# 8
# High Pressure Vessels

## 1.0. General

### 1.1. Introduction

Designers and engineers of pressure vessels must eventually become acquainted with the design of vessels for high pressure service. The immediate question is, "What constitutes high pressure"? For purposes of discussion here, high pressure begins around 3,000 PSI. This is an arbitrary value as there are no codes or standards that define where high pressure begins. Perhaps a better way to decide if a pressure vessel is high pressure is based on the R/t ratio. Ratios of R/t less than 10 should be considered as high pressure.

There are pressure apparatus that produce pressures up to 1.5 million PSI. Apparatus that produce artificial diamonds and gems operate in this pressure range. They are more technically "presses" and not pressure vessels, per se. Pressures in the range of 20,000 to 150,000 PSI are used for food sterilization. Isostatic pressing operates in the range from 15,000 to 100,000 PSI. Autoclave reactors for hydrothermal crystal growth or production of artificial quartz are used with pressures in the 15,000 to 25,000 PSI range. Polyethylene processes operate in the 20-35,000 PSI range.

So there are a number of industrial processes that use high pressure. This chapter attempts to give the designer and engineer some tools to develop a basic design in the 3,000 to 25,000 PSI range. Vessels that were built in this range would include;

1. Low Density Polyethylene (LDPE) Reactors (Autoclaves)
2. Isostatic Presses (Hot or Cold)
3. Urea Reactors (Autoclaves)
4. Polyethylene Separators
5. Ammonia Converters
6. Sterilization Vessels
7. Hydrothermal Synthesis
8. Autoclaves
9. High Pressure Extraction
10. Water Jet Cutting

High pressure vessels can be built to ASME Section VIII, Divisions 1, 2 or 3. People tend to discount Division 1 as a design code for high pressure vessels but this should not be the case, although Divisions 2 and 3 will produce a more economical design. Remember that Division 2 was not in existence before 1968 and Division 3 before 1997. Many thousands of high pressure vessels were built before these alternative divisions were issued.

High pressure vessels should be designed to either Division 2 or Division 3, Section VIII of the ASME Code. This chapter evaluates the criteria of each division and its impact on the wall thickness of the vessel shell. There are two basic criteria for each division. They are Design by Rules and Design by Analysis. It should be noted that only Division 3 allows credit for pre-stress (autofrettage) for layered construction.

The treatment and discussion of vessels for design of high pressure equipment may appear to be out of the ordinary. This is an illustrated approach, rather than an analytical one. It is a design approach, not an analytical one. This chapter does not focus on higher, more sophisticated, analytical methods such as elastic-plastic or limit analysis. Certainly these methods should be utilized to fine tune the design to reduce wall thickness and weight. The focus of this chapter is to enable the user to develop a design which will work. It may not be the most economical or efficient design, but it will work.

Often in our industry we are asked to come up with designs which can be priced in the marketplace to determine if a particular process is economically feasible. Alternatively, we are asked to determine the ultimate cost so that one process can be compared with another. Without making a research project out of this effort it is possible with the use of material presented here to come up with a design that will suit this work process.

In other words, the design developed by the procedures illustrated here, may not be the most economical design. They will provide a starting place for the "development" of the final design.

Designs with gasketed closures will not reduce the thickness of components. There is a certain force required to seat the gasket and resist the internal pressure. This does not vary with the analytical technique used. In fact, the "brute force" approach is the valid one. On the flip side, the shell design can be greatly reduced by these advanced tools.

For low and medium pressure vessels, the most economical L/D ratio is between 3 and 5. As pressures tend to increase, this ratio goes up to 10, 50 or even 100. This is because the economics will drive the designer to smaller diameters and greater length to achieve the same volume.

For this same reason, most high pressure vessels are built with an end closure to provide access to the inside of the vessel. It is economically unsound, and in many cases impossible to provide manways. In fact, due to the thick shells required, nozzles through the shell wall are discouraged. In high pressure vessels most nozzles go through the end closures, not through the shell.

This is why most high pressure vessels do not use typical pressure vessel heads such as hemispherical or semi ellipsoidal. These heads are impractical in high pressure applications. So the economics and mechanical limits of the materials will determine the ultimate shape of the vessel.

Thus, the focus of this chapter is on end closures and not on design of shells and flat heads. As shown here these are relatively simple calculations.

Diameters are deliberately kept small to keep the wall thickness and metal weight down. However, as pressures and diameters increase, and new industrial processes are developed, it will push open the envelope of our existing technologies. Yet, there is a physical limit to what steels can do. The highest strength for any material in ASME VIII, Division 3, is a forging material, SA-723. However this material cannot be used for welded construction, only machined applications.

### 1.2. ASME VIII, Division 3

In the mid 1970s it was realized that the current ASME Codes for pressure vessels were not adequate for vessel operating above about 20,000 PSI. ASME Section VIII, Division 2 did not have any allowance for vessels manufactured by the autofrettage technique. In November, 1979 the ASME board on Pressure Technology Codes and Standards established a special working group on high pressure vessels. The group was chartered to come up with a Code that would be applicable for vessels operating from 10,000 to 200,000 PSI. However no upper limit has been established.

This was quite challenging because of the diversity of the equipment required for this pressure range. For example, vessels in the lower end of the range typically are manufactured by welding. However equipment manufactured for the higher end of the range are typically manufactured by machining.

A fracture mechanics analysis is required for all vessels unless a leak before burst mode of failure can be assured. A fatigue analysis is required for all vessels. The determination of the number of cycles allowed by Division 3 is conservative but also allows for the recertification of equipment that has reached its design life.

Division 3 deliberately restricts the materials that are acceptable for use. Only 242 different types and grades of materials have been authorized. Allowable stresses have been based on yield strength. Toughness properties were considered critical for these applications and have required that the materials meet the required CIT values in the weak axis (transverse direction), perpendicular to the direction of rolling, wherever possible. The philosophy for setting high toughness criteria was based on the need for higher strength materials used in this equipment.

There are no tables of allowable stress in Section 2, Part D as there are for other sections of the Code. This is because thicknesses are based on ultimate tensile and yield strength. Properties for those materials acceptable in Division 3 are contained within the code itself in Article KM-4, "Material Design Data".

Division 3 provides special rules for layered and autofrettage vessels. Rules for wire wound vessels and interlocking strip wound vessels has also been included.

For designs that cannot prove leak-before-burst criteria, the vessel will be subject to fracture mechanics evaluation. Welded construction is permitted for all materials where ductility can be maintained. One material that has found favor for the highest pressure applications is SA-723 forgings. This material has incredible strength but welding is not allowed.

An inner core or liner used for corrosion resistance may or may not be included as part of the strength calculations. Weld overlay or cladding, specifically for corrosion resistance cannot be included in strength calculations.

All nozzles must be integrally reinforced. Reinforcing pads, fillet welds and partial penetration welds are not allowed. Threaded connections must be of the "straight" thread variety. Tapered threads, such as ordinary pipe threads are not allowed.

NDE requirements for Division 3 vessels and components was based on Section III, Class 1 rather than Division 2. UT is the preferred method of examination rather than RT for welded joints. All butt welds shall be 100% UT examined. In addition all plates and forgings must be 100% UT examined, not just those over 4 inches as in Division 2. 100% PT or MT is required for all internal and external surfaces of pressure boundary components.

A hydrotest is required for all vessels and is 1.25 times the design pressure times a stress material ratio, SMR. Unlike other divisions, Division 3 sets an upper limit on the hydrotest pressure.

Documentation and stamping follow ASME convention with regard to nameplate stamping, data reports and ASME third party inspection and witnessing of various hold points.

### 1.3. Types of Construction for High Pressure Vessels

There are two major categories for the manufacture of high pressure vessels as follows;

1. Monobloc
2. Layered
    a. Wound
        i. Wire or filament would
        ii. Interlocking strip wound
        iii. Flat steel ribbon wound
        iv. Plate wound
        v. Coil wound
    b. Autofrettage
        i. Hydraulically expanded type
        ii. Shrink fit
    c. Wrapped
        i. Spiral
        ii. Concentric

A detailed description of the various manufacturing techniques would be as follows;

1. Monobloc: Solid vessel wall by forging or rolled plate
2. Multilayer: Begins with a core of about ½ inches thick and successive layers are applied. Each layer is vented (except the core) and welded individually with no overlapping welds.
3. Multiwall: Begins with a core of 1½ inches to 2 inches thick. Outer layers of the same thickness are successively shrunk fit over the core. This creates compressive stress in the core, which is relaxed during pressurization. The process of compressing layers is called "autofrettage" from the French word meaning, self-hooping.
4. Multilayer autofrettage: Begins with a core about ½ inch thick. Bands of forged or welded rings are slipped over the core, and then the core is expanded hydraulically. The core is stressed into the plastic range but below ultimate strength. The outer rings are maintained at a margin below yield strength. The elastic deformation residual in the outer bands induces compressive stress in the core, which is relaxed during pressurization.
5. Wire Wrapped Vessels: Begin with an inner core of thickness, less than required for internal pressure. Core is wrapped with steel cables in tension until the desired thickness is reached.
6. Coil Wrapped Vessels: Begin with a core that is subsequently wrapped with thin steel sheet until the desired thickness is reached. Only two longitudinal welds are used, one attaching the sheet to the core. The other the final closure weld. Vessels 5 to 6 feet in diameter and pressure up to 5,000 psi have been made in this manner.

**HIGH PRESSURE VESSELS**

**METHODS OF MANUFACTURE OF SHELL**

- **MONOBLOC**
  - a. Forged
  - b. Machined
  - c. Cast
  - d. Welded
- **MULTI LAYER**
  - **SHRINK FIT (Autofrettage)**
    - a. Thermally
    - b. Hydraulically
    - c. Banded
  - **WOUND**
    - a. Ribbon
    - b. Wire
    - c. Filament
    - d. Interlocking Strip
    - e. Coil
    - f. Plate
  - **WRAPPED**
    - a. Spiral
    - b. Concentric

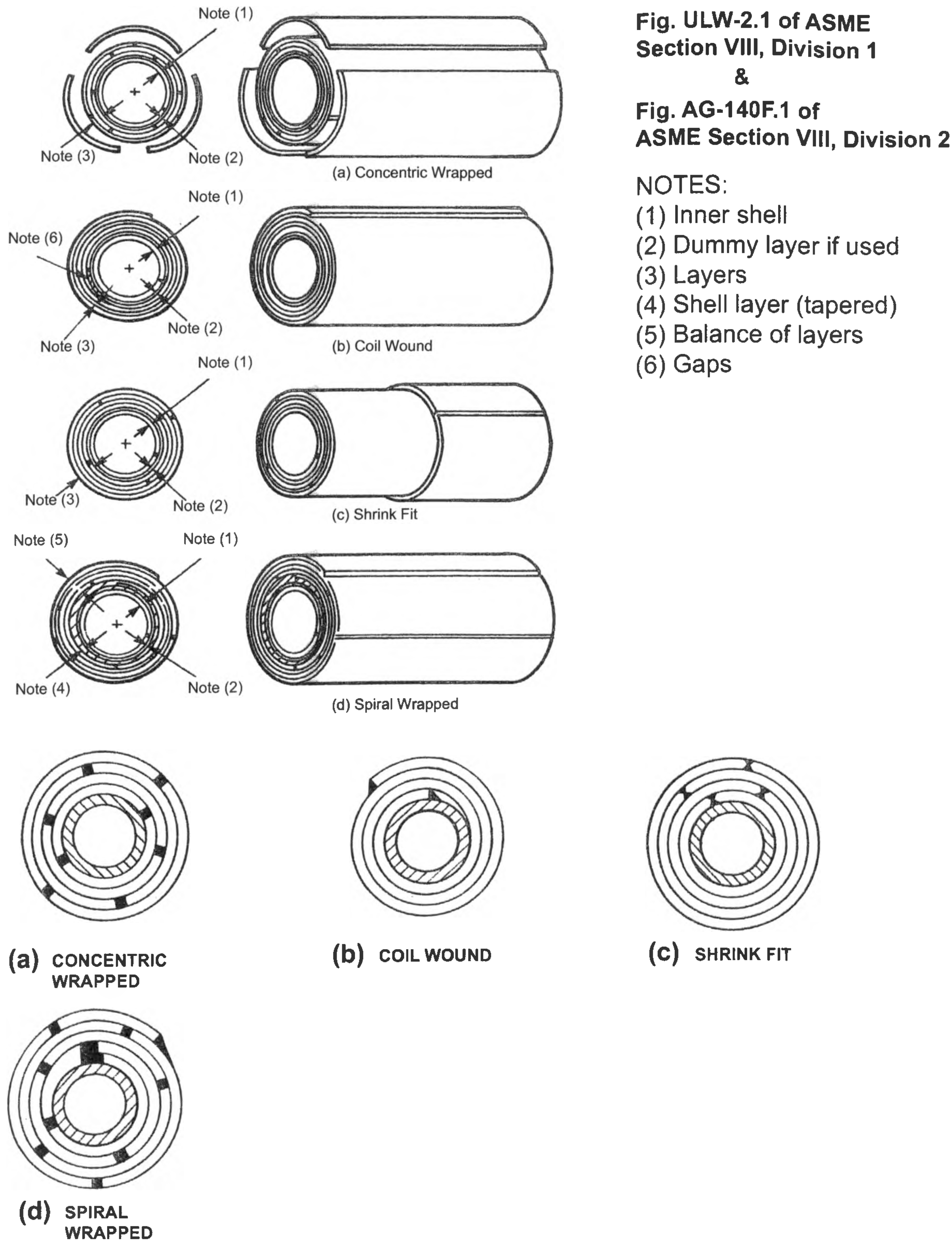


**Figure 8-1.** Types of layered construction.

## 1.4. Details of the Ribbon Wound Technique

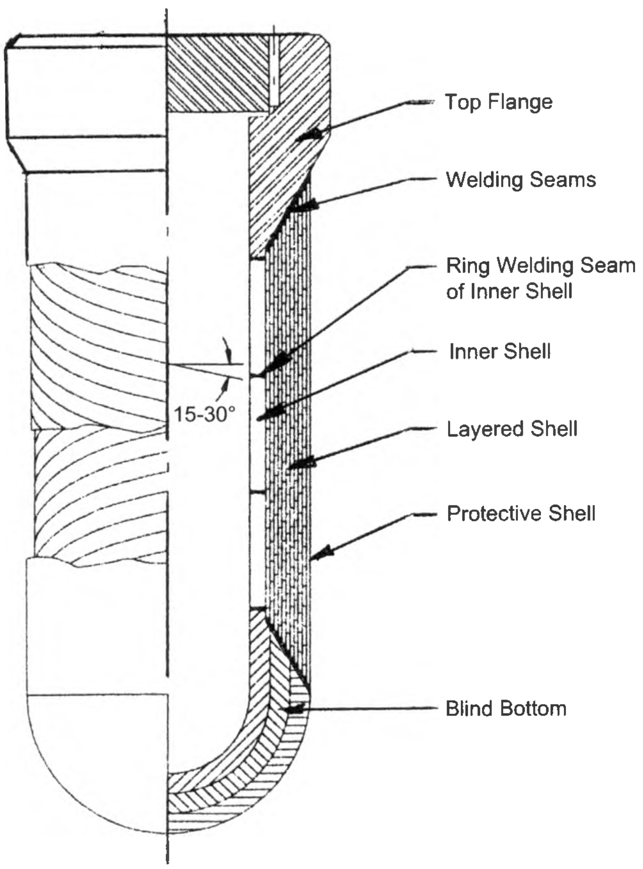


**Figure 8-2.** Flat steel ribbon wound pressure vessel.

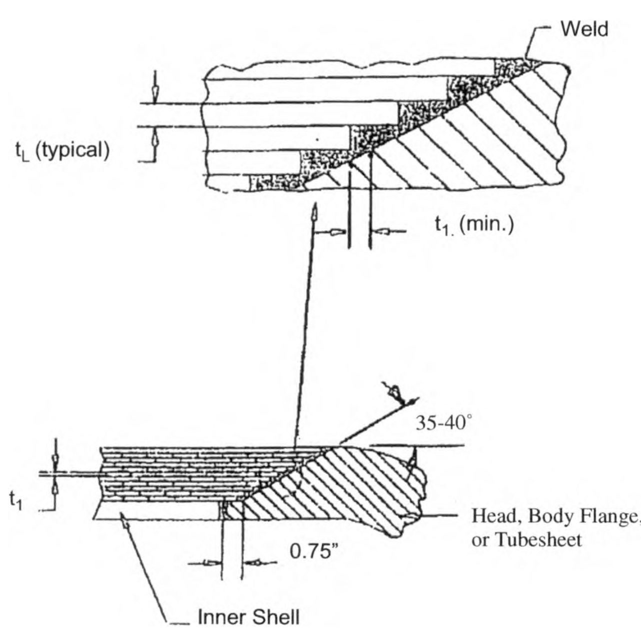


**Figure 8-3.** Typical head, body flange, or tubesheet attached to layered section.

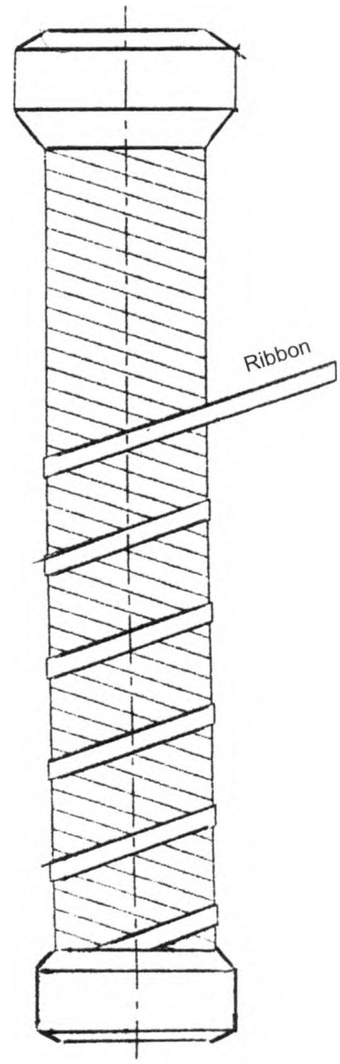


**Figure 8-4.** Schematic of winding process for the ribbon wound vessel.

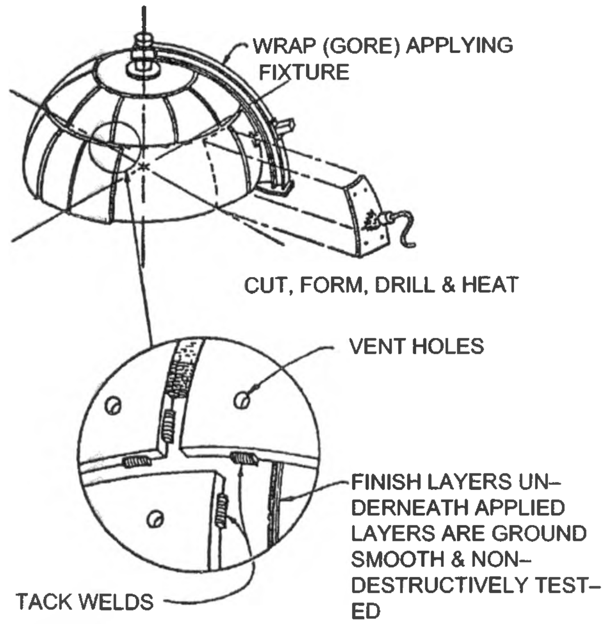


**Figure 8-5.** Layered hemispherical head fabrication procedure.

**Multi-Layer Pressure Vessels.** Multi-layer vessels have been known by various names over the years. The most common names are;

1. Multiwall Vessels
2. Banded Vessels
3. Layered Vessels

A multi-layer vessel is a vessel in which the cylindrical portion is made up of two or more contacting bands or layers. The inner shell is the innermost band of a multi-layer vessel and is made in the same manner as any solid wall vessel. The inner shell can be of any suitable material to resist corrosion by the contents. This is one of the unique advantages of a multi-layer vessel. Subsequent layers are added on top of this inner layer by a variety of techniques to achieve the ultimate wall thickness required.

### History

The concept for multi-layer construction was first developed in Germany around 1890. However it was not until the 1930s that the concept became practical with the advent of electric arc welding. In the 1930s the A.O. Smith Corporation pioneered the concept of concentric wrapped thin layers and eventually produced more than 10,000 multi-layered pressure vessels. The license was eventually sold to CB&I who produced multi-layer vessels under the patented Multilayer trademark until 1976, when the patents and machinery were subsequently sold to Nooter Corp. Although Nooter is now out of the fabrication business in the U.S.A., multi-layer vessels are still produced in Austria by a Nooter subsidiary, Schoeller-Bleckmann. Nooter marketed the multi-layer vessels under the trade name, Plywall.

In 1962 the autofrettage technique of shrink fitting successive layers was patented by Struthers Wells Corp. Over 800 such vessels were manufactured by the 1980s. Manufacture of these vessels was then transferred via license to Larson & Toubro, India, who has in turn produced more than 50 multi-layered, autofrettage vessels.

A Chinese manufacturing firm developed a ribbon wound technique for multi-layered vessel construction in the 1960s and has subsequently produced over 7,000 ribbon wound vessels. Kobe Steel in Japan, was originally a multi-layer vessel manufacturer and produced approximately 1,000 units of the concentrically wrapped types. They currently do not produce multi-layer vessels any longer but still engage in solid wall, monobloc construction.

In total, it is entirely probable that over 20,000 multi-layer vessels have been fabricated over the past 80 years, many of which are still in operation.

### General

Multi-layer vessels have been designed and built within the following ranges;

1. Diameters from 7 inches to 168 inches
2. Pressures from 1000 to 35,000 psi
3. Wall thickness' from 1½ inches to 18 inches
4. Temperatures from (-) 50 to 850°F
5. Lengths up to 100 feet
6. Weights up to 800 tons

In general, the thicker the vessel, and the longer the vessel, the more attractive the multi-layer option becomes. It is usually more economical to design a multi-layer vessel with a large L/D ratio for a given volume. The selection of the multi-layer option is usually determined by economics. However, once the practical manufacturing limits of solid wall construction are exceeded, multi-layer may be the only option.

Multi-layer pressure vessels can be manufactured by a variety of means. The most common categories or types of manufacture are;

1. Wrapped
2. Wound
3. Shrink Fit (autofrettage)

The most common applications;

a. Ammonia Converters
b. Methanol Converters
c. Ethanol Reactors
d. Urea Reactors
e. Urea Autoclaves
f. Storage Vessels for Helium, Nitrogen and Hydrogen
g. Hydrocracking Reactors
h. Polyethylene Reactors
i. High Pressure Separators
j. High Pressure Heat Exchangers
k. Hydraulic Cylinders
l. Accumulators
m. Air Receivers

### ASME Code

It was not until January 1979 that layered vessels were included in the ASME Code, almost 50 years after they were first introduced. Also in 1979 the ASME Policy Board, Codes and Standards, approved the establishment of a special working group for high pressure vessels under Section VIII. This effort would ultimately culminate in the issuance of Division 3 of Section VIII in 1998.

Multi-layer vessels can be built to any of the three sections of ASME Section VIII, Divisions 1, 2 or 3. However, only Division 3 allows the designer to take credit for the residual compressive stresses induced by the autofrettage technique. This is a big advantage in the design of the shell. The appropriate ASME Code Sections that apply to multi layer vessels are as follows;

1. Division 1: Part UWL
2. Division 2: Part 4.13
3. Division 3: KD-5 & KD-8

Code Cases 2229 and 2269 were issued in 1996 and 1997 respectively to allow for ribbon wound pressure vessels.

### Special Features/Advantages

1. Layered vessels are constructed by successively wrapping thin layers around a center core until the desired wall thickness is achieved. One advantage of this process over monobloc construction, is that the layers each have uniform chemical and mechanical properties. Optimum properties cannot always be achieved across thick sections or maintained during the whole fabrication sequence. With the multi-layer concept, no matter how thick the shell, it does not suffer from lack of material uniformity.
2. For corrosive environments, the inner core cylinder may be made of whatever alloy is necessary to provide corrosion resistance.
3. The use of high tensile material allows for reduced shell thickness.
4. High residual stresses from welding cannot develop across the multiple thin layers in multi-layered vessel shells. Any through thickness welds such as girth seams and nozzle welds do not develop high residual stresses because there is little "shear transfer" between the layers. As such, PWHT is seldom used and not desirable.
5. Vent holes are used in all the layers except the inner core layer. In the event of leakage in the inner cylinder, the leak may be detected without fear of failure of the vessel as a whole.
6. In the case of monobloc construction, the shell thicknesses are limited to ingot sizes, plate roll capacity or forging press capabilities. Layered construction does not have such limitations other than overall weight.

7. Butt welds in the inner core cylinder are radiographed in accordance with the Code. The balance of seams attaching each layer are magnetic particle tested.
8. The longitudinal welds are staggered.
9. Multi-layer vessels can be used in hot hydrogen service for hydrotreating and hydrocracking applications.
10. Multi-layer vessels can be field fabricated. Although this would be unusual, it is possible and has been done before.
11. Multi-layer vessels can be repaired in the field.
12. Multi-layer vessels can be rerated.
13. Nozzles can be welded through the shells of a multi-layer vessel but this is frequently avoided due to the extreme thickness. It is more common for all nozzles to be placed in the heads. In any case nozzles through multi-layered vessels should be minimized.

**Disadvantages**

1. Through thickness welds (welds that penetrate the multiple layers such as head attachment, circumferential welds between shell courses, or nozzle welds) cannot be easily examined by either UT or radiography.
2. Through thickness welds have different heat conduction properties than the layers to which they are attached. Therefore thermal stresses may create discontinuity stresses at locations already susceptible to higher stress levels.
3. Through thickness welds are not vented like the adjacent layers and therefore may be susceptible to hydrogen embrittlement.

**Autofrettage.** In thick walled cylindrical vessels, the resistance to yielding, fatigue crack initiation and propagation, and to fracture can be significantly increased by introducing residual compressive tangential stresses into the vessel wall near the inside surface. This effect occurs because the stress distribution in a thick walled cylinder by internal pressure consists of very high stress intensity near the inside surface. This stress intensity decreases rapidly with increasing radius and near the outside radius, the stress intensity is relatively small. This means that much of the material of the wall is not contributing significantly to the strength of the vessel.

Autofrettage is a method for increasing a cylindrical shells ability to withstand pressure by introducing residual compressive stress into the inner portion of the cylinder. Autofrettage is a French word that literally means, "self hooping". It is a method of prestressing the inner radius of a thick walled cylinder. The amount of prestressing causes plastic yielding of the inner membranes of the vessel wall and leaves this area in compression. During pressurization, the residual compression of the bore is initially relaxed before tension stress is achieved. This allows the cylinder to be made thinner since a significant portion of the compressive prestress must be overcome before tension even begins.

Yielding of the inner surface begins when the maximum shear stress is equal to the yield point shear stress. As the pressure, or force, is increased the plastic deformation penetrates farther into the vessel wall until it reaches the outer surface. At that point the entire shell has yielded. If the internal pressure is removed after the cylinder is in plastic state, a residual stress will remain in the wall. This residual stress allows the cylinder to contain more pressure then would be possible to without it.

Various manufacturing techniques have been developed to produce the residual stresses in the bore of thick walled cylinders. These processes are as follows;

1. Autofrettage (bore expansion technique)
   a. Hydraulic swage with tapered plug
   b. Mechanical pressure (swaging)
   c. Pressurized cold working
   d. Explosion technique
   e. Thermal gradient
2. Multiple Wall
   a. Welded layer – weld shrinkage
   b. Shrink fit
3. Wire or ribbon winding

The most common technique used in the manufacture of pressure vessels is the shrink fitting technique. In this process layers are successively shrunk, over an inner core, one cylinder over another. The outer layer is heated which causes it to expand. The inner layer is cooled which causes it to contract. Once the outer cylinder is slipped over the inner cylinder it contracts as it cools, putting the inner wall in compression. Successive layers can be applied which amplify the effect.

Shrink rings are used around other components to achieve the same effect. For example, studded flanges in high pressure applications, frequently have shrink rings placed in the area of high stress around the circumference of the flange to reduce stresses and wall thickness.

Another common practice is to hydraulically expand the inner cylinder beyond the plastic range. This was the first application of autofrettage used with cannon barrels. In this case the inner cylinder or core is expanded beyond the yield range. When the pressure is released the inner core goes into compression. There are several positive effects that result from this process. First, the work hardening that results from the material stretch contributes to the strength of the part. Secondly, cylinders expanded in this fashion are more resistant to fatigue failures then those that are not.

The third method utilized for creating a compressive prestress in the bore, is by wire or ribbon winding of the bore cylinder under tension. Although this is effective, there are limited suppliers who can actually manufacture equipment in this fashion.

Therefore the designer must allow for the various techniques for accomplishing the compressive prestress while allowing the flexibility of the supplier to apply their specialized technology.

