gasket seating surfaces are in good condition and have a fine tool finish:

Design Conditions: Tube Side: 710 PSI @ 1100°F

Shell Side: 585 PSI @ 975°F

Bolting: 36 — 1 1/4" 8 thds. on 34 1/8" B.C.

Bolt Material: ASTM A193 — B8C

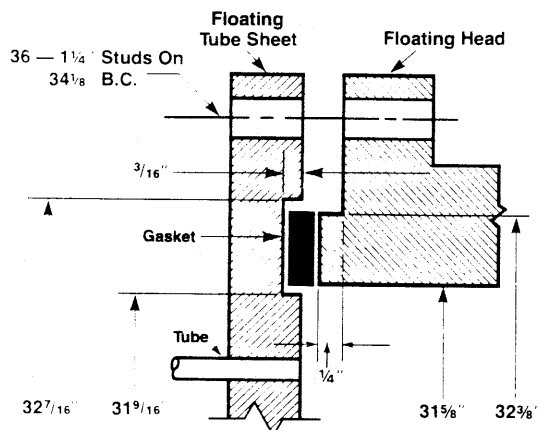
Flange Material: ASTM A312 Type 316 S.S.

Process Material: H₂ containing hydrocarbons

Allowable Bolt Stress @ Operating Temperature — 10,000 PSI (See Stress Table 1, Appendices) (Page 43)

Allowable bolt stress @ Ambient Temperature, according to Stress Table 1, Page 43 is only 15,000 PSI; however, to prevent leakage under hydrotest it is decided to tighten bolting to 20,000 PSI (See Note at bottom of Stress Table 1, Page 43; Appendix S, Page 32; and "Note", Page 29.

Pressure differential across joint: 125 PSI



DETAILS OF FLANGE

Analysis

- From Table 1, Page 4
 $m = 3$ $y = 10,000$ PSI
- In order to insure the spiral wound sealing member is seated against a flat surface, the gasket is sized 31 11/16" I.D. x 32 5/16" O.D. Thickness is selected as 0.175". In order to center the gasket on the tongue, provide spiral wound spacer rings on I.D. or O.D.

From Table 2, Page 5

$$\begin{aligned} N &= \frac{5}{16}'' = 0.3125'' \\ b_0 &= \frac{5}{32}'' = 0.15625'' \\ b &= 0.15625 \\ G &= 32 \end{aligned}$$

- From Formula on page 8

$$\begin{aligned} W_{m2} &= \pi b G y \\ &= 3.14 \times 0.15625'' \times 32'' \times 10,000 \text{ PSI} \\ &= 157,000 \text{ lbs.} \end{aligned}$$

$$\begin{aligned} W_{m1} &= \frac{\pi}{4} G^2 P + 2b\pi G m P \\ &= 0.785 \times (32'')^2 \times 125 \text{ PSI} + 2 \times 0.15625'' \\ &\quad \times 3.14 \times 32'' \times 3 \times 125 \text{ PSI} \\ &= 100,480 + 11,775 \\ &= 112,255 \text{ lbs} \end{aligned}$$

- From Chart in Appendix, page 42 and definition of A_b , Page 8.

$$A_b = 36 \times 0.929 \text{ sq. in.} = 33.444 \text{ sq in.}$$

Hence allowable bolt load:

@ Ambient Temperature:

$$20,000 \text{ PSI} \times 33.444 \text{ sq. in.} = 668,880 \text{ lbs.}$$

@ Design Temperature

$$10,000 \text{ PSI} \times 33.444 \text{ sq. in.} = 334,400 \text{ lbs.}$$

It is apparent adequate bolting is available for gasket seating, i.e. 668,880 lb. vs. 157,000 lbs. required. It is also adequate for design conditions, i.e. 334,440 lbs. vs. 112,255 lbs. required.

- When checking for the maximum unit stress $S_{g(max)}$ that could be exerted on the gasket bearing surface we find:

$$S_{g(max)} = \frac{\text{Maximum Bolt Load in Pounds}}{\text{Actual Gasket Area in Sq. In.}}$$

$$\begin{aligned} S_{g(max)} &= \frac{A_b S_a}{\frac{\pi}{4} [(OD - 0.125)^2 - (ID)^2]} \\ &= \frac{33.444 \text{ Sq. In.} \times 20,000 \text{ PSI}}{0.785 [(32.3125'' - 0.125'')^2 - (31.6875'')^2]} \\ &= \frac{668,880 \text{ lbs.}}{25.07 \text{ Sq. In.}} \\ &= 26,680 \text{ PSI} \end{aligned}$$

Experience indicates that $S_{g(max)}$ should not exceed 25,000 PSI. (See table page 9); therefore a positive stop must be provided either with a gauge ring or by altering the flange facing to provide limited compression.

There is insufficient space to supply a compression gauge ring and the customer does not want to remachine the flanges. As a consequence, the use of a spiral wound gasket is not recommended for this application and another type gasket should be used.

- The pressure-temperature conditions dictate the use of a metallic type gasket. A double-jacketed or double-jacketed corrugated gasket would be a good choice. Since the temperature is to be 1100°F a double-jacketed corrugated gasket with a corrugated filler is recommended (Lamons Style 340).

7. Standard practice is to size this type gasket equal to groove O.D. minus $\frac{1}{16}$ ", groove I.D. plus $\frac{1}{16}$ ". Good practice also indicates the metal should be 316 S.S. (same as flange material). Increasing the gasket area will reduce the load on the gasket slightly. Excessive loads on a double-jacketed corrugated gasket is not a problem.

The gasket recommendation would be:

Lamons Style 340 Gasket — 316 S.S. with 316 S.S. Insert.

$32\frac{3}{8} \times 31\frac{5}{8} \times \frac{1}{8}$

EXAMPLE B:

A designer wants a gasket recommendation for a special application sealing acid sludge at 600 psi and 500°F.

CONDITIONS:

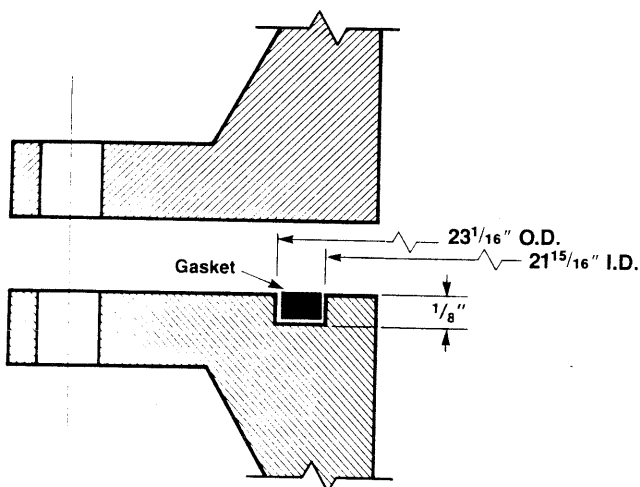
Design pressure — 600 psi

Test pressure — 900 psi

Design temperature — 500°F

Process material — Acid sludge

Flange details —



Details of Flange

Bolting — 24 — $\frac{1}{8}$ " — 8 thds.

Bolt Material — ASTM A193 — B7

Flange Material — ASTM A312 Type 316 S.S.

Allowable bolt stress @ Ambient Temperature, according to Stress Table 1, Page 43 is only 20,000 PSI; however, to prevent leakage under hydrotest it is decided to tighten bolting to 30,000 PSI (See Note at bottom of Stress Table 1, Page 43; Appendix S, Page 32; and "Note", Page 29.

Allowable "Stress @ 500°F — 20,000 PSI (see Stress Table 1 Appendices Page 43)

Analysis

The pressure-temperature conditions indicate a metallic type gasket should be used. The conditions appear to be suitable for a spiral wound gasket. The exact nature of the acid sludge is not known but since the flange material is 316 S.S., the logical choice for the metal in the gasket is 316 S.S. also. Since Blue African

asbestos has better acid resistance than white asbestos (chrysotile), the filler material is selected as Blue African (crocidolite).

- From Table 1, Page 4

$$m = 3$$

$$y = 10,000$$

- From page 24, "Sizing Spiral Wound Gaskets Confined on I.D. and O.D.", gaskets should have I.D. of 22" and O.D. of 23". Since facing is groove to flat face, gasket thickness must be $\frac{1}{8}$ ".

From Table 2, Page 5

$$N = \frac{1}{2} = 0.500"$$

$$b_o = 0.250"$$

$$b = 0.250"$$

$$G = 22.5"$$

- From formula on page 8.

$$\begin{aligned} W_{m2} &= \pi b G y \\ &= 3.14 \times 0.250" \times 22.5" \times 10,000 \text{ PSI} \\ &= 176,625 \text{ lbs.} \end{aligned}$$

$$W_{m1} = \frac{\pi}{4} G^2 P + 2b\pi G m P$$

$$\begin{aligned} W_{m1} (\text{Design}) &= 0.785 \times (22.5")^2 \times 600 \text{ PSI} + \\ &\quad 2 \times 0.250" \times 3.14 \times 22.5" \times 3 \\ &\quad \times 600 \text{ PSI} \\ &= 238,444 + 63,585 \\ &= 302,029 \text{ lbs.} \end{aligned}$$

$$\begin{aligned} W_{m1} (\text{Test}) &= 0.785 \times (22.5")^2 \times 900 \text{ PSI} + 2 \times \\ &\quad 0.250" \times 3.14 \times 22.5" \times 3 \times 900 \\ &\quad \text{PSI} \\ &= 357,666 + 95,378 \\ &= 453,043 \text{ lbs.} \end{aligned}$$

From Table on Page 42 and definition of A_b , page 8

$$A_b = 24 \times 0.728 = 17.472 \text{ sq. in.}$$

$$\text{Bolt load @ Test Condition: } 30,000 \times 17.472 = 524,160 \text{ lbs.}$$

$$\text{Bolt Load @ Design Condition: } 20,000 \times 17.472 \times 349,440 \text{ lbs.}$$

It is apparent adequate bolting is available to insure a tight joint. Minimum required bolt loading for gasket seating (W_{m2}) is 176,625 lbs. Available load for gasket seating is 524,160 lbs.

Minimum required bolt at design conditions is 302,029 lbs. and available load at design conditions is 349,440 lbs.

Note: required bolt load at test conditions is 453,043 lbs. and available bolt load at test conditions is 524,160 lbs.

- In order to check the maximum unit stress, $S_{g(\max)}$ at the gasket bearing surface solve the equation:

$$\begin{aligned} S_{g(\max)} &= \frac{A_b S_a}{\frac{\pi}{4} [(OD - 0.125)^2 - (ID)^2]} \\ &= \frac{17.472 \text{ Sq. In.} \times 30,000 \text{ PSI}}{0.785 [(23" - 0.125")^2 - (22")^2]} \\ &= \frac{524,160 \text{ lbs.}}{30.824 \text{ sq. in.}} \\ &= 17,005 \text{ PSI} \end{aligned}$$

The resulting unit load acting on the gasket at bolt up conditions is within the range for a spiral wound gasket. Since a positive stop is designed into the flange, i.e. groove to flat, no additional precautions are necessary. Any forces in excess of the force required to compress the gasket will be transmitted to the flange faces and gasket crushing cannot occur.

6. From the above analysis, it appears our original assumption is correct and the recommendation would be:

SpiraSeal Type W Gasket — 316 S.S./Blue
African Asbestos
22" ID × 23" OD × 0.175" Thick

In the design of spiral wound gaskets, as is pointed out in Example "B" paragraph 5 above, it is important

that the gasket must be able to carry the maximum compressive force that may be applied by the bolts unless provisions are made to use a compression stop in the flange design or a compression gauge ring on the gasket. If the value determined for $S_g(\max)$ exceeds those shown in Table 4 either a gasket with a wider flange width or a compression stop must be used.

*NOTE: Selection of gasket material and determination of facing type and dimensions is largely a matter of sound engineering judgment. Values of the four factors **N**, **b**, **y**, and **m**, given in these tables are identical with those appearing in the ASME Boiler and Pressure Vessel Code. They are not mandatory but represent suggested values and are based upon the best cross-section of successful design experience currently available. The designer is at liberty to use other factors should data to substantiate them be available.*

NOTE: On pages 10 and 11, the term "pressure-temperature conditions" was used indicating that this value determines the types of material and construction to be used in a gasket.

A "Rule-of-Thumb" guide for the selection of gasket materials has evolved over the years. This value is arrived at by multiplying operating pressure times operating temperature.

Temperature limits of gasketing materials are not absolute figures. Materials within any category may vary depending upon a manufacturer's processing techniques, grades and types of raw materials used, etc.

In addition, flange design and application peculiarities may influence the temperature limit of a material to a greater or lesser degree and, in certain instances, allow use of non-metallic gasketing material in excess of the maximum temperature limits above.

MATERIAL	MAXIMUM P × T	MAXIMUM TEMPERATURE OF MATERIALS, °F
Rubber	15,000	250
Vegetable Fiber	40,000	250
Solid Fluorocarbon	75,000	500
Rubberized Asbestos Cloth	125,000	500-600
Compressed Asbestos	250,000	650-1000
Metal Types	Over 250,000	Depends on Metal

SECTION II — SELECTION

SELECTING THE PROPER GASKET MATERIAL

The optimum gasket material would have the following characteristics. It would have the chemical resistance of PTFE, the heat resistance of graphite, the strength of steel, require a zero seating stress such as soft rubber and be inexpensive. Obviously there is no known gasket material that has all these characteristics and each material has certain limitations that restrict its use. It is possible to overcome limitations partially by several methods such as including the use of reinforcing inserts, combining it with other materials, varying the construction or density, or by designing the joint itself to overcome some of the limitations. Obviously, mechanical factors are important in the design of the joint but the primary selection of a gasket material is influenced by three factors,

- the temperature of the fluid or gas to be contained,
- the pressure of the fluid or gas to be contained,
- the corrosive characteristics of the fluid or gas to be contained.

Charts included in the appendix indicate some very general recommendations for non-metallic and metallic materials against various corrosive media. It should be pointed out that these charts are general recommendations and there are many additional factors that

can influence the corrosion resistance of a particular material at operating conditions. Some of these would include

- Concentration of the corrosive agent. (Full strength solutions are not necessarily more corrosive than those of dilute proportions and, of course, the reverse is also true.)
- The purity of a corrosive agent. For example, dissolved oxygen in otherwise pure water may cause rapid oxidation of steam generation equipment at high temperatures.
- The temperature of the corrosive agent. In general, higher temperatures of corrosive agents will accelerate corrosive attack.

As a consequence, it is often necessary to "field-test" materials for resistance to corrosion under normal operating conditions to determine if the material selected will have the required resistance to corrosion.

TYPES OF GASKETS

For the purposes of this bulletin, gaskets will be separated into two broad categories, non-metallic and metallic gaskets.

Of the two types, non-metallic gaskets are by far the most widely used. This discussion will cover the various types of non-metallic materials, general application data and temperature limitations.

NON-METALLIC GASKET MATERIALS

NATURAL RUBBER

Natural rubber has good resistance to mild acids and alkalis, salts and chlorine solutions. It has poor resistance to oils and solvents and is not recommended for use with ozone. Its temperature range is very limited and is suitable only for use from -70°F to 200°F.

SBR (STYRENE-BUTADIENE)

SBR is a synthetic rubber that has excellent abrasion resistance and has good resistance to weak organic acids, alcohols, moderate chemicals and ketones. It is not good in ozone, strong acids, fats, oils, greases and most hydrocarbons. Its temperature limitations are approximately -65°F to 250°F.

CR (CHLOROPRENE) (NEOPRENE)

Chloroprene is a synthetic rubber that is suitable for use against moderate acids, alkalis and salt solutions. It has good resistance to commercial oils and fuels. It is very poor against strong oxidizing acids, aromatic and chlorinated hydrocarbons. Its temperature range would be from approximately -60°F to 250°F.

BUNA-N RUBBER (NITRILE, NBR)

Buna-N is a synthetic rubber that has good resistance to oils and solvents, aromatic and aliphatic hydrocarbons, petroleum oils and gasoline over a wide range of temperature. It also has good resistance to caustics and salts but only fair acid resistance. It is poor in strong oxidizing agents, chlorinated hydrocarbons, ketones and esters. It is suitable over a temperature range of approximately -60°F to 250°F.

FLUOROCARBON (VITON)

Fluorocarbon elastomer has good resistance to oils, fuel, chlorinated solvents, aliphatic and aromatic hydro-

carbons and strong acids. It is not suitable for use against amines, esters, ketones or steam. Its normal temperature range would be between -40°F and 450°F.

CHLOROSULFONATED POLYETHELENE (HYPALON)

This material has good acid, alkali and salt resistance. It resists weathering, sunlight, ozone, oils and commercial fuels such as diesel and kerosene. It is not good in aromatics or chlorinated hydrocarbons and has poor resistance against chromic acid and nitric acid. Its normal temperature range would be between -50°F and 275°F.

SILICONES

Silicone rubbers have good resistance to hot air. They are unaffected by sunlight and ozone. They are not, however, suitable for use against steam, aliphatic and aromatic hydrocarbons. The temperature range would be between -150°F to 500°F.

EPDM (ETHYLENE PROPYLENE), MONOMER

This synthetic material has good resistance to strong acids, alkalis, salts and chlorine solutions. It is not suitable for use in oils, solvents or aromatic hydrocarbons. Its temperature range would be between -70°F and 350°F.

WOVEN ASBESTOS CLOTH

This material consists of woven long fiber white asbestos yarn usually reinforced with brass wire and impregnated with either rubber or TFE to minimize porosity. This type of material was developed primarily to handle applications where pitted, warped or rough flanges require a pliable material to adequately seal. It is

used against steam, gases and aqueous solutions. Its temperature range is usually limited to 600°F and it is not normally used in pressure applications over 300 psi.

GRAFOIL

This is an all graphite material containing no resins or inorganic fillers. It is available with or without a metal insertion, and in adhesive-back tape form for pipe gaskets over 24 inches in diameter. Grafoil has outstanding resistance to corrosion against a wide variety of acids, alkalis and salt solutions, organic compounds, and heat transfer fluids, even at high temperatures. It does not melt, but does sublime at temperatures over 6000°F. Its use against strong oxidizing agents at elevated temperatures should be investigated very carefully. In addition to being used as a gasket, Grafoil makes an excellent packing material and is also used as a filler material in spiral-wound gaskets.

CERAMIC FIBER

Ceramic fiber is available in sheet or blanket form and makes an excellent gasket material for hot air duct work with low pressures and light flanges. It is satisfactory for service up to approximately 2000°F. Ceramic material is also used as a filler material in spiral-wound gaskets.

PLASTICS

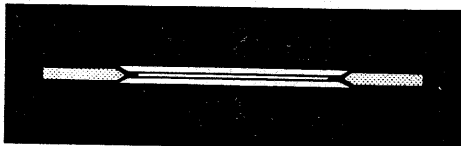
Of all the plastics, PTFE (polytetrafluoroethylene) has emerged as the most common plastic gasket material. PTFE's outstanding properties include resistance to temperature extremes from -423°F to 500°F. PTFE is highly resistant to chemicals, solvents, caustics and acids except free fluorine and alkali metals. It has a very low surface energy and does not adhere to the flanges. PTFE gaskets can be supplied in a variety of forms either as virgin material or reprocessed material and also with a variety of filler material such as glass, carbon, molybdenum disulfide, etc. The principal advantage in adding fillers to PTFE is to inhibit cold flow or creep relaxation which is a problem with the virgin sheet.

PTFE ENVELOPE GASKETS

Envelope gaskets utilizing a PTFE jacket have become popular for use in severely corrosive services because of their low minimum seating stresses, excellent creep resistance, high deformability and choice of a variety of filler materials to assure optimum performance on any specific application.

Fillers such as compressed asbestos, woven asbestos cloth, corrugated metal and rubber sheets are available. The sandwich constructions are becoming much more popular and if the corrugated metal insert is used with a correct asbestos sheet on each side it is possible to use these gaskets successfully at very high pressures approaching 5000 psi.

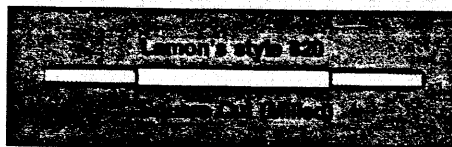
There are three basic designs of envelopes:



Slit Type

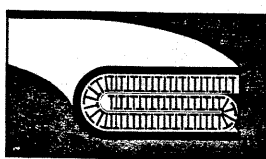
Slit envelopes are sliced from cylinders and split from the outside diameter to within approximately 1/16" of the inside diameter. The bearing surface is determined by

the filler dimensions. Clearance is required between the I.D. of the filler and the envelope I.D. The Gasket O.D. normally rests within the bolt hole circle and the I.D. is approximately equal to the nominal I.D. of pipe. Available in sizes to a maximum O.D. of 24".



Milled Type

Milled envelopes are machined from cylinder stock. The jacket is machined from the O.D. to within approximately 1/32" its I.D. The jacket's I.D. fits flush with pipe bore and its O.D. nests within the bolts. Available in sizes up to a maximum O.D. of 24". Milled envelopes are more expensive than slit type since considerably more material is lost in machining.



Formed Tape Type

Large diameter (over 12" N.P.S.) and irregularly shaped envelopes are formed from tape and heat sealed to produce a continuous jacket construction.

Filler Materials

The more popular fillers for envelope gaskets are:

- Compressed asbestos sheet
- Rubber sheet
- Compressed non-asbestos
- Corrugated metal inserts
- Felted asbestos sheet
- Sandwich constructions combining some of the above

The sandwich constructions, usually with layers of felted blue asbestos on each side of either a compressed asbestos or a corrugated metal insert, are widely used on glass lined piping and equipment.

On vacuum applications, double envelopes are frequently used where two jackets are overlapped to protect the O.D. as well as the I.D. They can be slit, milled or formed tape types.