#### History

The history of the autofrettage technique was developed for use in cannon barrels. The first autofrettaged gun barrel was produced in France in 1913. The process quickly was adopted by the U.S.A. and U.K. as well. During WWII, the availability of high strength steels was critical due to shortages of alloying elements and industrial capacity. With autofrettage, low strength steel could be utilized for gun barrels with better efficiency. Expansion of the barrels was accomplished by tapering the barrels and then hydraulically expanding the cylinders into carefully machined outer staves or dies. The expansion of the bore led to the work hardening benefits as well as the residual stresses from the autofrettage. This was known as the "container method".

After WWII, this method was dropped due to availability of alloy materials and the difficulty of expanding alloy barrels. The hydraulic intensifiers worked at an amazing 150,000 PSI. It was noted however that the autofrettage technique had a significant effect on the barrels ability to withstand fatigue. An alternative process was developed that swaged the bore by forcing a mandrel through the bore. The process became known as "hydraulic swage autofrettage" and is still in use today.

#### Autofrettage Pressure

Autofrettage pressure is the pressure that causes initial overstrain of the bore to occur. It is that pressure that results in plastic deformation, beginning at the inside surface of the cylinder. This is "initial autofrettage pressure" or "minimum autofrettage pressure". The "maximum autofrettage pressure" is the pressure resulting in plastic yielding across the entire wall thickness.

#### Overstrain Ratio

In order to calculate the residual stress produced by the autofrettage process, a measure of the extent of autofrettage must be known. The measure most frequently used is the "overstrain ratio". This is the ratio of the thickness of the plastically deformed portion of the shell to the total thickness.

#### Bauschinger Effect

The pressure at which the material of a previously autofrettaged cylinder first undergoes additional plastic deformation is called the re-yield pressure. Theoretically this pressure should be the same as the autofrettage pressure. However studies have shown that the re-yield pressure is actually less than the autofrettage pressure. This is because the unloading from the autofrettage pressure, does not follow linear elastic behavior. The reason for this phenomena has been termed the "Bauschinger Effect". The Bauschinger Effect is defined as a decrease in the yield strength of the material in compression as a result of prior deformation in tension. A Bauschinger Effect Factor (BEF) has been established to account for this condition.

When the yield strength is affected by over-strain, the BEF approaches 0. If the effect is very little, then the BEF approaches 1. High strength steels exhibit a significant Bauschinger Effect. Thus the material near the inner bore does not behave elastically during unloading, but actually reverse yields in compression., which in turn greatly reduces the tangential stress near the surface on the outside of the vessel.

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|---|---|---|---|---|---|
| **HIGH PRESSURE VESSELS** | | | | | |
| **DESIGN OPTIONS CHECKLIST** | | | | | |
| TYPE OF END CLOSURE | REMOVABLE | NON-REMOVABLE | | | |
| TYPE OF REMOVABLE END CLOSURE | THREADED | FLANGED | OTHER | | |
| TYPE OF GASKET | DELTA RING | BRIDGMAN | DOUBLE CONE | LENS RING | OTHER |
| MATERIAL OF CONSTRUCTION | | | | | |
| TYPE OF SHELL CONSTRUCTION | MONOBLOC | LAYERED | BANDED | WOUND | AUTOFRETTAGE |
| MATERIAL OF GASKET | | | | | |
| SHRINK SLEEVES | YES | NO | | | |
| SLEEVE NUTS | YES | NO | | | |
| STUD TIGHTENING METHOD | MECHANICAL | THERMAL | HYDRAULIC | | |
| THREADED CLOSURES - TYPE OF THREADS | ACME | BUTTRESS | MODIFIED | OTHER | |
| JACK SCREWS | YES | NO | | | |
| NOZZLE ATTACHMENTS - Type/Style | | | | | |
| CORROSION | | | | | |
| ORIENTATION - Vertical or Horizontal | | | | | |
| TYPE OF SUPPORT | | | | | |
| TYPE OF OPERATION - Continuous or Batch | | | | | |
| PRESSURE | | | | | |
| TEMPERATURE | | | | | |

| HIGH PRESSURE VESSELS - Checklist of Testing & NDE Requirements | | | | | | |
|---|---|---|---|---|---|---|
| | Description | Required | Before PWHT | After PWHT | After Hydro | Remarks/Notes |
| | ***NDE Requirements (Welds & Base Matl's)*** | | | | | |
| 1 | 100% RT all Cat A & B butt welds | | | | | |
| 2 | 100% RT all Cat C & D butt welds | | | | | |
| 3 | 100% UT all Cat A & B butt welds | | | | | |
| 4 | 100% UT all Cat C butt welds | | | | | |
| 5 | 100% UT all Cat D joints (butt and groove welds) | | | | | |
| 6 | Spot RT all back clad areas | | | | | |
| 7 | MT all plate edges and weld preps | | | | | |
| 8 | MT/PT all surfaces prior to WOL | | | | | |
| 9 | PT all attachments welded directly to cladding or WOL | | | | | |
| 10 | PT/MT all machined surfaces of forgings 100% | | | | | |
| 11 | PT all WOL/Clad surfaces | | | | | |
| 12 | UT all WOL/Clad surfaces | | | | | |
| 13 | MT backgouging of root passes | | | | | |
| 14 | MT of welds prior to backweld of WOL | | | | | |
| 15 | MT of main seams | | | | | |
| 16 | UT/MT of weld buildup after machining | | | | | |
| 17 | Acoustic emission | | | | | |
| | | | | | | |
| | | | | | | |
| | | | | | | |
| | ***Heat Treatment Requirements*** | | | | | |
| 1 | Plate or forgings for shell and heads Q&T | | | | | |
| 2 | Plate or forgings for shell and heads N&T | | | | | |
| 3 | PWHT | | | | | |
| 4 | Intermediate stress relieve (ISR) | | | | | |
| 5 | Dehydrogenation heat treatment (DHT) | | | | | |
| 6 | Step cooling | | | | | |
| 7 | Multiple HT cycles allowed for (maximum PWHT) | | | | | |
| | | | | | | |
| | | | | | | |
| | ***Fabrication/Design Requirements*** | | | | | |
| 1 | F' Type nozzles required > 3" | | | | | |
| 2 | All attachments full penetration | | | | | |

| | Description | Required | Before PWHT | After PWHT | After Hydro | Remarks/Notes |
|---|---|---|---|---|---|---|
| | | | | | | |
| | | | | | | |
| | ***Base Material Requirements*** | | | | | |
| 1 | Hot tensile tests | | | | | |
| 2 | Autoclave testing - disbonding | | | | | |
| 3 | Shell/head materials to SA-20, paragraphs S1, S2, S5 | | | | | |
| 4 | UT all plate and forgings > 4" | | | | | |
| 5 | X factor - weld filler metal | | | | | |
| 6 | J factor - plate, forgings, pipe and fittings | | | | | |
| 7 | Impact testing of base materials | | | | | |
| 8 | Production impact testing | | | | | |
| 9 | Copper sulfate examination - strip back of clad areas | | | | | |
| 10 | PMI | | | | | |
| 11 | Hardness | | | | | |
| 12 | Ferrite check - cladding and WOL | | | | | |
| 13 | Hot rolling of shell plates | | | | | |
| | | | | | | |
| | | | | | | |
| | ***Testing Requirements*** | | | | | |
| 1 | Hydrotest holding time | | | | | |
| 2 | Hydrotest water purity | | | | | |
| 3 | Hydrotest test temperature | | | | | |
| 4 | Chemical analysis of WOL | | | | | |
| 5 | Base line UT thickness readings | | | | | |
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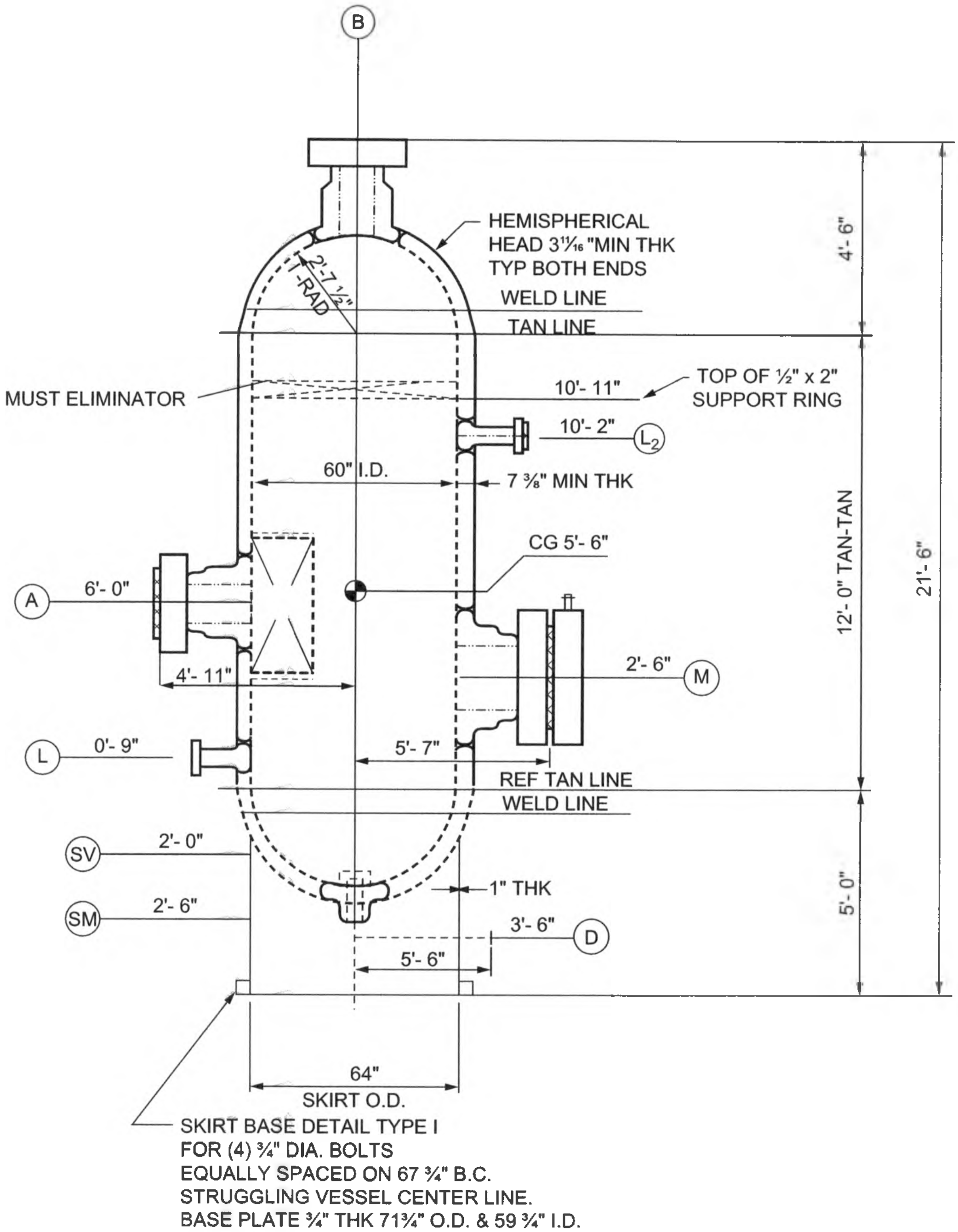
| | | | | | | |

## Section 1.5: Data for Examples - High Pressure Vessels

| Item | Description | Pressure, PSIG | Temp, °F | D, ID, in | Shell Thick. In | T-T OR OAL, Ft | L/D Ratio | $R_m$, in | $R_m$ /t Ratio | Weight (KIPS) | Shell Type | Matl | Closure Gasket Type/Matl |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | Compressor Suction Drum | 4960 | 250 | 60 | 7.375 | 12 | 2.4 | 33.68 | 4.6 | 105 | Monobloc | CS | NA |
| 2 | Receiver | 3800 | 350 | 22.83 | 2.283 | 11.22 | 5.9 | 12.56 | 5.5 | 11.2 | Monobloc | CS | NA |
| 3 | Urea Reactor | 4000 | 400 | 42 | 5.75 | 60 | 17.1 | 23.875 | 4.15 | 214 | Multilayer | CS | Bridgman-Alum |
| 4 | Ammonia Separator | 5800 | 90 | 30 | 6.25 | 8 | 3.2 | 18.125 | 2.9 | 42.2 | | CS | |
| 5 | Ammonia Converter | 10,000 | 300 | 29.75 | 4.375 | 37.83 | 15.3 | 17.06 | 3.9 | | | CS | Double Cone-Alum |
| 6 | H.P. Separators | 10,000 | 300 | 29.5 | 4.375 | 7.27 | 3.1 | 16.94 | 3.87 | | | CS | Double Cone - Annealed Copper |
| 7 | Liquid Phase Converter (1) | 10,300 | 1,000 | 32 | 8.75 | 42.58 | 16 | 20.375 | 2.3 | 190 | Monobloc | Cr-Mo | Delta Ring |
| 8 | Ammonia Converter | 10,000 | 300 | 29.75 | 4.375 | 37.83 | 15.3 | 17.06 | 3.9 | 32.9 | | Cr-Mo | Double Cone -Alum |

*Notes:*

1. Vessel is refractory lined. Weight includes refractory



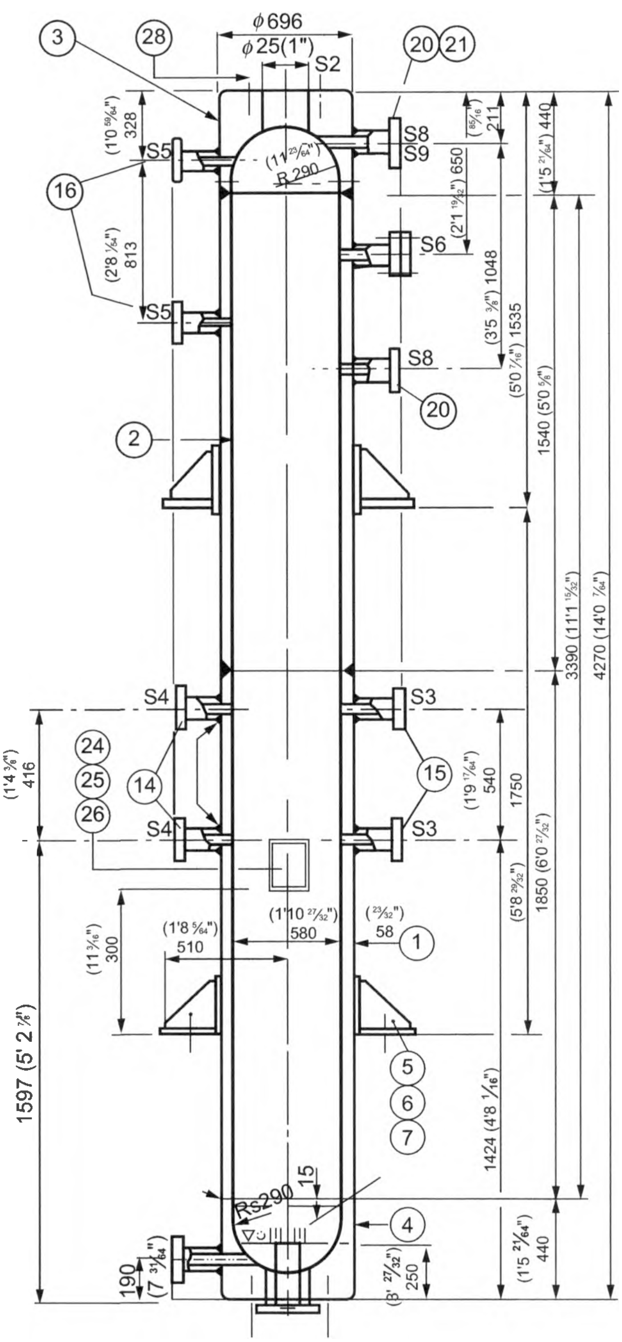

**EXAMPLE No.1**

Compressor Suction Drum

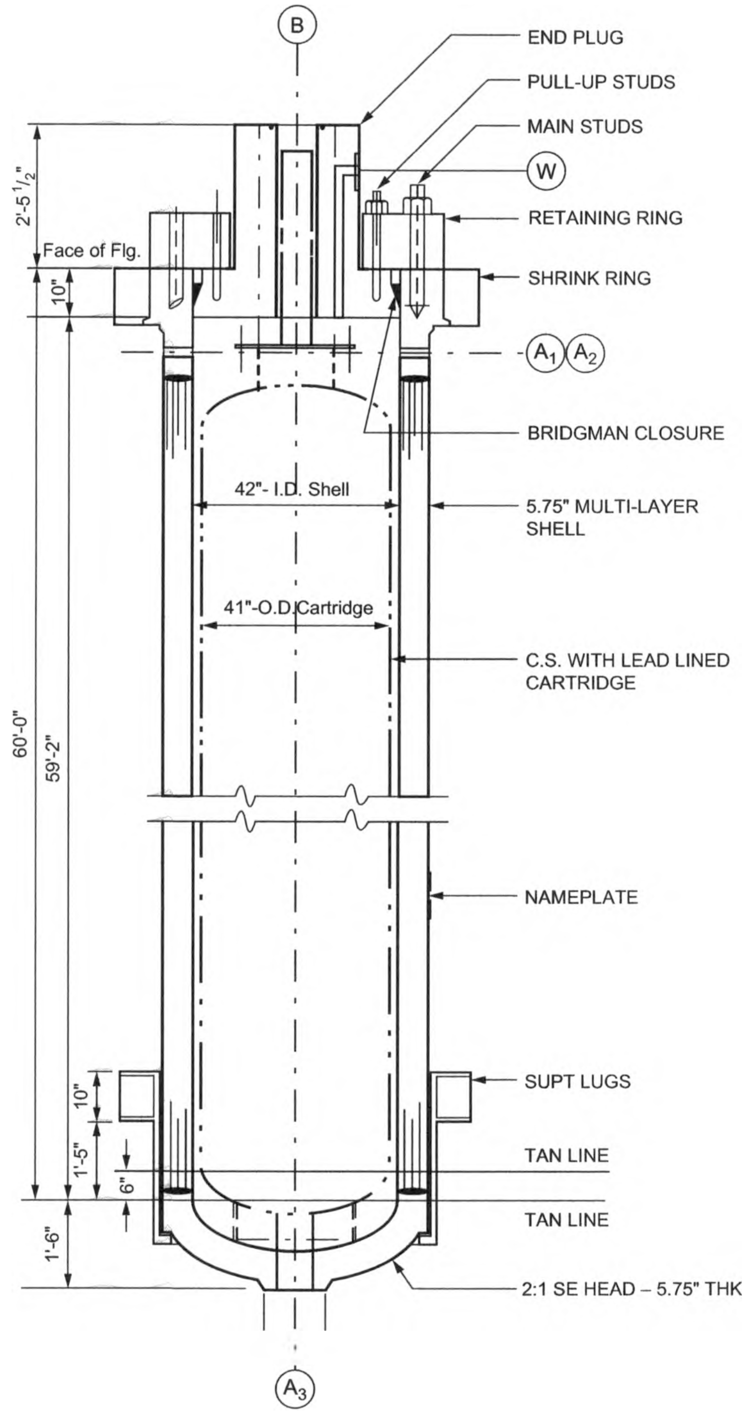
Matl : SA-105

ASME VIII-2

4960 PSIG @ 250°F

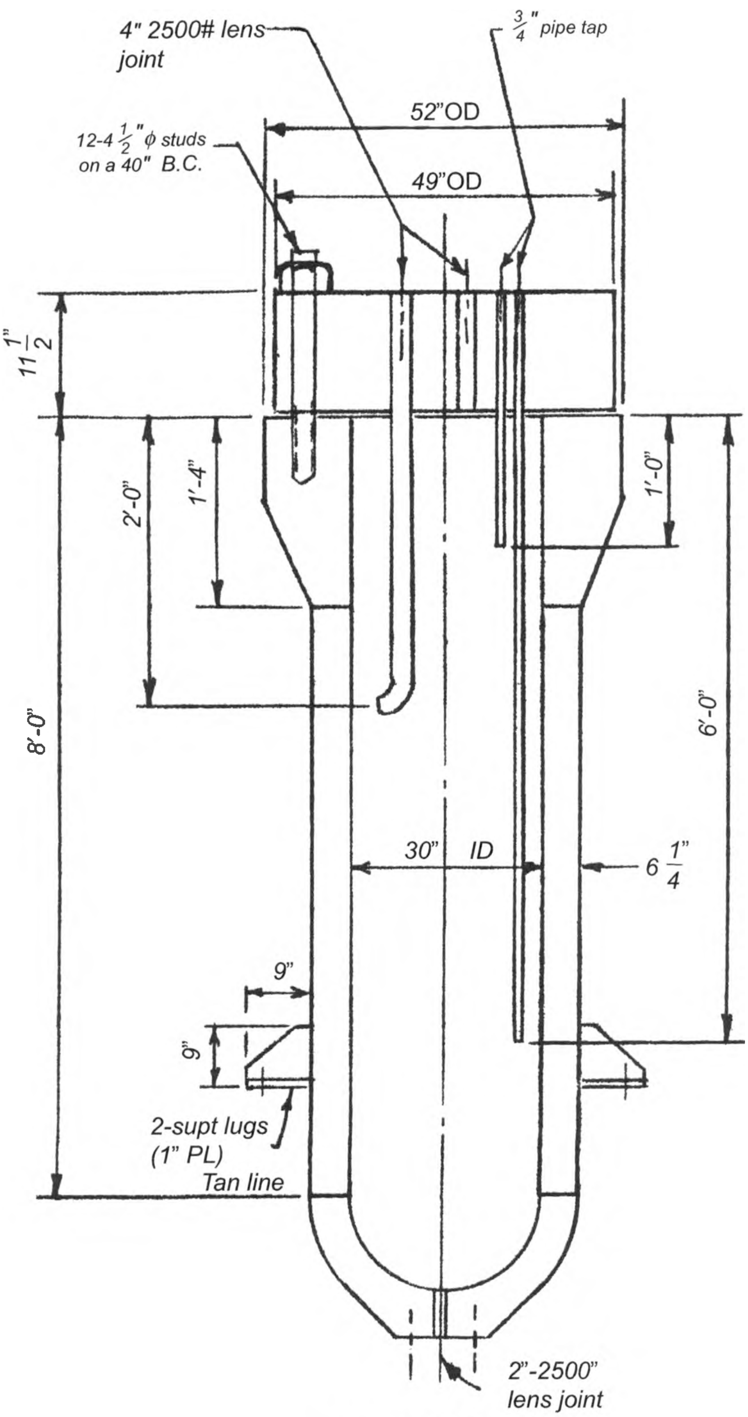



**EXAMPLE NO. 2**
Matl: SA-105
ASME VIII-1
3800 PSIG @ 350°F




**EXAMPLE No. 3**

Urea Reactor
Matl: SA-105
4,000 PSIG @ 400°F

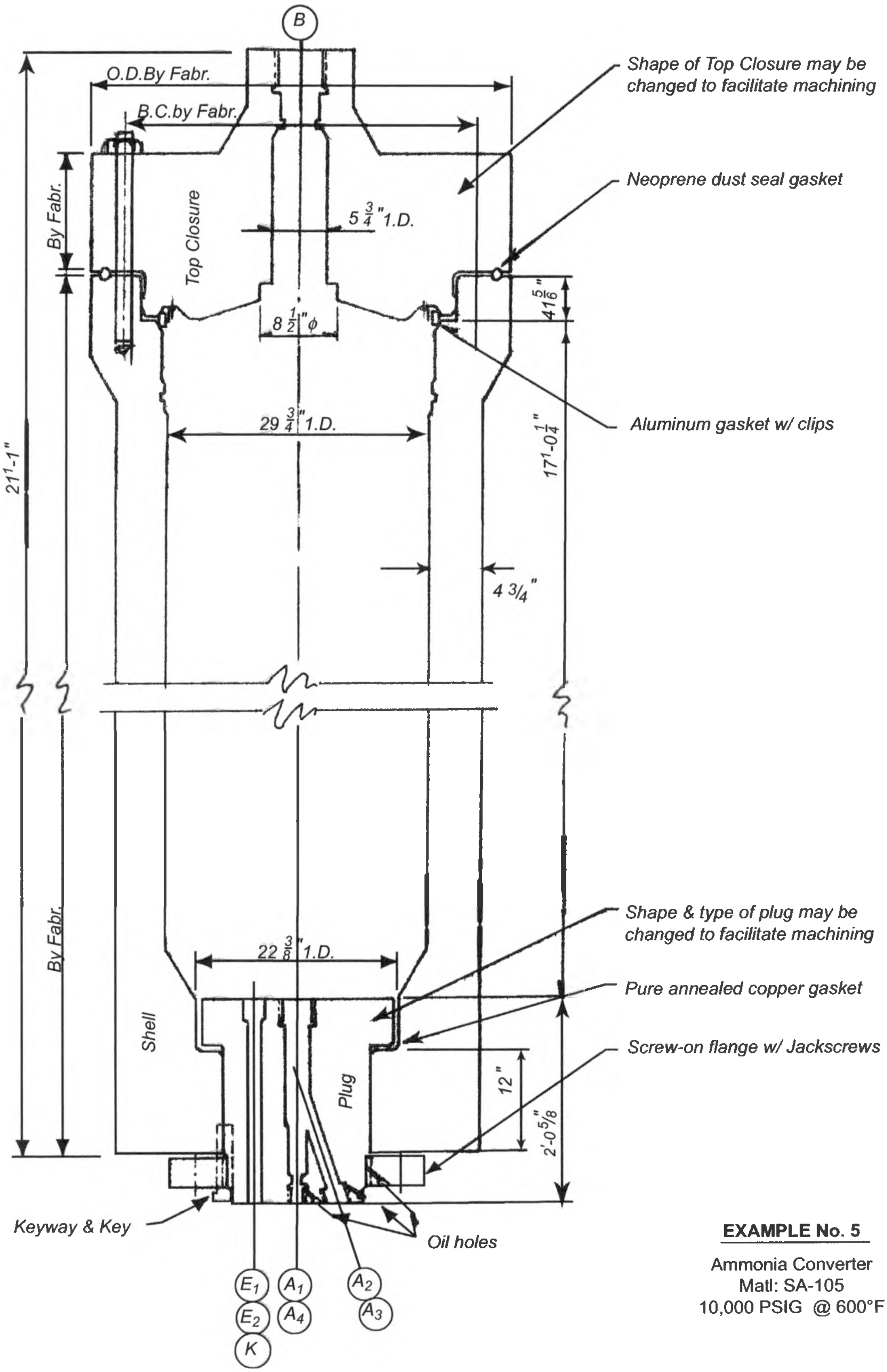



**EXAMPLE No. 4**

Ammonia Separator

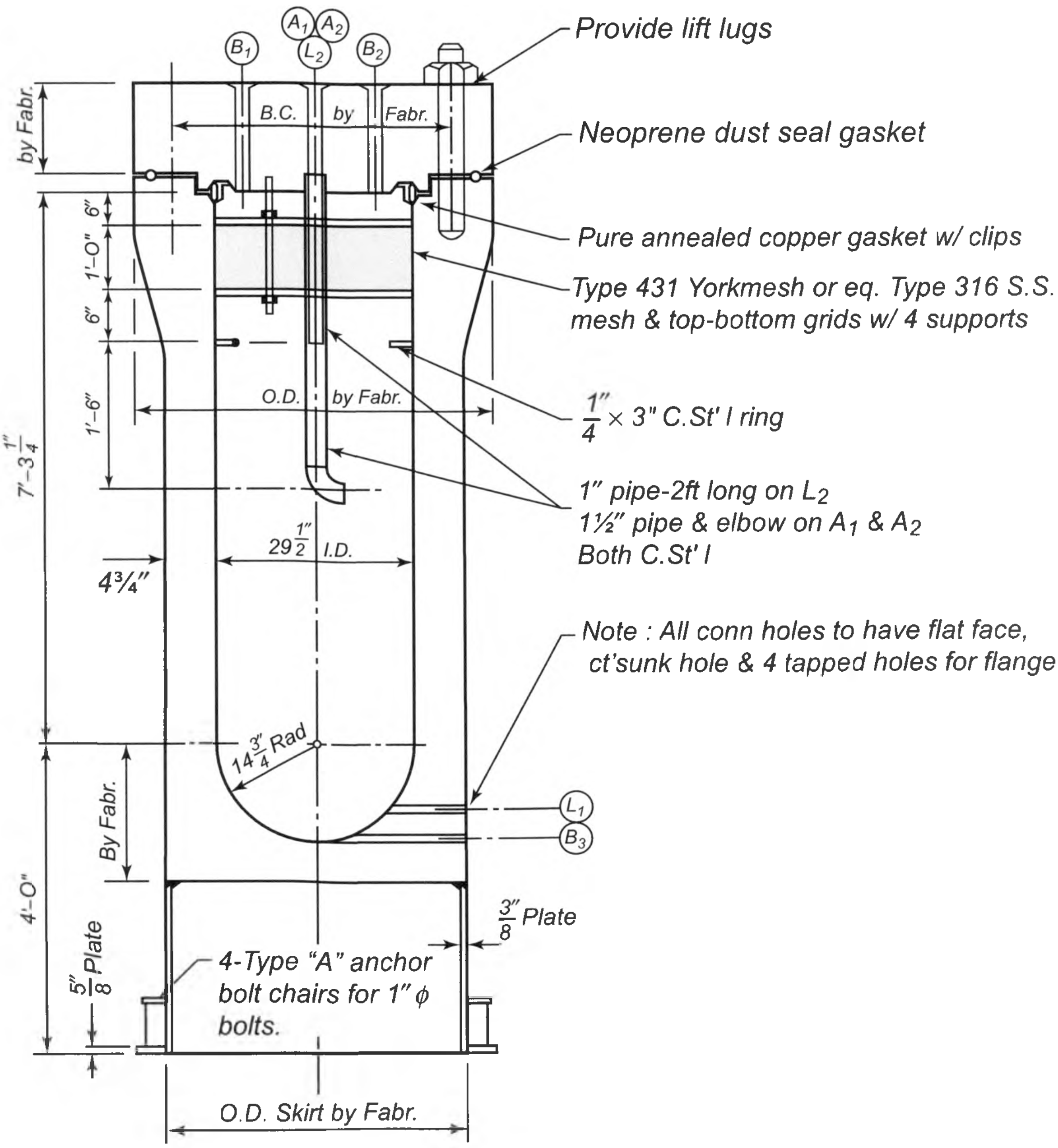
Matl: SA-105

5,800 PSIG @ 90°F



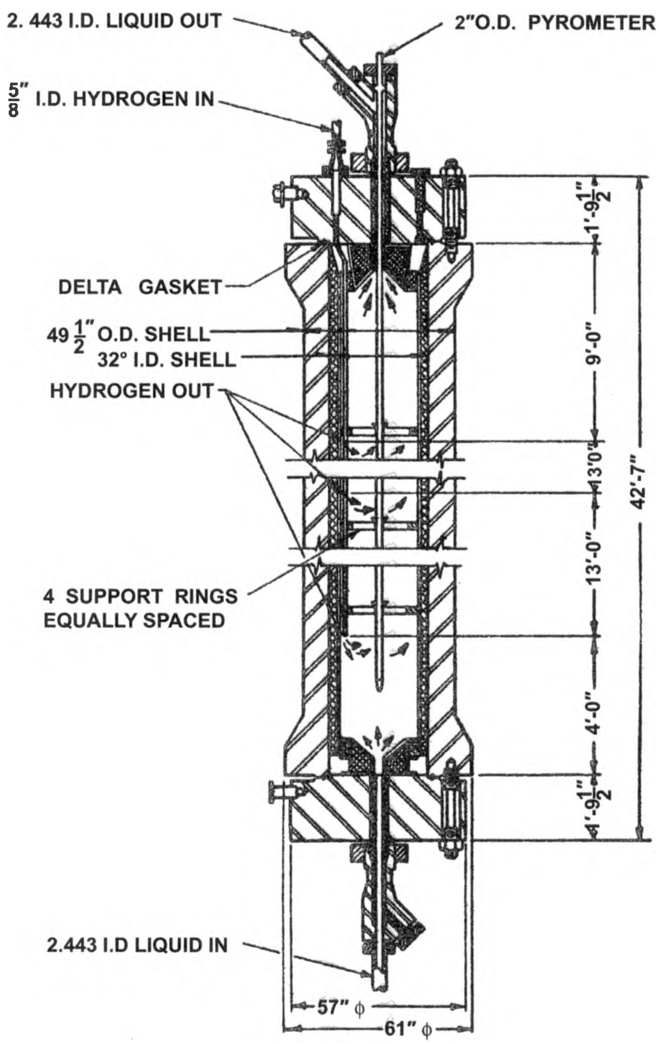

**EXAMPLE No. 5**

Ammonia Converter
Matl: SA-105
10,000 PSIG @ 600°F




**EXAMPLE No. 6**

High Pressure Separator
Matl: SA-105
10,000 PSIG @ 300°F




**EXAMPLE No. 7**

Liquid Phase Converter Assy
10,300 PSIG @ 1,000°F

*Shell material and properties*

3% Cr
.65% Ni
.3% Mo

$F_U$ = 100 KSI
$F_Y$ = 55 KSI
S = 17.5 KSI

Shell Thk: 8.75" solid wall (forging)

*Head material*

4-6% Cr
1-1.25% Ni
.4-.8% Mo

S= 17.5 KSI

Head Thk: 21.50" Thk

*Head Studs*

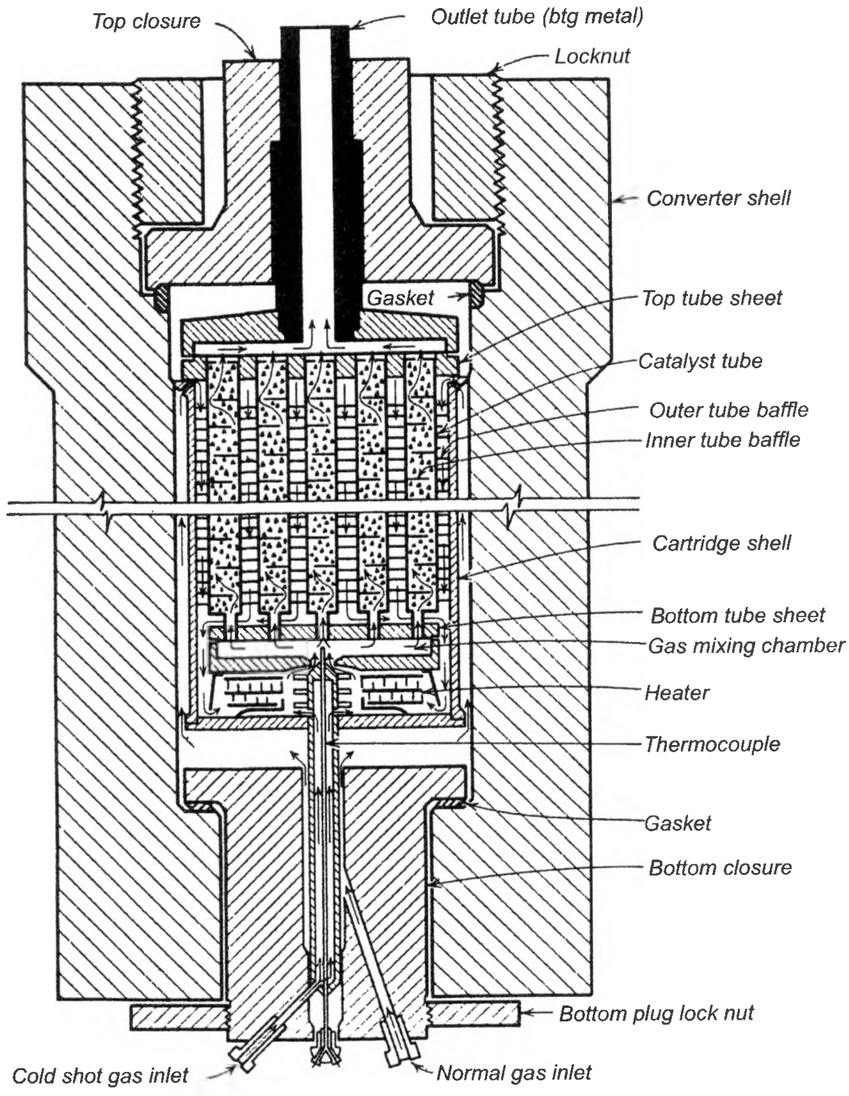
(12) 5.75" Dia on 45.50" B.C. SA-193-B16 (AISI 4340)

$F_U$ = 120 KSI
$F_Y$ = 90 KSI
S = 30.5 KSI

*Gaskets:* Annealed AISI 1020 Steel, Delta Ring

Misc Data:

- Vessel is refractory lined
- Shell Wt: 160 Kips
  Head Wt: 15 K (ea)
  Total Wt: 190 Kips
- No openings in the converter shell




**EXAMPLE NO.8**

Ammonia Converter
Matl: SA-105
10,000 PSIG @ 600°F

## 2.0. Shell Design

### 2.1. Introduction

When the thickness of a cylindrical pressure vessel becomes relatively large in comparison to the diameter ($t > .5R$), the variation of the stresses between the inner surface and outer surface becomes appreciable. At this point the ordinary membrane or "average" stress formulas are not a satisfactory indicator of the stresses.

In addition to the simple membrane stress of the cylinder, the shell is subjected to a radial stress due to the direct application of the pressure against the wall. This is a compressive stress and is insignificant for thin walled pressure vessels when compared to the other principal stresses. But the radial stress becomes more significant as the pressure and thus the thickness is increased.

In a thick walled cylinder, subjected to internal pressure, both circumferential and radial stresses are maximum on the inside surface. However, failure of the shell does not begin at the bore but in fibers along the outside surface of the shell. Although the fibers on the inside surface yield first, they are incapable of failing because they are restricted by the surrounding material, the outer regions of the shell. Above the elastic breakdown pressure, the region of plastic flow, or "overstrain" moves radially outward and causes the circumferential stress to reduce at the inner layers and increase in the outer layers. Thus the maximum hoop stress is reached there first and eventual failure begins there.

In 1833, Lamé developed a series of equations to find the greatest principal stresses in order to determine when the elastic limit was reached. Lamé proposed that elastic failure is considered to occur when the elastic limit of the material is reached. Beyond this limit, the material is permanently deformed or ruptured.

Coulomb and Tresca theorized, that the elastic limit was reached only when the shear stress reached it's maximum value. The basis of their theory was based on the actual failure mode of material. Material stretched beyond its elastic limit actually begins to fail along slip lines at 45 degrees from the applied force. The molecular bonds between adjoining atoms begin to slip along shear planes. The process is called twinning and results in Leuder's lines, visible at the point where failure begins, typically at some point of stress concentration.

**Safety Factors for High Pressure Design.** Designs below the creep range, are based on the elastic limit of the material. Historically, pressure vessel design for tensile requirements for continuous service was based on the lesser of;

a. ¼ Tensile (UTS)
b. ⅔ Yield
c. 1% strain for 100,000 hours for materials in the creep range

The safety factor for Division 1 was increased from 4:1 tensile to 3.5:1 in 2000. The Division 2 safety factor was historically 3:1 until 2007, when it was changed to 1:2.4. Historically, this translates to safety factors of 1.5 based on yielding or 4 based on bursting.

In the standard wall thickness calculation for ASME Section VIII, Division 1:

$$t = PR/(SE - .6P)$$

as the quantity SE - .6P approaches 0, the thickness approaches infinity. As the pressure increases, the allowable stress must be increased to higher than 60% of the design pressure for the equation to be valid. This becomes impractical for very high pressure applications, and a different theory of failure and design must be used.

Design of cylindrical shells is divided into the following categories;

a. Monobloc
b. Layered
c. Autofrettage

Monobloc cylindrical shells can be designed by one of three methods;

a. Standard method (KD-220)
b. Elastic-Plastic Analysis (KD-230)
c. Linear Elastic Analysis (KD-240)

ASME Section VIII, Division 3 does not give equations for calculating wall thickness as in other sections or divisions of the Code. Instead, an equation is given for maximum design pressure (KD-221.1). This equation can then be used to calculate the required thickness by iteration. A good starting point for determining wall thickness is to use the equation and allowable stresses for ASME Section VIII, Division 2 and then iterate the thickness until the maximum design pressure allowed exceeds the required design pressure.

## 2.2. Design of Thick Walled Cylinders

Thick walled cylinders can be designed to either ASME Section VIII, Divisions 1, 2 or 3. However, what is different in each of the Divisions is the allowable stress. Using the same allowable stress in the equations for the three divisions will yield approximately the same results. An example has been provided to illustrate this.

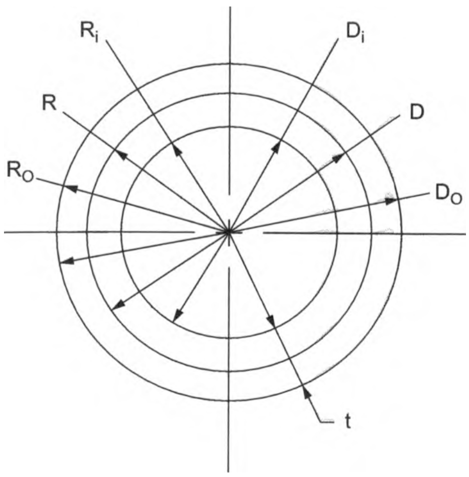
There are advantages to using Divisions 2 or 3 from a purely economic cost standpoint, since these two Divisions will yield lesser wall thickness, and therefore cost less to produce. However, this may not be the only consideration for choosing which Division to use. ASME Section VIII, Division 1 can be used economically up to about 10,000 PSI. However, if the designer wishes to reduce the wall thickness to the maximum extent possible, then there are advanced techniques in Division 2 and 3 that may be utilized. These advanced techniques are not covered in this book at this time.

These advanced techniques are as follows;

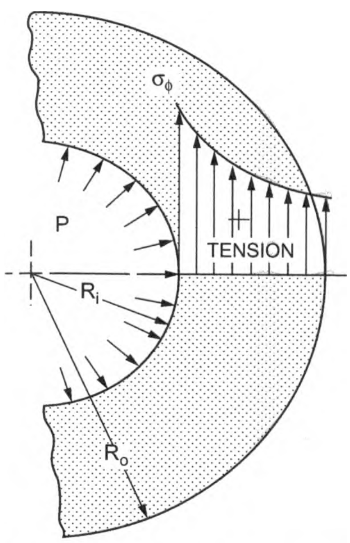
1. Autofrettage technique: Although multi-layer vessels can be built to either Divisions 1, 2 or 3, only Division 3 allows credit for the residual compressive stresses induced by the autofrettage technique.
2. Elastic Analysis
3. Limit Load Analysis
4. Elastic-Plastic

### Nomenclature

C.a. = Corrosion allowance, in
$D$ = Any intermediate diameter, in
DT = Design temperature, °F
$E$ = Modulus of elasticity, PSI
$F_y$ = Minimum specified yield strength, PSI
$P$ = Design internal pressure, PSIG
$P_b$ = Burst pressure, PSIG
$P_f$ = Pressure at which yielding occurs across the entire thickness, PSIG
$P_m$ = MAWP, PSIG
$P_y$ = Pressure at which yielding begins at the bore, PSIG
$R$ = Any intermediate radius, in
$S$ = Division 2 allowable stress, PSI
$S_T$ = Tensile stress at design temperature, PSI
$Y$ = Ratio, $D_o / D_i$
$Z$ = Ratio, $D_o / D$
$\delta$ = Radial expansion, in
$\varepsilon_X$ = Longitudinal strain, in/in
$\varepsilon_\phi$ = Circumferential strain, in/in
$\sigma_X$ = Longitudinal stress, PSI
$\sigma_\phi$ = Circumferential stress, PSI
$\sigma_r$ = Radial stress, PSI
$\sigma_{XR}$ = Longitudinal stress at radius R, PSI
$\sigma_{\phi R}$ = Circumferential stress at radius R, PSI
$\sigma_{rR}$ = Radial stress at radius R, PSI
$\tau$ = Shear stress, PSI
$\tau_R$ = Shear stress at radius R, PSI
$Z_1$ = ASME VIII-1 ratio

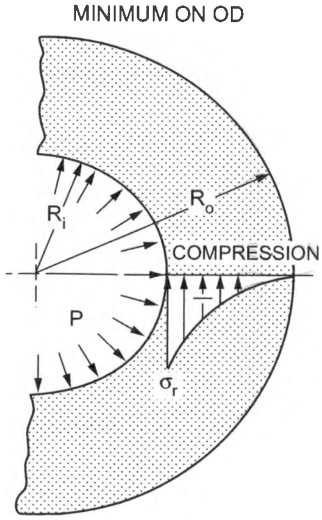



Dimensions of Thick Walled Cylinder




**(a)** Tangential stress distribution

MAXIMUM ON ID
MINIMUM ON OD




**(b)** Radial stress distribution

MAXIMUM ON ID
0 AT OUTSIDE

A summary of the requirements of the three Divisions of the ASME Section VIII is as follows;

## 1.0. ASME VIII-1

Assuming E (Joint efficiency) is one, a thick walled cylinder is defined in Appendix 1-2 as meeting either of the following parameters;

1. $P > .385\ S$
2. $t > .5\ R$

For applications below either of these parameters, the standard equation (UG-27) for thin walled cylinders will yield the same results.

The formulas are;

$$t_r = R_i \left[ Z_1^{1/2} - 1 \right] \text{ or}$$

$$t_r = \left[ R_O \left( Z_1^{1/2} - 1 \right) \right] / Z_1^{1/2}$$

$$P_m = S(Z_1 - 1)/(Z_1 + 1)$$

$$\text{Where } Z_1 = (S + P)/[(S - P)]$$

$$\text{or } Z_1 = ((R_i + t)/R_i)^2$$

$$\text{or } Z_1 = (R_O / R_i)^2$$

## 2.0. ASME VIII-2

$$t_r = R_i \left( e^{P/S} - 1 \right)$$

$$P_m = (S\ t)/(R_i + .5\ t)$$

$$\text{or } P_m = (S\ t)/(R_O - .5\ t)$$

## 3.0. ASME VIII-3

ASME VIII-3 does not give a direct equation for calculating wall thickness. Neither does it give allowable stresses for materials. Instead it gives one simple equation for calculating allowable pressure. By iterative trials, a suitable wall thickness is assumed that will yield a pressure above design. By multiple trials the minimum wall thickness may be determined.

$$P_m = .667\ F_y \text{ Ln } Y$$

$$\text{Where } Y = R_O/R_i \text{ or } D_O/D_i$$

$$\text{If } S = .667\ F_y$$

$$\text{And } Y = D_O/D_i = (D_i + 2\ t)/D_i = 1 + 2\ t/D_i$$

Then; $P_m = S \, Ln\, [1 + (2\,t)/D_i]$

And $e^{P/S} = 1 + 2\,t/D_i$

Solving for t:

$$t_r = .5\, D_i\left(e^{P/S} - 1\right)$$

Therefore the equations for Division 2 and 3 are identical.

### Comparison of ASME Sections Viii, Divisions 1, 2 & 3

If the same allowable stress is used it will provide a direct comparison between the various divisions;

Given;
Material: SA-105
$P = 8000$ PSIG
$DT = 400°F$
C.a. = 0.25 in
$F_y = 30.8$ KSI
$S = .667\, F_y = 20.53$ KSI
$R_i = 21.25$ in Corr
$D_i = 42.50$ in Corr

### ASME VIII – 1

$P/S = 8{,}000/20{,}530 = .3897$

Therefore this vessel qualifies as a "thick" cylindrical shell.

$$Z_1 = (S + P)\,/\,[(S - P)]$$
$$= (20530 + 8000)/(20530 - 8000) = 2.2769$$

$$t_r = R_i\left(Z_1{}^{1/2} - 1\right) = 21.25\left(2.2769^{1/2} - 1\right)$$
$$= 10.815 \text{ in}$$

### ASME VIII – 2

$P/S = .3897$

$$t_r = R_i\left(e^{P/S} - 1\right)$$
$$= 21.25\left(e^{.3897} - 1\right) = 10.125 \text{ in}$$

### ASME VIII – 3

Assume, $t = 10.125 + .25 = 10.375$

$D_O = 42 + 2\,(10.375) = 62.75$

$Y = D_O/D_i = 62.75/42.5 = 1.4765$

$$P_m = .667\, F_y\, Ln\, Y = .667(30800)\, Ln\, 1.4765$$
$$= 8001 \text{ PSI}$$

Therefore all three divisions yield very similar results if the allowable stress are the same. This proves that the basis of the equations for determination of wall thickness for all three divisions are practically the same.

### Lamé Equations

The equations shown on the following worksheet for determining shell thicknesses, are all based on Lamé equations. Reviewing the equations for various shell stresses show that they are in complete agreement with the ASME equations and yield the exact same results. The following equations serve to illustrate this point;

Lamé equations;

$$\sigma_\phi = \left[(P\,R_i{}^2)/(R_O{}^2 - R_i{}^2)\right]\left[1 + R_O{}^2/R_i{}^2\right]$$
$$\tau = (P\,R_O{}^2)/(R_O{}^2 - R_i{}^2)$$

Alternate equations;

$$\sigma_\phi = \left[P\,(Y^2 + 1)\right]/\left[Y^2 - 1\right]$$
$$\tau = (P\,Y^2)/(Y^2 - 1)$$

| WORKSHEET FOR DETERMINING THICKNESS OF CYLINDRICAL SHELL | | |
|---|---|---|
| | | |

| GIVEN | |
|---|---|
| MATERIAL | |
| P | |
| DT | |
| Ca | |
| $F_y$ | |
| S Div 2 | |
| $D_i$ | |
| $R_i$ | |
| E | |
| $S_T$ | |

| CALCULATE | | |
|---|---|---|
| TERM | EQUATION | VALUE |
| $t_r$ Div 2 (2) | $R_i\,[\,e^{P/S} - 1\,]$ | |
| t | Actual shell thickness used | |
| $D_O$ | $R_i + 2\ t$ | |
| Y | $D_O / D_i$ | |
| Z | $D_O / D$ | |
| $P_m$ | $.667\ F_y \ \mathrm{Ln}\ Y$ | |

| STRESSES | | |
|---|---|---|
| TERM | EQUATION | VALUE |
| $\sigma_x$ | $P\ /\ (Y^2 - 1)$ | |
| $\sigma_\phi$ | $P\,[\,(Y^2 + 1)\ /\ (Y^2 - 1)\,]$ | |
| $\sigma_r$ | $(-)P$ | |
| $\tau$ | $(P\ Y^2)\ /\ (Y^2 - 1)$ | |
| $P_y$ | $.577\ F_y\,[\,(Y^2 - 1)\,/\,Y^2]$ | |
| $P_f$ | $1.155\ F_y\ \mathrm{Ln}\ Y$ | |
| $P_b$ | $S_T[\,(Y^2 + 1)\ /\ (Y^2 - 1)\,]$ | |
| $\sigma_{XR}$ | $[\,P\,/\,(Y^2 - 1)\,]\,[\,1 + Z^2]$ | |
| $\sigma_{\phi R}$ | $[\,P(Z^2 + 1)\,]\,/\,(Y^2 - 1)$ | |
| $\sigma_{rR}$ | $[\,-P(Z^2 - 1)\,]\,/\,(Y^2 - 1)$ | |
| $\tau_R$ | $(P\,Z^2)\,/\,(Y^2 - 1)$ | |
| $\varepsilon_X$ | $[\,1.7\,P\,]\,/\,[\,E\,(Y^2 - 1)\,]$ | |
| $\varepsilon_\phi$ | $[\,.4\ P\,]\,/\,[\,E\,(Y^2 - 1)\,]$ | |
| $\delta$ | $t_\phi\ R_O$ | |
| | | |
| | | |
| | | |

TRIALS TO DETERMINE SHELL THICKNESS PER ASME VIII-3

| TRIAL | 1 | 2 | 3 | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|
| t | | | | | | | |
| $D_o$ | | | | | | | |
| Y | | | | | | | |
| $P_m$ | | | | | | | |

**NOTES:**

1. Y = Z at the bore, Z = 1 at the outside of the cylinder
2. Use thickness calculated from Division 2 as first trial. Continue trials until $P_m$ is close to P.

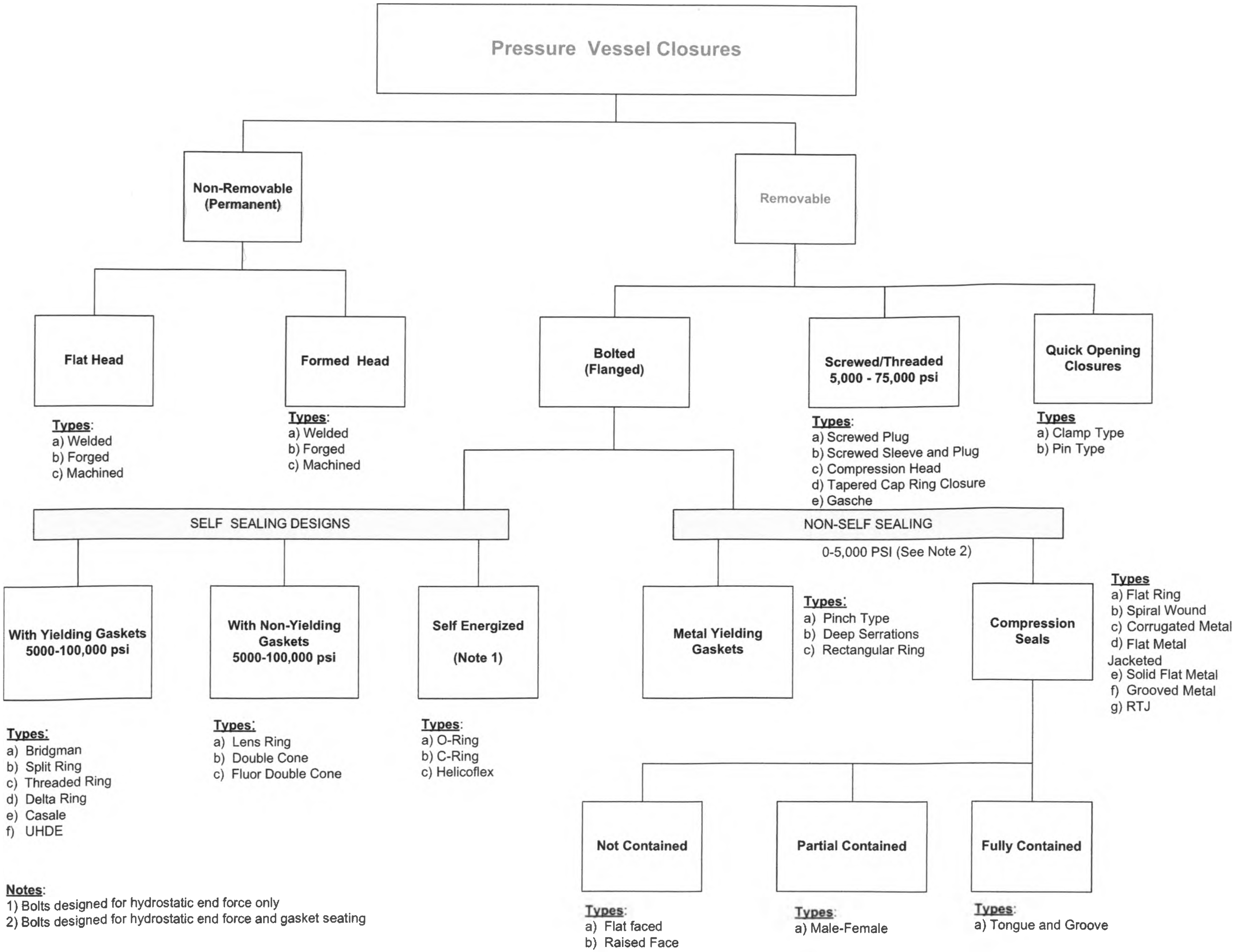
| WORKSHEET FOR DETERMINING THICKNESS OF CYLINDRICAL SHELL | | Example |
|---|---|---|
| | | |

| GIVEN | | | STRESSES | | |
|---|---|---|---|---|---|
| MATERIAL | SA-723-2-2 ( 2-3/4 Ni 1-1/2 Cr 1/2 Mo) | | TERM | EQUATION | VALUE |
| P | 15,000 PSIG | | $\sigma_x$ | $P / (Y^2 - 1)$ | 28,425 |
| DT | 600°F | | $\sigma_\phi$ | $P [ ( Y^2 + 1 ) / ( Y^2 - 1) ]$ | 71,850 |
| Ca | 0 | | $\sigma_r$ | (-)P | (-) 15,000 |
| $F_y$ | 106.7 KSI | | $\tau$ | $( P Y^2 ) / ( Y^2 - 1 )$ | 43,425 |
| S Div 2 | 56.3 KSI | | $P_y$ | $.577 F_y [ ( Y^2 - 1 ) / Y^2 ]$ | 21,270 |
| $D_i$ | 36 in | | $P_f$ | $1.155 F_y \ln Y$ | 26,110 |
| $R_i$ | 18 in | | $P_b$ | $S_T [ ( Y^2 + 1 ) / ( Y^2 - 1) ]$ | 646,658 |
| E | $25.1 \times 10^6$ PSI | | $\sigma_{XR}$ | $[ P / ( Y^2 - 1) ] [ 1 + Z^2 ]$ | 28,425 |
| $S_T$ | 135 KSI | | $\sigma_{\phi R}$ | $[ P ( Z^2 + 1) ] / (Y^2 - 1)$ | 63,196 |
| CALCULATE | | | $\sigma_{rR}$ | $[ - P ( Z^2 - 1) ] / (Y^2 - 1)$ | (-) 6345 |
| TERM | EQUATION | VALUE | $\tau_R$ | $( P Z^2 ) / ( Y^2 - 1 )$ | 34,771 |
| $t_r$ Div 2 (2) | $R_i [ e^{P/S} - 1 ]$ | 5.49 | $\varepsilon_X$ | $[ 1.7 P ] / [ E ( Y^2 - 1 ) ]$ | 0.001925 |
| t | Actual shell thickness used | 4.25 | $\varepsilon_\phi$ | $[ .4 P ] / [ E ( Y^2 - 1 ) ]$ | 0.000453 |
| $D_O$ | $R_i + 2 t$ | 44.5 | $\delta$ | $t_\phi R_O$ | |
| Y | $D_O / D_i$ | 1.236 | | | |
| Z | $D_O / D @ R_m$ | 1.106 | | | |
| $P_m$ | $.667 F_y \ln Y$ | 15,078 PSI | | | |

| TRIALS TO DETERMINE SHELL THICKNESS PER ASME VIII-3 | | | | | | | |
|---|---|---|---|---|---|---|---|
| TRIAL | 1 | 2 | 3 | 4 | 5 | 6 | 7 |
| t | 5 | 4.5 | 4.25 | | | | |
| $D_o$ | 46 | 45 | 44.5 | | | | |
| Y | 1.277 | 1.25 | 1.236 | | | | |
| $P_m$ | 17,436 | 15,872 | 15,078 | | | | |

**NOTES:**

1. Y = Z at the bore , Z = 1 at the outside of the cylinder
2. Use thickness calculated from Division 2 as first trial. Continue trials until $P_m$ is close to P.