COMPRESSED ASBESTOS SHEETING

Compressed asbestos sheeting is widely used in the soft gasketing field for industrial applications. It is available with a variety of elastomeric binders, the most common of which are SBR, CR, NBR and EPDM. Compressed asbestos sheet is available with either white (chrysotile) or blue African (crocidolite) asbestos. The asbestos and binders are bonded under pressure and vulcanized into a homogeneous sheet. The white asbestos would be the general service sheet and the blue African asbestos sheet is normally used for acid service. The strength of compressed asbestos sheet is basically controlled by the fiber length. In general, the longer the fiber length, the higher the tensile strength of the sheet. Compressed asbestos sheets contain ten to twenty percent of elastomeric binder which determines the suitability of any particular sheet for the intended service. White asbestos sheet with an SBR binder is normally used against steam, air, gas, oil, water, mild acids and chemicals. CR (neoprene) or NBR (nitrile)

binder sheets are usually recommended for gasolines, petroleum solvents, vegetable oils, refrigerants, etc. Severe acids or acid salt solutions may require blue African asbestos fiber with an acid resistant binder. White asbestos sheet with EPDM binder will have good acid resistant properties. Some of the principal advantages of compressed asbestos sheet are:

- a nonporous sheet strongly resistant to plastic flow under high bolt loads;
- suitable for high temperatures in the range of 700 to 1000°F depending on the particular formulation;
- low stress relaxation qualities;
- available in a variety of thicknesses from 1/64" to 1/4". Some manufacturers offer sheet sizes up to 150" x 150".

COMPRESSED NON-ASBESTOS SHEETING

Early efforts to replace asbestos resulted in the introduction and testing of compressed non-asbestos products in the 1970's. Many of these products have seen extensive use since that period however there have been enough problems to warrant careful consideration in choosing a replacement material for compressed asbestos. Most manufacturers of non-asbestos sheet materials use synthetic fibers, like Kevlar®, in conjunction with an elastomeric binder. The elastomeric binder makes up a larger percentage of this sheet and thereby

becomes a more important consideration when determining applications. The same binders offered in compressed asbestos sheeting are used for non-asbestos. Compressed non-asbestos materials usually have lower temperature limits than comparable asbestos products. Sheet sizes and thickness's are simily to compressed asbestos products.

® Kevlar is a registered trademark of E.I. DuPont Co.

ASBESTOS MILLBOARD

Asbestos millboard is made up relatively pure asbestos fibers bound with starch or sulphites. In gasket manufacture it is used as a soft filler for metal jacketed designs. Asbestos millboard is also used as a gasket material in low pressure, high temperature applications. The upper temperature limitation is normally 900°F.

VEGETABLE FIBER SHEET

Vegetable fiber sheet is a tough pliable gasket material manufactured by paper making techniques utilizing plant fibers and a glue-glycerine impregnation. It is widely used for sealing petroleum products, gases and a wide variety of solvents. Its maximum temperature limit is 250°F. If a more compressible material is required, a combination cork-fiber sheet is available. The cork-fiber sheet has the same maximum temperature limitation as the vegetable fiber sheet.

METALLIC GASKET MATERIALS

CARBON STEEL

Commercial quality sheet steel with an upper temperature limit of approximately 1000°F., particularly if conditions are oxidizing. Not suitable for handling crude acids or aqueous solutions of salts in the neutral or acid range. A high rate of failure may be expected in hot water service if the material is highly stressed. Concentrated acids and most alkalies have little or no action on iron and steel gaskets which are used regularly for such services. Brinell hardness is approximately 120.

304 STAINLESS STEEL

An 18-8 (Chromium 18-20%, Nickel 8-10%) Stainless with a maximum recommended working temperature of 1400°F. At least 80% of applications for non-corrosive services can use Type 304 Stainless in the temperature range of - 320°F. to 1000°F. Excellent corrosion resistance to a wide variety of chemicals. Subject to stress corrosion cracking and to intergranular corrosion at temperatures between 800°F. to 1500°F. in presence of certain media for prolonged periods of time. Brinell hardness is approximately 160.

304L STAINLESS STEEL

Carbon content maintained at a maximum of .03% Recommended maximum working temperature of 1400°F F. Same excellent corrosion resistance as Type 304. This low carbon content tends to reduce the precipitation of carbides along grain boundaries. Less subject to intergranular corrosion than Type 304. Brinell hardness is about 140

316 STAINLESS STEEL

An 18-12 Chromium-Nickel steel with approximately 2% of Molybdenum added to the straight 18-8 alloy which increases its strength at elevated temperatures and results in somewhat improved corrosion resistance. Has the highest creep strength at elevated temperatures of any conventional stainless type. Not suitable for extended service within the carbide precipitation range

of 800° to 1650°F. when corrosive conditions are severe. Recommended maximum working temperature of 1400° F. Brinell hardness is approximately 160.

316-L STAINLESS STEEL

Continuous maximum temperature range of 1400°-1500°F. Carbon content held at a maximum of .03%. Subject to a lesser degree of stress corrosion cracking and also to intergranular corrosion than Type 304. Brinell hardness is about 140.

321 STAINLESS STEEL

An 18-10 Chromium-Nickel steel with a Titanium addition. Type 321 stainless has the same characteristics as Type 347. The recommended working temperature is 1400° to 1500°F. and in some instances 1600°F. Brinell hardness is about 150.

347 STAINLESS STEEL

An 18-10 Chromium-Nickel steel with the addition of Columbium. Not as subject to intergranular corrosion as is Type 304. Is subject to stress corrosion. Recommended working temperature of 1400°-1500°F. and in some instances to 1700°F. Brinell hardness is approximately 160.

410 STAINLESS STEEL

A 12% Chromium-9% Nickel steel with a maximum temperature range of 1200°F. to 1300°F. Used for applications requiring good resistance to scaling at elevated temperatures. Is not recommended for use where severe corrosion is encountered but is still very useful for some chemical applications. May be used where dampness, alone or coupled with chemical pollution, causes steel to fail quickly. Brinell hardness is around 155.

502/501

4-6% Chromium and 1/2 Molybdenum alloyed for mild corrosive resistance and elevated service. Maximum working temperature is 1200°F. and has a Brinell hardness of around 130. If severe corrosion is anticipated, a better grade of stainless steel would probably be a better choice. Becomes extremely hard when welded.

ADMIRALTY

Arsenical Admiralty 443 has 71% Copper, 28% Zinc, 1% Tin and trace amounts of Arsenic. High corrosive resistance, holds up extremely well against salt and brackish waters, and water containing sulfides. Recommended maximum working temperature of 500° F. Ideal for carrying corrosive cooling waters at relatively high temperatures. Brinell hardness is about 64.

ALLOY 20

45% Iron, 24% Nickel, 20% Chromium, and small amounts of Molybdenum and Copper. Maximum temperature range of 1400°-1500° F. Developed specifically for applications requiring resistance to corrosion by sulphuric acid. Brinell hardness is about 160.

ALUMINUM

Alloy 1100 is commercially pure (99% minimum). Its excellent corrosion resistance and workability makes it ideal for double jacketed gaskets. The Brinell hardness is approximately 35. For solid gaskets, stronger alloys like 5052 and 3003 are used. Maximum continuous service temperature of 800° F.

BRASS

Yellow brass 268 has 66% Copper and 34% Zinc. Offers excellent to good corrosion resistance in most environments, but is not suitable for such materials as acetic acid, acetylene, ammonia, and salt. Maximum recommended temperature limit of 500° F. Brinell hardness is 58.

COPPER

Nearly pure copper with trace amounts of silver added to increase its working temperature. Recommended maximum continuous working temperature of 500° F. Brinell hardness is about 80.

CUPRO NICKEL

Contains 69% Copper, 30% Nickel, and small amounts of Manganese and Iron. Designed to handle high stresses, it finds its greatest application in areas where high temperatures and pressures combined with high velocity and destructive turbulence would rapidly deteriorate many less resistant alloys. Maximum recommended temperature limit of 500° F. Brinell hardness is about 70.

HASTELLOY B[®]

26-30% Molybdenum, 62% Nickel, and 4-6% Iron. Maximum temperature range of 2000° F. Resistant to hot, concentrated hydrochloric acid. Also resists the corrosive effects of wet hydrogen chlorine gas, sulphuric and phosphoric acids and reducing salt solutions. Useful for high temperature strength. Brinell hardness is approximately 230.

HASTELLOY C-276[®]

16-18% Molybdenum, 13-17.5% Chromium, 3.7-5.3% Tungsten, 4.5-7% Iron, and the balance is Nickel. Maximum temperature range of 2000° F. Very good in handling corrosives. High resistance to cold nitric acid of

varying concentrations as well as boiling nitric acid up to 70% concentration. Good resistance to hydrochloric acid and sulphuric acid. Excellent resistance to stress corrosion cracking. Brinell hardness is about 210.

INCONEL 600

Recommended working temperatures of 2000° F. and in some instances 2150° F. Is a nickel base alloy containing 77% Nickel, 15% Chromium and 7% Iron. Excellent high temperature strength. Frequently used to overcome the problem of stress corrosion. Has excellent mechanical properties at the cryogenic temperature range. Brinell hardness is about 150.

INCOLOY 800

32.5% Nickel, 46% Iron, 21% Chromium. Resistant to elevated temperatures, oxidation, and carburization. Recommended maximum temperature of 1600° F. Brinell hardness is about 150.

MONEL[®]

Maximum temperature range of 1500° F. Contains 67% Nickel and 30% Copper. Excellent resistance to most acids and alkalis, except strong oxidizing acids. Subject to stress corrosion cracking when exposed to fluorosilic acid, mercuric chloride and mercury, and should not be used with these media. With PTFE (Polytetrafluoroethylene), it is widely used for hydrofluoric acid service. Brinell hardness is about 120.

NICKEL 200[®]

Recommended maximum working temperature is 1400° F. and even higher under controlled conditions. Corrosion resistance makes it useful in caustic alkalis and where resistance in structural applications to corrosion is a prime consideration. Does not have the all-around excellent resistance of Monel. Brinell hardness is about 110.

PHOSPHOR BRONZE

90-95% Copper, 5-10% Tin, and trace amounts of phosphorus. Maximum temperature range of 500° F. Excellent cold working capacity. Limited to low temperature steam applications. Excellent corrosion resistance, but not suitable for acetylene, ammonia, chromic acid, mercury, and potassium cyanide. Brinell hardness is approximately 65.

TITANIUM

Maximum temperature range of 2000° F. Excellent corrosion resistance even at high temperatures. Known as the "Best solution" to chloride ion attack. Resistant to nitric acid in a wide range of temperatures and concentrations. Most alkaline solutions have little if any effect upon it. Outstanding in oxidizing environments. Brinell hardness is about 215.

Note

Maximum temperature ratings are based upon hot air constant temperatures. The presence of contaminating fluids and cyclic conditions may drastically affect the maximum temperature range.

MATERIAL HARDNESS CONVERSION SCALE

Rockwell "B"	Brinell 3000 Kg. Load
100	241
95	210
90	183
85	163
80	146
75	134
70	122
65	108
60	95
55	89
50	83
40	75
30	67
20	62
10	57

Brinell hardness figures are approximate guides only. Most materials ordered by Lamons are specified "dead soft"; however, different thicknesses and different heats of the same material will vary in hardness.

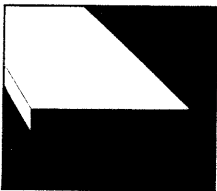
METAL GASKETS

Metallic gaskets are available in many forms including,

- solid metal gaskets that require very smooth, plain surface finishes and high clamping forces in order to seal,
- combinations with soft fillers such as double-jacketed and spiral-wound that can tolerate greater surface roughness and will seat with lesser compressive forces, and
- light cross section gaskets that are self-sealing and require minimum clamping forces for effective sealing.

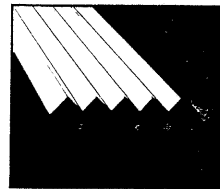
In all cases, however, careful attention must be given to machining details of the flanges and sizing of the gaskets.

SOLID METAL GASKETS



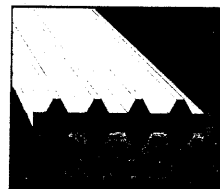
**PLAIN FLAT
METAL GASKETS**

Flat metal gaskets are best suited for applications such as valve bonnets, ammonia fittings, heat exchangers, hydraulic presses, tongue-and-groove joints. They can be used when compressibility is not required to compensate for flange surface finish, warpage or misalignment and where sufficient clamping force is available to seat the particular metal selected. They must be sealed by the flow of the gasket metal into the imperfections on the gasket seating surfaces of the flange. This requires heavy compressive forces and the load must exceed the tensile strength of the material. The hardness of gasket metal must be less than the hardness of the flanges to prevent damage to the gasket seating surface of the flange. Flat metal gaskets are relatively inexpensive to produce and can be made of virtually any material that is available in sheet form. Size limitation is normally restricted to the sheet size. Larger gaskets can be fabricated by welding.



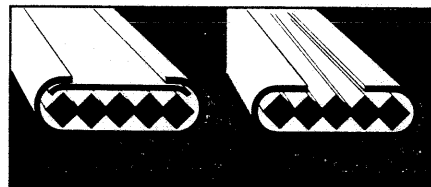
**PROFILE
GASKETS**

Profile type gaskets offer the desirable qualities of plain washer types and the added advantage of a reduced contact area provided by the V-shaped surface. It is used when a solid metal gasket is required because of pressure or temperature or because of the highly corrosive effect of the fluid to be contained and also when bolting is not sufficient to seat a flat washer.



**SERRATED
GASKETS**

Serrated gaskets are made of thick, solid metal machined with concentric rings. This type gasket is excellent for installations requiring a solid metal gasket with a reduced contact area. As is the case with a profile gasket, serrated gaskets are used when bolting is insufficient to flow or seat a flat washer type gasket.

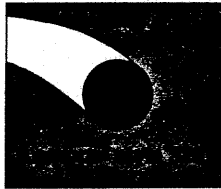


**A PROFILE
GASKET
WITH A
METAL
JACKET**

If flange conditions require a profile type gasket, but flange protection is required as well, the profile gasket may be supplied with either a single-jacketed or a double-jacketed shield. This will provide protection for the flanges and will minimize damage to the flange faces due to the profile surface.

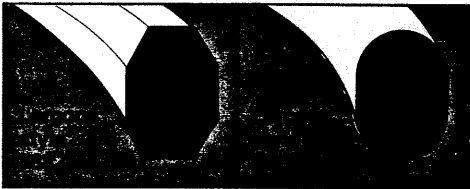
NOTE: Without exception all of the solid metal gaskets require a very fine surface finish on the flanges. A flange

with a flange surface roughness of 63 microinches or smoother is desired. Under no circumstances should the surface finish exceed 125 microinches. In addition, radial gouges or scores would be almost impossible to seal using solid metal gaskets.



**ROUND CROSS SECTION,
SOLID METAL
GASKETS**

Round cross section solid metal gaskets are used on specifically designed flanges grooved or otherwise faced to accurately locate the gasket during assembly. These gaskets seal by a line contact which provides an initial high seating stress at low bolt loads. This makes an ideal gasket for low pressures. The more common materials used for this type of gasket would be aluminum, copper, soft iron or steel, Monel®, nickel, and 300 series stainless steels. They are fabricated from wire formed to size and welded. The weld is then polished to the exact wire diameter.



**API RING
JOINT
GASKETS**

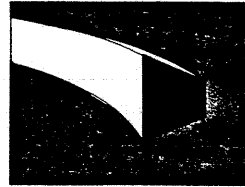
API ring joint gaskets come in two basic types, an oval cross section and an octagonal cross section. These basic shapes are used in pressures up to 10,000 psi. The dimensions are standardized and require specially grooved flanges. The octagonal cross section has a higher sealing efficiency than the oval and would be the preferred gasket. However, only the oval cross section can be used in the old type round bottom groove. The newer flat bottom groove design will accept either the oval or the octagonal cross section. The sealing surfaces on the ring joint grooves must be smoothly finished to 63 microinches and be free of objectionable ridges, tool or chatter marks. They seal by an initial line contact or a wedging action as the compressive forces are applied. The hardness of the ring should always be less than the hardness of the flanges. Dimensions for API ring joint gaskets and grooves are covered in ANSI B16.20 and API 6A.



**BX AND RX
RING
GASKETS**

The BX ring gasket differs from the standard oval or octagonal shape in that it is square in cross section and tapers in each corner. They can only be used in API 6BX flanges. RX ring gaskets are similar in shape to the standard octagonal ring joint gasket but their cross section is designed to take advantage of the contained fluid pressure in effecting a seal. They are both made to API 6A and interchangeable with standard octagonal rings for oil field drilling and production applications in API 6B

flanges. RX and BX gaskets are used at pressures up to 15,000 psi.



**LENS TYPE
GASKET**

A lens type gasket is a line contact seal for use in high pressure piping systems and in pressure vessel heads. The lens cross section is a spherical gasket surface and requires special machining on the flanges. These gaskets will seat with a small bolt load since the contact area is very small and gasket seating pressures are very high. Normally the gasket material should be softer than the flange. In ordering lens gaskets, complete drawings and material specifications must be supplied.



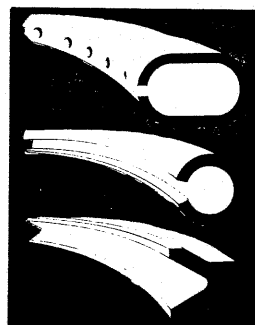
**DELTA
GASKET**

A delta gasket is a pressure actuated gasket used primarily on pressure vessels and valve bonnets at very high pressures in excess of 5000 psi. As with the lens gasket, complete drawings and material specifications must be supplied. Internal pressure forces the gasket material to expand when the pressure forces tend to separate the flanges. Extremely smooth surface finishes of 63 microinches or smoother are required when using this type of gasket.



**BRIDGEMAN
GASKET**

The Bridgeman gasket is a pressure activated gasket for use on pressure vessel heads and valve bonnets for pressures of 1500 psi and above. The cross section of the gasket is such that internal pressure acting against the ring forces it against the containing surface making a self-energized seal. Bridgeman gaskets are frequently silver plated or lead plated to provide a softer surface and minimize the force required to flow the gasket metal into the flange surface.



**MISCELLANEOUS
METAL
GASKETS**

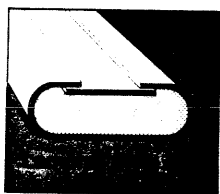
In addition to the commonly used, above-listed gaskets, there are specialty items available that, in specific applications, can provide a very effective seal. These

miscellaneous gaskets would include hollow metal O-rings, C-seals and V-seals, so-called because their cross section is essentially the same as the letters C & V. The hollow metal O-rings are available vented for high pressure applications and pressure filled for high temperature applications. They can be obtained with various platings in order to enhance their sealing abilities and to meet specific applications requirements. C-seals can be used either for vacuum applications or for high pressure applications. C-seals are self-energized gaskets requiring specific attention be paid to the design of the grooves to contain the gasket, and smooth surface finishes are a must. For large quantity applications, the C-seal can be a relatively low cost gasket. For small quantity applications, the cost can be rather high because of initial tooling requirements. V-seals are similar to the C-seal except for the fact that they are essentially machine components which makes the cost of the individual gasket rather high. They also require very fine surface finishes and specially designed grooves to effectively seal. All these specialty items do require initial consultation with the manufacturer in order to determine the practicability and the economies involved.

METAL JACKETED GASKETS

CONSTRUCTION OF JACKETED GASKETS

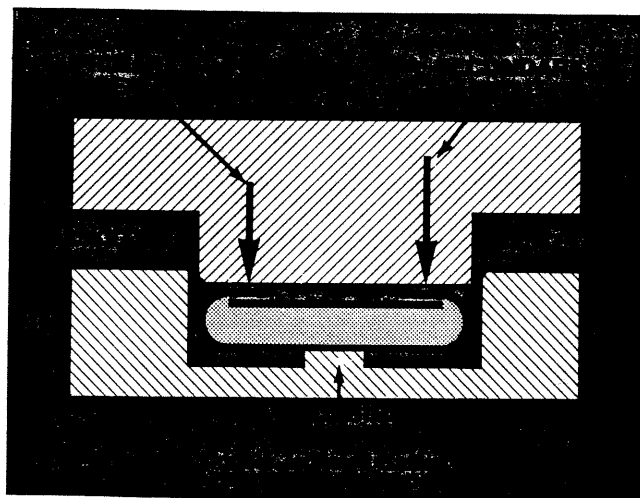
Lamons jacketed gaskets are normally supplied with a non-asbestos high temperature filter or asbestos millboard. The standard filler is normally sufficient for applications up to 1000°F. Other soft fillers are available for higher temperatures or special applications including Grafoil®. Standard metals used to make jacketed gaskets, regardless of the type, are aluminum, copper, the various brasses, soft steel, nickel, Monel®, Inconel® and stainless steel types 304, 316, 321, 347, 410, 502. Obviously the choice of the metal used for the jacketed part of the gasket would depend upon the corrosive conditions being encountered.



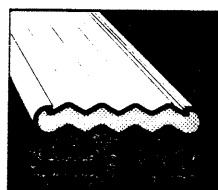
DOUBLE-JACKETED GASKET

Double-jacketed gaskets are probably the most commonly used style of gasket in heat exchanger applications. They are available in virtually any material that is commercially available in 26-gauge sheet. They are also extensively used in standard flanges where the service is not critical and at temperatures beyond which a soft gasket such as asbestos or rubber can be used. Since most double-jacketed gaskets are custom made, there is virtually no limit to the size, shape or configuration in which these gaskets can be made. This particular type of gasket is very versatile and can be used in a myriad of applications. Since the size and shape are not a problem and since most materials can be obtained

commercially, this particular gasket style is very popular. It must be remembered that the primary seal against Leakage, using a double-jacketed gasket, is the metal inner lap where the gasket is thickest before being compressed and densest when compressed. This particular section flows, effecting the seal. As a consequence the entire inner lap must be under compression. Frequently the outer lap is not under compression and does not aid in the sealing of the gasket. On most heat exchanger applications the outer lap is also under compression, providing a secondary seal. The intermediate part of a double-jacketed gasket does very little to effect the sealing capability of the gasket. In some cases nubbins are provided on heat exchanger designs to provide an intermediate seal. This nubbin is normally 1/64" high by 1/8" wide. Experience has indicated, however, that there is little advantage to this particular design. The primary seal is still dependent on the inner lap of the gasket doing the brute work and the secondary seal, when applicable, would be provided by the outer lap.