Pressure Vessel Closures
Non-Removable (Permanent)
Removable
Flat Head
Types:
a) Welded
b) Forged
c) Machined
Formed Head
Types:
a) Welded
b) Forged
c) Machined
Bolted (Flanged)
Screwed/Threaded 5,000 - 75,000 psi
Types:
a) Screwed Plug
b) Screwed Sleeve and Plug
c) Compression Head
d) Tapered Cap Ring Closure
e) Gasche
Quick Opening Closures
Types
a) Clamp Type
b) Pin Type
SELF SEALING DESIGNS
NON-SELF SEALING
0-5,000 PSI (See Note 2)
With Yielding Gaskets 5000-100,000 psi
Types:
a) Bridgman
b) Split Ring
c) Threaded Ring
d) Delta Ring
e) Casale
f) UHDE
With Non-Yielding Gaskets 5000-100,000 psi
Types:
a) Lens Ring
b) Double Cone
c) Fluor Double Cone
Self Energized (Note 1)
Types:
a) O-Ring
b) C-Ring
c) Helicoflex
Metal Yielding Gaskets
Types:
a) Pinch Type
b) Deep Serrations
c) Rectangular Ring
Compression Seals
Types
a) Flat Ring
b) Spiral Wound
c) Corrugated Metal
d) Flat Metal Jacketed
e) Solid Flat Metal
f) Grooved Metal
g) RTJ
Not Contained
Types:
a) Flat faced
b) Raised Face
Partial Contained
Types:
a) Male-Female
Fully Contained
Types:
a) Tongue and Groove
Notes:
1) Bolts designed for hydrostatic end force only
2) Bolts designed for hydrostatic end force and gasket seating

## High Pressure Closures

### 3.1. Introduction

Closures at the ends of vessels may be permanent, removable, screwed or flanged. The type of closure at the end of a pressure vessel depends on the frequency of opening, the size of the vessel, the temperature of the vessel, the corrosive conditions, openings and nozzles through the closure and the type of access provided by the removable head. Quick opening closures are available up to about 20,000 PSI.

**Flanged Closures.** Conventional flange seals are not particularly efficient as pressure seals. The reason is because as the pressure increases, it results in reduced gasket pressure. In order to maintain the seal, larger bolts are required. Larger bolts in turn result in a larger hub dimensions and thicker flanges to counteract the large bending moments. At some point the law of diminishing returns takes effect and the flange cannot be made large enough to resist the combined effect of bending plus hydrostatic end forces. Thus an alternate strategy is required. This alternate strategy is the principle of "yielding gaskets". Yielding gaskets can be either self-sealing or not self-sealing.

Standard flange design employs the principle of "non-yielding" gaskets. That is, a gasket is designed to have its surface yield, up to the point required to seal the joint. The flange face must be harder than the gasket material. As you approach the point of higher and higher pressure, you reach the point where the gasket and flange face cannot be made any harder. At this point the flanges become outlandishly large to uniformly distribute the gasket load.

For high pressure applications, the flange should be designed around a gasket that will yield. Yielding gaskets create a seal without the extreme force required to seat a gasket. Soft metal gaskets such as aluminum or copper have been used as yielding gaskets. A number of designs and methods have been used to cause a temporary stress concentration on the gasket, thus creating an initial flow of the material and thus a resultant perfect seal. Examples of yielding gaskets include pinch type, deep serrations, etc.

Self-sealing gaskets are those which are sealed by the system pressure. The system pressure acts to force the gasket to deform into a cavity that creates the seal. The seal remains intact by the system pressure.

Gaskets can be yielding or self-sealing or both. For example a delta gasket and Bridgman gasket are both yielding and self-missing sealing. A double cone and lens ring are both self-sealing but are non-yielding gaskets.

There are seven categories of removable end closures based on the type of gasket utilized. Some gasket types fit into more than one category. These are as follows;

1. Compression Seals - Standard Flanged Joints (not typically used for high pressure)
   a. Flat Ring
   b. Spiral Wound
   c. Corrugated Metal
   d. Flat Metal Jacketed
   e. Solid Flat Metal
   f. Grooved Metal
   g. Ring Joint
2. Types of Metal Yielding Gaskets
   a. Pinch Design
   b. Deep Serrations
   c. Delta Ring
   d. Rectangular Ring
   e. Lens Ring
3. Self-Sealing Closures with Yielding Gaskets
   a. Bridgman
   b. Split Ring Type
   c. Threaded Ring Type
   d. Delta Rings
4. Self-Sealing Closures with Non-Yielding Gaskets
   a. Lens Ring
   b. Double Cone
   c. Fluor Double Cone
5. Self Energized Gaskets (Self-Sealing)
   a. O-Ring
      1. Elastomeric
      2. Metal
         a. Plain
         b. Self Energized
         c. Pressure Filled
      3. Helicoflex
   b. Metal C-Ring
      1. For Internal Pressure
      2. For External Pressure
      3. For Axial Pressure
6. Breech Lock Closures
7. Other
   a. Outside Bump Ring
   b. Inside Bump Ring
   c. Gasche Spring Closure
   d. Casale Joint
   e. UHDE Type

**Materials.** Metal gaskets should be used for all applications over 930°F. A rough rule of thumb has been proposed by J. Harvey [1] that when the product of the pressure times the temperature exceeds 250,000, metal gaskets should be used.

Extreme care must be exercised in the selection of materials for sealing elements. The major considerations are as follows;

a. Ultimate strength of individual components
b. Compatibility of dissimilar metals
c. Difference in degree of component thermal expansion
d. Relative hardness and yield strength of mating components

### Flat Ring Gaskets

These are conventional seals or gaskets and are also known as compression seals. There are three methods of utilizing flat ring gaskets:

a. Non-retained
b. Partially retained
c. Fully retained

In high pressure service a non-retained style gasket should never be used. This is a conventional type gasket and the potential to blow out is too great. Spiral wound gaskets have been used up to about 2500 psi service without being fully contained. Examples of partially retained gaskets are male-female flange facings, since they are not confined on the ID. They do however provide sufficient blowout protection.

### Ring Joint Gaskets

Ring Joint Gaskets are neither yielding nor self-sealing. The sealing capabilities of RTJs are based on the sheer force required to seat the gasket and maintain a seal during operation. In the past, ring joint gaskets were primarily used for class 1500 or 2500 service. In overlayed, hydrogen service however the threat of cracks developing at the root of the groove has become highly probable and caused considerable concern within operating companies. The propagation of cracks in some cases has gone as deep as 7 inches into the base material. As a result, some owners are no longer specifying RTJ gaskets for new services, and are replacing existing flanges wherever possible. This includes both vessels as well as piping systems.

There are two types of ring joint gaskets. They are the oval ring and the octagonal ring. The oval ring provides a better seal but it is more difficult to produce the dimensional accuracy or surface finish desirable. Conversely, the octagonal ring can be produced more accurately, but requires more torque to cause the plastic flow required for sealing.

Ring type joints are generally considered only for a maximum of 4500 PSI. However API Spec 6A flanges have standard sizes for RTJ flanges to 20,000 PSI service.

Fortunately for ring joint gaskets it is extremely difficult to crush the gasket. Usually the bolts will elongate before the gasket is crushed. Gasket crushing is near the limits of the flanges ability to transmit the load. In addition the gaskets should always be softer than the flange itself. Then gaskets can plastically flow without damaging the surface of the groove.

Reusing of ring joint gaskets is not recommended due the probability of work hardening of the gasket material.

### Self-sealing closures with Yielding Types of Gaskets

Most gaskets utilized for high pressure applications are of the self-sealing or self-energized types. The amount of bolting preload required to produce a seal is quite small. However, sufficient bolting force must be applied to resist hydraulic loading. The bolting requirement for gasket seating is never governing for these types of gaskets. This is why these types of gaskets are used. The hydraulic loading, hydrostatic end force, will always be the governing condition for these gaskets. Thus in most calculations, the gasket seating condition is ignored. Types are as follows;

a. Bridgman Closure
b. Delta Ring

Bridgman Closure: The Bridgman Closure is an Axial type of gasket as opposed to a radial type. The pressure reacts on the exposed gasket area in an axial rather than a radial direction. These are generally only considered above 1500 PSI.

Delta Ring: The Radial pressure acting over the height of the gasket produces a sealing force reaction at the tips. Initial seating at the tips is produced by controlled mechanical interferences. These gaskets are used in pressures in the 5,000 to 20,000 psi range. Galling does not occur if the angle of the groove is slightly greater than the angle of the ring. They do not require any initial seating or bolt stress for gasket seating.

**Self-Energized Gaskets.** O-Rings: Elastomeric o-rings are used for high pressure but low temperature, generally below 400°F. Metal O-Rings have been used for very high temperatures (1500°F) but low pressures. A carefully dimensioned groove or slot is provided for the O-Rings. The mating surfaces are normally separated by .9 of the diameter of the O-Rings. Under pressure, the internal pressure forces the O-Rings to the side of the groove and into the small gap between the mating flanges, thus sealing the joint. Pressures as high as 60,000 psi have been obtained with O-Rings.

### Self-Sealing Closures with Non-Yielding Gaskets

Lens Ring: Normally used on small flanges. As the internal pressure increases, it acts on the inside surface of the ring, and forces it to the edge of the cone. Pressures as high as 45,000 psi have been obtained for very small sizes. Often considered one of the fastest opening closures.

Double Cone: Initial seating load is created by the stress to which the main bolts are tightened. The joints and gaskets must be machined to very high tolerances and very smooth finishes.

### Design Methods

Lens, delta, double cone and Bridgman gaskets are all exposed to the vessel internal pressure. There are no standard calculations for determining the forces and moment induced into the flange as a result of these loadings. Accordingly, a free-body diagram is required to determine the additional loads put into the flanges and bolting as a result of pressure on the gasket. Detailed procedures are given for determining these effects for the Bridgman gasket. Similar treatments can be developed for the lens and delta types.

Because of the small contact areas and yielding gaskets, the gasket seating forces will never govern design of these flanges and thus can be ignored. The bolting forces due to hydrostatic end force will govern these flanges.

**Table 8-1**
**Closures for high pressure vessels**

| Closure Type | Pressure Range PSIG | Inside Dia Range, in | Temperature Range, °F | Time for Opening | Ease of Fabrication | Remarks |
|---|---|---|---|---|---|---|
| Solid End - Bored | Unlimited | 6 | Metal | Permanent | Simple | Suitable for solid alloy or lined construction |
| Solid End - Forged | Unlimited | 6-72 | Metal | Permanent | Costly | Solid or electroplated |
| Flat Welded Head | 1,000 - 2,000 | 6 | Low | Permanent | Simple | Solid or thin metal liner |
| Formed Vessel Head - Welded Hemi or Ellipsoidal | 10-15,000 | Unlimited | Metal | Permanent | Simple | Solid, clad and weld overlay. Openings for ID access very expensive |
| Weld Neck Flg w/ Blind | 6,000 | 24 | Bolt Metal | Long | Simple | Solid, clad and weld overlay |
| Screwed Flg W/ Blind | 6,000 | 12 | Moderate | Long | Simple | Lining & clad difficult |
| Studding Flg W/ Blind | 10,000 | Unlimited | Metal | Long | Simple | Solid, clad and weld overlay |
| Clamp Type - Quick Opng Closure | 25,000 | >12 | Gasket Matl | Quick | Critical Machining. Many parts | Solid, clad and weld overlay |
| Screwed Bull Plug | 50,000 | 2 | Gasket Matl | Quick | Simple | |
| Screwed Sleeve End Plug | 5,000 | 3 | Gasket Matl | Quick | Careful Machine Work | Satisfactory for high temperatures |
| Threaded Closure | 100,000 (1) | 8 | Metal | Quick | Careful Machine Work | Solid, clad and weld overlay. Can use any type of self sealing gasket |
| | 50,000 (1) | 24 | Metal | Quick | Careful Machine Work | Solid, clad and weld overlay |
| Tapered Cap - Ring Closure | 5- 50,000 (1) | 6 - 16 | Stress in bolts | Quick | Careful Machine Work | Light weight, difficult to clea, economical for solid alloy construction |
| O-Ring | 30,000 | 12 | Gasket Matl | Depends on configuration | Relatively simple | Gasket easily replaced. Tight tolerances on o-ring groove |
| Bridgman | 100,000<br>30,000<br>15,000 | 1- 3<br>3-6<br>12- 48 | Gasket Matl<br>Gasket Matl<br>Gasket Matl | Long<br>Long<br>Long | Critical machining. Many parts. Tight tolerances and fine finishes | Careful cleaning and inspection after each opening. Use with flanged or threaded closure |
| Modified Bridgman | Unlimited | 12-48 | Gasket Matl | Long | | |
| Double Cone/Fluor Double Cone | 20,000 | 12-36 | Gasket Matl | Depends on configuration | Lapping of groove required | Use with either threaded or flanged closure |
| Delta Ring | 50,000 | 2-36 | Gasket Matl | Depends on configuration | Tight tolerances and fine finishes required on groove | Use with flanged closure only |
| Lens Ring | 45,000 | | Gasket Matl | Depends on configuration | Tight tolerances and fine finishes required on gasket & groove | Mainly used on piping systems |

*Notes:*
*Reference: Adapted from High Pressure Technology, E.W.Comings, McGraw-Hill Book Co, 1956*
*1. Pressure limit is for Self Sealing Gasket*

### 3.2. Gaskets

Most gaskets utilized for high pressure applications are of the self-sealing or self-energized types. The amount of bolting preload required to produce a seal is quite small. However, sufficient bolting force must be applied to resist hydraulic loading. The bolting requirement for gasket seating is never governing for these types of gaskets. This is why these types of gaskets are used. The hydraulic loading, hydrostatic end force, will always be the governing condition for these gaskets. Thus in most calculations, the gasket seating condition are ignored.

1. Delta Gaskets: These are self sealing gaskets sealed by internal pressure. They do not require any initial seating or bolt stress. Good for pressures greater than 5,000 PSI.
2. Bridgman Type: Sealed by internal pressure. Used for applications above 1500 PSI.
3. Lens Ring: Self sealing and one of the fastest opening closures.
4. All metal gaskets required where temperatures exceed 930°F.
5. As a general rule, all nozzles should go through the top and bottom closures and not through the shell.
6. Ring type joint gaskets (RTJ) are only considered to be practical to 4500 PSI. RTJ gaskets are not sealed by internal pressure and require large bolt areas for gasket seating. RTJs are not effective in sealing where there is a rapid decrease in temperature. The gasket area is very small in relation to the flanges, and therefore contracts more quickly then the flanges and bolts. This reduces the pre-load and can lead to leakage.

**Metal Gaskets.** Selection of the material for metal gaskets is based on the following considerations;

- Temperature
- Coefficient of expansion (COE)
- Ductility
- Resistance to corrosion

**Table 8-2**
**Properties of metal gaskets**

| Material | Max Temp °F | Max Recommended Temp °F | Thermal Conductivity, K | Coe, $\alpha$ |
|---|---|---|---|---|
| Lead | 200 | 100 | 240 | 16.3 |
| Brass | 500 | 250 | 803-1100 | 11.6 |
| Copper | 600 | 300 | 2700 | 9.8 |
| Aluminum | 800 | 400 | 840-1500 | 13.7 |
| 304 SST | 1000 | 500 | 150 | 9.1 |
| 316 SST | 1000 | 500 | 150 | 9.1 |
| Soft Iron, Low CS | 1000 | 500 | 360 | 8.6 |
| Titanium | 1000 | 500 | 105 | 7.1 |
| 410 SST | 1200 | 600 | 170 | 6.5 |
| Silver | 1200 | 600 | 2900 | 10.9 |
| 430 SST | 1400 | 700 | 181 | 6 |
| Nickel | 1400 | 700 | 637 | 9.2 |
| Monel | 1500 | 750 | 173 | 9.5 |
| 321/347 SST | 1600 | 800 | 112 | 10.2 |
| Inconel | 2000 | 1000 | 104 | 7.7 |
| Hastelloy | 2000 | 1000 | 87 | 7.9 |

*Notes:*

1. Units for Conductivity: BTu-in/hr/ft$^2$/°F
2. Units for coefficient of thermal expansion (COE): in/in/°F $\times 10^{-6}$

As a rough guide to determine whether metallic or nonmetallic gaskets should be used, multiply the operating pressure in psi times the operating temperature in °F. If the result exceeds 250,000, then metallic gaskets should be used. In general, nonmetallic gaskets should not be used above 850°F, or pressures above 1200 psi.

Of the above considerations, the coefficient of thermal expansion is most frequently overlooked. The importance of this consideration is best described by a comparison of the properties of the most common metals used (See Table 8-2). The coefficient of thermal expansion becomes important for applications where rapid temperature fluctuations or thermal gradients are present. Leakage can occur if the thermal coefficient of expansion between the gasket and flange material are too far apart.

In ammonia synthesis, copper was the preferred material for use with steel or low alloy flanges. Copper expands and contracts 9 times faster than steel. Leakages were found due to the large differences of the thermal expansion coefficients between the metals.

Although silver is used for special applications, copper, aluminum, steel and stainless steel are more common. Most alloys of aluminum should be limited to 400°F because the material becomes soft and has a tendency to extrude at under pressure through very small clearances.

Stainless steel has the problem of work hardening.

**O-Rings.** O-rings are known as self energized gaskets. That is, they use the system pressure to obtain a sealing force. A small amount of excess bolt load is required to prevent gasket blow-out. O-ring types are as follows;

1. Elastomeric
2. Metal
3. Helicoflex

### Rubber or Elastomeric O-Rings

Those made of elastomeric materials work on the basis where the gasket is confined in a small groove or recess and the pressure attempts to extrude it out. The tightness of the gasket is dependent on the viscous strength of the seal material and the exit gap created by the machining tolerances and/or a separation of the flange and shoulder. A back-up ring may be used to restrict the extrusion of the softer o-ring material. The back-up ring can be made of nylon or metal. Pressures up to 30,000 PSI have been sealed in this manner. The density of the elastomeric material varies with the pressure being sealed.

### Metal O-Rings

Metal o-rings are used in the plated or un-plated variety. Un-plated types are used in many liquid sealing applications. Platings and/or coatings are preferred for most gas applications. Metal o-rings can be used in temperature ranges from cryogenic to 2,000°F. As a rough rule of thumb, metal o-rings should always be used if the product of pressure and temperature is greater than 250,000.

There are three major types of metal o-rings;

1. Plain Type: Used for low to moderate pressures. Made of metal tubing and available in most alloys. These are the most economical and used to a pressure of about 100 PSI.
2. Self Energized: This type is utilized for high pressure applications. The surface exposed to the highest pressure, usually the inner periphery, is vented with small holes or slots. The pressure inside the vessel enters the o-ring and reduces the differential pressure across the seal.
3. Pressure Filled: Pressure filled o-rings are designed for high temperatures, from 800–2,000°F. The rings are filled with an inert gas to 600 PSIG. At high temperatures, the gas pressure increases and offsets the loss of strength of the tubing.

### Helicoflex

Helicoflex o-rings are metal gaskets with a helical spring on the inside of the o-ring. This is a variation of the other types of o-rings based upon the plastic deformation of a jacket of greater ductility than the flange material. The close wound helical spring is selected to have a specific compression resistance. During compression, the pressure forces the jacket to yield and fill the flange imperfections. Each coil of the helical spring acts independently and allows the seal to conform to flange imperfections. These o-rings can be used from cryogenic applications up to 1800°F and pressures to 50,000 PSI and higher for special applications.

Jacket materials can be made of aluminum, silver, copper, soft iron, mild steel, nickel, monel, tantalum, stainless steel, inconel or titanium.

## 3.3. Bolted Flat Covers

Flat, unstayed covers may be integral or loose. If they are welded in place the construction is considered as

**Table 8-3**
**Gasket allowable stresses**

| | Matl | | $F_y$ (PSI) | $S_g$ (PSI) | $S_s$ (PSI) |
|---|---|---|---|---|---|
| 1 | Soft Copper or Brass | | 13,000 | 19,500 | 6,500 |
| 2 | Soft Aluminum | | 8,800 | 13,200 | 4,400 |
| 3 | Iron or Soft Steel | | 18,000 | 27,000 | 9,000 |
| 4 | SST | T-304 | 30,000 | 45,000 | 15,000 |
| 5 | | T-304L | 25,000 | 37,500 | 12,500 |
| 6 | | T-316 | 30,000 | 45,000 | 15,000 |
| 7 | | T-321 | 30,000 | 45,000 | 15,000 |
| 8 | | T-347 | 30,000 | 45,000 | 15,000 |
| 9 | 4-6% Cr | | 21,800 | 32,700 | 10,900 |
| 10 | Nickel 200 | | 15,000 | 22,500 | 7,500 |
| 11 | Monel 400 | | 21,800 | 32,700 | 10,900 |
| 12 | Inconel 600 | | 35,000 | 52,500 | 17,500 |
| 13 | Incoloy 800 | | 30,000 | 45,000 | 15,000 |
| 14 | | | | | |
| 15 | | | | | |

*Notes:*

1. The values in Table are for ambient conditions only! The gasket material selection is based on the design temperature and the process conditions. The properties for seating and sealing of the gasket should be based on the ambient properties.
2. Verify properties with gasket manufacturer prior to design.
3. $S_g = 1.5\ F_y$ $S_s = .5\ F_y$

integral. These are commonly referred to as flat heads. Conversely, bolted type heads are basically blind flanges.

Flat heads may be circular or non-circular. This section does not cover non-circular heads because this would be a very rare application for a high pressure vessel. The ASME Code, Section VIII, Division 2 has formulas to follow for fixed, non-circular, flat heads.

Heads may also be what is known as an end plug such as is used in Bridgman closures and threaded closures. The end plug is peculiar to these designs and the design of these is handled in their respective sections.

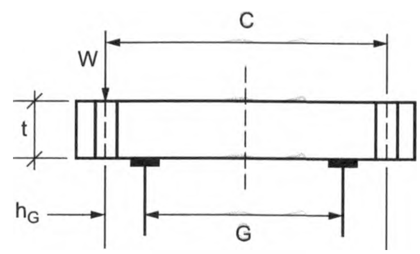
The thickness of flat, bolted heads is governed by bending, and not by tension. Thus, the formulas for the design of these components is the same for Division 1 or Division 2.

For high pressure vessels, the nozzles are typically located in the end closures, and not in the shells. Frequently they have centrally located openings with either integral or non-integral (loose) attachments for nozzles. This procedure provides for the design of both types of construction.

If the centrally located opening exceeds ½ the inside diameter of the vessel, then the design procedure reverts to a standard flange design.

## BOLTED FLAT COVERS

### CASE 1: NO OPENING



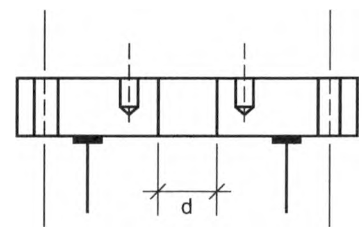

Thickness is greater of the following assuming $C_1$ attachment factor is equal to .3

$$t_G = G\sqrt{\frac{1.9Wh_G}{S_{fa}G^3}}$$

$$t_o = G\sqrt{\frac{.3P}{S_{fo}} + \frac{1.9W_{m1}h_G}{S_{fo}G^3}}$$

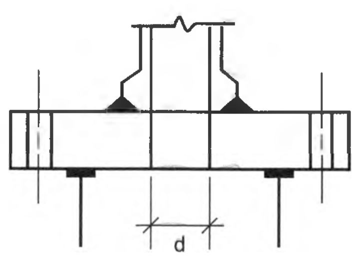
Terms $W_{m1}$, W, $S_{fo}$ and $S_{fa}$ are defined in the flange design section

### CASE 2: CENTRAL OPENING, NON-INTEGRAL CONSTRUCTION, d < .5D



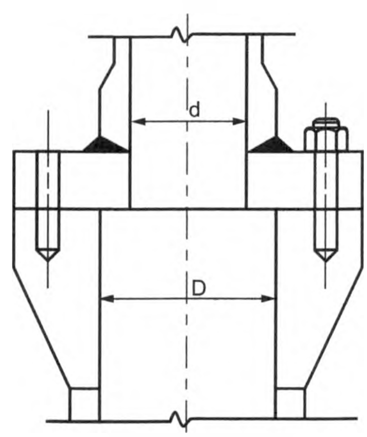

1. If d < .5D then area replacement is .5d ($t_r$). Increase flange thickness to compensate for area replacement.
2. Alternative design procedures;
   a. Design using $C_1$ = .6
   b. Increase flange thickness by 1.414

### CASE 3: CENTRAL OPENING, INTEGRAL CONSTRUCTION, d < .5D




1. If d < .5D then area replacement is .5d ($t_r$). Increase flange thickness to compensate for area replacement.
2. Alternative design procedures;
   a. Design using $C_1$ = .6
   b. Increase flange thickness by 1.414

### CASE 4: CENTRAL OPENING, INTEGRAL CONSTRUCTION, d > .5D




When d > .5D design the bolted flat cover as an ordinary weld neck flange.

### 3.4. Lens Ring Closure

The lens ring is one of the most common types of joints that originated in Germany at about 1920 with ammonia synthesis. The lens ring is used primarily on piping systems with some applications for pressure vessel heads.

The lens ring is a line contact seal for high pressure piping systems. The design is based on an unsupported area or self-sealing principal where the internal pressure acts on the inside surface of the gasket and forces it towards the edge of the cone. As the gasket expands, the sealing surface is forced further into the wedge of the seat. The force is applied radially.

There are many modifications of the basic lens ring. The most popular lens ring has spherical faces and is used between flanges with straight, tapered (20°) faces. The line of contact between the gasket and the flange faces is approximately 1/3 of the way across the flange face.

For hard materials the lens ring is not completely self-sealing like the Bridgman or delta ring, but tends to act more like a ring joint gasket rather than the other self-sealing gaskets. For softer materials like copper or aluminum, it can be considered as fully deformable, and therefore a self sealing type.

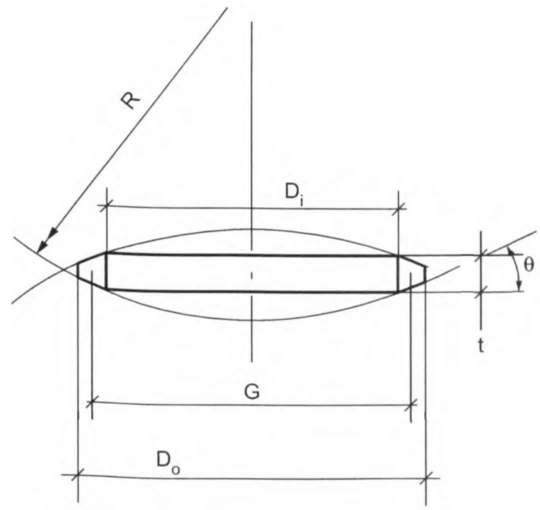
The gasket consists of a lens shaped ring of heat treated low alloy steel or some other metal. The ring should be softer than the flange face. Hardness of the conventional lens ring varies with the metal required for the service condition. This joint is ideal for pipe and tube applications. It can be used for small connections and can be used for pressure vessel closures but is better suited for piping applications.

Much of the problems associated with these joints is the non-uniform take up of the flange bolts as well as misalignment of the piping. The lens ring is one of the fastest opening closures. Lens ring gaskets have been used up to about 45,000 PSI.

Lens rings have been made with stiffening rings added to the basic lens ring, but the stiffening rings seem to be of little value. Hollowed out lens rings, lens rings with a groove cut on the inner periphery, have been used on the theory that internal pressure will "balloon out" the ring and increase its effectiveness. Hollowed out lens rings work satisfactorily, but their tolerances and hardness are very critical.