Always install double jacketed gasket with smooth side toward the nubbin.

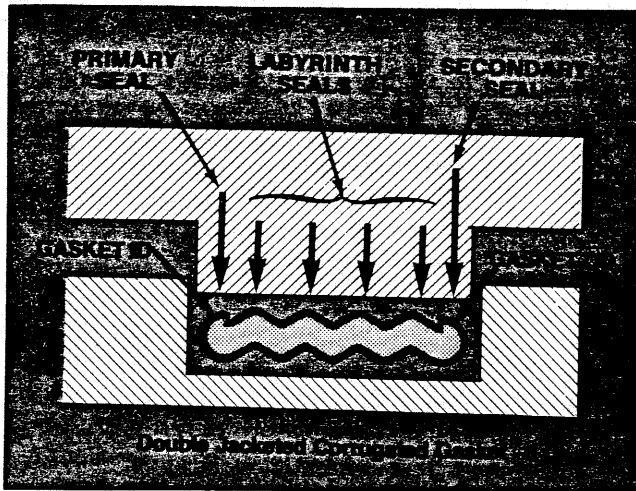


DOUBLE-JACKETED CORRUGATED GASKETS

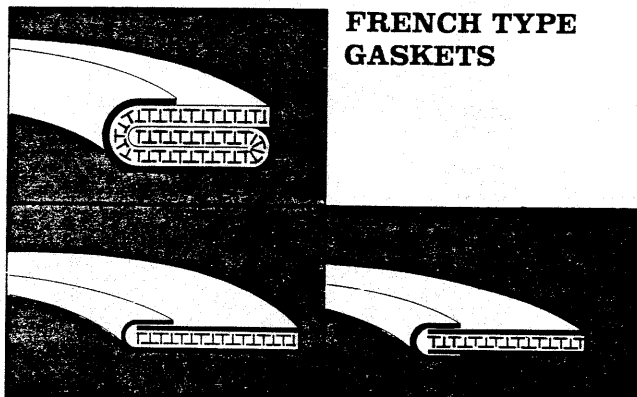
The double-jacketed corrugated gasket is an improvement on a plain jacketed gasket in that the corrugations on the gasket will provide an additional labyrinth seal. It also provides the advantage of reducing the contact area of the gasket, enhancing its compressive characteristics. A double-jacketed corrugated gasket still relies on the primary seal on the inner lap.

Note: Double-jacketed gaskets are sometimes used with a very-light coating of gasket cement or lubricant which will assist in flowing the metal portion of the gasket into the tool marks on the flange seating surface.

(Cont.)

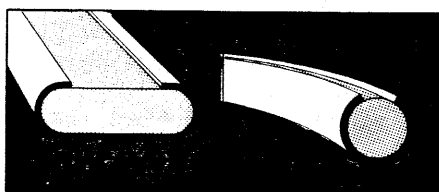


When using a gasket compound or lubricant it is important to remember to use only a very light coating. Excessive amounts of lubricant or compound may cause total gasket failure if the joint is exposed to high temperature and/or pressure.



FRENCH TYPE GASKETS

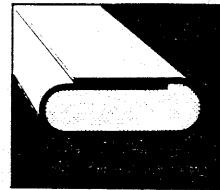
French type gaskets are available in a one-piece jacketed construction for narrow radial widths not exceeding 1/4" and in two- and three-piece constructions, as shown in the sketches, for wider applications. This type of gasket can also be used with the jacket on the external edge of the gasket when the application requires the outer edge of the gasket to be exposed to fluid pressure. The most widely used French type gaskets are fabricated using a copper sheath. The double-jacketed construction is preferred over the French or single-jacketed construction, where practical, since it provides a totally sheathed gasket with none of the soft filler exposed.



SINGLE-JACKETED GASKET

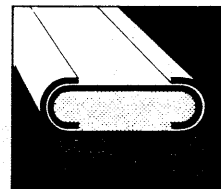
Single-jacketed gaskets are normally used for relatively narrow applications similar to the French type.

They are made by encasing a soft filler on one face, both edges and a portion of the other face with a metal. The majority of applications for single-jacketed gaskets are normally 1/4" or less in radial width. This type of gasket is widely used in air tool applications and engine applications where space is limited, gasket seating surfaces are narrow and relatively low compressive forces are available for seating the gasket. For applications in excess of 1/4", a double-jacketed gasket or double-jacketed corrugated gasket is normally recommended. Most single-jacketed gaskets are supplied with copper as the jacketing material, however, other materials are available.



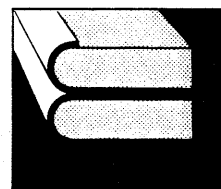
SINGLE-JACKETED OVERLAP

In the single-jacketed overlap construction the maximum flange width is approximately 1/4". This type of gasket is used when total enclosure of the soft filler material is required and when the flange width makes it impractical to use a double-jacketed gasket.



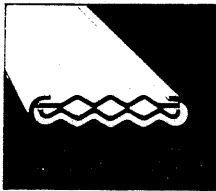
DOUBLE-JACKETED DOUBLE-SHELL GASKET

The double-jacketed, double-shelled gasket is similar to the double-jacketed gasket except that instead of using a shell and a washer, two shells are used in the fabrication of the gasket. It has the advantage of a double lap at both the I.D. and the O.D. of the gasket, adding greater stability to the gasket. The construction will withstand higher compressive loads. Double-shell gaskets are normally restricted to use in high pressure applications. Its temperature limitations depend upon the type of metal and filler used in construction.



MODIFIED FRENCH TYPE

This particular type of gasket is normally used with very light flanges on duct work handling hot gases. Its construction consists of two French type shields welded together with a soft filler material on either side of the metal. Metal thickness is normally 26 gauge, rolled on the I.D. to act as a shield. The soft gasket materials available are Cerafelt, asbestos millboard, compressed asbestos sheet, etc.



DOUBLE-JACKETED CORRUGATED GASKET WITH A CORRUGATED METAL FILLER

At temperatures in excess of the range of 900°F to 1000°F where the standard soft filler is normally not recommended, a double-jacketed corrugated metal gasket with a corrugated metal filler is frequently used. This construction has all the advantages of the double-jacketed corrugated metal gasket and, in addition, since the filler is normally the same material as the gasket itself, the upper temperature limit would be determined by the metal being used. This type of gasket, depending upon metal selected, makes an excellent heat exchanger gasket for high pressure, high temperature applications. As in the case of double-jacketed metal gaskets and double-jacketed corrugated metal gaskets, the primary seal would be the inner lap of metal, the secondary seal would be the outer lap of metal and some degree of labyrinth sealing can be achieved with the corrugations.

CORRUGATED AND CORRUGATED INLAID GASKETS STYLE



Lamons corrugated gaskets are economical for use on relatively low pressure applications that require low

bolt loads for gasket seating. Because of the corrugations and thin metal thicknesses (.010" to .031"), relatively light bolt forces are required to flow the gasket materials at the points of contact with the flange. Required bolt loads are substantially less than solid metal types such as plain flat metal, profile or serrated fabricated of the same material. The corrugations do afford some degree of resilience depending upon their pitch, depth and the thickness of the material used in fabrication. With the corrugations, a labyrinth effect is created. They are corrugated concentric to the I.D. and are used either plain, with a coating of sealing or gasket compound, or with an asbestos or ceramic cord cemented into the corrugations. It should be pointed out, however, when using a gasket compound on the surface of the gasket that a very thin coat should be applied to prevent a "pumping" action from taking place. The temperature range for corrugated gaskets would depend upon the selection of the material. The upper temperature range with only a metal component would be the same as for a solid metal gasket. When using an asbestos cord cemented to the corrugations it is generally limited to an upper temperature range of approximately 900°F. At temperatures in excess of 900°F, ceramic cord may be used in place of asbestos cord. This would raise the upper temperature limit for the corrugated gasket to approximately 2000°F, depending upon the metal being used. Corrugated gaskets with asbestos cord or with ceramic cord make particularly effective gaskets on large diameter flanges handling flue and combustion gases at high temperatures. They can be fabricated in virtually any shape including rectangular. There are no limitations to the size other than problems that would be associated with shipping. Corrugated gaskets with asbestos cord or ceramic cord are available with a special aluminum paint which minimizes the tendency to stick to the flanges.

— SIZING METAL JACKETED GASKETS —

The following sizings and tolerances are not mandatory but are suggested values based upon experience.

GASKETS CONFINED ON O.D. AND I.D.

Gasket I.D. = Groove I.D. + 1/16"
Gasket O.D. = Groove O.D. - 1/16"

GASKETS CONFINED ON O.D. ONLY

Gasket I.D. = Bore + minimum 1/8"
Gasket O.D. = Recess O.D. - 1/16"

GASKETS UNCONFINED ON O.D. AND I.D.

Gasket I.D. = Bore + minimum 1/8"
Gasket O.D. = Up to a maximum of the bolt hole circle diameter minus one bolt hole diameter unless gasket is to be full face. If gasket is to be full face, then the following must be specified:
(a) Bolt hole circle diameter
(b) Bolt hole diameter
(c) Number of bolt holes
(d) Desired gasket O.D.

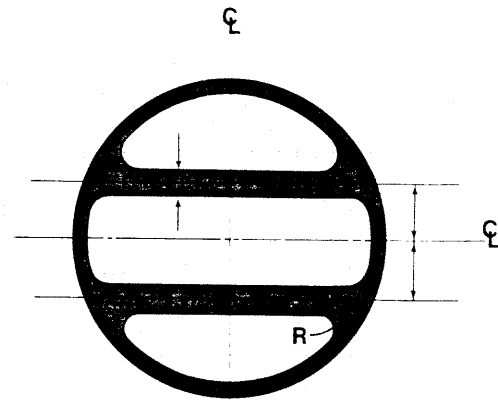
STANDARD TOLERANCES

Gasket Diameter	I.D.	O.D.
Up to 6"	+ 1/32" - 0	+ 0 - 1/32"
6" to 60"	+ 1/16" - 0	+ 0 - 1/16"
60" and Above	+ 3/32" - 0	+ 0 - 3/32"

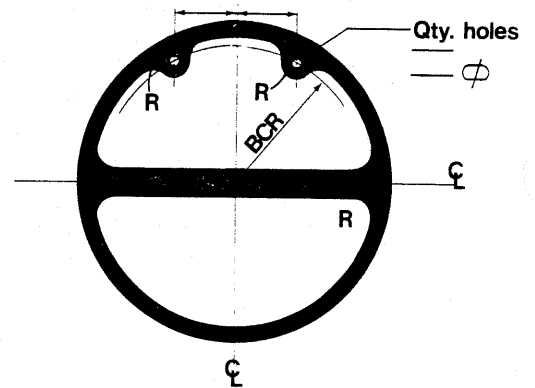
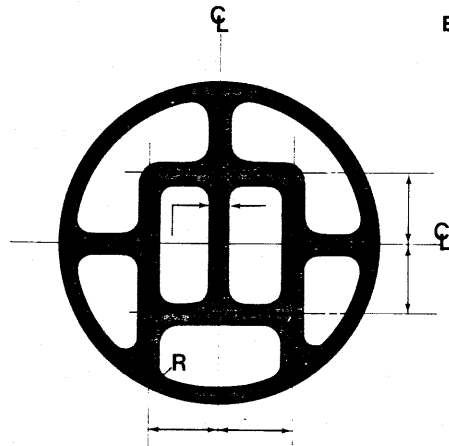
LAMONS **METAL CLAD AND SOLID METAL HEAT EXCHANGER GASKETS**

INFORMATION NEEDED TO FILL AN ORDER:

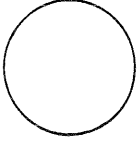
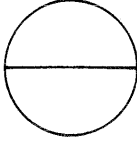
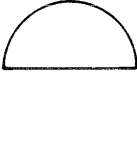
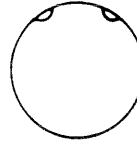
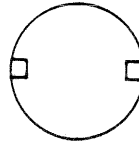
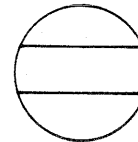
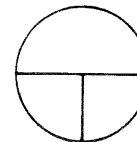
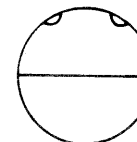
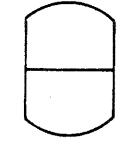
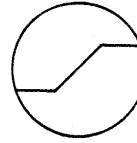
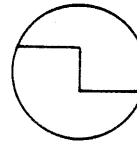
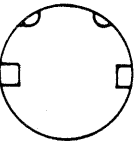
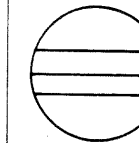
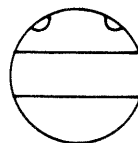
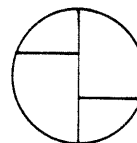
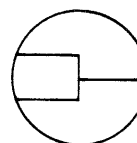
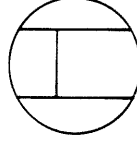
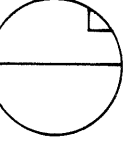
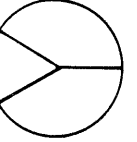
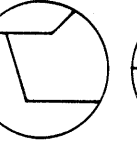
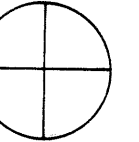
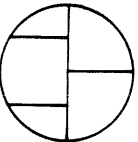
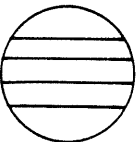
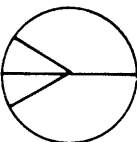
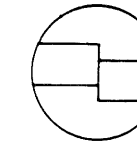
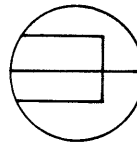
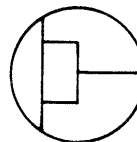
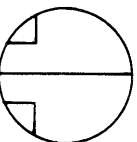
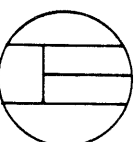
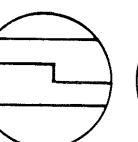
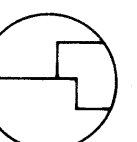
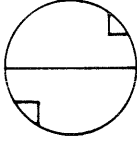
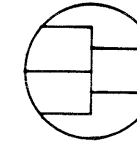
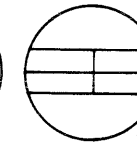
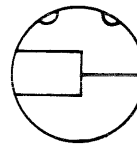
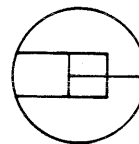
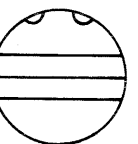
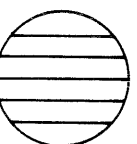
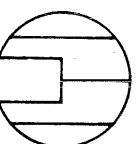
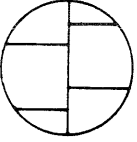
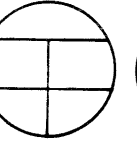
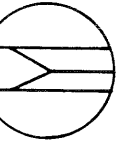
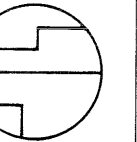
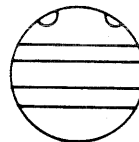
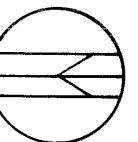
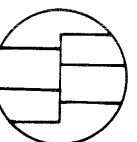
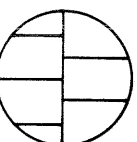
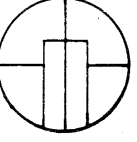
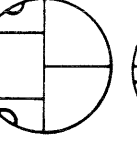
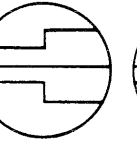
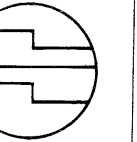
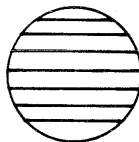
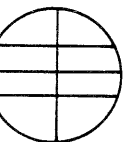
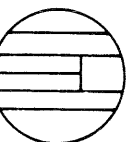
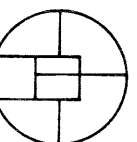
1. Outside diameter.
2. Inside Diameter
3. Shape per Standard Shapes Index
4. Lamons style per catalog, or type of construction
5. Thickness
6. Materials (metal or metal and filler)
7. Rib size
8. Distance from centerline of gasket to centerline of ribs
9. Radii



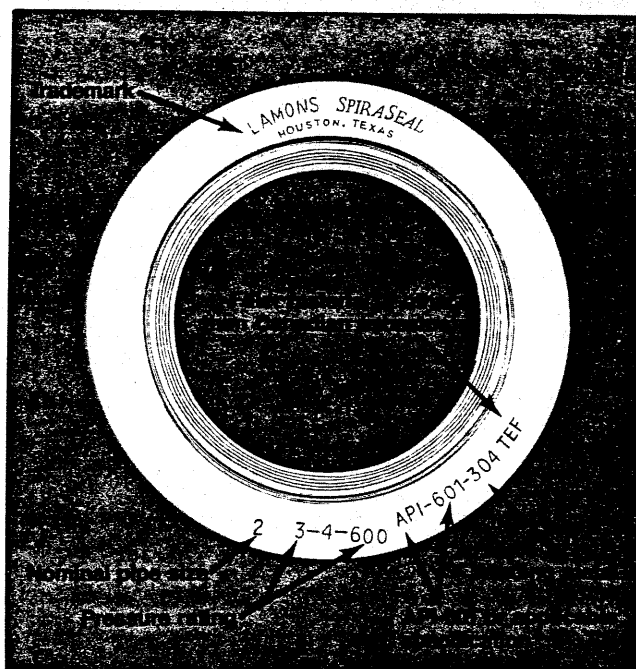
Examples:



LAMONS HEAT EXCHANGER GASKETS — STANDARD SHAPE INDEX

 R	 C-1	 C-2	 D-1	 D-2	 E-1	 E-2	 E-3
 E-4	 F-1	 F-2	 F-3	 G-1	 G-2	 G-3	 G-4
 G-5	 G-6	 G-7	 G-8	 G-9	 H-1	 H-2	 H-3
 H-4	 H-5	 H-6	 H-7	 H-8	 H-9	 H-10	 H-11
 H-12	 I-1	 I-2	 I-3	 I-4	 I-5	 I-6	 I-7
 I-8	 I-9	 I-10	 I-11	 J-1	 J-2	 J-3	 J-4
 J-5	 J-6	 J-7	 J-8	 K-1	 K-2	 K-3	 K-4

SPIRAL-WOUND GASKETS



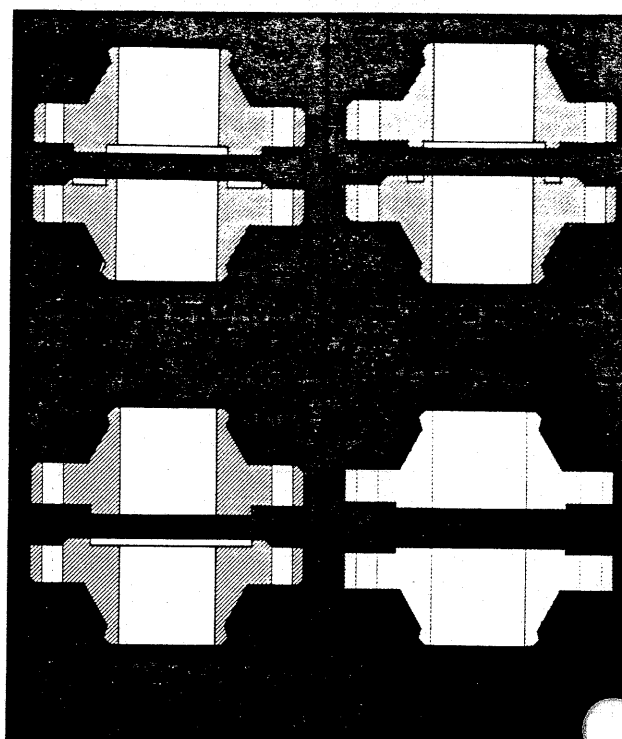
Spiral-wound gaskets have become extremely popular due to the wide variety of available styles and sizes. Spiral-wound gaskets can be fabricated of any metal which is available in thin strip and which can be welded; therefore, they can be used against virtually any corrosive medium dependent upon the choice of the metal and filler. They can be used over the complete temperature range from cryogenic to approximately 2000°F. This type gasket can be used in all pressures from vacuum to the standard 2500 psi flange ratings and on special design flanges up to 10,000 psi. They are more resilient than any other type of metallic gasket with the exception of pressure sealing metal gaskets and, as a consequence, can compensate for flange movement that may occur due to temperature gradients, variations of pressure and vibration. Spiral-wound gaskets can also be manufactured with variable densities, i.e. relatively low density gaskets for vacuum service up to extremely high density gaskets having a seating stress of approximately 30,000 psi. The softer gaskets would require a seating stress in the range of 10,000 psi.

VARIABLE DENSITY

Spiral-wound gaskets are manufactured by alternately winding strips of metal and soft fillers on the outer edge of winding mandrels that determine the inside dimensions of the wound component. In the winding process, the alternating plies are maintained under pressure. Varying the pressure during the winding operation and/or the thickness of the soft filler, the density of the gasket can be controlled over a wide range. As a general rule, low winding pressure and thick soft fillers are used for low pressure applications. Thin fillers and high pressure loads are used for high pressure applications. This of course would account for the higher bolt loads that have to be applied to the gasket in high pressure applications. In addition to all these advantages of the spiral-wound gasket, they are a relatively low cost. When special sizes are required, tooling costs are very nominal.

SIZING SPIRAL WOUND GASKETS

Spiral-wound gaskets must be sized to ensure the spiral-wound component is seated between flange faces. If it protrudes beyond a raised face or into a flange bore, mechanical damage and leakage may occur.