Types of lens ring gaskets:

a. Standard
b. Bellows
c. Stiffened Lens Ring
d. Modified Type




LENS RING GASKET

#### Data

$A_b$ = Area of bolts, in$^2$, $R_a \times n$
$F_y$ = Minimum specified yield strength, PSI
$G$ = Mean gasket diameter, in
$n$ = Number of bolts
$N$ = Gasket seating width, in
$R_a$ = Root area of one bolt, in$^2$
$S_b$ = Allowable stress, bolt, PSI
$\theta$ = Angle of friction, (use 20° for mild steel)

#### Dimensions

- Gasket OD, $D_O$

$$D_O = \left[(4\,A_b/\pi) + D_i^2\right]^{1/2}$$

- Mean gasket diameter, G

$$G = D_i + [(D_O - D_i)/3]$$

- Gasket radius, R

$$R = .5\,G/\sin\theta$$

- Minimum seating width, N

$$N = (1.5\,A_b\,S_b)/(3\,\pi\,G\,F_y)$$

- Width of gasket, w

$$w = .5(D_O - D_i)$$

- Basic gasket seating width, $b_O$

$b_O$ = Lesser of .25 N or .125 w

- Effective gasket seating width, b

$b = b_O$ when $b_O \le 0.25$ in

$b = .5\,b_O^{1/2}$ when $b_O > 0.25$ in

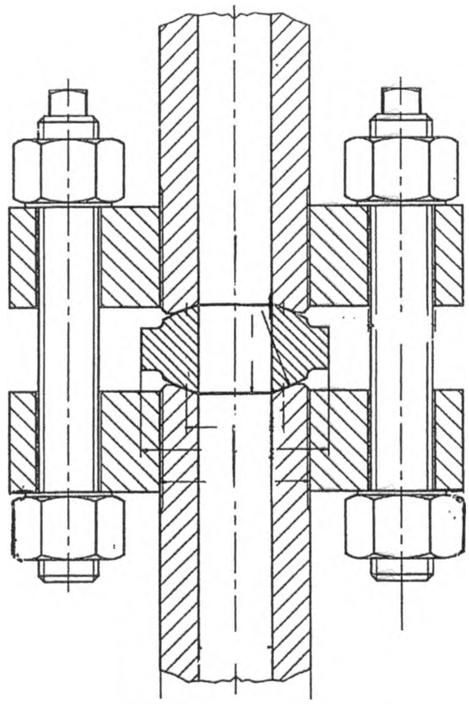


**Figure 8-6.** Lens ring closure.

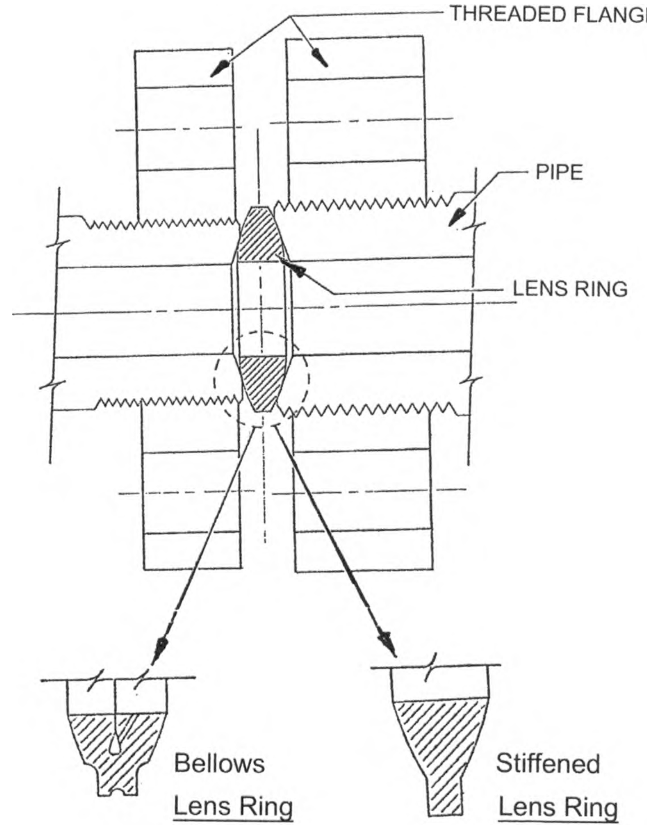



**Figure 8-8.** Lens ring closure.

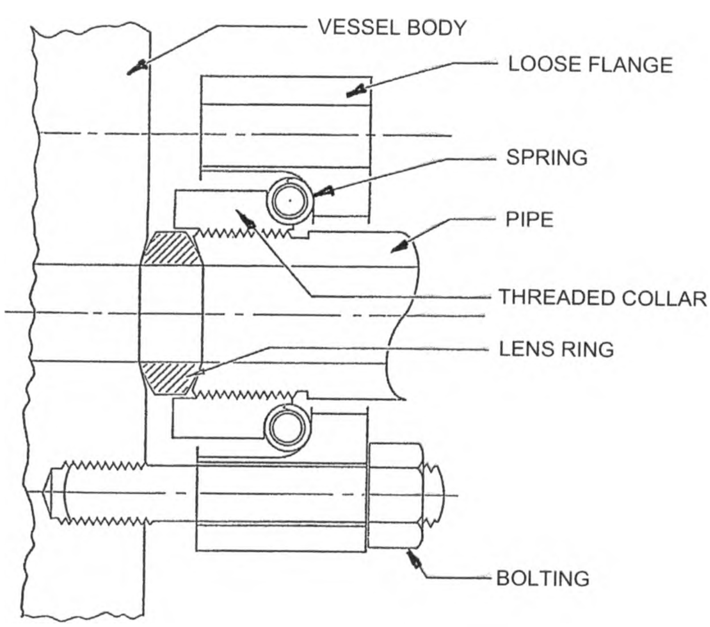
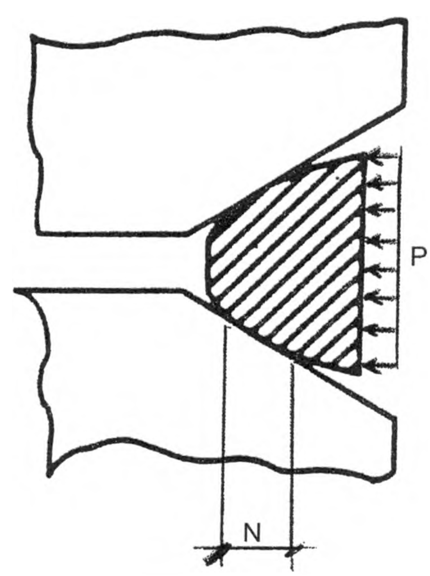



**Figure 8-7.** Modified lens ring closure.

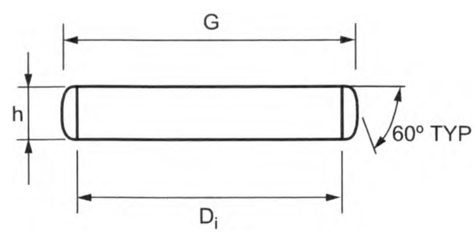



P = Radial Pressure

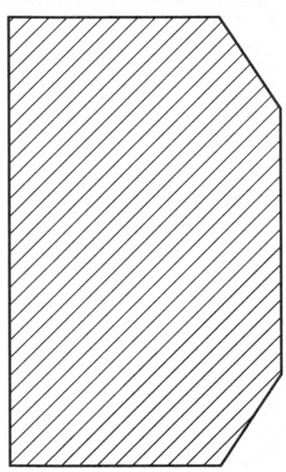
**Figure 8-9.** Lens ring closure.

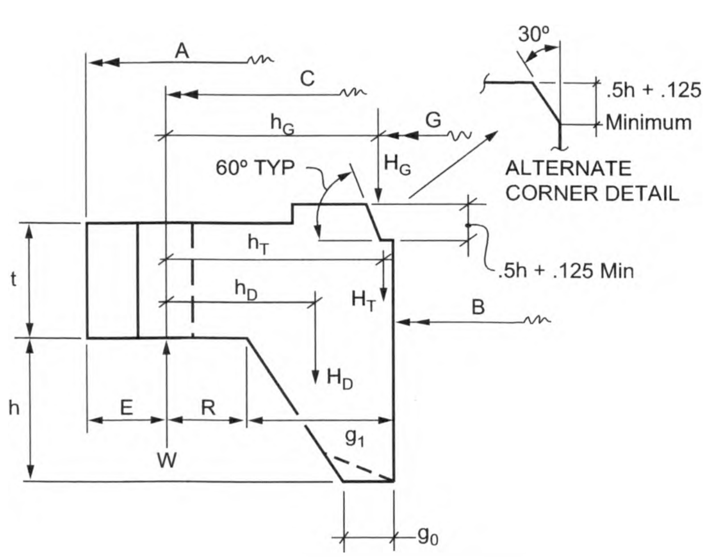
### 3.5. Double Cone Closures

The double cone ring type seal utilizes the principle of pressure sealing. This gasket type is good for high pressure, high temperature service. The double ring closure has the advantage of providing good seals for larger size vessel openings. As the pressure increases, the pressure sealing characteristics compensate for the effects of bolt elongation. They also allow for relatively easy assembly and disassembly.




DOUBLE CONE GASKET






GASKET DIMENSIONS AND LOADS

Like most high pressure joints, the double cone requires precision machined surfaces. In addition, proper bolt loads and tightening procedures are critical. Double cone gaskets can be used with either a bolted flat head or threaded closure type configuration.

#### Data

$A_b$ = Area of bolts, $in^2$, $R_a \times n$

$F_y$ = Minimum specified yield strength, PSI

G = Mean gasket diameter, in

n = Number of bolts

N = Gasket seating width, in

P = Design pressure, PSIG

$R_a$ = Root area of one bolt, $in^2$

$S_b$ = Allowable stress, bolt, PSI

y = Design seating stress of gasket, PSI

#### Dimensions

- Gasket OD, $D_O$

$$D_O = \left[\left(4\,A_b/\pi\right) + D_i^{\,2}\right]^{1/2}$$

- Minimum seating width, N

$$N = \left(1.5\,A_b\,S_b\right)/\left(3\cos 60\,\pi\,G\,F_y\right)$$

- Mean gasket diameter, G

$$G = D_O - .5\,N$$

- Gasket radius, R

$$R = .5\,G/\sin\theta$$

- Height of gasket, h

$$h = (y\,N\sin 60)/P$$

- Width of gasket, w

$$w = .5\,(D_O - D_i)$$

- Basic gasket seating width, $b_O$

$$b_O = \text{Lesser of } .25\,N \text{ or } .125\,w$$

- Effective gasket seating width, b

$$b = b_O \text{ when } b_O \le 0.25 \text{ in}$$

$$b = .5\,b_O^{\,1/2} \text{ when } b_O > 0.25 \text{ in}$$

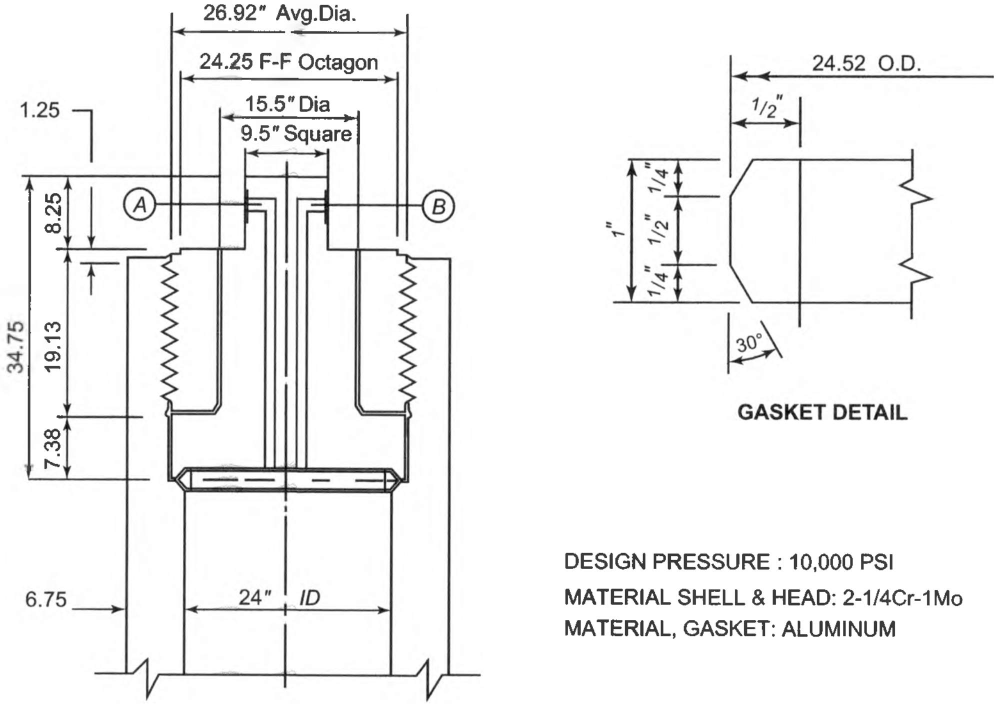



**Figure 8-10.** Threaded closure with double cone gasket

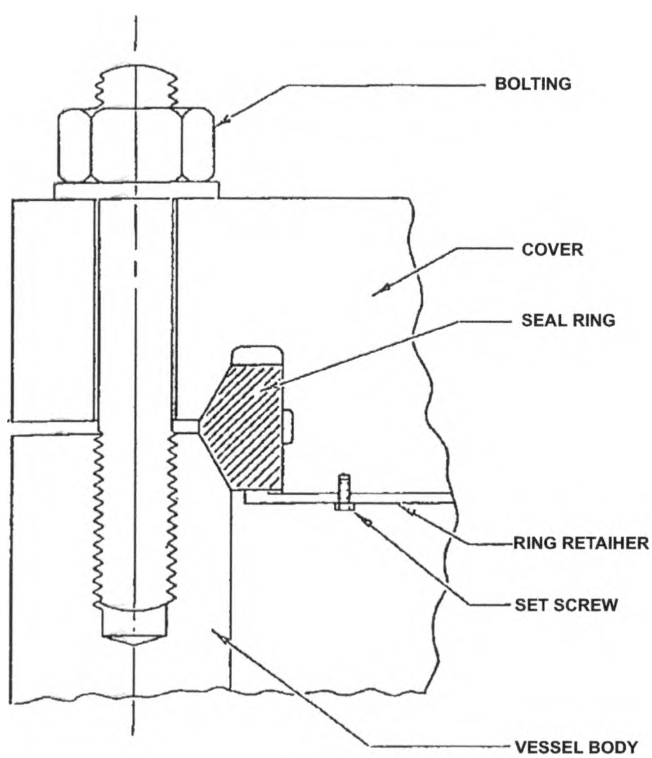



**Figure 8-11.** Double cone closure

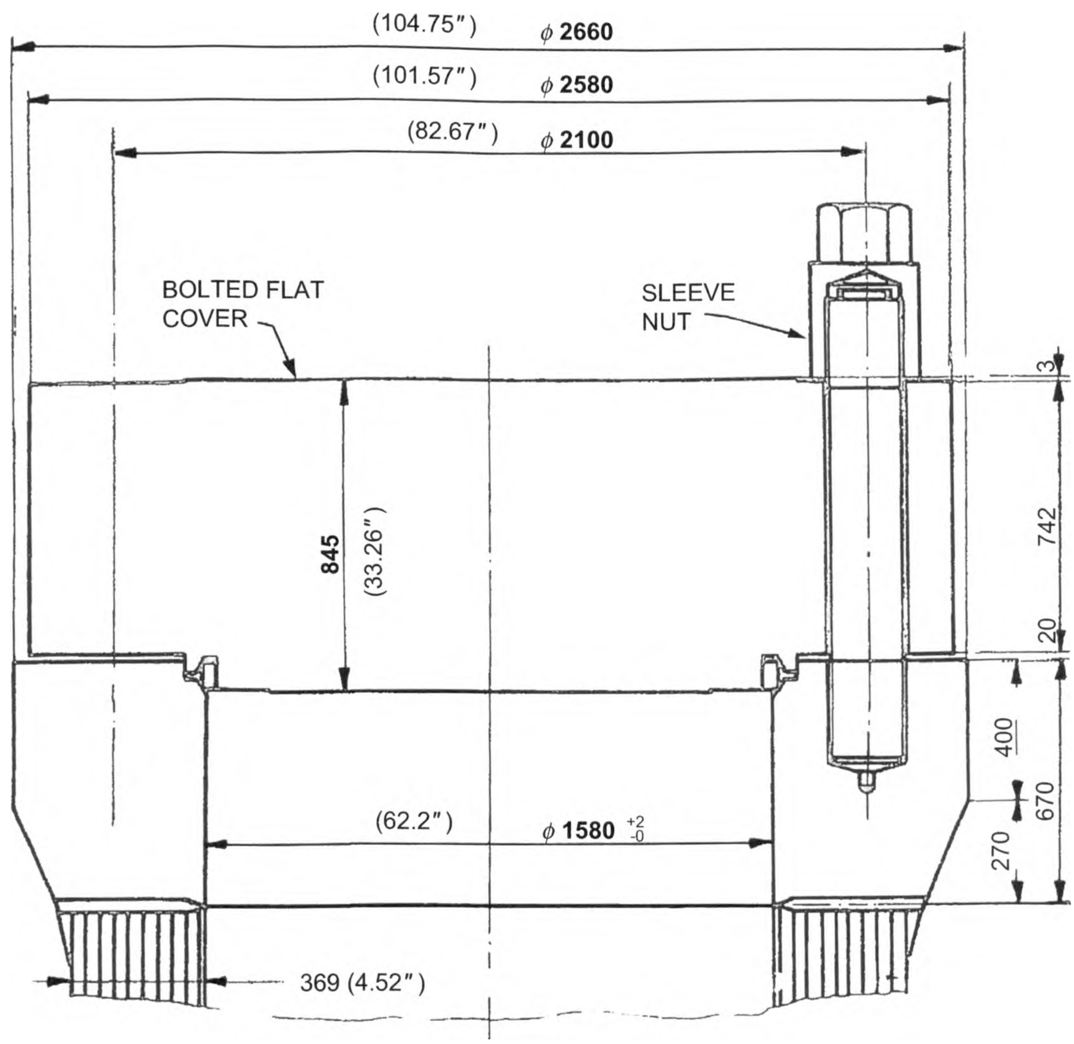



**Figure 8-12.** Example of double cone closure 10,000 PSI (680 ATM G) @ (-) 27 °F, (-) 33 °C.

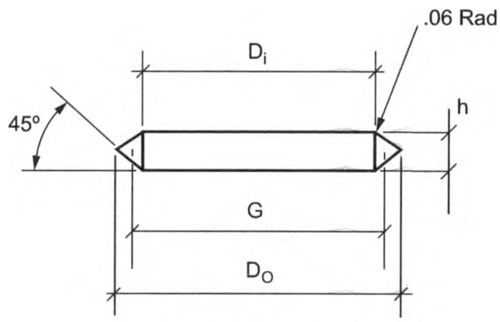
### 3.6. Delta Ring Closure

Delta rings were first developed in 1938. These gaskets are self-sealing based on the unsupported area principle. The gasket is triangular in cross section and is slightly wider than the combined depth of the gasket grooves. Initial sealing is obtained at the edges of the gasket during bolt-up and further sealing occurs during operation when the internal pressure forces the gasket further into the groove.

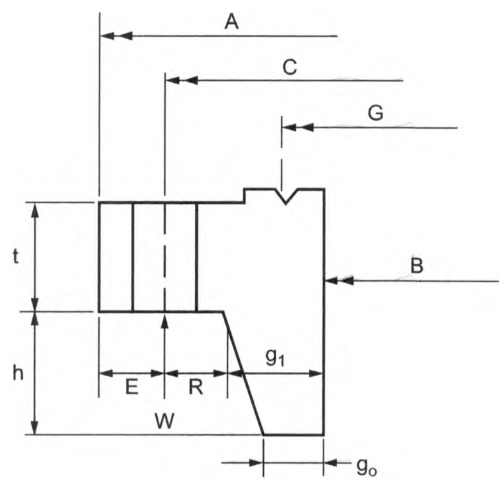
The delta ring gasket may not be suitable for very high cycle loadings such as on rotating machinery. This is due to the relative knife edge of the gasket and the high stresses at this interface point. In addition, the ring and groove dimensional control are too precise and costly to be used as a piping gasket.

These gaskets are used in the range from 2,500 to 30,000 PSI and diameters from 6 inches to 36 inches. This gasket is used more for vessels than on piping. The seating surfaces require very fine finishes, on the order of 15 AARH. Because it is self-sealing, severe tightening of the bolts is unnecessary. This in turn allows for the bolt loads to be used for the hydrostatic end load. It can be used for high temperature and corrosive service as well.

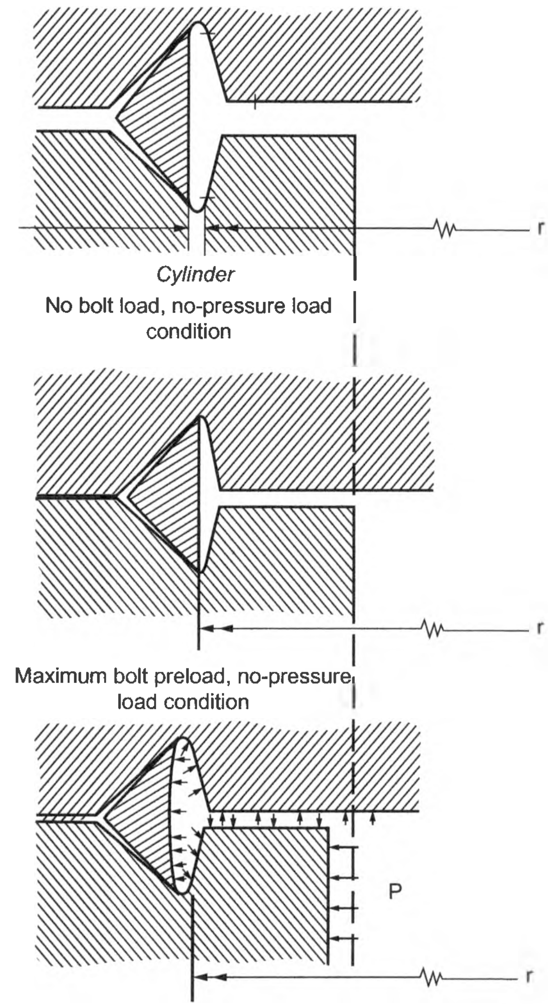
The delta joint has been used on converters, autoclaves, separators, etc.




**GASKET DIMENSIONS**




**DIMENSIONS**

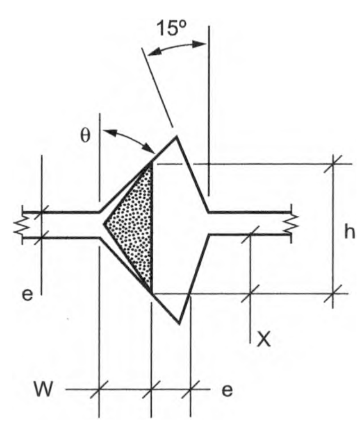



No bolt load, no-pressure load condition

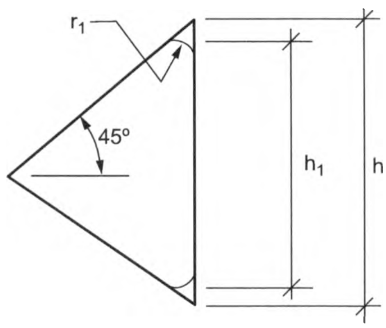
Maximum bolt preload, no-pressure load condition

Maximum bolt load, maximum-pressure load condition

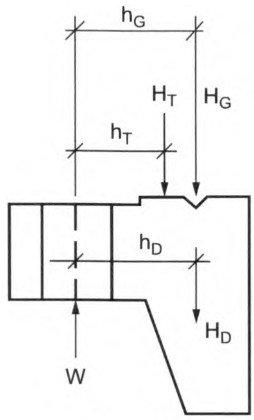
**APPLICATION OF PRESSURE ON GASKET**



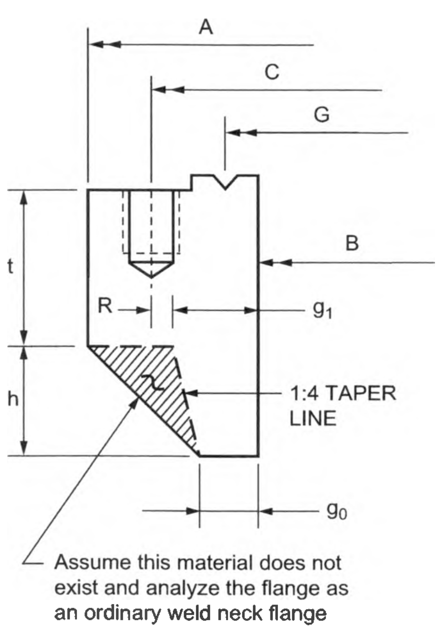

**GASKET CAVITY DIMENSIONS**




GASKET DIMENSIONS




FORCES




$h_G = .5\,(C - G)$
$h_T = .5\,(R + g_1 + h_G)$
$h_D = R + .5g_1$

## Studded Flange Details

### Data

$A_b$ = Area of bolts, in$^2$, $R_a \times n$
$F_y$ = Minimum specified yield strength, PSI
G = Mean gasket diameter, in
P = Design pressure, PSIG
E = Modulus of elasticity of gasket material, PSI
N = Gasket seating width, in
$R_a$ = Root area of one bolt, in$^2$
$S_b$ = Allowable stress, bolt, PSI
y = Design seating stress of gasket, PSI

**Table 8-4**
**Coefficient of friction, $\mu$**

| | |
|---|---|
| Steel - Steel | .8 |
| Steel - Aluminum | .61 |
| Steel - Copper | .53 |
| Steel - Brass | .51 |
| Inconel - Monel | .25 - .4 |

**Table 8-5**
**Properties of gaskets**

| MATERIAL | M | y |
|---|---|---|
| Aluminum | 3.25 | 5,500 |
| Copper | 3.50 | 6,500 |
| Brass | 3.50 | 6,500 |
| Iron | 3.75 | 7,600 |
| Monel | 3.75 | 9,000 |
| SST | 4.25 | 10,000 |

### Dimensions

- Mean gasket diameter, G

  $G = D_i + .125$
- Width of gasket, w

  $w = .5\ h$
- Basic gasket seating width, $b_O$

  $b_O$ = Lesser of .25 N or .125 w
- Effective gasket seating width, b

  $b = b_O$ when $b_O \leq .25''$

  $b = .5\ b_O{}^{1/2}$ when $b_O > .25''$

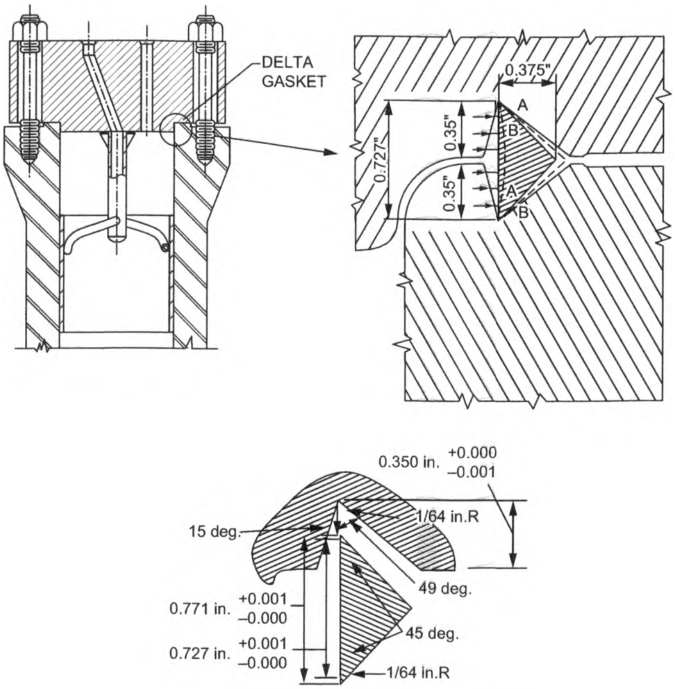



**Figure 8-13.** Delta gasket assembly.

### 3.7. Bridgman Closure

This is one of the first self-sealing joints based on the unsupported area principle. This joint was developed by Dr. P.W. Bridgman who was the author of the book, "The Physics of High Pressure", published in 1931. This type of sealing configuration has been used many times since 1929 and is highly reliable. Pressures up to 100,000 PSI have been achieved for small closures and pressures in the range of 25,000 PSI are readily available.

The Bridgman design is an axial type of self-sealing closure. In contrast, the delta ring and the lens ring closures are radial type. In the axial type, the force of the internal pressure acts in the axial direction when applied to the floating head. In the radial types the pressure acts radially to the gasket and the force is transmitted directly into radial contact with the gasket seat by a slight stretch of the ring.

The Bridgman closure can be used in either the bolted or threaded versions. The threaded version utilizes a main nut to provide the initial gasket seating contact before pressure is applied. In the threaded version the threads of the main nut take up the hydrostatic end force. In the bolted type, the main studs take the hydrostatic end force.

In the bolted head version, the pressure load is sustained by a large ring bolted to the top of the vessel. To obtain fluid tightness, the floating head (end plug) is pulled up against the triangular shaped gasket by bolts to get initial sealing. After this, pressure takes over and forces the seal into contact with the seat. It is the internal pressure that seats the gasket, not the force applied by either the main studs or the main nut.

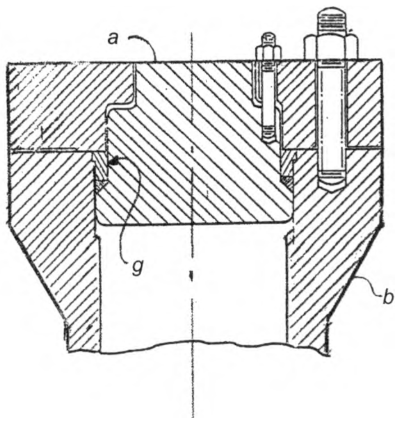
The chief drawbacks of this closure are:

a. High initial cost due to the exact machining tolerances of the gasket and seat.
b. Tendency of the gasket seat to become deformed and require additional machining.
c. Difficulty in adapting it to existing equipment.
d. When the gasket extrudes it causes the closure to stick and complicates removal of the head. Thus jack bolts are frequently utilized to allow for mechanical prying action to dislodge the extruded joint. Brass rings have been used to reduce the amount of extrusion of the gasket.

There are various options when specifying a Bridgman closure. Options include the use of jack bolts and shrink ring. Jack bolts are used for the removal of the end plug from the assembly. Expansion of the gasket during operation can extrude the gasket and make removal difficult. The jack bolts remedy this situation by providing a built-in method for removing the end plug. Jack bolts are not required for all cases. Jack bolts are never applicable for a threaded closure.

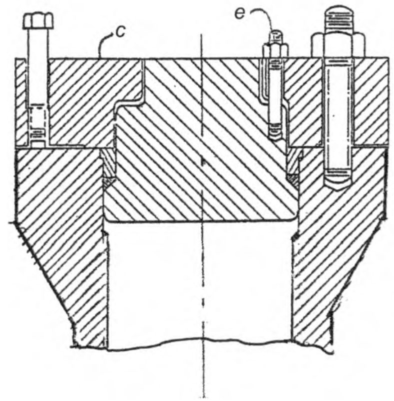
A shrink ring is used when the sum of all radial forces beyond the bolt circle cause excessive circumferential stresses in the outer ring section of the flange. A shrink ring provides residual compressive stress to reduce the tensile stress in this outer ring. The decision of whether to add a shrink ring, or increase the diameter of the flange forging is strictly one of economics.

## BRIDGMAN CLOSURE



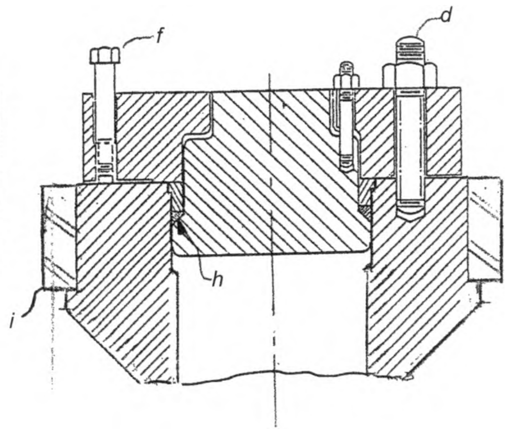

### TYPE 1

- For lower pressure applications
- No jack bolts required




### TYPE 2

- For medium pressure applications
- Jack bolts required




### TYPE 3

- For higher pressure applications
- Jack bolts required
- Shrink ring required

### Nomenclature

a. End Plug (Floating Head)
b. Flange
c. Retaining Ring
d. Main Studs
e. Pullup Studs
f. Jack Bolts
g. Follower Ring
h. Gasket
i. Shrink Ring

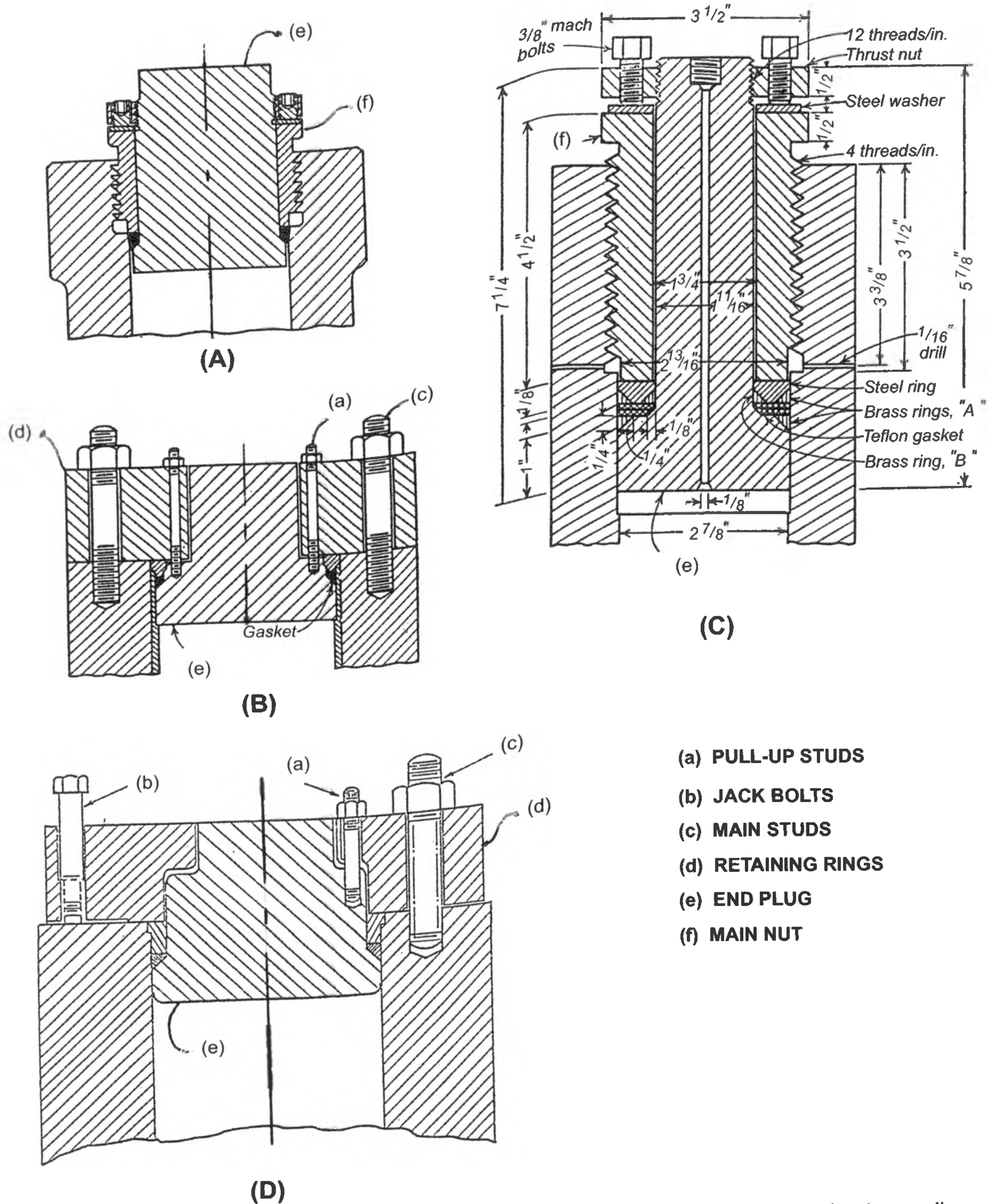



**Figure 8-14.** Bridgman type closures (A) Threaded plug type (B) Floating head type for large diameter heads (C) Modified type using rings to prevent extrusion of the gasket (D) Studded outlet with floating head.

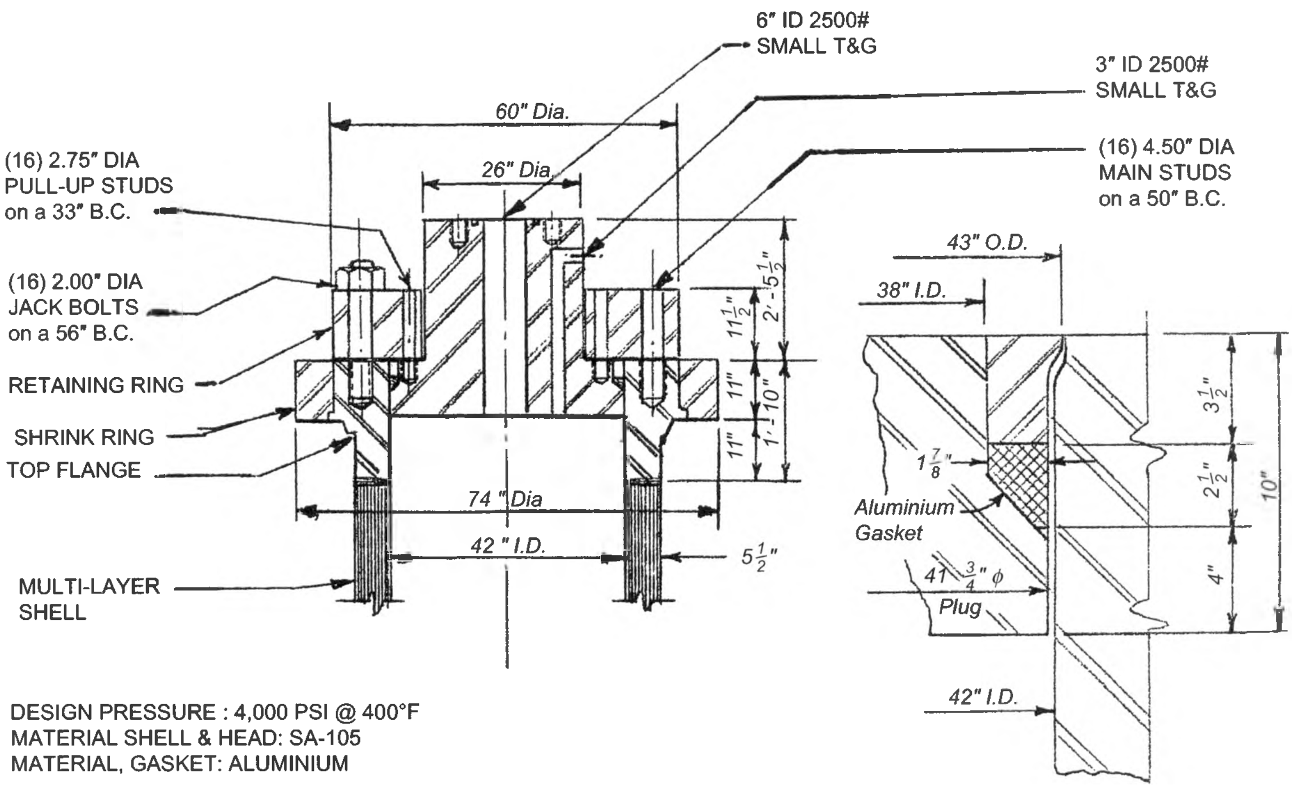



**Figure 8-15.** Bridgman joint, plug type, with shrink ring.

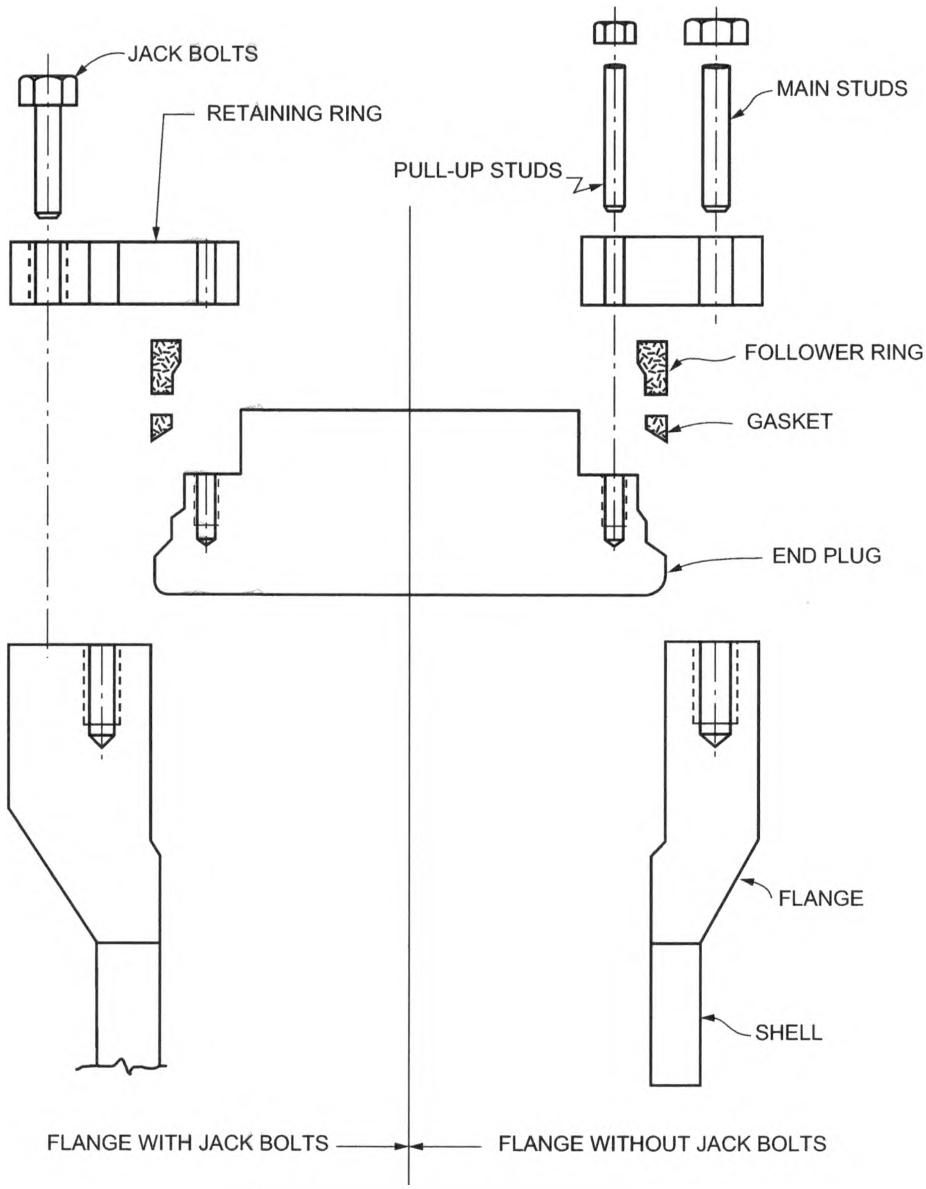



**Figure 8-16.** Exploded view of typical Bridgman closure.

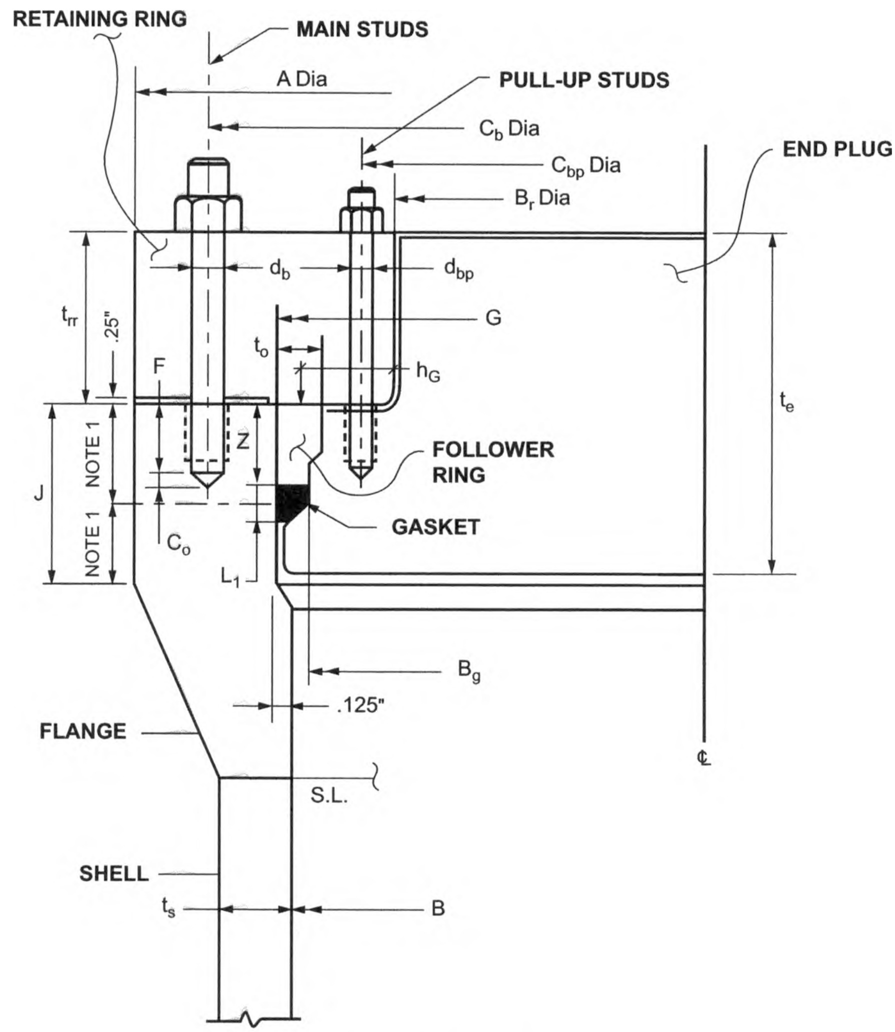
## Bridgman Closure

### Design Procedure

#### Notation

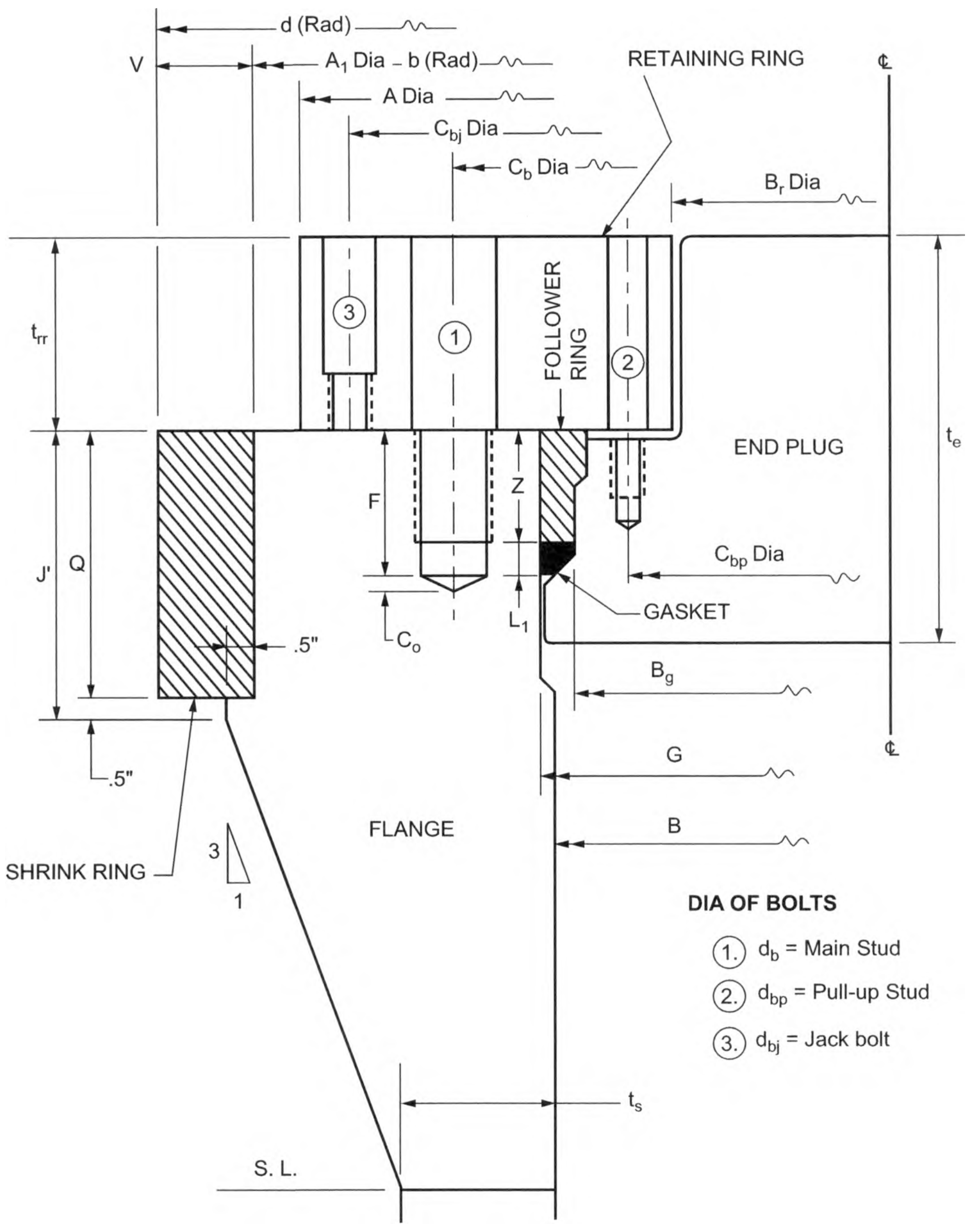
$A_b$ = Area required, main studs, $in^2$
$A_{bj}$ = Area required, jack bolts, $in^2$
$A_{bp}$ = Area required, pull up studs, $in^2$
$A_{Fr}$ = Area required, follower ring, $in^2$
$A_g$ = Actual gasket area, $in^2$
$A_r$ = Area required, $in^2$
B = Vessel ID, in
$B_S$ = Bolt spacing, in
C = Cover factor, .3
C.a. = Corrosion allowance, in
$C_W$ = Wrench clearance, nut, in
DT = Design temperature, °F
$D_m$ = Mean diameter of follower ring, in
$d_S$ = Diameter, main stud, in
$d_{bj}$ = Diameter, jack bolt, in
$d_{bp}$ = Diameter, pull up stud, in
E = Joint efficiency
$E_M$ = Modulus of elasticity, PSI
$f_{a1}$, $f_{a2}$, $f_{a3}$, $f_{aT}$, $f_{an}$ = Loads in flange, Lbs/in
FS = Factor of safety
$F_y$ = Min specified yield strength, PSI
G = Mean gasket diameter, in
$G_O$, $G_I$ = Gasket OD/ID, in
H = Hydrostatic end force, Lbs
J = Thickness flange, in
K = Ratio
$M_r$ = Moment in retaining ring, in-Lbs
N = Quantity of bolts or studs
P = Internal pressure, PSIG
$P_e$ = Equivalent internal pressure, PSIG
R = Vessel inside radius, in
$R_a$ = Root area of bolt or stud, $in^2$
$R_1$ = Radial load due to gasket reaction, Lbs/in
$R_2$ = Radial load due to pressure, Lbs/in
$R_T$ = Total combined radial load, Lbs/in
$S_{if}$ = Hoop stress in flange at flange ID, PSI
$S_{ri}$ = Hoop stress in flange at inner surface of main stud hole, PSI
$S_{ro}$ = Hoop stress in flange at outer surface of main stud hole, PSI
$S_{of}$ = Hoop stress at flange OD, PSI
$S'_{if}$ = Corrected hoop stress at flange ID, PSI
$t_e$ = Thickness, end plug, in
$t_r$ = Thickness required, in
$t_{rr}$ = Thickness, retaining ring, in
$t_s$ = Thickness, shell, in
W = Main stud bolt load, Lbs
$W_C$ = Width across corners of nuts, in
$W_F$ = Width across flats of nuts, in
$W_G$ = Pull up stud load, Lbs
$W_J$ = Load on jack bolts, lbs
$W_{ga}$ = Actual gasket load, Lbs
$Y_1$ = Correction ratio
Y = value of ASME VIII-1, Fig 2-7.1

Allowable Stresses: See Table 8-6




BRIDGMAN CLOSURE W/O SHRINK RING

Note 1: Centerline of gasket should correspond with centerline of flange thickness, "J"



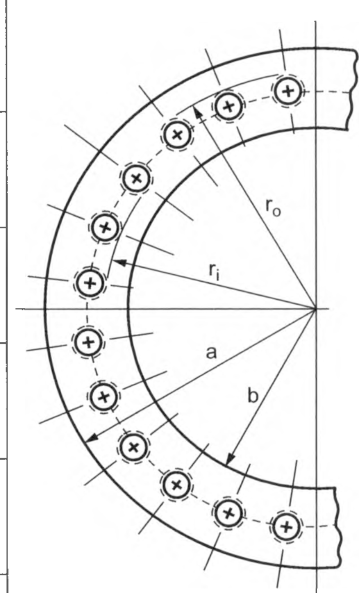

**BRIDGMAN CLOSURE W/ SHRINK RING**

**TABLE 8-6**
**Summary of materials, stress & allowable stresses**

| | Symbol | Component | Temp | Basis/Description | Material | $F_y$ (PSI) | Allow Stress (PSI) |
|---|---|---|---|---|---|---|---|
| 1 | $S_a$ | Main Studs | Ambient | ASME Code Allowable Stress, Tension | | | |
| 2 | $S_b$ | Main Studs | Design Temp | ASME Code Allowable Stress, Tension | | | |
| 3 | $S_c$ | End Plug | Design Temp | Combined Stress | | | |
| 4 | $S_e$ | End Plug | Design Temp | ASME Code Allowable Stress, Tension | | | |
| 5 | $S_f$ | Flange | Design Temp | ASME Code Allowable Stress, Tension | | | |
| 6 | $S_g$ | Gasket | Design Temp | See Table 8-3 | | | |
| 7 | $S_r$ | Retaining Ring | Design Temp | ASME Code Allowable Stress, Tension | | | |
| 8 | $S_s$ | Gasket | Design Temp | Shear Stress, See Table 8-3 | | | |
| 9 | $S_{bj}$ | Jack Bolts | Design Temp | Bearing Stress, Use .9 $F_y$ of Flange Material | | | |
| 10 | $S_{bp}$ | Pull-Up Studs | Design Temp | ASME Code Allowable Stress, Tension | | | |
| 11 | $S_{rb}$ | Retaining Ring | Design Temp | Bearing Stress, Use 1.6 X Sr | | | |
| 12 | $S_{if}$ | Flange | Design Temp | Hoop Stress in Flange @ inner surface | | | |
| 13 | $S_{ri}$ | Flange | Design Temp | Hoop Stress in Flange @ inner surface of main stud hole | | | |
| 14 | $S_{ro}$ | Flange | Design Temp | Hoop Stress in Flange @ outer surface of main stud hole | | | |
| 15 | $S_{of}$ | Flange | Design Temp | Hoop Stress in Flange @ outer surface of Flange | | | |
| 16 | | Follower Ring | Design Temp | | | | |
| 17 | $S_{EB}$ | End Plug | Design Temp | Bending Stress | | | |
| 18 | $S_{ES}$ | End Plug | Design Temp | Shear Stress | | | |

*Design Temp* = ____________________

| WORKSHEET FOR BRIDGMAN TYPE STUDDING FLANGES | | | | | |
|---|---|---|---|---|---|
| **GIVEN** | | **LOADS & STRESSES** | | | |
| FLANGE OD, A OR $A_1$ | | $R_1 = [(H + W_{ga})L_1]/A_{ga}$ | | $f_{an} = f_{at} - [(F + .5\,C_O)\,f_{a2}/J]$ | |
| FLANGE ID, B | | | | | |
| BOLT CIRCLE, $C_b$ | | $R_2 = P[J - (Z + L_1)]$ | | $Y_1 = f_{at} / f_{an}$ | |
| HYDROSTATIC END FORCE, H | | | | | |
| GASKET LOAD, $W_{ga}$ | | $R_T = R_1 + R_2$ | | $S'_{if} = Y_1\,S_{if}$ | |
| AREA OF GASKET, $A_{ga}$ | | | | | |
| DIAMETER OF STUDS, $d_S$ | | $P_e = R_T / J$ | | **NOTES:** | |
| HEIGHT OF METAL GASKET, $L_1$ | | | | 1. If $S'_{if} \leq$ Allowable Stress, then the design is OK | |
| PRESSURE, P | | $S_{if} = P_e[(a^2 + b^2)/(a^2 - b^2)]$ | | 2. If $S'_{if} >$ Allowable Stress, then implement a or b below | |
| THICKNESS, t | | | | | |
| **DIMENSIONS** | | $S_{ri} = [P_e\,b^2/(a^2 - b^2)]\,[1 + a^2/r_i^2]$ | | a. Increase flange proportions | |
| $a = .5\,(A \text{ or } A_1)$ | | | | b. Add a shrink ring | |
| $b = .5\,B$ | | $S_{ro} = [P_e\,b^2/(a^2 - b^2)]\,[1 + a^2/r_o^2]$ | | | |
| $r_i = .5\,(C_b - d_S)$ | | | | | |
| $r_o = .5\,(C_b + d_S)$ | | $S_{of} = (2\,P_e\,b^2)/(a^2 - b^2)$ | | | |
| $F = L + .25"$ or $(1.5\,d_S$ min) | | | | | |
| $C_O = .288\,d_S$ | | $f_{a1} = .5\,(r_i - b)\,(S_{if} + S_{ri})$ | | | |
| $J = F + .5\,(a - r_o)/2$ | | | | | |
| $Z = .5\,J + .5\,L_1$ | | $f_{a2} = .5\,(r_o - r_i)\,(S_{ri} + S_{ro})$ | | | |
| | | $f_{a3} = .5\,(a - r_o)\,(S_{ro} + S_{of})$ | | | |
| | | $f_{aT} = f_{a1} + f_{a2} + f_{a3}$ | | | |




Plan View-Dimensions

## Bridgman Closure

### Calculation Procedure

1.0 GIVEN;

$B = ID =$ ______________

$R = IR =$ ______________

$P =$ Internal Pressure = __________

DT = Design Temperature = _________

Corrosion Allowance = ____________

FS = Factor of Safety = ____

E = Joint Efficiency = _____

$E_m$ = Modulus of Elasticity = _______

$t_S$ = Shell Thickness = __________

C = Cover Factor = ________

2.0 MAIN STUDS;

Material: __________________

$S_b =$ ________________

a. Mean gasket diameter, G

$$G = B + 0.25 \text{ in}$$

b. Hydrostatic end force, H

$$H = .25(\pi G^2 P)$$

c. Area required, studs, $A_b$

$$A_b = H / S_b$$

d. Quantity of studs required, N
Always use multiples of 4, $N = A_b / R_a$

| TRIAL | $d_b$ | $R_a$ | N | USE |
|---|---|---|---|---|
| 1 | | | | |
| 2 | | | | |
| 3 | | | | |
| 4 | | | | |

e. Determine bolt circle, $C_b$

$$C_b\text{min} = B + d_b + 2 \text{ in}$$

f. Check stud spacing, $B_S$

Max : $(\pi C_b)/N < 2\,d_b + J$

Min : > $B_S$ from Table 8-7

g. Main stud bolt load, W

$$W = N R_a S_b$$

3.0 GASKET

1. Determine gasket material and properties;
   a. Material ________
   b. $S_g$ from Table 8-3
   c. $F_y$ from Table 8-3
2. Determine gasket proportions;
   a. Area of gasket required;

$$A_g = (2\,H)/(2\,S_g - F_y)$$

   b. Pull-up stud load required to deform gasket;

$$W_g = .5\,A_g\,F_y$$

   c. Set gasket OD equal to G;

$$G_O = G$$

   d. Determine maximum gasket ID;

$$G_i \max = \left[G_O^2 - (4\,A_g)/\pi\right]^{1/2}$$

   Use $G_i =$ ____________

3. Based on selected gasket dimensions determine;
   a. Actual gasket area;

$$A_{ga} = \left(.25\,\pi\,G_O^2 - .25\,\pi\,G_i^2\right)$$

   b. Actual gasket load;

$$W_{ga} = .5\,A_{ga}\,F_y$$

4. Gasket dimensions;

$$w = .5\,(G_O - G_i)$$

$$L_i = w + 0.25 \text{ in Min}$$

4.0 PULL-UP STUDS;

Material: __________

$S_{bp} =$ ___________

a. Required stud area;

$$A_{bp} = W_{ga}/S_{bp}$$

b. Determine quantity of pull-up studs;
Note: The quantity should be the same as the number of main studs if possible.

$$N = A_{bp}/R_a$$

Use: $d_{bp} =$ ________

c. Determine bolt circle;
Minimum:

$$C_{bp} = G_I - 2(d_{bp} - 0.125 \text{ in})$$

Note: For pull-up studs less than 1.5 inches diameter, do not subtract 0.125 inches in the above equation.