GASKET CONFINED ON I.D. AND O.D.

Gasket I.D. = Groove I.D. + 1/16"

Gasket O.D. = Groove O.D. - 1/16"

GASKET CONFINED ON O.D. ONLY

Gasket I.D. = Bore + Minimum 1/4"

Gasket O.D. = Recess O.D. - 1/16"

GASKET UNCONFINED I.D. AND O.D.

Gasket I.D. = Seating Surface I.D. + Minimum 1/4"

Gasket O.D. = Seating Surface O.D. - Minimum 1/4"

Centering Guide O.D. = Bolt Circle Diameter - Diameter of Bolt

STANDARD TOLERANCES

Gasket Diameter	I.D.	O.D.
Up to 1"	+3/64 -0	+0 -1/32
1" to 24"	+1/32 -0	+0 -1/32
24" to 36"	+3/64 -0	+0 -1/16
36" to 60"	+1/16 -0	+0 -1/16
60" and above	+3/32 -0	+0 -3/32

Thickness +.010 — .005 on gaskets with:

a. less than 1" I.D. greater than 26" I.D.

b. teflon fillers

c. compressed asbestos fillers

d. 1" or larger flange width

Thickness ±.005 on all others

AVAILABLE SIZES AND THICKNESSES

Lamons spiral-wound gaskets are available in thicknesses of .0625", .100", .125", .175", .250", and .285". The following chart indicates the size range that can normally be fabricated in the various thicknesses along with the recommended compressed thickness of each and the maximum flange width.

LIMITATIONS OF SIZE AND THICKNESS

Gasket Thickness	Maximum I.D.*	Maximum Flange Width*	Recommended Compressed Thickness
.0625"	9"	3/8"	.050/.055"
.100"	12"	1/2"	.075/.080"
.125"	40"	3/4"	.090/.100"
.175"	75"	1"	.125/.135"
.250"	160"	1 1/4"	.180/.200"
.285"	160"	1 1/4"	.200/.220"

*These limitations are intended as a general guide only. Materials of construction and flange width of gasket can drastically affect the limitations listed.

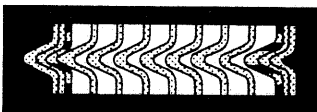
FLANGE SURFACE FINISH

Use of spiral-wound gaskets gives the designer and the user a wider tolerance for flange surface finishes than other metallic gaskets. While they can be used against most commercially available flange surface finishes, experience has indicated that the optimum flange surface finishes used with spiral-wound gaskets are as follows:

- 150 and 300 psi service, general purpose, 500 microinches AARH
- 400 psi and above, general purpose, 250 microinches AARH
- hot hydrogen, hazardous materials and hard to hold fluids, 125 microinches.
- For vacuum service, an 80 microinch AARH finish is recommended.

AVAILABLE SPIRAL SEAL STYLES

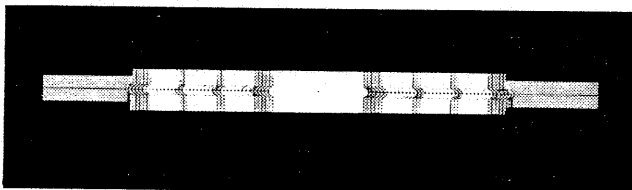
Lamons spiral-wound gaskets are available in a variety of styles to suit the particular flange facing being utilized on the flanges.



LAMONS' STYLE W

Style W is a spiral-wound sealing component only that is normally used on tongue and groove joints, male and female flange facings and groove to flat flange facings.

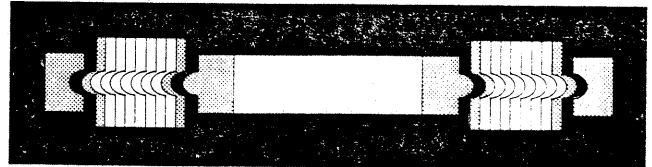
LAMONS' STYLE WR



Style WR gaskets consist of a spiral-wound sealing component with a solid metal outer guide ring. These gaskets are used on plain flat face flanges and on raised face flanges. The outer guide ring serves to center the

gasket properly in the flange joint, acts as an anti-blowout device, provides radial support for the spiral-wound component, and acts as a compression gauge to prevent the spiral-wound component from being crushed. Normally the outer guide rings are furnished in mild steel, but can be supplied in other metals when required by operating conditions.

LAMONS' STYLE WRI



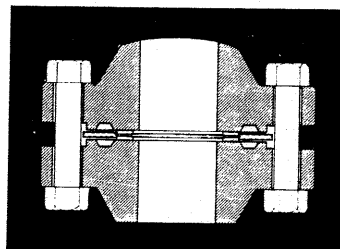
Style WRI is identical to style WR with the addition of an inner ring. The inner ring serves several functions. It provides radial support for the gasket on the I.D. to prevent the occurrence of any buckling or imploding. Its I.D. is normally sized slightly larger than the I.D. of the flange bore, minimizing turbulence in process flow. After the gasket is compressed, the flanges would normally be in contact with the inner ring and hence erosion and corrosion of the flange surface between the I.D. of the sealing component and the flange bore is minimized. The inner rings are normally supplied in the same material as the spiral-wound component. Refer to table below for dimensions of inner ring I.D.'s for flanges up to 24-inch diameter and 2500 PSI.

STANDARD INSIDE DIMENSIONS OF INNER RINGS AS SUGGESTED BY THE FLUID SEALING ASSOCIATION

Nom. Pipe Size	150#	300#	400#	600#	900#	1500#	2500#
1-1/4"	1.50	1.50	1.50	1.50	1.3125	1.3125	1.3125
1-1/2"	1.75	1.75	1.75	1.75	1.63	1.63	1.63
2"	2.18	2.18	2.18	2.18	2.06	2.06	2.06
2-1/2"	2.63	2.63	2.63	2.63	2.50	2.50	2.50
3"	3.19	3.19	3.19	3.19	3.19	3.19	3.19
3-1/2"	3.69	3.69	3.69	3.69	—	—	—
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"	8.13	8.13	8.13	8.13	8.13	8.13	8.13
10"	10.15	10.15	10.15	10.15	10.15	10.15	10.15
12"	12.06	12.06	12.06	12.06	12.06	12.06	12.06
14"	13.25	13.25	13.25	13.25	13.25	13.25	—
16"	15.13	15.13	15.13	15.13	15.13	15.13	—
18"	17.00	17.00	17.00	17.00	17.00	17.00	—
20"	18.94	18.94	18.94	18.94	18.94	18.94	—
24"	22.75	22.75	22.75	22.75	22.75	22.75	—

NOTES:

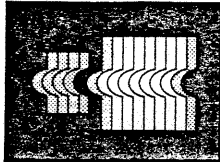
- 1 Dimensions shown are in inches with a tolerance of plus/minus 1/32" for sizes up to 4" ips and plus/minus 1/16" for sizes over 4" ips.
- 2 All sizes suitable for schedule 40 pipe or heavier.
- 3 Inner Ring dimensions for 1/2", 3/4" and 1" NPS, all pressure series are not considered standard and should be specified by the buyer.
- 4 Inner Ring I.D. dimensions shown for 1-1/4", 1-1/2", 2", and 2-1/2", 900#, 1500#, and 2500# series will produce inner ring flange widths of 1/8", a practical minimum for production purposes.



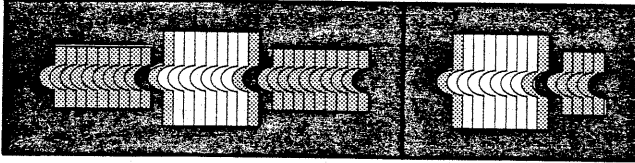
LAMONS' STYLE WR-RJ

This style gasket is identical to a Style WR in construction features but is specially sized to be used as a replacement gasket for flanges machined to accept oval

or octagonal ring joint gaskets. The sealing component is located between the I.D. of the groove machined in the flange and the flange bore. These are intended to be used as replacement parts and are considered a maintenance item. In new construction, where spiral-wound gaskets are intended to be used, raised face flanges should be utilized. Refer to Lamons General Catalog for dimensions of Style WR-RJ gaskets for flanges up to 24-inch diameter and 1500 psi.

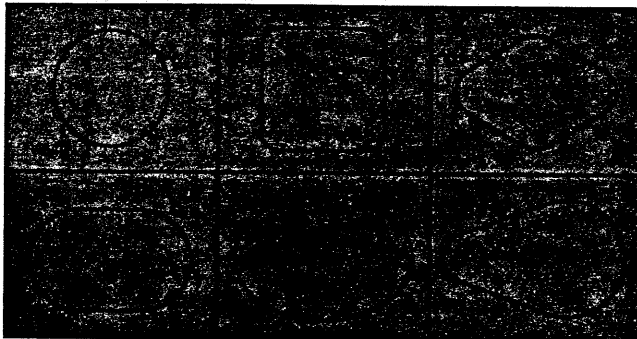


GASKETS WITH WOUND GAUGE RINGS

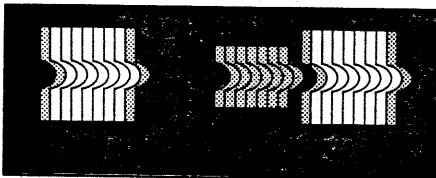


When a guide ring is required that is too narrow for practical fabrication of solid metal guide rings, Lamons spiral-wound gaskets are available with a guide made entirely of spiral metal windings. These spiral metal windings serve the same basic purpose as the solid metal ring, that is as a compression limiting and a centering device. The spirally wound ring is normally supplied in the same metal as the metal in the gasket. This type of wound guide ring is normally limited to a 1/4" radial width or less.

LAMONS' STYLE H



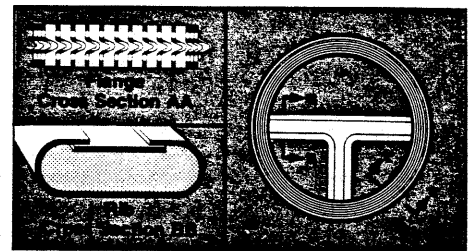
Style H gaskets are for use on boiler handhole and tubecap assemblies. They are available in round, square, rectangular, diamond, obround, oval and pear shapes. The Lamons Gasket Company has tooling available for manufacturing most of the standard handhole and tubecap sizes of the various boiler manufacturers. (Refer to our General Catalog.) These are also available in special sizes and shapes. To order special gaskets, dimensional drawings or sample cover plates should be provided in order to assure proper fit.



LAMONS' STYLE MW AND MWC

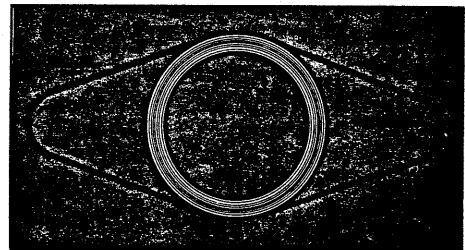
These gaskets are available in round, obround, and oval shapes and are used for standard manhole cover plates. (Refer to Lamons General Catalog for standard available shapes and sizes.) When special gaskets are required, it is necessary to submit complete information including a sketch or blueprint or a sample cover plate on which the gasket is to be used.

NOTE: When spiral-wound handhole and manhole gaskets with a straight side are required it is necessary that some curvature be given to the flat or straight side to prevent buckling of the gasket. This is due to the fact that spiral-wound gaskets are wrapped under tension and therefore tend to buckle inward when the gaskets are removed from the winding mandrel. As a rule of thumb, the ratio of the long ID to the short ID should not exceed 3 to 1.



LAMONS' STYLE W OR WRP

These gaskets are similar to Style W and Style WRP with the addition of pass partitions for use with shell and tube heat exchangers. Partitions are normally supplied with a double-jacketed construction of the same material as the spiral-wound component. The partition strips can be soft soldered, tack welded or silver soldered to the spiral-wound component. The double-jacketed partition strips are normally slightly thinner than the spiral-wound component in order to minimize the bolt loading required to properly seat the gasket.



LAMONS' STYLE L

The Lamons Style L gasket is available for raised face and flat face applications where it is not practical to supply an outer gauge ring. The spiral-wound components of Style L are identical to those of Style W and in addition have a wire loop welded to the outer periphery of the gasket, sized so as to fit over diametrically opposite bolts, for proper centering of the spiral-wound component on the gasket seating surface. Whenever possible, it is recommended that a Style WR gasket be used in lieu of a Style L gasket because of the obvious advantages of the outer solid metal gauge ring. The Style L is considerably more difficult to produce than Style WR and therefore more expensive.

SECTION III — INSTALLATION

INSTALLATION AND MAINTENANCE TIPS FOR ALL GASKETS

All too often we hear "the gasket leaks". This is not strictly true. It is the joint that leaks and the gasket is one component of several that make up the joint. Unfortunately, the gasket is expected to make up for any and all deficiencies in design, improper installation procedures, and to compensate for all flange movement due to thermal changes, pressure changes, vibrations, etc. In many cases the gasket will do these things but only when careful attention is given to all the aspects of gasket selection, design and installation.

Experience in investigating leaky joints over the years has indicated the most common cause of leaky joints is improper installation procedures being followed.

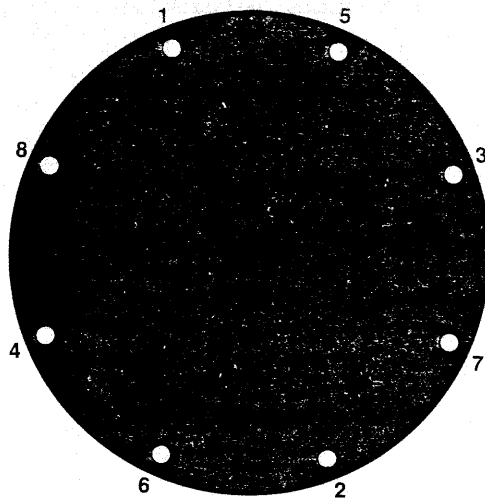
GASKET INSTALLATION PROCEDURES

Regardless of the type of gasket being used or the materials of construction, certain basic procedures must be followed if the joint is to be assembled, tested and put into operation with minimum difficulty. While these comments may seem elementary, they are extremely important in achieving a satisfactory seal and minimizing the time required to successfully make up joints. These procedures should be followed whether bolt stress will be achieved with ordinary stud wrenches, preheating studs, using tensioning devices, using torque wrenches, or using hydraulic wrenches. This is so very important that it bears repeating. Regardless of the method used to apply stress to the studs, these fundamentals should be very carefully followed:

1. Inspect the gasket seating surfaces. Look for tool marks, cracks, scratches or pitting by corrosion and make sure that the gasket seating surface is proper for the type of gasket being used. Radial tool marks on a gasket seating surface are virtually impossible to seal regardless of the type gasket being used, therefore every attempt must be made to minimize these. If remachining of flanges is not possible, investigate the use of patching cements such as Devcon which can be fairly effective in repairing the gasket seating surfaces.
2. Inspect the gasket. Make sure the material is as specified, look for any possible defects or damage in the gasket.
3. Inspect and clean each stud or bolt, each nut, each washer, and the facing on the flanges against which the nuts will rotate. Look for severe galling, pitting, etc. If any of the above mentioned items are damaged beyond repair, replace that item.
4. Lubricate all thread contact areas and nut facings. The importance of proper lubrication cannot be overstressed. No joint should be made up without the proper lubricant being applied to the threaded surfaces and to the nut facings. When flanges will be subjected to high temperatures, the use of an anti-seize compound should be considered to facilitate subsequent disassembly. There are available on the market today a vast variety of suitable lubricants that can be selected to provide the necessary low coefficient of friction for installation and adequate anti-seize properties for high temperature application. The better the lubricant, the more consistent will be the actual achieved bolt stress at installation.
5. With raised face and flat face installation, loosely install the stud bolts on the lower half of the flange. Insert the gasket between the flange facing to allow the bolts to center the gasket on the assembly. Install the balance of the bolts and nuts and bring all to a hand-tight or snug condition.
6. If the gasket is being installed in a recess or a groove, center the gasket midway into the recess or the groove. If the joint is vertical it may be necessary to use some cup grease or a few dabs of gasket cement or some other adhesive compatible with the process fluids, to keep the gasket in position until the flanges are tightened.
7. Torque the bolts up to a maximum of thirty percent of the final torque value required following the sequence recommended. (See charts for bolting sequence.) Number bolts so that torquing requirements can be followed. With any gasket material, it is extremely important to follow a proper bolting sequence. If this sequence is not followed, the flanges can be cocked. Then, regardless of the amount of subsequent torquing, they cannot be brought back parallel. This problem, of course, is maximized on metallic gaskets more so than on non-metallic.
8. Repeat step 7, increasing the torque to approximately 50 to 60 percent of the final torque required.
9. Continue with a rotational pattern of retorquing all studs or bolts to the desired amount until no further rotation of the nuts can be achieved. This may require several retorquings since as one stud is torqued it will relieve the stress on the adjacent stud until such time as equilibrium is achieved.
10. On high-pressure, high-temperature applications, it is recommended that the flanges be retorqued to the required stress after 24 hours at operating pressures and temperatures to compensate for any relaxation or creep that may have occurred.

BOLT TORQUE SEQUENCE

8-Bolts



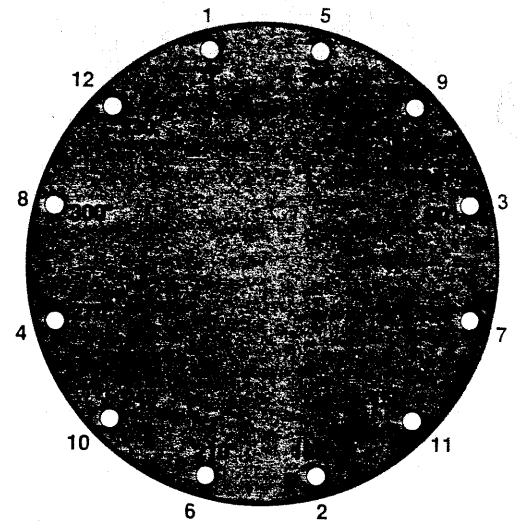
Sequential Order

1-2
3-4
5-6
7-8

Rotational Order

1
5
3
7
2
6
4
8

12-Bolts



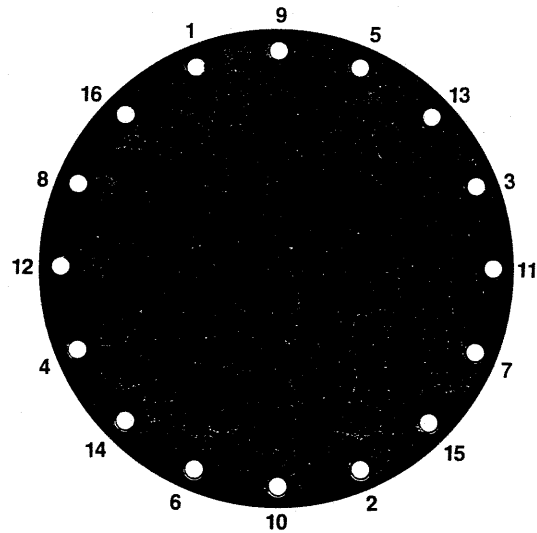
Sequential Order

1-2
3-4
5-6
7-8
9-10
11-12

Rotational Order

1
5
9
3
7
11
2
6
10
4
8
12

16-Bolts



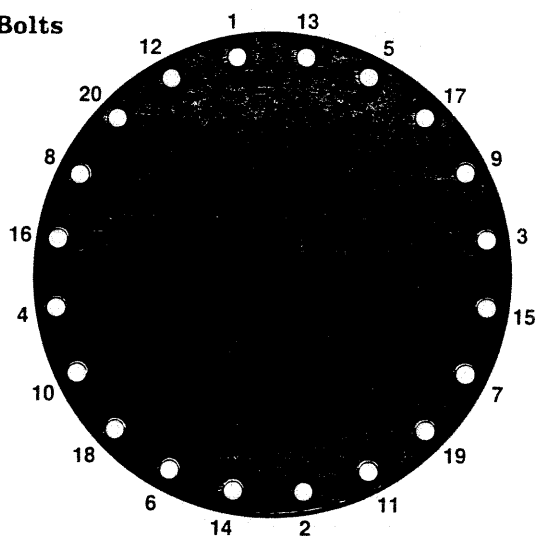
Sequential Order

1-2
3-4
5-6
7-8
9-10
11-12
13-14
15-16

Rotational Order

1
9
5
13
3
11
7
15
2
10
6
14
4
12
8
16

20-Bolts



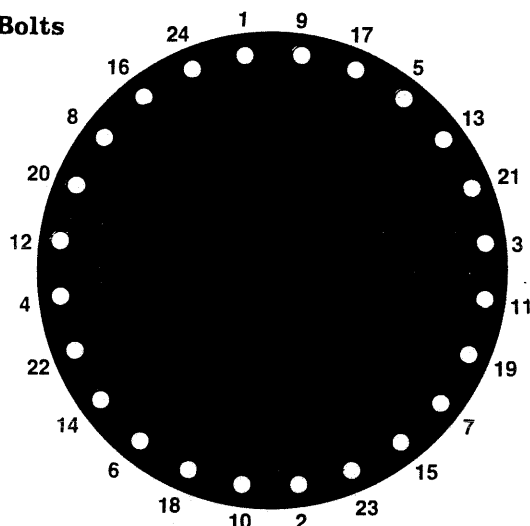
Sequential Order

1-2
3-4
5-6
7-8
9-10
11-12
13-14
15-16
17-18
19-20

Rotational Order

1
13
5
17
9
3
15
7
19
11
2
14
6
18
10
4
16
8
20
12

24-Bolts



Sequential Order

1-2
3-4
5-6
7-8
9-10
11-12
13-14
15-16
17-18
19-20
21-22
23-24

Rotational Order

1
9
17
5
13
21
3
11
19
7
15
23
2
10
18
16
8
20
12
4
14
22
6

sequence the trend in industry today is the use of torque wrenches, tensioning devices, hydraulic wrenches, or drilling the studs and inserting heaters to preheat the stud to a specific temperature that will ultimately create the proper tension on the bolt. The use of manpower to tighten the bolts, by sledgehammers, striking wrenches and pieces of pipe on the end of the wrench is becoming less and less a standard practice. It is time-consuming, strenuous and is a very dangerous practice. The newer techniques are much more reliable.

NOTE: Allowable bolt stresses. Section VIII of the ASME Pressure Vessel Code, Appendix S, specifically recognizes the problem of initial bolt stresses. For example, a flange designer will determine his required bolting for a 600 psi application at a given operating temperature specifically in accordance with allowable stresses for the bolt material at the operating temperature. These allowable stresses are based on the particular material and their strength at operating temperature. In addition, the same bolt material will have an allowable stress at ambient conditions as specified. As a consequence, in most cases the design of the flange is based upon the allowable bolt stress of the particular material at design temperature and at the design or operating pressure. However, in most cases, the hydrostatic test pressure that the flange joint must pass is one and a half times the design pressure. As a consequence, any joint that is designed in strict accordance with the ASME Pressure Vessel Code and is subjected to hydrostatic tests in excess of the design pressure, will require a higher initial stress on the stud to successfully pass the hydrostatic test. Appendix S of Section 8 of the ASME Pressure Vessel Code speaks in great length on this problem and, in essence, states, that in order to pass hydrostatic tests, bolts may be stressed to whatever level is required to satisfactorily pass the test. This introduces additional problems. In cases where low yield bolt material is being used, the stresses required in bolts sufficient to satisfactorily pass the test may exceed the yield point of the bolt material. Once this occurs, no additional stressing of the bolt will alleviate the problem of leakage. As a consequence it may be necessary to use high tensile bolts or studs in order to achieve a satisfactory test. When this is required, the following procedures should be followed. (See Page 32)

- Use high tensile bolts or studs for hydrostatic tests following the procedures outlined above for gasket installation. After a successful hydrostatic test has been achieved, relieve the bolts to approximately 50 percent of the prestress required.
- Replace the bolts or studs one at a time with the proper grade bolt for operating conditions. As each bolt is replaced, torque it to the value of the other bolts.
- After all the bolts have been replaced, retorque the bolts to 100% of the allowable stress for the particular grade material. (Once again it is imperative that a proper lubricant be used on the bolts when replacement is being made.)

TORQUE VALUES

Probably the only true measurement of bolt stress is by bolt or stud elongation. In practice, however, this would be an extremely costly and impractical approach to determine the true measure of bolt stress. As a con-

TROUBLE SHOOTING LEAKING JOINTS

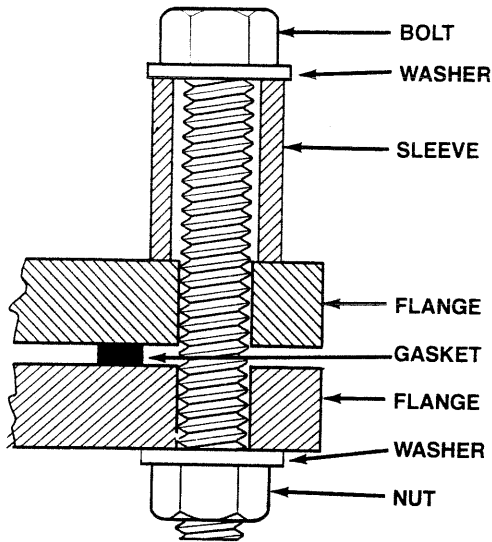
One of the best available tools to aid in determining the cause of leakage is a careful examination of the gasket in use when leakage occurred.

Observation	Possible Remedies
Gasket badly corroded	Select replacement material with improved corrosion resistance.
Gasket extruded excessively	Select replacement material with better cold flow properties, select replacement material with better load carrying capacity — i.e., more dense.
Gasket grossly crushed	Select replacement material with better load carrying capacity, provide means to prevent crushing the gasket by use of a stop ring or re-design of flanges.
Gasket mechanically damaged due to overhang of raised face or flange bore.	Review gasket dimensions to insure gaskets are proper size. Make certain gaskets are properly centered in joint.
No apparent gasket compression achieved.	Select softer gasket material. Select thicker gasket material. Reduce gasket area to allow higher unit seating load.
Gasket substantially thinner on O.D. than I.D.	Indicative of excessive "flange rotation" or bending. Alter gasket dimensions to move gasket reaction closer to bolts to minimize bending movement. Provide stiffness to flange by means of back-up rings. Select softer gasket material to lower required seating stresses. Reduce gasket area to lower seating stresses.
Gasket unevenly compressed around circumference	Improper bolting up procedures followed. Make certain proper sequential bolt up procedures are followed.
Gasket thickness varies periodically around circumference.	Indicative of "flange bridging" between bolts or warped flanges. Provide reinforcing rings for flanges to better distribute bolt load. Select gasket material with lower seating stress. Provide additional bolts if possible to obtain better load distribution. If flanges are warped, re-machine or use softer gasket material.

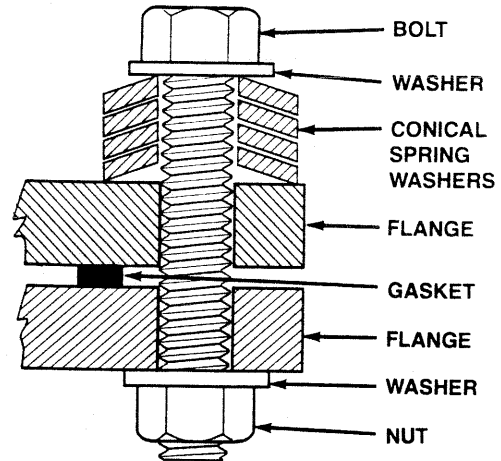
OTHER PROBLEM AREAS

JOINT MUST COMPENSATE FOR WIDE TEMPERATURE VARIATIONS:

Solution: Consider use of sleeve around bolts to increase effective bolt length:

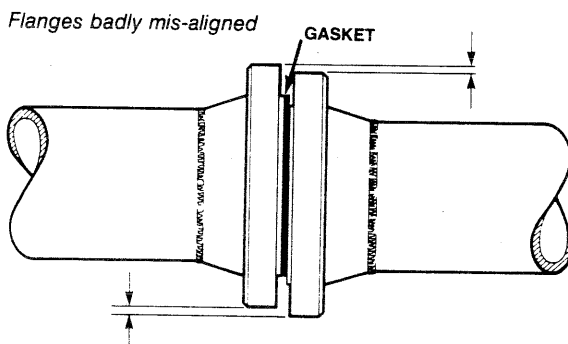
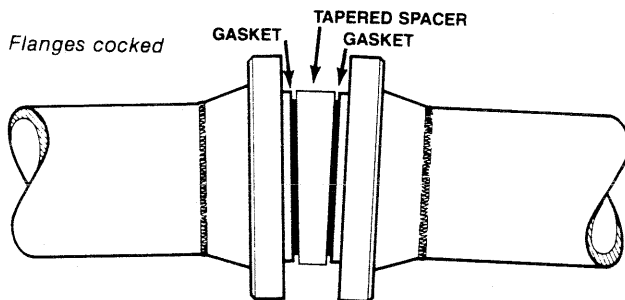
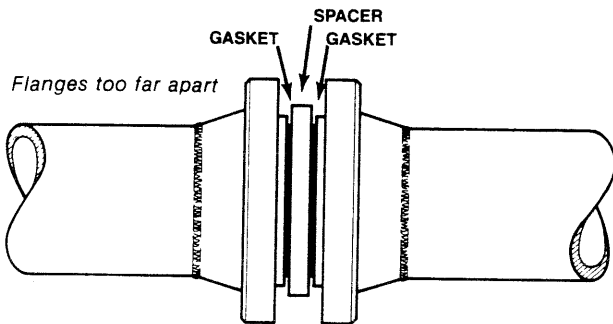


Or consider use of conical spring washers in place of sleeve to eliminate torque losses over wide temperature ranges.

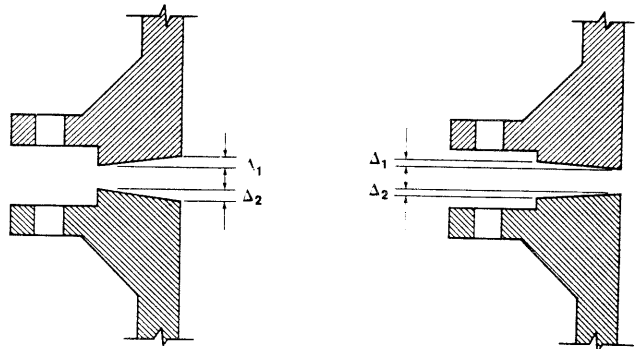


FLANGES BADLY COCKED OR SEPARATED TOO FAR:

Solution: *Do not* try to correct problem with flange bolts — can overstress.
Do use spacers to correct problem with gasket on each side.



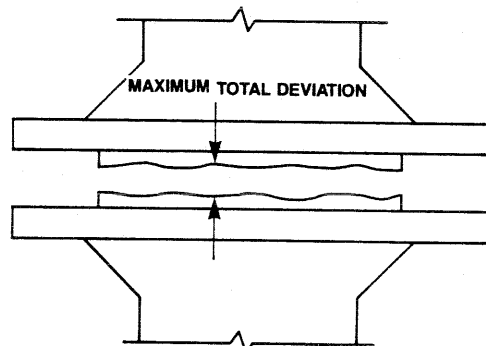
FLANGES OUT OF PARALLEL:



Total allowable out of parallel: $\Delta_1 + \Delta_2 = 0.015''$

Note — Deviation on right is less critical than deviation on left since bolt tightening will tend to bring flanges parallel due to flange bending.

WAVY SURFACE FINISH



Note:

1. If using jacketed or spiral wound gaskets — deviation should not exceed 0.015"
2. If using solid metal gaskets — deviation should not exceed 0.005"
3. If using rubber or compressed asbestos, more leeway is possible — perhaps total of 0.030".

SECTION IV — APPENDIX

APPENDIX S ASME SECTION VIII DIVISION I PRESSURE VESSELS DESIGN CONSIDERATIONS FOR BOLTED FLANGE CONNECTIONS

The primary purpose of the rules for bolted flange connections in Parts A and B of Appendix II is to insure safety, but there are certain practical matters to be taken into consideration in order to obtain a serviceable design. One of the most important of these is the proportioning of the bolting, i.e., determining the number and size of the bolts.

In the great majority of designs the practice that has been used in the past should be adequate, viz., to follow the design rules in Appendix II and tighten the bolts sufficiently to withstand the test pressure without leakage. The considerations presented in the following discussion will be important only when some unusual feature exists, such as a very large diameter, a high design pressure, a high temperature, severe temperature gradients, an unusual gasket arrangement, and so on.

The maximum allowable stress values for bolting given in the various tables of Subsection C are design values to be used in determining the minimum amount of bolting required under the rules. However, a distinction must be kept carefully in mind between the design value and the bolt stress that might actually exist or that might be needed for conditions other than the design pressure. The initial tightening of the bolts is a prestressing operation, and the amount of bolt stress developed must be within proper limits, to insure, on the one hand, that it is adequate to provide against all conditions that tend to produce a leaking joint, and on the other hand, that it is not so excessive that yielding of the bolts and/or flanges can produce relaxation that also can result in leakage.

The first important consideration is the need for the joint to be tight in the hydrostatic test. An initial bolt stress of some magnitude greater than the design value therefore must be provided. If it is not, further bolt strain develops during the test, which tends to part the joint and thereby to decompress the gasket enough to allow leakage. The test pressure is usually $1\frac{1}{2}$ times the design pressure, and on this basis it may be thought that 50 percent extra bolt stress above the design value will be sufficient. However, this is an oversimplification, because, on the one hand, the safety factor against leakage under test conditions in general need not be as great as under operating conditions. On the other hand, if a stress-strain analysis of the joint is made, it may indicate that an initial bolt stress still higher than $1\frac{1}{2}$ times the design value is needed. Such an analysis is one that considers the changes in bolt elongation, flange deflection, and gasket load that take place with the application of internal pressure, starting from the prestressed condition. In any event, it is evident that an initial bolt stress higher than the design value may and, in some cases, must be developed in the tightening operation, and it is the intent of this Division of Section VIII that such a practice is permissible, provided it includes necessary and appropriate provision to insure against excessive flange distortion and gross crushing of the gasket.

It is possible for the bolt stress to decrease after initial tightening, because of slow creep or relaxation of the gasket, particularly in the case of the "softer" gasket materials. This may be the cause of leakage in the hydrostatic test, in which case it may suffice merely to retighten the bolts. A decrease in bolt stress can also occur in service at elevated temperatures, as a result of creep in the bolt and/or flange or gasket material, with consequent relaxation. When this results in leakage under service conditions, it is common practice to retighten the bolts, and sometimes a single such operation, or perhaps several repeated at long intervals, is sufficient to correct the condition. To avoid chronic difficulties of this nature, however, it is advisable when designing a joint for high-temperature service to give attention to the relaxation properties of the materials involved, especially for temperatures where creep is the controlling factor in design. This prestress should not be confused with initial bolt stress, S_1 , used in the design of Part B flanges.

In the other direction, excessive initial bolt stress can present a problem in the form of yielding in the bolting itself, and may occur in the tightening operation to the extent of damage or even breakage. This is especially likely with bolts of small diameter and with bolt materials having a relatively low yield strength. The yield strength of mild carbon steel, annealed austenitic stainless steel, and certain of the nonferrous bolting materials can easily be exceeded with ordinary wrench effort in the smaller bolt sizes. Even if no damage is evident, any additional load generated when internal pressure is applied can produce further yielding with possible leakage. Such yielding can also occur when there is very little margin between initial bolt stress and yield strength.

An increase in bolt stress, above any that may be due to internal pressure, might occur in service during startup or other transient conditions, or perhaps even under normal operation. This can happen when there is an appreciable differential in temperature between the flanges and the bolts, or when the bolt material has a different coefficient of thermal expansion than the flange material. Any increase in bolt load due to this thermal effect, superposed on the load already existing, can cause yielding of the bolt material, whereas any pronounced decrease due to such effects can result in such a loss of bolt load as to be a direct cause of leakage. In either case, retightening of the bolts may be necessary, but it must not be forgotten that the effects of repeated retightening can be cumulative and may ultimately make the joint unserviceable.

In addition to the difficulties created by yielding of the bolts as described above, the possibility of similar difficulties arising from yielding of the flange or gasket material, under like circumstances or from other causes, should also be considered.

Excessive bolt stress, whatever the reason, may cause the flange to yield, even though the bolts may not yield. Any resulting excessive deflection of the flange, accompanied by permanent set, can produce a leaking

joint when other effects are superposed. It can also damage the flange by making it more difficult to effect a tight joint thereafter. For example, irregular permanent distortion of the flange due to uneven bolt load around the circumference of the joint can warp the flange face and its gasket contact surface out of a true plane.

The gasket, too, can be overloaded, even without excessive bolt stress. The full initial bolt load is imposed entirely on the gasket, unless the gasket has a stop ring or the flange face detail is arranged to provide the equivalent. Without such means of controlling the compression of the gasket, consideration must be given to the selection of gasket type, size and material that will prevent gross crushing of the gasket.

From the foregoing, it is apparent that the bolt stress can vary over a considerable range above the design stress value. The design stress values for bolting in Subsection C have been set at a conservative value to provide a factor against yielding. At elevated temperatures, the design stress values are governed by the creep rate and stress-rupture strength. Any higher bolt stress existing before creep occurs in operation will have already served its purpose of seating the gasket and holding the hydrostatic test pressure, all at atmospheric temperature, and is not needed at the design pressure and temperature.

Theoretically, the margin against flange yielding is not as great. The design values for flange materials may be as high as five-eighths or two-thirds of the yield strength. However, the highest stress in a flange is usually the bending stress in the hub or shell, and is more or less localized. It is too conservative to assume that local yielding is followed immediately by overall yielding of the entire flange. Even if a "plastic hinge" should develop, the ring portion of the flange takes up the portion of the load the hub and shell refuse to carry. Yielding is far

more significant if it occurs first in the ring, but the limitation in the rules on the combined hub and ring stresses provides a safeguard. In this connection, it should be noted that a dual set of stresses is given for some of the materials in Table UHA-23, and that the lower values should be used in order to avoid yielding in the flanges.

Another very important item in bolting design is the question whether the necessary bolt stress is actually realized, and what special means of tightening, if any, must be employed. Most joints are tightened manually by ordinary wrenching, and it is advantageous to have designs that require no more than this. Some pitfalls must be avoided, however. The probable bolt stress developed manually, when using standard wrenches, is:

$$S = \frac{45,000}{\sqrt{d}}$$

where S is the bolt stress and d is the nominal diameter of the bolt. It can be seen that smaller bolts will have excessive stress unless judgment is exercised in pulling up on them. On the other hand, it will be impossible to develop the desired stress in very large bolts by ordinary hand wrenching. Impact wrenches may prove serviceable, but if not, resort may be had to such methods as preheating the bolt, or using hydraulically powered bolt tensioners. With some of these methods, control of the bolt stress is possible by means inherent in the procedure, especially if effective thread lubricants are employed, but in all cases the bolt stress can be regulated within reasonable tolerances by measuring the bolt elongation with suitable extensometer equipment. Ordinarily, simple wrenching without verification of the actual bolt stress meets all practical needs, and measured control of the stress is employed only when there is some special or important reason for doing so.

CHEMICAL RESISTANCE CHART — GASKET METALS

A — Good Resistance
B — Moderate Resistance
U — Unsatisfactory

Media	Alumi- num	Alloy 20	Copper	Hastel- loy	Inconel 600	Monel 400	Nickel 200	304 S.S.	316 S.S.	410 S.S.	Steel
Acetic Acid											
Room Temp.	A	A	A	A	B	B	B	A	A	A	U
Acetic Anhydride											
Room Temp.	A	A	A	A	B	B	A	A	A	A	B
Acetone	A	A	A	A	A	A	A	A	A	A	A
Aluminum Chloride											
Room Temp.	U	A	B	A	—	B	B	U	U	U	U
Aluminum Fluoride											
Room Temp.	B	A	B	A	—	B	B	U	U	U	B
Aluminum Sulphate	B	A	B	A	B	B	B	A	A	B	U
Ammonia (Anhydrous)	A	A	A	A	A	B	B	A	A	A	B
Ammonium Chloride	U	A	U	A	A	B	B	U	B	B	B
Ammonium Hydroxide	B	A	U	A	A	U	U	A	A	A	A
Ammonium Nitrate	A	A	U	A	B	U	U	A	A	A	A
Ammonium Phosphate	A	A	B	A	B	B	B	A	A	A	U
Ammonium Sulphate	U	A	B	A	B	B	B	U	A	A	A
Amyl Acetate	A	A	A	A	A	A	A	A	A	—	B
Aniline	B	A	A	A	B	B	B	A	A	A	A
Barium Chloride	B	A	B	A	A	—	B	B	A	A	B
Beer	A	A	A	A	A	A	A	A	A	A	A
Benzene	A	A	A	A	A	A	B	A	A	A	A
Benzol	A	A	A	A	B	A	B	A	A	A	A
Borax	A	—	A	A	A	A	—	A	A	A	A
Boric Acid	A	A	A	A	B	B	B	A	A	A	U
Bromine	A	A	A	A	A	A	A	U	U	U	U
Butyl Alcohol	A	A	A	A	A	A	A	A	A	A	A
Calcium Carbonate	A	A	A	A	A	A	A	A	A	A	A
Calcium Chloride	B	A	A	A	A	B	B	B	A	U	A
Calcium Hydroxide	B	A	A	A	B	B	B	B	B	A	A
Calcium Hypochlorite	U	U	U	A	U	U	B	B	A	B	U
Carbolic Acid	A	A	A	A	B	B	B	A	A	U	U
Carbon Tetrachloride	B	A	B	A	A	A	A	A	A	A	U
Chlorine-Dry	A	A	A	A	A	A	A	U	U	U	A
Chlorine-Wet	U	U	U	U	B	B	B	U	U	U	U
Chromic Acid	B	A	U	A	B	U	U	A	A	B	—
Citric Acid	A	A	A	A	B	B	B	A	A	A	U
Copper Chloride	U	—	U	A	U	U	U	U	B	B	B
Copper Sulphate	U	A	B	A	B	B	B	A	A	A	U
Creosote (Coal Tar)	B	A	A	—	B	B	B	A	A	—	A
Crude Oil	A	A	B	A	—	B	—	A	A	A	A
Ether	A	A	A	A	B	B	B	A	A	A	A
Ethyl Acetate	A	A	A	A	A	A	A	A	A	—	A
Ethyl Chloride	B	A	A	A	—	B	B	A	A	A	A
Ferric Chloride	U	U	U	A	U	U	U	U	U	U	U
Ferric Sulphate	B	A	B	A	U	U	U	A	A	A	U
Formaldehyde	B	A	A	A	A	A	A	A	A	A	B
Formic Acid	U	A	A	A	B	B	B	B	A	U	U
Fuel Oil	A	A	A	A	B	B	A	A	A	—	A
Fuel Oil (Acid)	B	A	B	A	U	B	U	U	B	—	B
Furfural	A	A	A	A	B	A	B	A	A	—	A
Gasoline	A	A	A	A	A	A	A	A	A	A	A
Glue	A	A	A	A	A	A	A	A	A	A	A
Glycerin	A	A	A	A	A	A	A	A	A	A	A
Hydrobromic Acid	U	U	U	A	U	U	U	U	U	U	U
Hydrochloric Acid											
Room Temp. 150°F	U	U	U	A	U	U	U	U	U	U	U
Hydrocyanic Acid	A	A	C	A	—	B	—	A	A	U	B
Hydrofluoric Acid	U	U	U	A	A	A	A	U	U	U	U
Hydrofluosilicic Acid	—	A	U	A	B	—	B	U	U	—	U
Hydrogen Peroxide	A	A	C	A	B	B	B	A	A	A	U
Hydrogen Sulphide	A	A	A	A	B	B	B	A	A	A	U
Kerosene	A	A	A	A	A	A	A	A	A	A	A
Lactic Acid	B	A	A	A	B	U	U	B	B	A	A
Linseed Oil	A	A	B	A	A	A	A	A	A	A	A

CHEMICAL RESISTANCE CHART — GASKET METALS (CONT.)