Maximum:

$C_{bp} = B_r + 3\ d_{bp}$

Use: __________

4.0 FOLLOWER RING

Retainer Ring Material: ________

$S_r =$ ________

- Allowable bearing stress;

$S_{rb} = 1.6\ S_r$

- Minimum area at top of follower ring;

$A_{Fr} = (H + W_{ga})/S_{rb}$

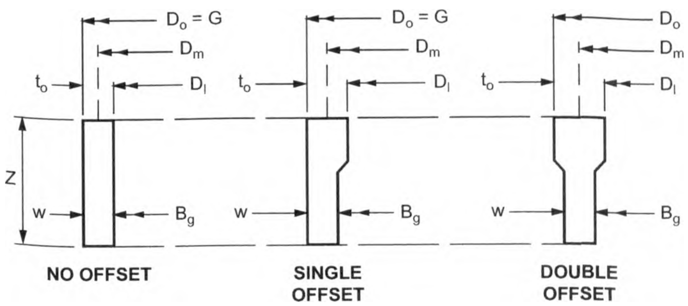
- As a first trial, assume $D_O = G$ and solve for $D_i$;

$D_i = \left[D_O{}^2 - (4\ A_{Fr})/\pi\right]^{1/2}$

Notes for design of Follower Rings;

1. The offset options are determined such that the bearing stress on the retainer ring is not exceeded.
2. The bottom of the follower ring shall have the same ID and OD as the gasket.
3. The maximum offset shall not exceed 25% of width "w".
4. It is preferable to set the OD of the follower ring as equal to the OD of the gasket.
5. The double offset option should only be utilized when the follower ring width, $t_o$, exceeds the 25% limit of the gasket width "w".
6. Width of ring, $t_o$

   $t_o = .5\ (D_o - D_i)$
7. If $t_o < w$ then design is OK as is.
8. If $t_o > w$ then the designer must decide whether a single or double offset is required.
9. Begin with a single offset and check that offset does not exceed .25 w.
10. If the single offset exceeds .25 w, then a double offset is required.
11. Check the width of ring, $t_o$, to ensure that neither the inner nor outer offset exceeds the 25% w criteria. This effectively sets a max limit of $t_o$ as lesser than or equal to 1.5 w.
12. If $t_o$ exceeds 1.5 w then either the ring width, w, must be made wider or the material must be changed to increase the allowable bearing strength.




5.0 JACK BOLTS

Notes:

1. The jack bolts are used to remove the end plug in the event that the gasket becomes fused to the inside wall of the flange.
2. For design purposes assume that 65% of the gasket is welded to the flange.
3. The jack bolts must be capable of exerting enough force on the end of the flange to break the weld in shear.
4. A copper gasket is the most easily fused.
5. Use the same quantity of jack bolts as main studs as a starting point.
6. Jack bolts should be a minimum of 1-1/2" diameter.
7. Check wrench clearances between all top nuts and bolts.
8. Stagger jack bolts in between the main studs to obtain adequate wrench clearance.
9. It is common practice to leave the jack bolts in the retainer ring during operation.
10. Maintain at least one bolt diameter between the jack bolt circle and the OD of the retaining ring.

- Design load on jack bolts;

  $W_J = .65\, S_S\, G_O\, \pi\, L_i$
- Allowable bearing stress, flange material;

  $S_{bj} = .9\, F_y$
- Area required, bolts;

  $A_{bj} = W_J / S_{bj}$
- Actual bolting used;

  N = __________

  $d_{bj}$ = __________
- Determine bolt circle required;

The bolt circle required for the jack bolts, $C_{bj}$, is dependent on wrench clearances and spacing between nuts. Recommended wrench clearance and dimensions of nuts are contained in the following Table.

**Table 8-7**
**Nut and wrench clearance**

| Item | Bolt Size (in) | | | | |
|---|---|---|---|---|---|
| | 1.5 | 1.75 | 2 | 2.25 | 2.5 |
| $C_W$ | 3.375 | 4.125 | 4.5 | 4.875 | 5.375 |
| $W_C$ | 2.742 | 3.175 | 3.608 | 4.041 | 4.474 |
| $W_F$ | 2.375 | 2.75 | 3.125 | 3.5 | 3.875 |
| | Bolt Size (in) | | | | |
| | 2.75 | 3 | 3.5 | 4 | 4.5 |
| $C_W$ | 5.875 | 6.125 | 7 | 7.66 | 8.5 |
| $W_C$ | 4.907 | 5.34 | 5.928 | 6.755 | 8 |
| $W_F$ | 4.25 | 4.625 | 5.375 | 6.125 | 7 |

Parameters;

$\alpha = .5\,(360 / N)$
$a = .5 \cos \alpha\, C_b$
$b = .5 \sin \alpha\, C_b$
$c = .5C_{bj} - a$
$d = (b^2 + c^2)^{1/2}$
$d > .5\,(C_W + W_C)$

For first trial assume $C_{bj} = C_b + C_W$

6.0 RETAINING RING

With jack bolts;

- OD of retaining ring, $A_r$

  $A_r = C_{bj} + 2\, d_{bj}$

Without jack bolts;

- OD of retaining ring, $A_r$

  $A_r = C_b + 2\, d_b$
- ID of retaining ring, $B_r$

  $B_r = C_{bp} - 2d_{bp}$
- Calculate ratio K;

  $K = A_r / B_r$
- Determine value Y from Fig 8-18

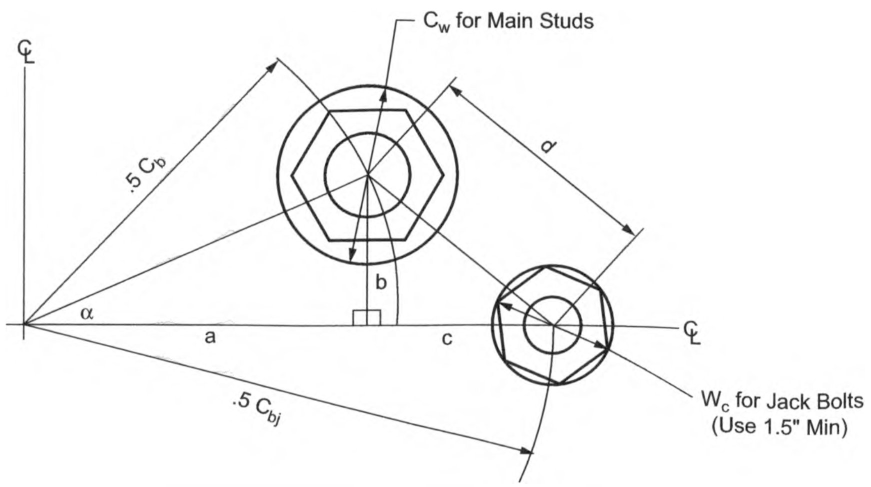



**Figure 8-17.** Nut-wrench clearances/proportions.

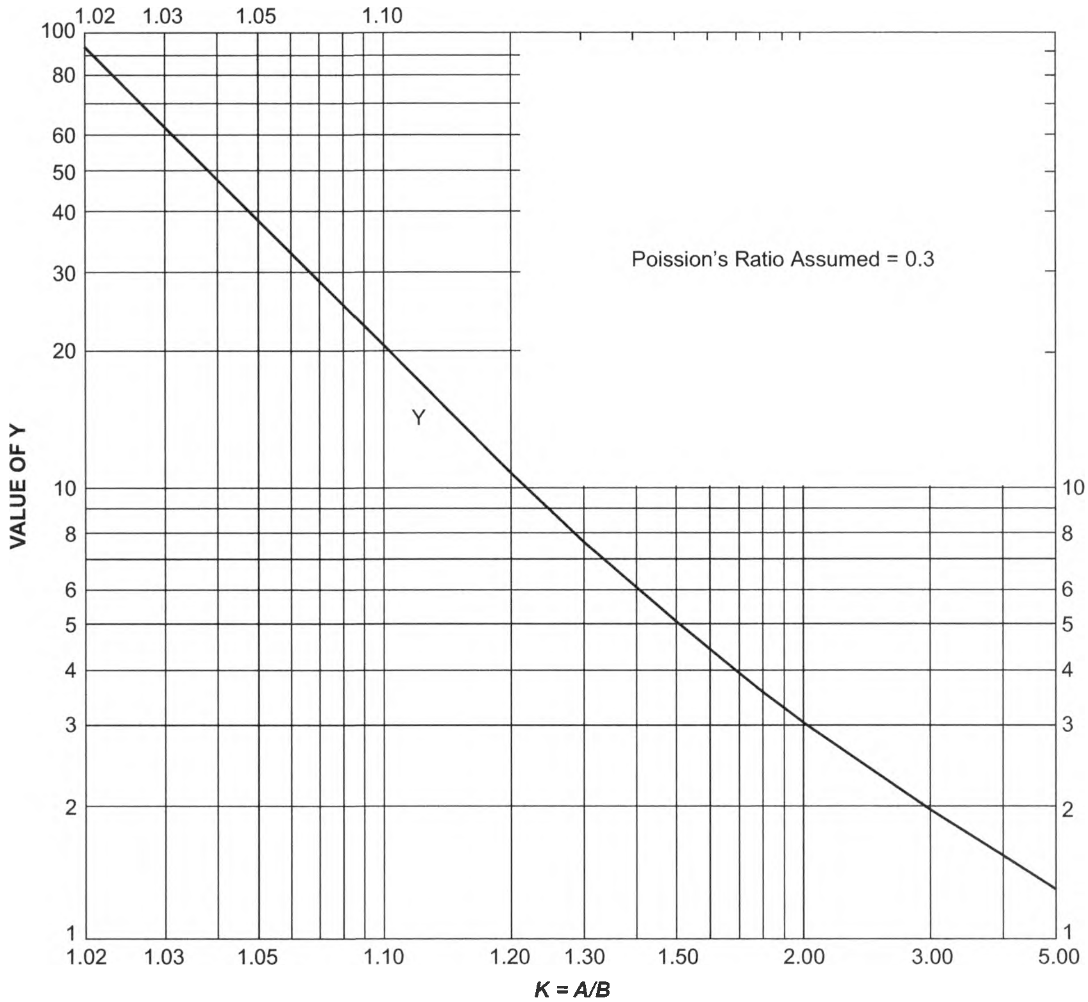



**Figure 8-18.** Value of Y.

- Mean diameter of follower ring, $D_m$

$D_m = .5(D_O - D_i)$

- Determine lever arm, $h_G$

$h_G = .5(C_b - D_m)$

- Bending moment in retaining ring, $M_r$

$M_r = (W\,h_G)/B_r$

- Thickness required, retaining ring, $t_r$

$t_r = [(M_r\,Y)/S_r)] + C.a.$

Use $t_{rr} =$ _________

## 7.0 END PLUG

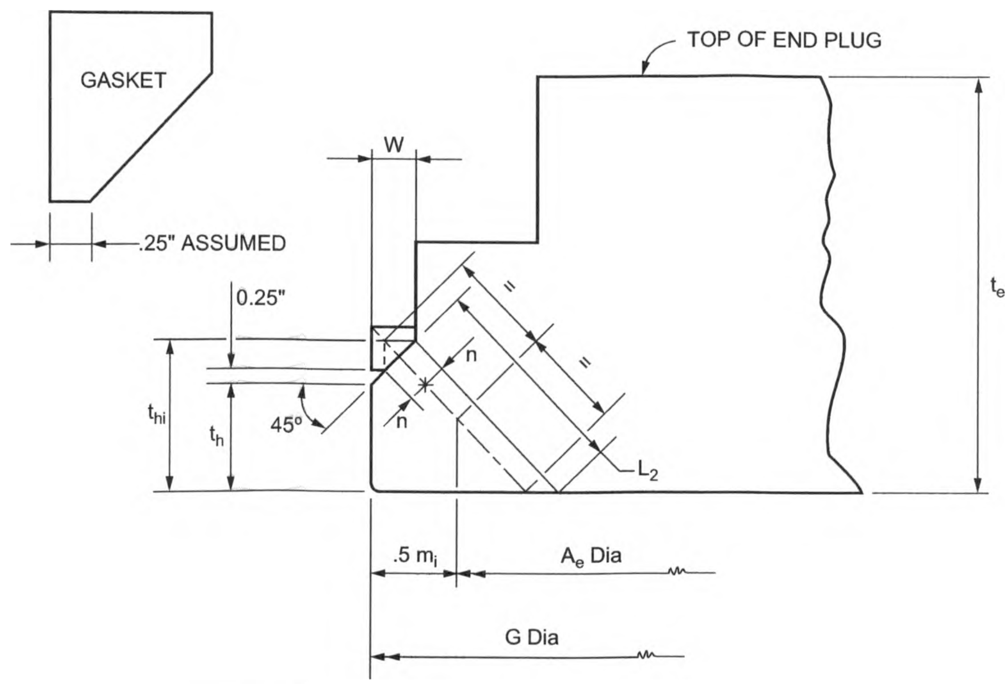



**Figure 8-19.** Dimensional Proportions of end plug/gasket.

DIMENSIONS;

$$t_{hi} = (H + W_{ga}) / [.6\ \pi\ S_e (G - 2\ w)]$$

Round up to the nearest 0.25 inches.

$$t_h = t_{hi} - w$$
$$n = .707(w - 0.25\ in)$$
$$L_2 = 1.414\ t_{hi}$$
$$m_i = 2(.354\ L_2 + 0.25\ in)$$
$$A_e = G - m_i$$

STRESS;

- Bending stress, $S_{EB}$

$$S_{EB} = [6(H + W_{ga})n] / [\pi\ A_e\ L_2^2\ \mathrm{Sin}\ 45]$$

- Shear stress, $S_{ES}$

$$S_{ES} = (H + W_{ga}) / (\pi\ A_e\ L_2\ \mathrm{Sin}\ 45)$$

- Combined stress, $S_{comb}$

$$S_{comb} = (S_{EB}^2 + S_{ES}^2)^{1/2}$$

- Allowable stress, $S_e$

$S_e =$ ________

- Required thickness, end plug, $t_r$

$$t_r = G\left[ (C\ P / S_e)^{1/2} \right] + \mathrm{C.a.}$$

Notes:

1. The top of the end plug should be flush with the top of the retaining ring.
2. If the opening in the end plug is greater than 2″ NPS but less than .5 G, either sufficient integral reinforcement shall be provided or the thickness may be multiplied by 1.414.

- Actual thickness used;

$$t_e = t_{rr} + Z + L_1 + X$$

Use ____________

## 8.0 FLANGE

Dimensions;

A = Larger of...

a. $C_b + 2\ (d_b - 0.125\ in)$
b. $B + 2\ t_f + 2\ d_b$

$a = .5\ A$
$b = .5\ B$
$r_i = .5\ (C_b - d_b)$
$r_O = .5\ (C_b + d_b)$
$J_{min} = F + .5\ (a - r_O)$
Note: The gasket should be positioned such that the centerline of the gasket corresponds to the centerline of dimension J.

$Z = .5\ J - .5\ L_1$

Adjust to the nearest 1/8 inch.

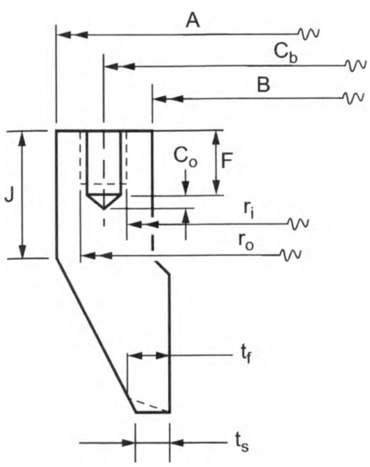



**Figure 8-20.** Flange dimensions.

Use worksheet for Bridgman type studding flanges to determine all stresses.

**Sample Problem**

1.0 GIVEN

$B = 24$ in
$R = 12$ in
$P = 3{,}000$ PSI
$DT = 860°F$
$C.a. = 0.1875$ in
$FS = 4$
$E_m = 25 \times 10^6$ PSI
$F_y = 26{,}000$ PSI

2.0 SHELL THICKNESS

$P/S = 3000/15{,}950 = .188$

Since this is less than .385, thin wall formulas for VIII-1 apply.

**ASME VIII-1**

Thin Wall formula...

$$t = (P\,R)/(SE - .6\,P)$$
$$= (3000)(12.1875)/(15950 - 1800)$$
$$= 2.58 + .1875 = 2.77 \text{ in}$$

Thick Wall formula...

$$Z_1 = (S + P)/(S - P)$$
$$= (15950 + 3000)/(15950 - 3000) = 1.46$$
$$t = R_i\left[(Z_1)^{1/2} - 1\right] = 12.1875\left(1.46^{1/2} - 1\right)$$
$$= 2.55 + .1875 = 2.743 \text{ in}$$

**ASME VIII-2**

$$t = R_i\left[e^{P/S} - 1\right) = 12.1875\left[e^{.188} - 1\right]$$
$$= 2.52 + .1875 = 2.71 \text{ in}$$

**ASME VIII-3**

Assume $t = 2.5$ new
$t = 2.5 - .1875 = 2.3125$ in
$D_O = 24 + 2\ (2.5) = 29$ in
$Y = D_O / D_i = 29 / 24.375 = 1.189$

$$P_m = .667\ F_y \text{Ln } Y = .667\ (26{,}000)\ \text{Ln } 1.189$$
$$= 3011 \text{ PSI}$$

In summary....

| | |
|---|---|
| ASME VIII–1 – Thin: | 2.77 |
| ASME VIII–1 – Thick: | 2.743 |
| ASME VIII–2: | 2.71 |
| ASME VIII–3: | 2.5 |
| Use ________ | |

3.0 MAIN STUDS

Matl: SA-193-B16

$S_b = 22{,}900$ PSI

- Mean gasket diameter, G

$$G = B + 0.25 \text{ in}$$
$$= 24 + .25 = 24.25 \text{ in}$$

- Hydrostatic end force, H

$$H = .25(\pi G^2 P) = .25(\pi 24.25^2(3000))$$
$$= 1,385,589 \text{ Lbs}$$

- Area required, studs, $A_b$

$$A_b = H/S_b = 1,385,589/22900 = 60.5 \text{ in}^2$$

- Quantity of studs required, N

Always use multiples of 4, $N = A_b / R_a$

| Trial | $d_b$ | $R_a$ | N | Use |
|---|---|---|---|---|
| 1 | 2.5 | 3.715 | 16.28 | 20 |
| 2 | 2.25 | 3.02 | 20.03 | 24 |
| 3 | 2 | 2.3 | 26.3 | 28 |
| 4 | 2.25-8 | 3.423 | 17.67 | 20 |

Use Trial 4,

- Determine bolt circle, $C_b$

$$C_b\text{min} = B + d_b + 2 \text{ in} = 24 + 2.25 + 2$$
$$= 28.25 \text{ in}$$

- Check stud spacing, $B_S$

$$\text{Max}: (\pi C_b) / N < 2 d_b + J \pi 28.25 / 20$$
$$= 4.43 \text{ in}$$

Min: $> B_S$ from Table 8-7 $= 4.75$ in

Increase $C_b$ to accommodate minimum bolt spacing;

$$C_b = B_S N/\pi = 4.75(20)/\pi = 30.23 \text{ in}$$

Use 30.25 in

- Main stud bolt load, W

$$W = N R_a S_b = 20(3.423)22,900$$
$$= 1,567,734 \text{ Lbs}$$

4.0 GASKET
Matl: T-347 SST

$F_y = 30,000$ PSI

$S_g = 45,000$ PSI

$S_S = 15,000$ PSI

- Determine gasket proportions;
  a. Area of gasket required;

$$A_g = (2H)/(2S_g - F_y)$$
$$= (2(1,385,589))/(2(45,000) - 30,000)$$
$$= 46.19 \text{ in}^2$$

- Pull-up stud load required to deform gasket;

$$W_g = .5 A_g F_y = .5(46.19)30,000$$
$$= 692,850 \text{ Lbs}$$

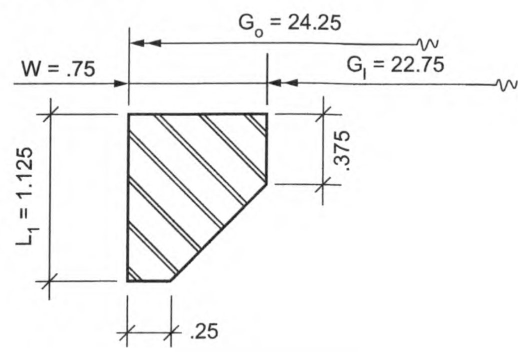
- Set gasket OD equal to G;

$G_O = G = 24.25$ in

- Determine maximum gasket ID;

$$G_i \max = \left[G_O^2 - (4 A_g)/\pi\right]^{1/2}$$
$$= \left[24.25^2 - (4)(46.19)/\pi\right]^{1/2}$$
$$= 23.00 \text{ in}$$

Use 22.75 in




BRIDGMAN GASKET

- Based on selected gasket dimensions determine;
  a. Actual gasket area;

$$A_{ga} = .25 \pi G_O^2 - .25 \pi G_i^2) = 55.37 \text{ in}^2$$

  b. Actual gasket load;

$$W_{ga} = .5 A_{ga} F_y = .5(55.37)30,000$$
$$= 830,550 \text{ Lbs}$$

$$\gg W_g \text{ OK}$$

5.0 PULL-UP STUDS
Matl: SA-193 B7

$S_{bp} = 25,000$ PSI

- Required stud area;

$$A_{bp} = W_{ga}/S_{bp} = 830,558/25,000 = 33.22 \text{ in}^2$$

- Determine quantity of pull-up studs;
  Note: The quantity should be the same as the number of main studs if possible.

$N = 20$

$$A_r = A_{bp}/N = 33.22/20 = 1.66 \text{ in}^2$$

Use : $d_{bp} = 1.625 \; R_a = 1.68 \text{ in}^2$

- Determine bolt circle;

Minimum: $C_{bp} = G_I - 2(d_{bp} - 0.125 \text{ in})$

$= 22.75 - 2(1.625) = 19.5$

Maximum: $C_{bp} = B_r + 3d_{bp}$

$= 17 + 3(1.625) = 21.875$ in

Use: $C_{bp} = 20$ in

6.0 FOLLOWER RING
Ring material: SA-182–F21

$S_r = 15{,}950$ PSI

- Allowable bearing stress, $S_{rb}$

$S_{rb} = 1.6\, S_r = 1.6\,(15{,}950) = 25{,}520$ PSI

- Minimum area required at top of follower ring, $A_{Fr}$

$A_{Fr} = (H + W_{ga})/S_{rb}$

$= (1{,}385{,}587 + 830{,}558)/25{,}520$

$= 86.84 \text{ in}^2$

- Assume D = G = 24.25
Find $D_i$

$D_i = \left[D_O{}^2 - (4A_{Fr})/\pi\right]^{1/2}$

$= \left[24.25^2 - (4)(86.84)/\pi\right]^{1/2} = 21.85$ in

- Width of ring, $t_o$

$t_o = .5\,(D_o - D_i) = .5(24.25 - 21.85) = 1.12$ in

Therefore a double offset is required.