A — Good Resistance
B — Moderate Resistance
U — Unsatisfactory

Media	Alumi-num	Alloy 20	Copper	Hastel-loy	Inconel 600	Monel 400	Nickel 200	304 S.S.	316 S.S.	410 S.S.	Steel
Lye (Caustic)	U	A	B	A	A	A	A	A	A	B	A
Manganese Carbonate	A	A	A	—	B	B	B	A	A	A	—
Manganese Chloride	U	B	B	—	B	B	B	A	A	—	—
Magnesium Carbonate	B	A	A	A	A	A	A	A	A	A	—
Magnesium Chloride	B	A	B	A	A	A	A	A	A	U	B
Magnesium Hydroxide	U	A	A	A	A	A	A	A	A	—	A
Magnesium Nitrate	A	A	B	A	B	B	—	A	A	A	B
Magnesium Sulphate	A	A	A	A	B	B	B	A	A	A	A
Methylene Chloride	U	U	U	A	U	U	U	U	U	U	B
Mercuric Chloride	U	U	U	A	U	U	U	U	U	U	U
Mercury	U	A	U	A	A	B	B	A	A	A	A
Muriatic Acid	U	U	U	A	U	U	U	U	U	U	U
Nitric Acid-Diluted	U	A	U	A	U	U	U	A	A	A	U
Nitric Acid-Concentrated	A	A	U	A	U	U	U	A	A	A	U
Nitrous Acid	B	A	B	—	B	—	—	A	A	A	—
Nitrous Oxide	A	A	A	A	U	A	U	—	—	—	B
Oleic Acid	A	A	A	A	A	B	B	A	A	B	B
Oxalic Acid	B	A	A	A	B	B	B	A	A	—	U
Petroleum Oils-Crude	A	A	U	A	A	A	A	A	A	A	A
Phosphoric Acid	U	A	B	A	B	B	B	A	A	B	U
Picric Acid	A	A	C	A	U	U	U	A	A	A	A
Potassium Bromide	B	A	A	A	B	A	B	B	A	—	B
Potassium Carbonate	B	A	A	A	B	B	B	A	A	A	B
Potassium Chloride	B	A	B	A	B	B	B	A	A	A	A
Potassium Cyanide	U	A	U	A	B	B	B	A	A	A	A
Potassium Hydroxide	U	A	U	A	B	A	A	B	A	—	B
Potassium Sulphate	A	A	A	—	B	B	B	A	A	—	A
Sea Water	B	A	B	A	B	B	B	A	A	U	B
Sewage	B	—	B	—	—	—	—	A	A	—	B
Silver Nitrate	U	A	U	B	B	U	U	A	A	A	U
Soaps	B	A	B	A	A	A	A	A	A	A	A
Sodium Bicarbonate	B	A	A	A	A	A	A	A	A	A	B
Sodium Bisulphate	B	A	B	A	B	B	B	A	A	—	U
Sodium Bromide	B	A	A	A	B	B	B	A	A	—	B
Sodium Carbonate	B	A	A	A	B	B	B	A	A	A	A
Sodium Chloride	B	A	A	A	A	A	A	A	A	—	A
Sodium Hydroxide	U	A	B	A	A	A	A	A	A	A	A
Sodium Hyperchlorite	U	B	U	A	U	U	U	B	A	A	U
Sodium Nitrate	A	A	A	A	A	B	B	A	A	A	A
Sodium Peroxide	A	A	B	A	B	B	B	A	A	—	B
Sodium Phosphate	A	A	A	A	B	B	B	A	A	—	B
Sodium Silicate	B	A	A	A	—	B	—	A	A	A	A
Sodium Sulphate	A	A	A	A	B	B	B	A	A	A	A
Sodium Sulphide	U	A	U	A	B	—	B	B	—	A	A
Soy Bean Oil	A	A	A	A	—	B	—	A	A	—	—
Steam	A	A	B	—	A	A	A	A	A	A	A
Stearic Acid	A	A	A	A	B	B	B	A	A	A	B
Stannic Chloride	U	A	U	A	B	B	B	A	A	U	—
Sulphur Chloride	U	A	A	A	—	B	B	U	U	U	B
Sulphur Dioxide-Dry	A	A	A	A	A	A	A	A	A	—	A
Sulphuric Acid—<10%-Cold	B	A	B	A	U	B	B	U	B	U	U
Sulphuric Acid—<10%-Hot	U	B	U	A	U	B	U	U	U	U	U
Sulphuric Acid—10-50%-Cold	U	A	U	A	U	B	U	U	U	—	U
Sulphuric Acid—10-50%-Hot	U	U	U	A	U	U	U	U	U	U	U
Sulphuric Acid-Fuming	A	A	U	A	U	U	U	A	A	—	B
Sulphurous Acid	B	A	U	A	U	U	U	U	B	U	A
Sulphur-Molten	A	A	U	A	A	U	U	A	A	A	A
Tannic Acid	B	A	A	A	B	B	B	A	A	A	U
Tartaric Acid	B	A	A	A	B	B	B	A	A	U	U
Vinegar	B	A	B	A	A	A	A	A	A	A	B
Zinc Chloride	U	A	B	B	B	B	B	U	U	U	B
Zinc Sulphate	B	A	A	A	B	B	B	A	A	A	B

METALS

SUGGESTED MAXIMUM SERVICE TEMPERATURES IN AIR

TYPE	CONTINUOUS SERVICE	
	°C	°F
Carbon Steel	538	1000
304 S.S.	925	1700
309 S.S.	1095	2000
310 S.S.	1150	2100
316 S.S.	925	1700
321 S.S.	925	1700
347 S.S.	925	1700
410 S.S.	705	1300
430 S.S.	815	1500
501 S.S.	649	1200
Alloy 20	815	1500
Aluminum	427	800
Brass	260	500
Copper	260	500
Hastelloy B & C®	1095	2000
Inconel 600®	1095	2000
Incolloy 800®	871	1600
Monel®	815	1500
Nickel	760	1400
Phosphor Bronze	260	500
Tantalum	1649	3000
Titanium	1095	2000

Note: Maximum temperature ratings are based upon hot air constant temperatures. The presence of contaminating fluids and cyclic conditions may drastically affect the maximum temperature range.

CHEMICAL RESISTANCE CHART

VEGETABLE FIBER SHEET — GLUE-GLYCERIN BINDER

Vegetable fiber sheet is a tough, pliable and compressible protein bonded sheet that is suitable for the services listed below to a maximum temperature limit of 250°F. For unusual concentrations, pressures or temperatures, further investigation is indicated.

Suitable for use with:

Acetone	Fuel Oil	Propylene Glycol
Alcohol	Gas Illuminating	Pyranol A13B3B
Animal Fats & Oils	Gasoline	Skydrol 500B
Benzene (Benzol)	Greases	Skydrol 7000 Abs.
Benzine (Gasoline)	Hydrogen	Soap
Bunker Oil	Hydrogen Sulphide	Sperry Oil
Butane	Inerteen 70-30	Sulphur Dioxide
Butyl Acetate	Inks	Super VM&P Naphtha
Carbon Dioxide	Kerosene	Toluol
Carbon Tetrachloride	Lacquers and Thinners	Transformer Oil
Cresol	Lubricating Oil	Trichloroethylene
Dibutyl Phthalate	Methyl Chloride (Refrigerant)	Tricresyl Phosphate
DOP (Diocetyl Phthalate)	Methyl Ethyl Ketone (MEK)	Triethylene Glycol (Neutral Grade)
Dry Cleaning Fluid	Methyl Isobutyl Detone (MIBK)	Turpentine
Ether	Naphtha, Petroleum	Varnish
Ethyl Acetate	Naphtha, Coal Tar	Vegetable Oil
Ethylene Glycol	Paints	Water
Formaldehyde	Petroleum	Wood Alcohol
Freon	Prestone (Antifreeze)	Xylol

Not suitable for use with:

Acids (Inorganic)	Nitro Benzine
Alkalies	Oxygen
Hydrochloric Acid	Silicate of Soda
Nitric Acid (Dilute)	Sulphuric Acid (Dilute)

COMPRESSED ASBESTOS SHEET

CHEMICAL RESISTANCE CHART

A — Suitable
 B — Depends upon operating conditions
 C — Not suitable

Medium	White Canadian Asbestos (Chrysotile)/NBR Binder	White Canadian Asbestos (Chrysotile)/Neo-prene Binder	Blue (Crocidolite) SBR Binder	White Canadian Asbestos (Chrysotile)/SBR Binder
Acetaldehyde	B	C	A	A
Acetic Acid 100%	A	A	A	A
Acetic Acid 10%	A	A	A	A
Acetic Ether	A	A	B	B
Acetone	B	B	A	A
Acetylene	A	A	A	A
Adipic Acid	A	A	A	A
Air	A	A	A	A
Alum	A	A	A	A
Aluminum Acetate	A	A	B	A
Aluminum Chloride	A	A	A	A
Ammonia	A	A	A	A
Ammonium Bicarbonate	A	A	A	A
Ammonium Chloride	A	A	A	A
Ammonium Diphosphate	A	A	A	A
Ammonium Hydroxide	A	A	A	A
Amyl Acetate	A	A	B	B
Aniline	C	C	C	A
Aviation Fuels	A	A	C	C
Barium Chloride	A	A	A	A
Benzene	A	A	C	C
Benzoic Acid	A	A	A	A
Bleach Solutions	A	A	A	A
Boiler Feed Water	A	A	A	B
Borax	A	A	A	A
Boric Acid	A	A	A	A
Butane	A	A	B	C
Butanone (M.E.K.)	B	B	B	B
Butyl Acetate	A	A	B	B
Butyl Alcohol (Butanol)	A	A	A	A
Butyric Acid	A	A	A	A
Calcium Chloride	A	A	A	A
Calcium Hydroxide	A	A	A	A
Calcium Hypochlorite	A	A	A	A
Carbon Dioxide	A	A	A	A
Carbon Disulphide	A	A	C	C
Carbon Tetrachloride	A	B	C	C
Castor Oil	A	A	A	B
Chlorine (Dry)	A	A	B	B
Chlorine (Wet)	B	B	B	C
Chloroform	B	B	C	C
Chromic Acid	B	B	B	C
Citric Acid	A	A	A	A
Clophen T .64	A	A	C	C
Condensate	A	A	A	B
Copper Sulphate	A	A	A	A
Creosote	B	C	C	B
Cresol	B	B	C	B
Cyclohexanol	A	A	B	B

COMPRESSED ASBESTOS SHEET (CONT.)

A — Suitable
B — Depends upon operating conditions
C — Not suitable

Medium	White Canadian Asbestos (Chrysotile)/NBR Binder	White Canadian Asbestos (Chrysotile)/Neo-prene Binder	Blue (Crocidolite) SBR Binder	White Canadian Asbestos (Chrysotile)/SBR Binder
Decalin	A	B	C	C
Dibenzylether	B	C	C	C
Dibutylphthalate	A	A	C	C
Diesel Oil	A	A	C	C
Dimethylformamide	C	B	C	B
Diphyl, Dowtherm A	A	B	C	C
Dye Liquors (alkaline, neutral, acid)	A	A	B	A
Ethane	A	A	B	A
Ethyl Acetate	A	A	C	B
Ethyl Alcohol, Ethanol	A	A	A	A
Ethyl Chloride	B	B	C	C
Ethylene	A	A	A	A
Ethylene Chloride	B	B	C	B
Ethylene Glycol	A	A	A	A
Ethyl Ether	A	A	C	A
Freon 12	A	A	C	C
Freon 22	B	A	C	C
Formaldehyde	A	A	A	A
Formic Acid 10%	A	A	A	A
Formic Acid 85%	A	A	A	B
Glycerine	A	A	A	A
Heating Oil	A	A	C	C
Heptane	A	A	C	C
Hydraulic Oil (Glycol Based)	A	A	A	A
Hydraulic Oil (Mineral)	A	A	C	C
Hydraulic Oil (Phosphate Ester Based)	B	B	B	B
Hydrochloric Acid 20%	B	B	B	C
Hydrochloric Acid 37%	B	B	B	C
Hydrofluoric Acid 10%	C	C	B	C
Hydrofluoric Acid 40%	C	C	B	C
Hydrogen	A	A	A	A
Hydrogen Peroxide (up to 6% w/w)	A	A	A	A
Hydrogen Chloride (Dry)	A	A	A	A
Iso-octane	A	A	B	B
Iso-propyl Alcohol	A	A	A	A
Kerosene	A	A	C	C
Lactic Acid 50%	A	A	A	A
Linseed Oil	A	A	B	B
Magnesium Sulphate	A	A	A	A
Mallic Acid	A	A	A	A
Methane	A	A	A	A
Methyl Alcohol	A	A	A	A
Methyl Chloride	B	B	B	C
Methylene Chloride	C	C	C	C
Methyl Ethyl Ketone	B	B	B	B
Mineral Oil	A	A	C	C
Mineral Oil Type ASTM 1	A	A	C	C
Mineral Oil Type ASTM III	A	A	C	C
Monochlor Methane	B	B	C	C
Nitrogen	A	A	A	A
Naphtha	A	A	C	C
Nitric Acid 20%	B	B	B	C
Nitric Acid 40%	B	B	B	C
Nitric Acid 96%	C	C	B	C

COMPRESSED ASBESTOS SHEET (CONT.)

A — Suitable B — Depends upon operating conditions C — Not suitable

Medium	White Canadian Asbestos (Chrysotile)/NBR Binder	White Canadian Asbestos (Chrysotile)/Neo- prene Binder	Blue (Crocidolite) SBR Binder	White Canadian Asbestos (Chrysotile)/SBR Binder
Octane	A	A	B	C
Oleic Acid	A	A	B	B
Oxalic Acid	B	B	A	C
Palmitic Acid	A	A	B	B
Pentane	A	A	C	C
Perchloroethylene	B	B	C	C
Petroleum	A	A	C	C
Petroleum Ether	A	A	B	B
Phenol	B	B	B	B
Phosphoric Acid (all concentrations)	A	A	A	B
Phthalic Acid	A	A	A	A
Potassium Acetate	A	A	A	A
Potassium Carbonate	A	A	A	A
Potassium Chlorate	A	A	A	A
Potassium Chloride	A	A	A	A
Potassium Cyanide	A	A	A	A
Potassium Dichromate	A	A	A	A
Potassium Hydroxide	A	A	B	B
Potassium Hypochlorite	A	A	A	A
Potassium Nitrate	A	A	A	A
Potassium Permanganate	A	A	A	A
Producer Gas	A	A	A	A
Propane	A	A	B	B
Pydrol	A	B	C	C
Pyridine	C	C	C	B
Rape Seed Oil	A	A	B	B
Silicone Oil	A	A	A	A
Sea Water	A	A	A	A
Sodium Aluminate	A	A	A	A
Sodium Bicarbonate	A	A	A	A
Sodium Bisulphite	A	A	A	A
Sodium Chloride	A	A	A	A
Sodium Hydroxide	A	A	B	B
Sodium Silicate	A	A	A	A
Sodium Sulphate	A	A	A	A
Sodium Sulphide	A	A	A	A
Steam	A	A	C	A
Stearic Acid	A	A	B	B
Sulphur Dioxide	B	B	B	C
Sulphuric Acid 20%	B	B	B	C
Sulphuric Acid 50%	B	B	B	C
Sulphuric Acid 96%	B	B	C	C
Sulphurous Acid	B	B	A	C
Tannic Acid	A	A	A	A
Tartaric Acid	A	A	A	A
Tetrachlorethane	B	B	C	C
Tetralin	A	A	C	C
Toluene	A	A	B	C
Town Gas	A	A	B	B
Transformer Oil	A	A	B	B
Trichlor Ethylene	B	B	B	C
Turpentine	A	A	B	C
Vinyl Acetate	A	A	C	C
Water	A	A	A	A
White Spirit	A	A	C	C
Xylene	A	A	C	C

CIRCUMFERENCES AND AREAS OF CIRCLES

Diam.	Circ.	Area	Diam.	Circ.	Area	Diam.	Circ.	Area	Diam.	Circ.	Area	Diam.	Circ.	Area
$\frac{1}{32}$	.0981	.00076	8	25.13	50.265	17	53.40	226.98	26	81.68	530.93	35	109.9	962.11
$\frac{1}{16}$	.1963	.00306	$\frac{1}{8}$	25.52	51.848	$\frac{1}{8}$	53.79	230.33	$\frac{1}{8}$	82.07	536.04	$\frac{1}{8}$	110.3	968.99
$\frac{1}{8}$	.3926	.01227	$\frac{1}{4}$	25.91	53.456	$\frac{1}{4}$	54.19	233.70	$\frac{1}{4}$	82.46	541.18	$\frac{1}{4}$	110.7	975.90
$\frac{3}{16}$	.5890	.02761	$\frac{3}{8}$	26.31	55.088	$\frac{3}{8}$	54.58	237.10	$\frac{3}{8}$	82.85	546.35	$\frac{3}{8}$	111.1	982.84
$\frac{1}{4}$	.7854	.04908	$\frac{1}{2}$	26.70	56.745	$\frac{1}{2}$	54.97	240.52	$\frac{1}{2}$	83.25	551.54	$\frac{1}{2}$	111.5	989.80
$\frac{5}{16}$	.9817	.07669	$\frac{5}{8}$	27.09	58.426	$\frac{5}{8}$	55.37	243.97	$\frac{5}{8}$	83.64	556.76	$\frac{5}{8}$	111.9	996.78
$\frac{3}{8}$	1.178	.1104	$\frac{3}{4}$	27.47	60.132	$\frac{3}{4}$	55.76	247.45	$\frac{3}{4}$	84.03	562.00	$\frac{3}{4}$	112.3	1003.7
$\frac{7}{16}$	1.374	.1503	$\frac{7}{8}$	27.88	61.862	$\frac{7}{8}$	56.16	250.94	$\frac{7}{8}$	84.43	567.26	$\frac{7}{8}$	112.7	1010.8
$\frac{1}{2}$	1.570	.1963	9	28.27	63.617	18	56.54	254.46	27	84.82	572.55	36	113.0	1017.8
$\frac{9}{16}$	1.767	.2485	$\frac{1}{8}$	28.66	65.396	$\frac{1}{8}$	56.94	258.01	$\frac{1}{8}$	85.21	577.87	$\frac{1}{8}$	113.4	1024.9
$\frac{5}{8}$	1.963	.3097	$\frac{1}{4}$	29.05	67.200	$\frac{1}{4}$	57.33	261.58	$\frac{1}{4}$	85.60	583.20	$\frac{1}{4}$	113.8	1032.0
$\frac{11}{16}$	2.159	.3712	$\frac{3}{8}$	29.45	69.029	$\frac{3}{8}$	57.72	265.18	$\frac{3}{8}$	86.00	588.57	$\frac{3}{8}$	114.2	1039.1
$\frac{3}{4}$	2.356	.4417	$\frac{1}{2}$	29.84	70.882	$\frac{1}{2}$	58.11	268.80	$\frac{1}{2}$	86.39	593.95	$\frac{1}{2}$	114.6	1046.3
$\frac{13}{16}$	2.552	.5184	$\frac{5}{8}$	30.23	72.759	$\frac{5}{8}$	58.51	272.44	$\frac{5}{8}$	86.78	599.37	$\frac{5}{8}$	115.0	1053.5
$\frac{7}{8}$	2.748	.6013	$\frac{3}{4}$	30.63	74.662	$\frac{3}{4}$	58.90	276.11	$\frac{3}{4}$	87.17	604.80	$\frac{3}{4}$	115.4	1060.7
$\frac{15}{16}$	2.945	.6902	$\frac{7}{8}$	31.02	76.588	$\frac{7}{8}$	59.29	279.81	$\frac{7}{8}$	87.57	610.26	$\frac{7}{8}$	115.8	1067.9
1	3.141	.7854	10	31.41	78.539	19	59.69	283.52	28	87.96	615.75	37	116.2	1075.2
$\frac{1}{8}$	3.534	.9940	$\frac{1}{8}$	31.80	80.515	$\frac{1}{8}$	60.08	287.27	$\frac{1}{8}$	88.35	621.26	$\frac{1}{8}$	116.6	1082.4
$\frac{1}{4}$	3.927	1.227	$\frac{1}{4}$	32.20	82.516	$\frac{1}{4}$	60.47	291.03	$\frac{1}{4}$	88.75	626.79	$\frac{1}{4}$	117.0	1089.7
$\frac{3}{8}$	4.319	1.484	$\frac{3}{8}$	32.59	84.540	$\frac{3}{8}$	60.86	294.83	$\frac{3}{8}$	89.14	632.35	$\frac{3}{8}$	117.4	1097.1
$\frac{1}{2}$	4.712	1.767	$\frac{1}{2}$	32.98	86.590	$\frac{1}{2}$	61.26	298.64	$\frac{1}{2}$	89.53	637.94	$\frac{1}{2}$	117.8	1104.4
$\frac{5}{8}$	5.105	2.073	$\frac{5}{8}$	33.37	88.664	$\frac{5}{8}$	61.65	302.48	$\frac{5}{8}$	89.92	643.54	$\frac{5}{8}$	118.2	1111.8
$\frac{3}{4}$	5.497	2.405	$\frac{3}{4}$	33.77	90.762	$\frac{3}{4}$	62.04	306.35	$\frac{3}{4}$	90.32	649.18	$\frac{3}{4}$	118.6	1119.2
$\frac{7}{8}$	5.890	2.761	$\frac{7}{8}$	34.16	92.885	$\frac{7}{8}$	62.43	310.24	$\frac{7}{8}$	90.71	654.83	$\frac{7}{8}$	118.9	1126.6
2	6.283	3.141	11	34.55	95.033	20	62.83	314.16	29	91.10	660.52	38	119.3	1134.1
$\frac{1}{8}$	6.675	3.546	$\frac{1}{8}$	34.95	97.205	$\frac{1}{8}$	63.22	318.09	$\frac{1}{8}$	91.49	666.22	$\frac{1}{8}$	119.7	1141.5
$\frac{1}{4}$	7.068	3.976	$\frac{1}{4}$	35.34	99.402	$\frac{1}{4}$	63.61	322.06	$\frac{1}{4}$	91.89	671.95	$\frac{1}{4}$	120.1	1149.0
$\frac{3}{8}$	7.461	4.430	$\frac{3}{8}$	35.73	101.62	$\frac{3}{8}$	64.01	326.05	$\frac{3}{8}$	92.23	677.71	$\frac{3}{8}$	120.5	1156.6
$\frac{1}{2}$	7.854	4.908	$\frac{1}{2}$	36.12	103.86	$\frac{1}{2}$	64.40	330.06	$\frac{1}{2}$	92.67	683.49	$\frac{1}{2}$	120.9	1164.1
$\frac{5}{8}$	8.246	5.411	$\frac{5}{8}$	36.52	106.13	$\frac{5}{8}$	64.79	334.10	$\frac{5}{8}$	93.06	689.29	$\frac{5}{8}$	121.3	1171.7
$\frac{3}{4}$	8.639	5.939	$\frac{3}{4}$	36.91	108.43	$\frac{3}{4}$	65.18	338.16	$\frac{3}{4}$	93.46	695.12	$\frac{3}{4}$	121.7	1179.3
$\frac{7}{8}$	9.032	6.491	$\frac{7}{8}$	37.30	110.75	$\frac{7}{8}$	65.58	342.25	$\frac{7}{8}$	93.85	700.98	$\frac{7}{8}$	122.1	1186.9
3	9.424	7.068	12	37.69	113.00	21	65.97	346.36	30	94.24	706.86	39	122.5	1194.5
$\frac{1}{8}$	9.817	7.669	$\frac{1}{8}$	38.09	115.46	$\frac{1}{8}$	66.36	350.49	$\frac{1}{8}$	94.64	712.76	$\frac{1}{8}$	122.9	1202.2
$\frac{1}{4}$	10.21	8.295	$\frac{1}{4}$	38.48	117.85	$\frac{1}{4}$	66.75	354.65	$\frac{1}{4}$	95.03	718.69	$\frac{1}{4}$	123.3	1209.9
$\frac{3}{8}$	10.60	8.946	$\frac{3}{8}$	38.87	120.27	$\frac{3}{8}$	67.15	358.84	$\frac{3}{8}$	95.42	724.64	$\frac{3}{8}$	123.7	1217.6
$\frac{1}{2}$	10.99	9.621	$\frac{1}{2}$	39.27	122.71	$\frac{1}{2}$	67.54	363.05	$\frac{1}{2}$	95.81	730.61	$\frac{1}{2}$	124.0	1225.4
$\frac{5}{8}$	11.38	10.320	$\frac{5}{8}$	39.66	125.18	$\frac{5}{8}$	67.93	367.28	$\frac{5}{8}$	96.21	736.61	$\frac{5}{8}$	124.4	1233.1
$\frac{3}{4}$	11.78	11.044	$\frac{3}{4}$	40.05	127.67	$\frac{3}{4}$	68.32	371.54	$\frac{3}{4}$	96.60	742.64	$\frac{3}{4}$	124.8	1240.9
$\frac{7}{8}$	12.17	11.793	$\frac{7}{8}$	40.44	130.19	$\frac{7}{8}$	68.72	375.82	$\frac{7}{8}$	96.99	748.69	$\frac{7}{8}$	125.2	1248.7
4	12.65	12.566	13	40.84	132.73	22	69.11	380.13	31	97.38	754.76	40	125.6	1256.6
$\frac{1}{8}$	12.95	13.364	$\frac{1}{8}$	41.23	135.29	$\frac{1}{8}$	69.50	384.46	$\frac{1}{8}$	97.78	760.86	$\frac{1}{8}$	126.0	1264.5
$\frac{1}{4}$	13.35	14.186	$\frac{1}{4}$	41.62	137.88	$\frac{1}{4}$	69.90	388.82	$\frac{1}{4}$	98.17	766.99	$\frac{1}{4}$	126.4	1272.3
$\frac{3}{8}$	13.74	15.033	$\frac{3}{8}$	42.01	140.50	$\frac{3}{8}$	70.29	393.20	$\frac{3}{8}$	98.56	773.14	$\frac{3}{8}$	126.8	1280.3
$\frac{1}{2}$	14.13	15.904	$\frac{1}{2}$	42.41	143.13	$\frac{1}{2}$	70.68	397.60	$\frac{1}{2}$	98.96	779.31	$\frac{1}{2}$	127.2	1288.2
$\frac{5}{8}$	14.52	16.800	$\frac{5}{8}$	42.80	145.80	$\frac{5}{8}$	71.07	402.03	$\frac{5}{8}$	99.35	785.51	$\frac{5}{8}$	127.6	1291.2
$\frac{3}{4}$	14.92	17.720	$\frac{3}{4}$	43.19	148.48	$\frac{3}{4}$	71.47	406.49	$\frac{3}{4}$	99.74	791.73	$\frac{3}{4}$	128.0	1304.2
$\frac{7}{8}$	15.31	18.665	$\frac{7}{8}$	43.58	151.20	$\frac{7}{8}$	71.86	410.97	$\frac{7}{8}$	100.1	797.97	$\frac{7}{8}$	128.4	1312.2
5	15.70	19.635	14	43.98	153.92	23	72.25	415.47	32	100.5	804.24	41	128.8	1320.2
$\frac{1}{8}$	16.10	20.629	$\frac{1}{8}$	44.37	156.69	$\frac{1}{8}$	72.64	420.00	$\frac{1}{8}$	100.9	810.45	$\frac{1}{8}$	129.1	1328.3
$\frac{1}{4}$	16.49	21.647	$\frac{1}{4}$	44.76	159.48	$\frac{1}{4}$	73.04	424.55	$\frac{1}{4}$	101.3	816.86	$\frac{1}{4}$	129.5	1336.4
$\frac{3}{8}$	16.88	22.690	$\frac{3}{8}$	45.16	162.29	$\frac{3}{8}$	73.43	429.13	$\frac{3}{8}$	101.7	823.21	$\frac{3}{8}$	129.9	1344.5
$\frac{1}{2}$	17.27	23.758	$\frac{1}{2}$	45.55	165.13	$\frac{1}{2}$	73.82	433.73	$\frac{1}{2}$	102.1	829.57	$\frac{1}{2}$	130.3	1352.6
$\frac{5}{8}$	17.57	24.850	$\frac{5}{8}$	45.94	167.98	$\frac{5}{8}$	74.21	438.30	$\frac{5}{8}$	102.4	835.97	$\frac{5}{8}$	130.7	1360.8
$\frac{3}{4}$	18.06	25.967	$\frac{3}{4}$	46.33	170.87	$\frac{3}{4}$	74.61	443.01	$\frac{3}{4}$	102.8	842.39	$\frac{3}{4}$	131.1	1369.0
$\frac{7}{8}$	18.45	27.108	$\frac{7}{8}$	46.73	173.78	$\frac{7}{8}$	75.00	447.69	$\frac{7}{8}$	103.2	848.83	$\frac{7}{8}$	131.5	1377.2
6	18.84	28.274	15	47.12	176.71	24	75.39	452.39	33	103.6	855.30	42	131.9	1385.4
$\frac{1}{8}$	19.24	29.464	$\frac{1}{8}$	47.51	179.67	$\frac{1}{8}$	75.79	457.11	$\frac{1}{8}$	104.0	861.79	$\frac{1}{8}$	132.3	1393.7
$\frac{1}{4}$	19.63	30.679	$\frac{1}{4}$	47.90	182.72	$\frac{1}{4}$	76.18	461.86	$\frac{1}{4}$	104.4	868.30	$\frac{1}{4}$	132.7	1401.9
$\frac{3}{8}$	20.02	31.919	$\frac{3}{8}$	48.30	185.66	$\frac{3}{8}$	76.57	466.63	$\frac{3}{8}$	104.8	874.88	$\frac{3}{8}$	133.1	1410.2
$\frac{1}{2}$	20.42	33.183	$\frac{1}{2}$	48.69	188.69	$\frac{1}{2}$	76.96	471.43	$\frac{1}{2}$	105.2	881.41	$\frac{1}{2}$	133.5	1418.6
$\frac{5}{8}$	20.81	34.471	$\frac{5}{8}$	49.08	191.74	$\frac{5}{8}$	77.36	476.25	$\frac{5}{8}$	105.6	888.00	$\frac{5}{8}$	133.9	1426.9
$\frac{3}{4}$	21.20	35.784	$\frac{3}{4}$	49.48	194.82	$\frac{3}{4}$	77.75	481.10	$\frac{3}{4}$	106.0	894.61	$\frac{3}{4}$	134.3	1435.3
$\frac{7}{8}$	21.57	37.122	$\frac{7}{8}$	49.87	197.73	$\frac{7}{8}$	78.14	485.97	$\frac{7}{8}$	106.4	901.25	$\frac{7}{8}$	134.6	1443.7
7	21.90	38.484	16	50.26	201.06	25	78.54	490.87	34	106.8	907.92	43	135.0	1452.2
$\frac{1}{8}$	22.38	39.871	$\frac{1}{8}$	50.65	204.21	$\frac{1}{8}$	78.93	495.79	$\frac{1}{8}$	107.2	914.61	$\frac{1}{8}$	135.4	1460.6
$\frac{1}{4}$	22.77	41.282	$\frac{1}{4}$	51.05	207.39	$\frac{1}{4}$	79.32	500.74	$\frac{1}{4}$	107.5	921.32	$\frac{1}{4}$	135.8	1469.1
$\frac{3}{8}$	23.16	42.718	$\frac{3}{8}$	51.44	210.59	$\frac{3}{8}$	79.71	505.71	$\frac{3}{8}$	107.9	928.06	$\frac{3}{8}$	136.2	1477.6
$\frac{1}{2}$	23.56	44.178	$\frac{1}{2}$	51.83	213.82	$\frac{1}{2}$	80.10	510.70	$\frac{1}{2}$	108.3	934.82	$\frac{1}{2}$	136.6	1486.1
$\frac{5}{8}$	23.95	45.663	$\frac{5}{8}$	52.22	217.07	$\frac{5}{8}$	80.50	515.72	$\frac{5}{8}$	108.7	94			

CIRCUMFERENCES AND AREAS OF CIRCLES (CONT.)

Diam.	Circ.	Area	Diam.	Circ.	Area	Diam.	Circ.	Area	Diam.	Circ.	Area	Diam.	Circ.	Area
44	138.2	1520.5	53	166.5	2206.1	62	194.7	3019.0	71	223.0	3959.2	84	263.8	5541.7
1/8	138.6	1529.1	1/8	166.8	2216.6	1/8	195.1	3031.2	1/8	223.4	2973.1	1/4	264.6	5574.8
1/4	139.0	1537.8	1/4	167.2	2227.0	1/4	195.5	3043.4	1/4	223.8	3987.1	1/2	265.4	5607.9
3/8	139.4	1546.5	3/8	167.6	2237.5	3/8	195.9	3055.7	3/8	224.2	4001.1	3/4	226.2	5641.1
1/2	139.8	1555.2	1/2	168.0	2248.0	1/2	196.3	3067.9	1/2	224.6	4015.1			
5/8	140.1	1564.0	5/8	168.4	2258.5	5/8	196.7	3080.2	5/8	225.0	4029.2	85	267.0	5674.5
3/4	140.5	1572.8	3/4	168.8	2269.0	3/4	197.1	3092.5	3/4	225.4	4043.2	1/4	267.8	5707.9
7/8	140.9	1581.6	7/8	169.2	2279.6	7/8	197.5	3104.8	7/8	255.8	4067.3	1/2	268.6	5741.4
												3/4	269.3	5775.0
45	141.3	1590.4	54	169.6	2290.2	63	197.9	3117.2	72	226.1	4071.5	86	270.1	5808.8
1/8	141.7	1599.2	1/8	170.0	2300.8	1/8	198.3	3129.6	1/8	226.5	4085.6	1/4	270.9	5842.6
1/4	142.1	1608.1	1/4	170.4	2311.4	1/4	198.7	3142.0	1/4	226.9	4099.8	1/2	271.7	5876.5
3/8	142.5	1617.0	3/8	170.8	2322.1	3/8	199.0	3144.4	3/8	227.3	4114.0	3/4	272.5	5910.5
1/2	142.9	1625.9	1/2	171.2	2332.8	1/2	199.4	3166.9	1/2	227.7	4128.2			
5/8	143.3	1634.9	5/8	171.6	2343.5	5/8	199.8	3179.4	5/8	228.1	4142.5	87	273.3	5944.6
3/4	143.7	1643.8	3/4	172.0	2354.2	3/4	200.2	3191.9	3/4	228.5	4156.7	1/4	274.1	5978.9
7/8	144.1	1652.8	7/8	172.3	2365.0	7/8	200.6	3204.4	7/8	228.9	4171.0	1/2	274.8	6013.2
												3/4	275.6	6047.6
46	144.5	1661.9	55	172.7	2375.8	64	201.0	3216.9	73	229.3	4185.3	88	276.4	6082.1
1/8	144.9	1670.9	1/8	173.1	2386.6	1/8	201.4	3229.5	1/8	229.7	4199.7	1/4	277.2	6116.7
1/4	145.2	1680.0	1/4	173.5	2397.4	1/4	201.8	3242.1	1/4	230.1	4214.1	1/2	278.0	6151.4
3/8	145.6	1689.1	3/8	173.9	2408.3	3/8	202.2	3254.8	3/8	230.5	4228.5	3/4	278.8	6186.2
1/2	146.0	1698.2	1/2	174.3	2419.2	1/2	202.6	3267.4	1/2	230.9	4242.9			
5/8	146.4	1707.3	5/8	174.7	2430.1	5/8	203.0	3280.1	5/8	231.3	4257.3	89	279.6	6221.1
3/4	146.8	1716.5	3/4	175.1	2441.0	3/4	203.4	3292.8	3/4	231.6	4271.8	1/4	280.3	6256.1
7/8	147.2	1725.7	7/8	175.5	2452.0	7/8	203.8	3305.5	7/8	232.0	4286.3	1/2	281.1	6291.2
												3/4	281.9	6326.4
47	147.6	1734.9	56	175.9	2463.0	65	204.2	3318.3	74	232.4	4300.8	90	282.7	6361.7
1/8	148.0	1744.1	1/8	176.3	2474.0	1/8	204.5	3331.0	1/8	232.8	4315.3	1/4	283.5	6397.1
1/4	148.4	1753.4	1/4	176.7	2485.0	1/4	204.9	3343.8	1/4	233.2	4329.9	1/2	284.3	6432.6
3/8	148.8	1762.7	3/8	177.1	2496.1	3/8	205.3	3356.7	3/8	233.6	4344.5	3/4	285.1	6468.2
1/2	149.2	1772.0	1/2	177.5	2507.1	1/2	205.7	3369.5	1/2	234.0	4359.1			
5/8	149.6	1781.3	5/8	177.8	2518.2	5/8	206.1	3382.4	5/8	234.4	4373.8	91	285.8	6503.8
3/4	150.0	1790.7	3/4	178.2	2529.4	3/4	206.5	3395.3	3/4	234.8	4388.4	1/4	286.6	6539.6
7/8	150.4	1800.1	7/8	178.6	2540.5	7/8	206.9	3408.2	7/8	235.2	4403.1	1/2	287.4	6575.5
												3/4	288.2	6611.5
48	150.7	1809.5	57	179.0	2551.7	66	207.3	3421.2	75	235.6	4417.8	92	289.0	6647.6
1/8	151.1	1818.9	1/8	179.4	2562.9	1/8	207.7	3434.1	1/8	236.4	4447.3	1/4	289.8	6683.8
1/4	151.5	1828.4	1/4	179.8	2574.1	1/4	208.1	3447.1	1/4	237.1	4476.9	1/2	290.5	6720.0
3/8	151.9	1837.9	3/8	180.2	2585.4	3/8	208.5	3460.1	3/8	237.9	4506.6	3/4	291.3	6756.4
1/2	152.3	1847.4	1/2	180.6	2596.7	1/2	208.9	3473.2				93	292.1	6792.0
5/8	152.7	1856.9	5/8	181.0	2608.0	5/8	209.3	3486.3	76	238.7	4536.4	1/4	292.9	6829.4
3/4	153.1	1866.5	3/4	181.4	2619.3	3/4	209.7	3499.3	1/4	239.5	4566.3	1/2	293.7	6866.1
7/8	153.5	1876.1	7/8	181.8	2630.7	7/8	210.0	3512.5	1/2	240.3	4596.3	3/4	294.5	6902.9
									3/4	241.1	4626.4			
49	153.9	1885.7	58	182.2	2642.0	67	210.4	3525.6	77	241.9	4666.6	94	295.3	6939.7
1/8	154.3	1895.3	1/8	182.6	2653.4	1/8	210.9	3538.8	1/4	242.6	4686.9	1/4	296.0	6976.7
1/4	154.7	1905.0	1/4	182.9	2664.9	1/4	211.2	3552.0	1/2	243.4	4717.3	1/2	296.8	7013.8
3/8	155.1	1914.7	3/8	183.3	2676.3	3/8	211.6	3565.2	3/4	244.2	4747.7	3/4	297.6	7050.9
1/2	155.5	1924.4	1/2	183.7	2687.8	1/2	212.0	3578.4				95	298.4	7088.2
5/8	155.9	1934.1	5/8	184.1	2690.3	5/8	212.4	3591.7	1/4	245.0	4778.3	1/4	299.2	7125.5
3/4	156.2	1943.9	3/4	184.5	2710.8	3/4	212.8	3605.0	1/2	245.8	4809.0	1/2	300.0	7163.0
7/8	156.6	1953.6	7/8	184.9	2722.4	7/8	213.2	3618.3	3/4	246.6	4839.8	3/4	300.8	7200.5
									3/4	247.4	4870.7			
50	157.0	1963.5	59	185.3	2733.9	68	213.6	3631.6	79	248.1	4901.6	96	301.5	7238.2
1/8	157.4	1973.3	1/8	185.7	2745.5	1/8	214.0	3645.0	1/4	248.9	4932.7	1/4	302.3	7275.9
1/4	157.8	1983.1	1/4	186.1	2757.1	1/4	214.4	3658.4	1/2	249.7	4963.9	1/2	303.1	7313.8
3/8	158.2	1993.0	3/8	186.5	2768.8	3/8	214.8	3671.8	3/4	250.5	4995.1	3/4	303.9	7341.7
1/2	158.6	2002.9	1/2	186.9	2780.5	1/2	215.1	3685.2				97	304.7	7389.8
5/8	159.0	2012.8	5/8	187.3	2792.2	5/8	215.5	3698.7	1/4	251.3	5026.5	1/4	305.5	7427.9
3/4	159.4	2022.8	3/4	187.7	2803.9	3/4	215.9	3712.2	1/2	252.1	5058.0	1/2	306.3	7466.2
7/8	159.8	2032.8	7/8	188.1	2815.6	7/8	216.3	3725.7	3/4	252.8	5089.5	3/4	307.0	7504.4
									3/4	253.6	5121.2			
51	160.2	2042.8	60	188.4	2827.4	69	216.7	3739.2	81	254.4	5153.0	98	307.8	7542.9
1/8	160.6	2052.8	1/8	188.8	2839.2	1/8	217.1	3752.8	1/4	255.2	5184.8	1/4	308.6	7581.5
1/4	161.0	2062.9	1/4	189.2	2851.0	1/4	217.5	3766.4	1/2	256.0	5216.8	1/2	309.4	7620.1
3/8	161.3	2072.9	3/8	189.6	2862.8	3/8	217.9	3780.0	3/4	256.8	5248.8	3/4	310.2	7658.8
1/2	161.7	2083.0	1/2	190.0	2874.7	1/2	218.3	3793.6				99	311.0	7697.7
5/8	162.1	2093.2	5/8	190.4	2886.6	5/8	218.7	3807.3	1/4	257.6	5281.0	1/4	311.8	7736.6
3/4	162.5	2103.3	3/4	190.8	2898.5	3/4	219.1	3821.0	1/2	258.3	5313.2	1/2	312.5	7775.6
7/8	162.9	2113.5	7/8	191.2	2910.5	7/8	219.5	3834.7	3/4	259.1	5345.6	3/4	313.3	7814.7
									3/4	259.9	5378.0			
52	163.3	2123.7	61	191.6	2922.4	70	219.9	3848.4	82	257.6	5281.0	100	314.1	7853.0
1/8	163.7	2133.9	1/8	192.0	2934.4	1/8	220.3	3862.2	1/4	258.3	5313.2	1/4	314.9	7893.3
1/4	164.1	2144.1	1/4	192.4	2946.4	1/4	220.6	3875.9	1/2	259.1	5345.6	1/2	315.7	7932.7
3/8	164.5	2154.4	3/8	192.8	2958.5	3/8	221.0	3889.8	3/4	260.7	5410.6	3/4	316.4	7972.2
1/2	164.9	2164.7	1/2	193.2	2970.5	1/2	221.4	3903.6	1/4	261.5	5443.2			
5/8	165.3	2175.0	5/8	193.6	2982.6	5/8	221.8	3917.4	1/2	262.3	5476.0			
3/4	165.7	2185.4	3/4	193.9	2994.6	3/4	222.2	3931.3	3/4	263.1	5508.8			
7/8	166.1	2195.7	7/8	194.3	3006.9	7/8	222.6	3945.2						

TORQUE REQUIRED TO PRODUCE BOLT STRESS

The torque or turning effort required to produce a certain stress in bolting is dependent upon a number of conditions, some of which are:

1. Diameter of bolt.
2. Type and number of threads on bolt.
3. Material of bolt.
4. Condition of nut bearing surfaces.
5. Lubrication of bolt threads and nut bearing surfaces.

The tables below reflect the results of many tests to determine the relation between torque and bolt stress. Values are based on steel bolting well lubricated with a heavy graphite and oil mixture.

It was found that a non-lubricated bolt has an efficiency of about 50 percent of a well lubricated bolt and also that different lubricants produce results varying between the limits of 50 and 100 percent of the tabulated stress figures.

Data for Use with Machine Bolts and Cold Rolled Steel Stud Bolts Load in Pounds on Bolts and Stud Bolts when Torque Loads Are Applied

NOMINAL DIAMETER OF BOLT (Inches)	NUMBER OF THREADS (Per Inch)	DIAMETER AT ROOT OF THREAD (Inches)	AREA AT ROOT OF THREAD (Sq. Inch)	STRESS					
				7,500 PSI		15,000 PSI		30,000 PSI	
				Torque Ft. Lbs.	Compres- sion, Lbs.	Torque Ft. Lbs.	Compres- sion, Lbs.	Torque Ft. Lbs.	Compres- sion, Lbs.
1/4	20	.185	.027	1	203	2	405	4	810
5/16	18	.240	.045	2	338	4	675	8	1350
3/8	16	.294	.068	3	510	6	1020	12	2040
7/16	14	.345	.093	5	698	10	1395	20	2790
1/2	13	.400	.126	8	945	15	1890	30	3780
9/16	12	.454	.162	12	1215	23	2430	45	4860
5/8	11	.507	.202	15	1515	30	3030	60	6060
3/4	10	.620	.302	25	2265	50	4530	100	9060
7/8	9	.731	.419	40	3143	80	6285	160	12570
1	8	.838	.551	62	4133	123	8265	245	16530
1 1/8	7	.939	.693	98	5190	195	10380	390	20760
1 1/4	7	1.064	.890	137	6675	273	13350	545	26700
1 3/8	6	1.158	1.054	183	7905	365	15810	730	31620
1 1/2	6	1.283	1.294	219	9705	437	19410	875	38820
1 5/8	5 1/2	1.389	1.515	300	11363	600	22725	1200	45450
1 3/4	5	1.490	1.744	390	13080	775	26160	1550	52320
1 7/8	5	1.615	2.049	525	15368	1050	30735	2100	61470
2	4 1/2	1.711	2.300	563	17250	1125	34500	2250	69000

Data for Use with Alloy Steel Stud Bolts Load in Pounds on Stud Bolts when Torque Loads Are Applied

NOMINAL DIAMETER OF STUD (Inches)	NUMBER OF THREADS (Per Inch)	DIAMETER AT ROOT OF THREAD (Inches)	AREA AT ROOT OF THREAD (Sq. Inch)	STRESS					
				30,000 PSI		45,000 PSI		60,000 PSI	
				Torque Ft. Lbs.	Compres- sion, Lbs.	Torque Ft. Lbs.	Compres- sion, Lbs.	Torque Ft. Lbs.	Compres- sion, Lbs.
1/4	20	.185	.027	4	810	6	1215	8	1620
5/16	18	.240	.045	8	1350	12	2025	16	2700
3/8	16	.294	.068	12	2040	18	3060	24	4080
7/16	14	.345	.093	20	2790	30	4185	40	5580
1/2	13	.400	.126	30	3780	45	5670	60	7560
9/16	12	.454	.162	45	4860	68	7290	90	9720
5/8	11	.507	.202	60	6060	90	9090	120	12120
3/4	10	.620	.302	100	9060	150	13590	200	18120
7/8	9	.731	.419	160	12570	240	18855	320	25140
1	8	.838	.551	245	16530	368	24795	490	33060
1 1/8	8	.963	.728	355	21840	533	32760	710	43680
1 1/4	8	1.088	.929	500	27870	750	41805	1000	55740
1 3/8	8	1.213	1.155	680	34650	1020	51975	1360	69300
1 1/2	8	1.338	1.405	800	42150	1200	63225	1600	84300
1 5/8	8	1.463	1.680	1100	50400	1650	75600	2200	100800
1 3/4	8	1.588	1.980	1500	59400	2250	89100	3000	118800
1 7/8	8	1.713	2.304	2000	69120	3000	103680	4000	138240
2	8	1.838	2.652	2200	79560	3300	119340	4400	159120
2 1/4	8	2.088	3.423	3180	102690	4770	154035	6360	205380
2 1/2	8	2.338	4.292	4400	128760	6600	193140	8800	257520
2 3/4	8	2.588	5.259	5920	157770	8880	236655	11840	315540
3	8	2.838	6.324	7720	189720	11580	284580	15440	379440

Bolting Materials* (UCS-, UHA-, UNF-23)
Stress Table 1

ASTM Specification			Maximum Allowable Stress Values (psi) For Metal Temperatures Not Exceeding Deg. F										
Number	Grade	Notes	– 20 to 650	700	750	800	850	900	950	1000	1050	1100	
A307	B	(1)	—	—	—	—	—	—	—	—	—	—	
A325	—	(2)(3)	18,750	17,200	15,650	—	—	—	—	—	—	—	
A354	BB	(2)(3)	18,750	17,200	15,650	—	—	—	—	—	—	—	
	BC	(2)(3)	20,000	18,400	16,750	—	—	—	—	—	—	—	
	BD	(2)(3)	20,000	18,400	16,750	—	—	—	—	—	—	—	
A193	B7	(2)(3)	20,000	20,000	20,000	20,000	16,250	12,500	8,500	4,500	—	—	
	B5	(2)(3)	20,000	20,000	20,000	20,000	17,250	13,750	10,300	7,300	4,800	2,750	
	B14	(2)(3)	20,000	20,000	20,000	20,000	18,750	16,650	14,250	11,000	6,250	2,750	
B16	(2)(3)	20,000	20,000	20,000	20,000	20,000	18,750	16,650	14,250	11,000	6,250	2,750	
ASTM A193	Maximum Allowable Stress Values (psi) For Metal Temperatures Not Exceeding Deg. F												
Grade	Notes	– 20 to 100	200	300	400	500	600	650	700	750	800	850	900
B6	(2)	20,000	19,300	18,700	18,300	17,850	17,000	16,500	15,750	14,900	13,800	12,500	11,000
B8	(2)(4)(5)	15,000	13,300	12,000	10,900	10,000	9,300	8,950	8,650	8,300	8,000	7,750	7,500
B8C	(2)(4)(5)	15,000	15,000	13,600	12,650	12,200	11,900	11,850	11,800	11,750	11,650	11,450	11,300
B8T	(2)(4)(5)	15,000	15,000	13,600	12,650	12,200	11,900	11,850	11,800	11,750	11,650	11,450	11,300
		For Metal Temperatures Not Exceeding Deg. F											
		950	1000	1050	1100	1150	1200	1250	1300	1350	1400	1450	1500
B6	(2)	—	—	—	—	—	—	—	—	—	—	—	—
B8	(2)(4)(5)	7,250	7,050	6,800	6,300	5,750	4,500	3,250	2,450	1,800	1,400	1,000	750
B8C	(2)(4)(5)	11,100	10,800	10,500	10,000	8,000	5,000	3,600	2,700	2,000	1,550	1,200	1,000
B8T	(2)(4)(5)	11,100	10,800	10,500	10,000	8,000	5,000	3,600	2,700	2,000	1,550	1,200	1,000
ASTM A320													
Grade		Notes											
L7, L9, L10 B8F		(2)(6) (2)(4)(7)	These materials are for low temperature service. Tensile range given in Materials Table 2 (page 6), is based on bolt diameter. Refer to ASTM Specification A320 for details										

- (1) Not permitted above 450F; allowable stress value 7,000 psi. (Table UCS-23.)
- (2) These stress values are established from a consideration of strength only and will be satisfactory for average service. For bolted joints, where freedom from leakage over a long period of time without retightening is required, lower stress values may be necessary as determined from the relative flexibility of the flange and bolts, and corresponding relaxation properties. (Tables UCS-23 and UHA-23.)
- (3) Between temperatures of - 20F to 400F, stress values equal to the lower of the following will be permitted: 20% of the specified tensile

- strength, or 25% of the specified yield strength. (Table UCS-23.)
- (4) These stress values permitted for material that has been carbide-solution treated. (Table UHA-23.)
- (5) These stress values apply only when the carbon is 0.04% or higher. Table UHA-23.)
- (6) For temperatures below 400F, stress values equal to 20% of the specified minimum tensile strength will be permitted. (Table UCS-23.)
- (7) For temperatures below 100F, stress values equal to 20% of the specified minimum tensile strength will be permitted. (Table UHA-23.)

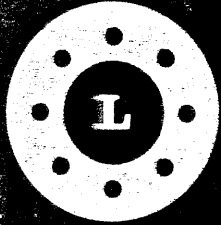
Note:

* It is often necessary to tighten bolting to much higher stresses than those given in the Table in order to prevent leakage under hydrotest and also to obviate frequent retightening due to relaxation. The Code does not prohibit this practice and the stress values listed are rather to be considered as applying in the design of flanges.

BOLTING DATA FOR STANDARD FLANGES

NOMINAL PIPE SIZE (Inches)	150 PSI SERIES				300 PSI SERIES				400 PSI SERIES				600 PSI SERIES			
	Diam. of Flange (Inches)	Num- ber of Bolts	Diam. of Bolts (Inches)	Bolt Circle (Inches)	Diam. of Flange (Inches)	Num- ber of Bolts	Diam. of Bolts (Inches)	Bolt Circle (Inches)	Diam. of Flange (Inches)	Num- ber of Bolts	Diam. of Bolts (Inches)	Bolt Circle (Inches)	Diam. of Flange (Inches)	Num- ber of Bolts	Diam. of Bolts (Inches)	Bolt Circle (Inches)
1/4	3 3/8	4	1/2	2 1/4	3 3/8	4	1/2	2 1/4	3 3/8	4	1/2	2 1/2	3 3/8	4	1/2	2 1/4
1/2	3 1/2	4	1/2	2 3/8	3 3/4	4	1/2	2 5/8	3 3/4	4	1/2	2 5/8	3 3/4	4	1/2	2 5/8
3/4	3 7/8	4	1/2	2 3/4	4 5/8	4	5/8	3 1/4	4 5/8	4	5/8	3 1/4	4 5/8	4	5/8	3 1/4
1	4 1/4	4	1/2	3 1/8	4 7/8	4	5/8	3 1/2	4 7/8	4	5/8	3 1/2	4 7/8	4	5/8	3 1/2
1 1/4	4 5/8	4	1/2	3 1/2	5 1/4	4	5/8	3 7/8	5 1/4	4	5/8	3 7/8	5 1/4	4	5/8	3 7/8
1 1/2	5	4	1/2	3 3/8	6 1/8	4	3/4	4 1/2	6 1/8	4	3/4	4 1/2	6 1/8	4	3/4	4 1/2
2	6	4	5/8	4 3/4	6 1/2	8	5/8	5	6 1/2	8	5/8	5	6 1/2	8	5/8	5
2 1/2	7	4	5/8	5 1/2	7 1/2	8	3/4	5 7/8	7 1/2	8	3/4	5 7/8	7 1/2	8	3/4	5 7/8
3	7 1/2	4	5/8	6	8 1/4	8	3/4	6 5/8	8 1/4	8	3/4	6 5/8	8 1/4	8	3/4	6 5/8
3 1/2	8 1/2	8	5/8	7	9	8	3/4	7 1/4	9	8	7/8	7 1/4	9	8	7/8	7 1/4
4	9	8	5/8	7 1/2	10	8	3/4	7 7/8	10	8	7/8	7 7/8	10 3/4	8	7/8	8 1/2
5	10	8	3/4	8 1/2	11	8	3/4	9 1/4	11	8	7/8	9 1/4	13	8	1	10 1/2
6	11	8	3/4	9 1/2	12 1/2	12	3/4	10 5/8	12 1/2	12	7/8	10 5/8	14	12	1	11 1/2
8	13 1/2	8	3/4	11 3/4	15	12	7/8	13	15	12	1	13	16 1/2	12	1 1/8	13 3/4
10	16	12	7/8	14 1/4	17 1/2	16	1	15 1/4	17 1/2	16	1 1/8	15 1/4	20	16	1 1/4	17
12	19	12	7/8	17	20 1/2	16	1 1/8	17 3/4	20 1/2	16	1 1/4	17 3/4	22	20	1 1/4	19 1/4
14	21	12	1	18 3/4	23	20	1 1/8	20 1/4	23	20	1 1/4	20 1/4	23 3/4	20	1 3/8	20 3/4
16	23 1/2	16	1	21 1/4	25 1/2	20	1 1/4	22 1/2	25 1/2	20	1 3/8	22 1/2	27	20	1 1/2	23 3/4
18	25	16	1 1/8	22 3/4	28	24	1 1/4	24 3/4	28	24	1 3/8	24 3/4	29 1/4	20	1 5/8	25 3/4
20	27 1/2	20	1 1/8	25	30 1/2	24	1 1/2	27	30 1/2	24	1 1/2	27	32	24	1 5/8	28 1/2
24	32	20	1 1/4	29 1/2	36	24	1 1/2	32	36	24	1 3/4	32	37	24	1 7/8	33

NOMINAL PIPE SIZE (Inches)	900 PSI SERIES				1500 PSI SERIES				2500 PSI SERIES			
	Diam. of Flange (Inches)	Number of Bolts	Diam. of Bolts (Inches)	Bolt Circle (Inches)	Diam. of Flange (Inches)	Number of Bolts	Diam. of Bolts (Inches)	Bolt Circle (Inches)	Diam. of Flange (Inches)	Number of Bolts	Diam. of Bolts (Inches)	Bolt Circle (Inches)
1/2	4 3/4	4	3/4	3 1/4	4 3/4	4	3/4	3 1/4	5 1/4	4	3/4	3 1/2
3/4	5 1/8	4	3/4	3 1/2	5 1/8	4	3/4	3 1/2	5 1/2	4	3/4	3 3/4
1	5 7/8	4	7/8	4	5 7/8	4	7/8	4	6 1/4	4	7/8	4 1/4
1 1/4	6 1/4	4	7/8	4 3/8	6 1/4	4	7/8	4 3/8	7 1/4	4	1	5 1/8
1 1/2	7	4	1	4 7/8	7	4	1	4 7/8	8	4	1 1/8	5 3/4
2	8 1/2	8	7/8	6 1/2	8 1/2	8	7/8	6 1/2	9 1/4	8	1	6 3/4
2 1/2	9 5/8	8	1	7 1/2	9 5/8	8	1	7 1/2	10 1/2	8	1 1/8	7 3/4
3	9 1/2	8	7/8	7 1/2	10 1/2	8	1 1/8	8	12	8	1 1/4	9
4	11 1/2	8	1 1/8	9 1/4	12 1/4	8	1 1/4	9 1/2	14	8	1 1/2	10 3/4
5	13 3/4	8	1 1/4	11	14 3/4	8	1 1/2	11 1/2	16 1/2	8	1 3/4	12 3/4
6	15	12	1 1/8	12 1/2	15 1/2	12	1 3/8	12 1/2	19	8	2	14 1/2
8	18 1/2	12	1 3/8	15 1/2	19	12	1 5/8	15 1/2	21 3/4	12	2	17 1/4
10	21 1/2	16	1 3/8	18 1/2	23	12	1 7/8	19	26 1/2	12	2 1/2	21 1/4
12	24	20	1 3/8	21	26 1/2	16	2	22 1/2	30	12	2 3/4	24 3/8
14	25 1/4	20	1 1/2	22	29 1/2	16	2 1/4	25				
16	27 3/4	20	1 5/8	24 1/4	32 1/2	16	2 1/2	27 3/4				
18	31	20	1 7/8	27	36	16	2 3/4	30 1/2				
20	33 3/4	20	2	29 1/2	38 3/4	16	3	32 3/4				
24	41	20	2 1/2	35 1/2	46	16	3 1/2	39				



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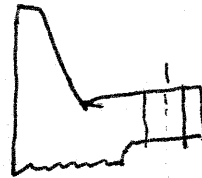
THE LAMONS WARRANTY

Goods sold are warranted to be free from defect in material and workmanship, but this express warranty is in lieu of and excludes all other warranties. Defective goods may be returned to Seller after inspection by Seller and upon receipt of definite instruction from Seller. Goods so returned and found to be defective will be replaced or repaired without charge, but Seller shall not be liable for loss or damage directly or indirectly arising from the use of the goods or from any other cause; Seller's liability being expressly limited to the replacement or repair of defective goods. Every claim on account of defective goods, short count, or for any other cause, shall be deemed waived by Purchaser, unless made in writing within thirty (30) days from the receipt of goods to which such claims relates.

FLANGE RATINGS

By Mr. Mike Vetter.

Tue. Oct. 4 : PM.



Outline

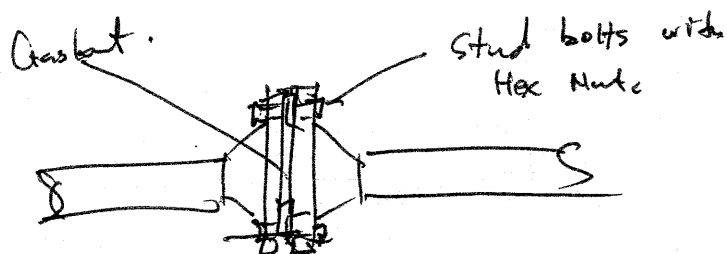
- A. Flange Ratings
- B. ANSI B16.5
 - a. Flanges
- D. Flange Design.
- E. Gasket.

- What is a flange?

A flange is a steel forging, casting, or plate used to connect piping or equipment by means of bolting.

- Why use a flange?

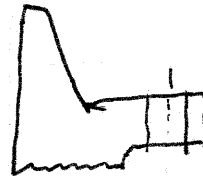
- Breakpoint
- Thermal expansion.



FLANGE RATINGS

By Mr. Mike Vetter.

Tue. Oct. 4 3 PM.



Outline

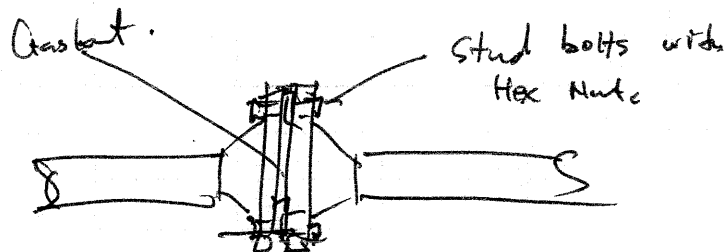
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WHY USE CODES ?

- Safety
- Economics
- Separate pressure vessels

CODES

1. Chemical plant and Petrochemical Refining Piping
(ANSI / ASME B31.3)
2. Pipe Flange and Flanged Fittings
(ANSI B16.5)
3. ASME Boiler and Pressure Vessel Section VIII Div.1

ANSI

A - American
N - National
S - Standard
I - Institute.

ASME

American Society Mechanical Engineers

ANSI B16.5 (Latest edition - 1988)

PIPE FLANGE AND FLANGED FITTINGS

ANSI B16.5

SCOPE

1. Pressure - Temperature Ratings
2. Materials
3. Dimensions
4. Tolerances
5. Markings
6. Testing.

Ratings are maximum allowable non-shock working gage pressure at the ~~the~~ temperature shown in Table 2. for the applicable material and rating.

SIZE RANGES

NPS $\frac{1}{2}$ " to NPS 24"

NPS = Nominal Pipe Size.

CAST or FORGED

Blind Flanges and Reduced Flange (without hubs)

MAYBE MADE FROM PLATE

INTEGRAL FLANGE

- Welding neck
- Weld neck
- Ring.

LOOSE FLANGE

- Slip-on
- Lap joint. : low pressure

PRESSURE - TEMPERATURE RATINGS

CLASS 150, 300, 400, 600, 900, 1500, 2500
CL 600

OLD DESIGNATIONS;

600 lb 600# , 600 psi.

FLANGE RATING

1. Determine Material Group Number,
2. State Design pressure and Design temperature
3. Choose class
4. Check hydrostatic Test pressure

EXAMPLE

Material A182-F11
($\frac{1}{2}$ Cr - $\frac{1}{2}$ Mo)

1-9.
CL 600

Design press. : $P_D = 400 \text{ psig}$

Design Temp. : 1000°F

Allowable stress :

@ 1000°F

$S_H = 6900 \text{ psi}$

@ ATM

$S_C = 17500 \text{ psi}$

$$P_T = (1.5) P_D \times \frac{S_C}{S_H}$$

$$= (1.5) (400) \left(\frac{17500}{6900} \right)$$

$$\therefore P_T = 1521.5 \text{ psi} < 2250 \text{ psi}$$

1500 $\times 1.5$
 Allowable pressure @ ambient

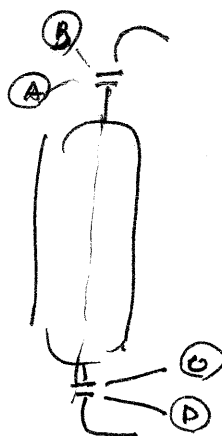
Flange Material

(A)(C)(D) A182-F11 (1/2 Cr - 1/2 Mo)

(B) A182-F316 (16 Cr - 12 Ni + 2 Mo)

Design Press. $P_D = 850 \text{ psi}$

Design Temp. $T_D = 850^\circ \text{F}$



(A)(D) : CL. 600 , 1.9
 (A)(B) : CL. 900 , 2.2.

Allowable Stress

A182-F11 @ 850°F $S_H: 17100$

@ ATM. $S_C: 17500$

A182-F316 @ 850°F $S_H: 10900$

@ ATM. $S_H: 17500$

$$P_T = (850) (1.5) \left(\frac{17500}{10900} \right) = 2047 < 3240$$

$$2150 \times 1.5 = 3240$$