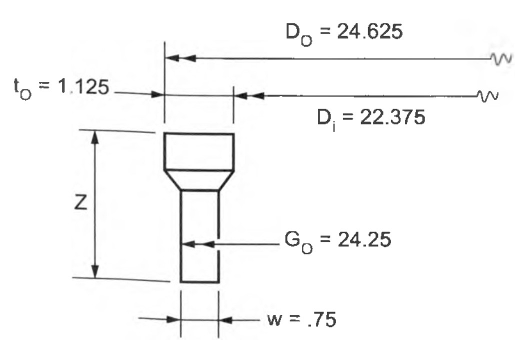
- Max offset allowed;

$.25\, w = .25\,(75) = .188$ in

- Check width of ring for double offset;

$w = t_O - 2 \times \text{offset}$

$= 1.125 - 2(.188) = 0.75$ in OK




FOLLOWER RING

- Find length of ring required, Z

$Z = 4w = 4\,(.75) = 3$ in minimum

7.0 JACKBOLTS

- Design load on jackbolts;

$W_J = .65\, S_S\, G_O\, \pi\, L_i$

$= .65\,(15{,}000)\, 24.25\,(\pi)\,(1.125)$

$= 835{,}639$ Lbs

- Allowable bearing stress, flange material;

$S_{bj} = .9F_y = .9\,(26{,}000) = 23{,}400$ PSI

- Area required, bolts;

$A_{bj} = W_J/S_{bj} = 835{,}639/23{,}400 = 35.71 \text{ in}^2$

- Actual bolting used;

$N = 20$

$d_{bj} = 1.75$ in

$R_a = 1.98 \text{ in}^2$

- Determine bolt circle required;
Given;

$C_b = 30.25$ in

$C_{WM} = 4.875$ in

$C_{WJ} = 4.125$ in

$W_{CJ} = 3.175$ in

Calculate;

$\alpha = .5(360/N) = .5\,(360/20) = 9^\circ$

$a = 0.5 \cos\alpha\, C_b = 0.5 \cos 9^\circ\,(30.25) = 14.94$ in

$b = 0.5 \sin\alpha\, C_b = 0.5 \sin 9^\circ(30.25) = 2.37$ in

$X = .5(C_{WM} + C_{CJ}) = .5\,(4.875 + 3.175)$

$= 4.025$ in

Assume values for bolt circle, $C_{bj}$, and calculate corresponding distances "c" and "d". Select bolt circle, $C_{bj}$, where d > x

$c = .5C_{bj} - a$

$d = \left(b^2 + c^2\right)^{1/2}$

| $C_{bj}$ | c | d |
|---|---|---|
| 36.00 | 3.06 | 3.87 |
| 36.25 | 3.185 | 3.97 |
| 36.50 | 3.31 | 4.07 |
| 36.75 | 3.435 | 4.17 |
| 37.00 | 3.56 | 4.27 |

Use $C_{bj} = 36.50$ in

8.0 RETAINING RING

- OD of retaining ring, A

  $A = C_{bj} + 2\,d_{bj} = 36.5 + 2(1.75) = 39.75$ in Dia

- ID of retaining ring, $B_r$

  $B_r = C_{bp} - 2\,d_{bp} = 20 - 2(1.625)$

  $= 16.75$ in Dia

- Calculate ratio K;

  $K = A/B_r = 39.75/16.75 = 2.37$

- Determine value Y from Fig 8-18

  $Y = 2.47$

- Mean diameter of follower ring, $D_m$

  $D_m = .5(D_O - D_i) = .5\ (24.625 + 22.375)$

  $= 23.5$ in Dia

- Determine lever arm, $h_G$

  $h_G = .5(C_b - D_m) = .5(30.25 - 23.5) = 3.375$ in

- Bolt load, W

  $W = N\,R_a\,S_b = 20(3.423)22{,}900$

  $= 1{,}567{,}734$ Lbs

- Bending moment in retaining ring, $M_r$

  $M_r = (W\,h_G)/B_r = (1{,}567{,}734(3.375))/16.75$

  $= 315{,}887$ in − Lbs

- Thickness required, retaining ring, $t_r$

  $t_r = [\ (M_r Y)/S_r)] + C.a.$

  $= [\ (315{,}887(2.47))/17{,}100] + .1875 = 6.94$ in

  Use $t_{rr} = 7$ in

9.0 END PLUG

- Required thickness, end plug, $t_r$

  $t_r = G\ [(C\ P/S_e)]^{1/2} + C.a.$

  $= 24.25\left[.3\left(3000\right)/17{,}100\right] + .1875 = 5.75$ in

Given;

$H = 1{,}385{,}589$ Lbs

$W_{ga} = 830{,}558$ Lbs

$G = 24.25$ in

$w = .75$ in

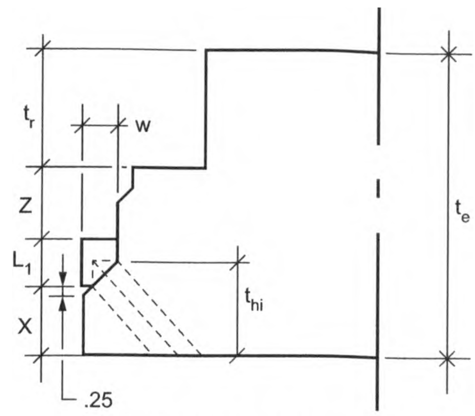
- Calculate distance, $t_{hi}$

  $t_{hi} = (H + W_{ga})/[.6\ \pi\ S_e(G - 2w)]$

  $= (1{,}385{,}589 + 830{,}558)/$

  $\times\ (.6\ \pi\ 17{,}100(24.25 - 1.5))$

  $= 3.022$(Round to nearest 1/4 inch) $= 3.25$ in




END PLUG AND GASKET

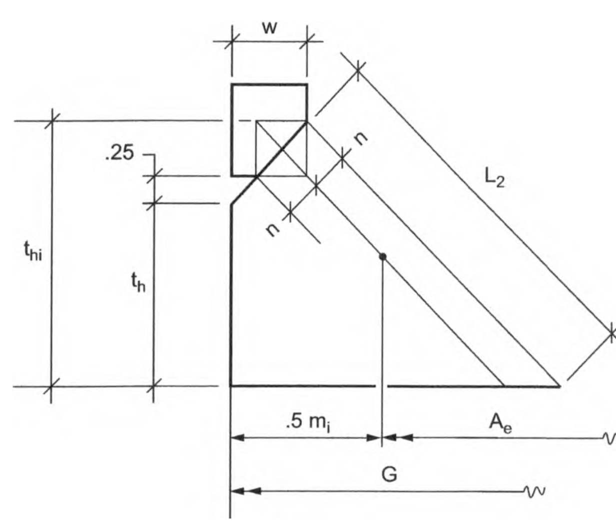
- Distance X

  $X = t - w + 0.25 = 3.25 - .75 + .25 = 2.75$ in

- Actual thickness, $t_e$

  $t_e = t_{rr} + Z + L_1 + X\ 7 + 3 + 1.125 + 2.75$

  $= 13.875$ in




DIMENSIONAL PROPORTIONS

$t_h = t_{hi} - w = 3.25 - .75 = 2.5$ in

$n = .707(w - .25) = .707(.75 - .25) = .3535$ in

$L_2 = 1.414 t_{hi} = 1.414\ (3.25) = 4.596$ in

$$m_i = 2(.354\ L_2 + .25) = 2(.354(4.596) + .25) = 3.75 \text{ in}$$

$A_e = G - m_i = 24.25 - 3.75 = 20.5$ in

- Bending stress, $S_{EB}$

$$S_{EB} = \left[6\left(H + W_{ga}\right)n\right] / \left[\pi A_e\ L_2{}^2\ \text{Sin } 45\right]$$
$$= \left[6\left(1{,}385{,}589 + 830{,}558\right)3535\right] / \left[\pi(20.5)\left(4.596^2\right)\text{Sin } 45\right]$$
$$= 4896 \text{ PSI}$$

- Shear stress, $S_{ES}$

$$S_{ES} = \left(H + W_{ga}\right) / \left(\pi\ A_e\ L_2\ \text{Sin } 45\right)$$
$$= \left(1{,}385{,}589 + 830{,}558\right) / \times \left[\pi(20.5)\ \left(4.596^2\right)\ \text{Sin } 45\right]$$
$$= 10{,}588 \text{ PSI}$$

- Combined stress, $S_{comb}$

$$S_{comb} = \left(S_{EB}{}^2 + S_{ES}{}^2\right)^{1/2}$$
$$= \left[4896^2 + 10{,}588^2\right]^{1/2} = 11{,}661 \text{ PSI}$$

- Allowable stress, $S_e$

$S_e = 17{,}100$ PSI

## 10.0 FLANGE DESIGN

See worksheet

**Table 8-8**
**Summary of materials, stress & allowable stresses - Example**

| | Symbol | Component | Temp | Basis/Description | Material | $F_y$ (PSI) | Allow Stress (PSI) |
|---|---|---|---|---|---|---|---|
| 1 | $S_a$ | Main Studs | Ambient | ASME Code Allowable Stress, Tension | SA-193-B16 | | 25 KSI |
| 2 | $S_b$ | Main Studs | Design Temp | ASME Code Allowable Stress, Tension | SA-193-B16 | | 22.9 KSI |
| 3 | $S_c$ | End Plug | Design Temp | Combined Stress | SA-182-F21 | 26 KSI | 17.1 KSI |
| 4 | $S_e$ | End Plug | Design Temp | ASME Code Allowable Stress, Tension | SA-182-F21 | 26 KSI | 17.1 KSI |
| 5 | $S_f$ | Flange | Design Temp | ASME Code Allowable Stress, Tension | SA-182-F21 | 26 KSI | 17.1 KSI |
| 6 | $S_g$ | Gasket | Design Temp | | T-347 | | 45 KSI |
| 7 | $S_r$ | Retaining Ring | Design Temp | ASME Code Allowable Stress, Tension | SA-182-F21 | 26 KSI | 17.1 KSI |
| 8 | $S_s$ | Gasket | Design Temp | Shear Stress | T-347 | | 15 KSI |
| 9 | $S_{bj}$ | Jack Bolts | Design Temp | Bearing Stress, Use .9 $F_y$ of Flange Material | SA-193-B7 | | 23.4 Ksi |
| 10 | $S_{bp}$ | Pull-Up Studs | Design Temp | ASME Code Allowable Stress, Tension | SA-193-B7 | | 25 KSI |
| 11 | $S_{rb}$ | Retaining Ring | Design Temp | Bearing Stress, Use 1.6 X $S_r$ | SA-182-F21 | 26 KSI | 27.36 KSI |
| 12 | $S_{if}$ | Flange | Design Temp | Hoop Stress in Flange @ inner surface | SA-182-F21 | 26 KSI | 17.1 KSI |
| 13 | $S_{ri}$ | Flange | Design Temp | Hoop Stress in Flange @ inner surface of main stud hole | SA-182-F21 | 26 KSI | 17.1 KSI |
| 14 | $S_{ro}$ | Flange | Design Temp | Hoop Stress in Flange @ outer surface of main stud hole | SA-182-F21 | 26 KSI | 17.1 KSI |
| 15 | $S_{of}$ | Flange | Design Temp | Hoop Stress in Flange @ outer surface of Flange | SA-182-F21 | 26 KSI | 17.1 KSI |
| 16 | | Follower Ring | Design Temp | | SA-182-F21 | 26 KSI | 17.1 KSI |
| 17 | $S_{EB}$ | End Plug | Design Temp | Bending Stress | SA-182-F21 | 26 KSI | 17.1 KSI |
| 18 | $S_{ES}$ | End Plug | Design Temp | Shear Stress | SA-182-F21 | 26 KSI | 17.1 KSI |

*Design Temp = 860°F*

## WORKSHEET FOR BRIDGMAN TYPE STUDDING FLANGES EXAMPLE

### GIVEN

| | |
|---|---|
| FLANGE OD, A OR $A_1$ | 44.25 |
| FLANGE ID, B | 24.25 |
| BOLT CIRCLE, $C_b$ | 30.25 |
| HYDROSTATIC END FORCE, H | 1,385,589 |
| GASKET LOAD, $W_{ga}$ | 830,558 |
| AREA OF GASKET, $A_{ga}$ | 55.37 |
| DIAMETER OF STUDS, $d_S$ | 2.25 |
| HEIGHT OF METAL GASKET, $L_1$ | 1.125 |
| PRESSURE, P | 3000 |
| THICKNESS, t | |

### DIMENSIONS

| | |
|---|---|
| $a = .5\,(A \text{ or } A_1)$ | 22.125 |
| $b = .5\,B$ | 12.125 |
| $r_i = .5\,(C_b - d_S)$ | 14 |
| $r_O = .5\,(C_b + d_S)$ | 16.25 |
| $F = L + .25"$ or ($1.5\,d_S$ min) | 3.375 |
| $C_O = .288\,d_S$ | 0.648 |
| $J = F + (a - r_o)/2$ | 6.3125 |
| $Z = .5\,J + .5\,L_1$ | 3 |

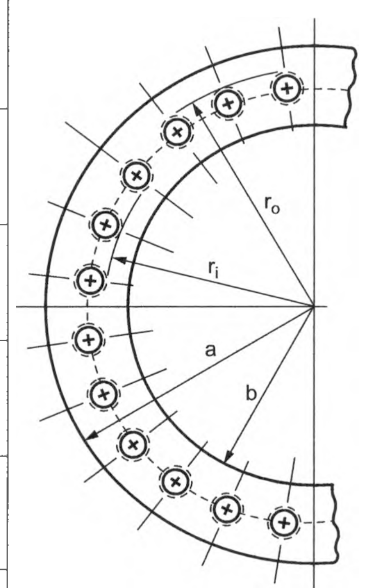
### LOADS & STRESSES

| | |
|---|---|
| $R_1 = [(H + W_{ga})\,L_1]\,/A_{ga}$ | 45,027 |
| $R_2 = P\,[J - (Z + L_1)]$ | 6,563 |
| $R_T = R_1 + R_2$ | 51,590 |
| $P_e = R_T\,/\,J$ | 8,173 |
| $S_{if} = P_e\,[(a^2 + b^2)\,/\,(a^2 - b^2)]$ | 15,189 |
| $S_{ri} = [P_e\,b^2/(a^2 - b^2)]\,[1 + a^2/r_i^2]$ | 12,269 |
| $S_{ro} = [P_e\,b^2/(a^2 - b^2)]\,[1 + a^2/r_o^2]$ | 10,012 |
| $S_{of} = (2\,P_e\,b^2)\,/\,(a^2 - b^2)$ | 7,016 |
| $f_{a1} = .5\,(r_i - b)\,(S_{if} + S_{ri})$ | 25,741 |
| $f_{a2} = .5\,(r_o - r_i)\,(S_{ri} + S_{ro})$ | 25,066 |
| $f_{a3} = .5\,(a - r_o)\,(S_{ro} + S_{of})$ | 50,020 |
| $f_{aT} = f_{a1} + f_{a2} + f_{a3}$ | 100,827 |

| | |
|---|---|
| $f_{an} = f_{at} - [(F + .5\,C_O)\,f_{a2}\,/\,J]$ | 86,138 |
| $Y_1 = f_{at}\,/\,f_{an}$ | 1.17 |
| $S'_{if} = Y_1\,S_{if}$ | 17,778 |

**NOTES:**

1. If $S'_{if} \leq$ Allowable Stress, then the design is OK
2. If $S'_{if} >$ Allowable Stress, then implement a or b below
   a. Increase flange proportions
   b. Add a shrink ring




Plan View-Dimensions

### 3.8. Threaded Closures

Threaded end closure is the nomenclature used to describe configurations where the end plug is secured in place with a threaded main nut. The main nut is threaded into the body of the vessel or end flange as applicable. In other styles of end closures, the end plug is secured with a retainer ring that is connected to the body of the vessel or end flange with studs. The studs are threaded into the end of the shell or flange. To open a threaded closure the entire end plug must be unscrewed out of the body of the vessel using the main nut.

From a design standpoint, the total hydrostatic end force, H, is taken by the threads in a threaded closure, or by the studs in other designs. Various types of gaskets can be used with either design but all are the self-sealing type such as Bridgman, delta or double cone.

Extremely tight fits of the threads should be avoided, and the pitch of the male and female threads should be the same within very close limits. Threads should be generously lubricated with graphite paste for lower temperatures or moly-sulfide for higher temperatures. Coarse threads do not show as much tendency to gall as fine threads. A difference in hardness of mating parts will help to prevent seizure.

#### Notation

$A_m$ = Area between OD and pitch diameter, $in^2$
$A_P$ = Equivalent chamber area, $in^2$
$C_f$ = Body flexibility factor
$C_T$ = Thread flexibility factor
$D_i$ = Inside diameter, in
$D_L$ = Loading diameter, in
$D_o$ = Outside diameter, in
$D_r$ = Diameter at root of thread, in
e = Tooth thickness, in
F = End load per inch at pitch diameter, Lbs/in
$f_b$ = Bending stress, PSI
$F_{avg}$ = Average load on threads, Lbs
$F_i$ = Load on any given thread, Lbs
$F_{sum}$ = Cumulative load on threads, Lbs
$F_y$ = Minimum specified yield stress, PSI
h = Length from root of last thread to end of vessel, in
$h_t$ = Height of thread tooth, in
$K_n$, $K_T$, $K_f$ = Stress concentration factors
$M_X$ = Longitudinal bending moment, in-Lbs
n = Number of threads
$N_R$ = Neutral radius, in
P = Internal design pressure, PSI
$P_t$ = Thread pitch, in
$R_L$ = Loading radius, in
$R_o$ = radius, outside, in
$R_r$ = Radius at root of thread, in
$S_{bL}$ = Combined bending and longitudinal stress at root of thread, PSI
$S_T$ = Combined stress intensity at root of thread, PSI
$S_{tt}$ = Thread tooth stress intensity, PSI
$\sigma_C$ = Combined stress, PSI
$\sigma_X$ = Average longitudinal stress, PSI
$\sigma_{tt}$ = Thread tooth bending stress, PSI
$\nu$ = Poissons ratio
$\beta$ = Damping factor

#### Design Procedure

##### 1.0. Stress in Shell

- Average longitudinal stress across vessel wall, $\sigma_X$

$$\sigma_x = (D_i^2\,P)/(D_o^2 - D_r^2)$$

- Damping factor, $\beta$

$$B = 1.285/(R_r\,t)^{1/2}$$

- Load diameter, $D_L$

$$D_L = .5(D_r + D_i)$$

$$R_L = .5\,D_L$$

- Neutral radius, $N_R$

$$N_R = \left[.5\left(R_o^2 + R_r^2\right)\right]^{1/2}$$

- End load, F, Lbs/in at pitch diameter

$$F = (D_i^2 P)/(4D_L)$$

- Longitudinal bending moment, $M_X$

$$X = \left[\frac{1}{1 + \frac{\beta h}{2} + \frac{1 - v^2}{2\beta_r \beta}\left(\frac{h}{t}\right)^3 \ln\left(\frac{R_o}{R_r}\right)}\right]$$

$$M_X = F(N_R - R_L)X$$

- Bending stress, $f_b$

$$f_b = (6M_X)/t^2$$

- Combined bending and longitudinal stress at root of thread, $S_{bL}$

$$S_{bL} = K_T(\sigma_X + f_b)$$

**2.0. Thread Load Distribution**

- Area between OD and pitch diameter, $A_m$

$$A_m = .25\ \pi(D_o^2 - D_L^2)$$

- Equivalent chamber area at pitch diameter, $A_P$

$$A_P = .25\ \pi\ D_L^2$$

- Body flexibility factor, $C_f$

$$C_f = p_t(1/A_m + 1/A_P)$$

- Thread flexibility factor, $C_t$

$$C_t = 2/D_L$$

- Calculate ratio $C_f / C_t$

$$C_f/C_t =$$

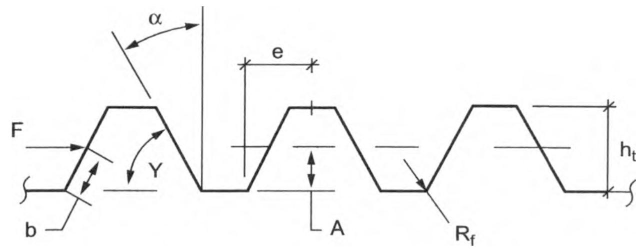
And complete Table below....

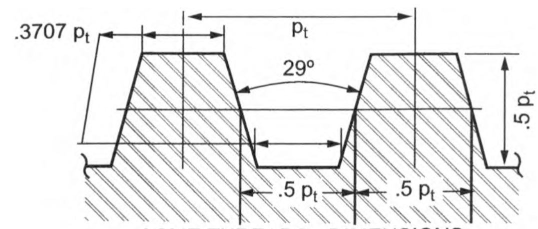
| Thread Load Distribution | | | |
|---|---|---|---|
| Thread | $F_i$ | $F_{sum}$ | % |
| 1 | | | |
| 2 | | | |
| 3 | | | |
| 4 | | | |
| 5 | | | |
| 6 | | | |
| 7 | | | |
| 8 | | | |
| | | | |
| | | | |
| Σ | | | 100 |

Where;

$$F_i = F_{i-1} + C_f/C_t(F_{sum})$$

And % = $F_i$ / $F_{sum}$ (Innermost thread)




ACME THREADS - DIMENSIONS

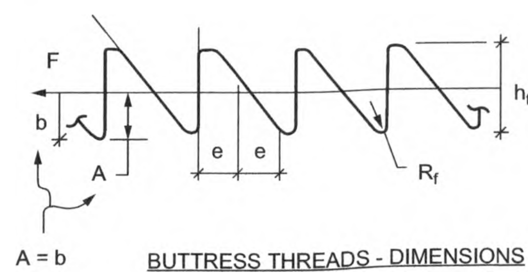
$\gamma = 90° - \alpha$

$$h_t = .5\ p_t$$

$$A = .25\ p_t$$

$$e = (.5\ p_t\ \mathrm{Tan}\ \alpha) + .1853\ p_t$$

$$b = A/\cos\alpha$$




BUTTRESS THREADS - DIMENSIONS

$\gamma = 90°$

**3.0. Thread Root Tooth Bending Stress**

- Thread tooth bending stress, $\sigma_{tt}$

$$\sigma_{tt} = \frac{F}{n}\left[\frac{1.5A}{e^2} + \frac{\cos\gamma}{2e} + \frac{.45}{\sqrt{b\ e}}\right]$$

- Stress concentration factor, $K_n$

  $K_n = \left[1 + .26\ (e/Rf)^{.7}\right]$

- Highest load distribution factor, $K_f$

  $K_f = n\ (\%\ Max/100)$

- Thread tooth intensity, $S_{tt}$

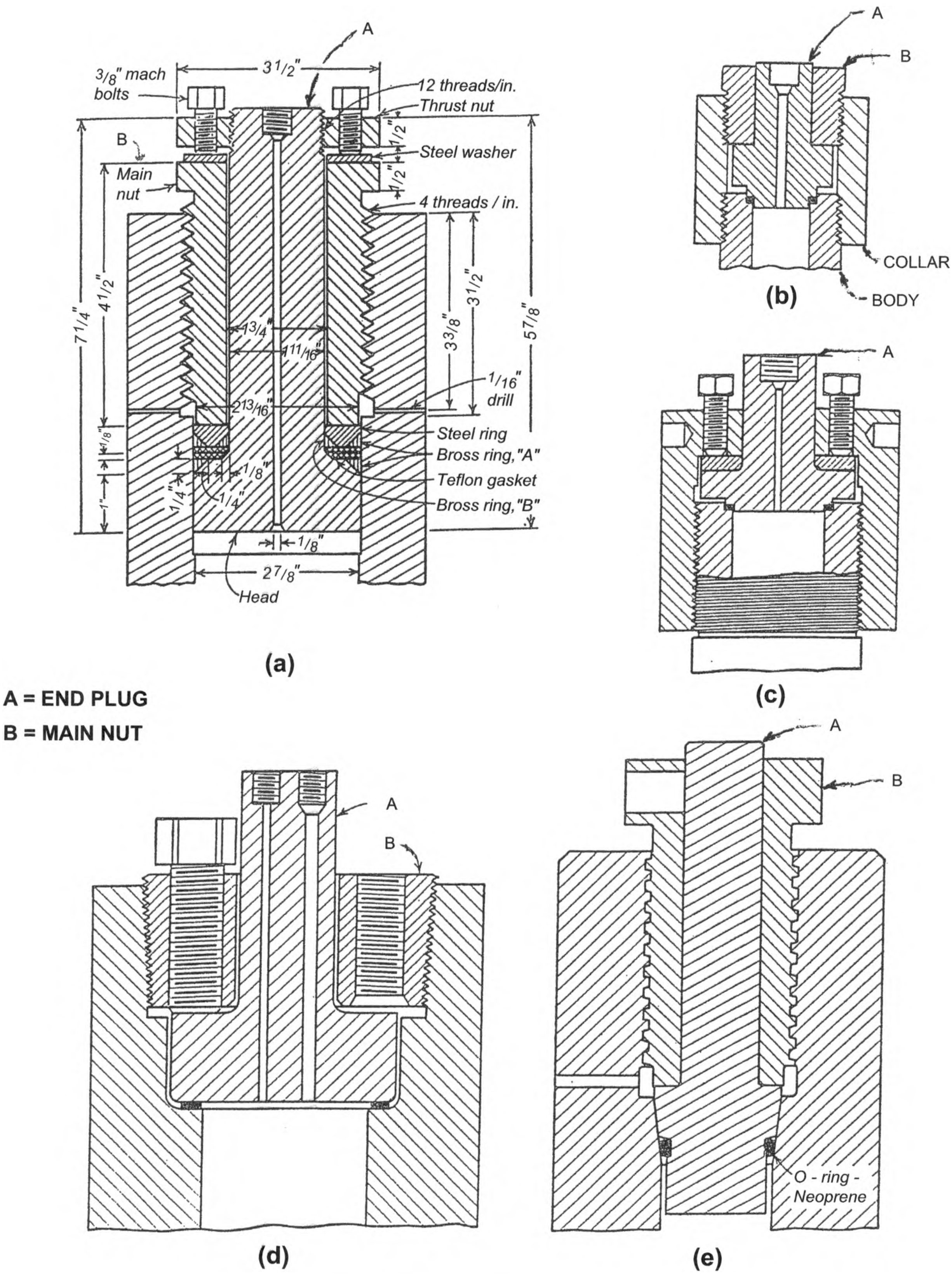
  $S_{tt} = K_f\ K_T\ \sigma_{tt}$

- Combined stress, $S_T$

  $S_T = S_{bL} + S_{tt}$

Notes:

1. Threads are not uniformly loaded
2. The threads closest to the inside (innermost) are the most highly loaded.
3. The maximum load is typically about 2 to 4 times the "average stress load"
4. The practical limit on the number of engaged threads is 10. Threads beyond this will carry negligible load.
5. The innermost threads will carry approximately 20% of the load and the first 4 threads will carry approximately 50%. Conversely, the outermost threads carry only approximately 3 % of the total load.
6. Total stress for fatigue evaluation should be based on the most highly stressed region of the vessel. If a threaded closure is utilized, this worst stress case is at the root of the most highly loaded thread. This stress should be combined with bending and longitudinal shell stresses at the same location.
7. Load distribution depends on a number of factors such as;
   a. Form of threads
   b. Thickness of walls supporting the threads
   c. Pitch of threads
   d. Number of engaged threads
   e. Boundary conditions
8. Although the last threads carry very little load, their contribution in increasing height, h, is significant in reducing the bending moment in the shell carrying the threads.




## THREADED CLOSURES

(a) Modified Bridgman Closure using rings to prevent extrusion of gasket (b) Screwed sleeve end plug (c) Compression Head (d) Tapered Cap –Ring Closure (e) O-ring Closure

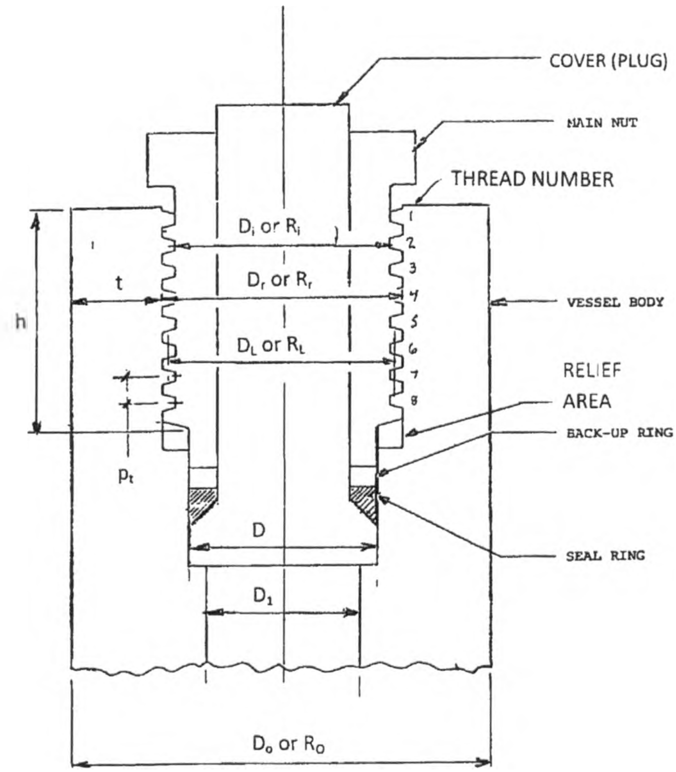



**Figure 8-21.** Bridgman closure, threaded with ACME threads.

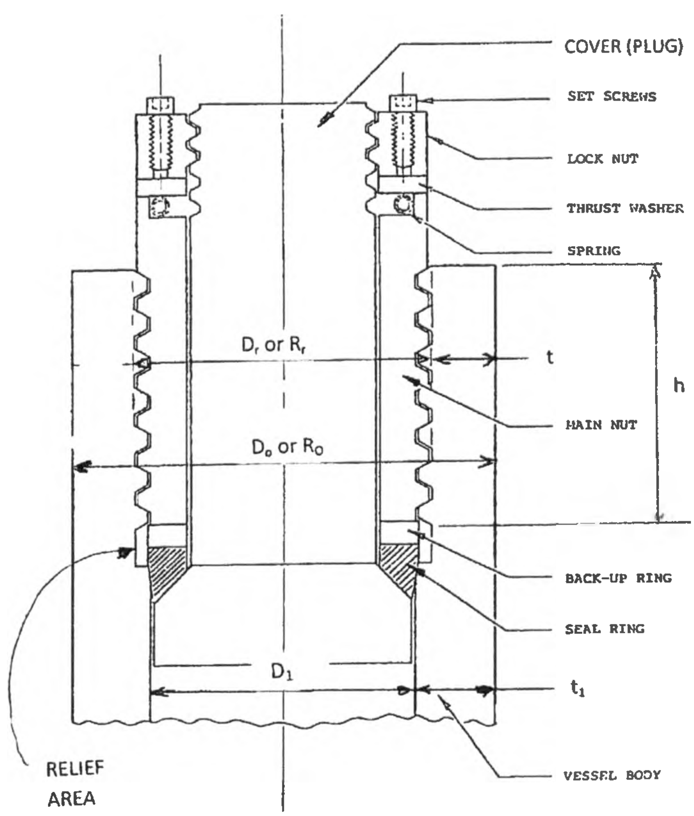



**Figure 8-22.** Modified Bridgman closure, threaded, with ACME threads.

### Threaded Closure With Bridgman Gasket

**EXAMPLE No. 1**

1.0 GIVEN

$D_i = 17$ in
$D_O = 36$ in
$D_r = 18$ in
$F_y = 54$ KSI
$h = 12$ in
$K_T = 1.25$
$n = 12$
$P = 25{,}000$ PSI
$p_T = 1.00$ in
$R_f = 0.125$ in
$R_o = 18$ in
$R_r = 9$ in
$t = R_o - R_r = 9$ in
$\alpha = 14.5^o$
$\gamma = 90 - 14.5 = 75.5^o$
$\nu = .3$

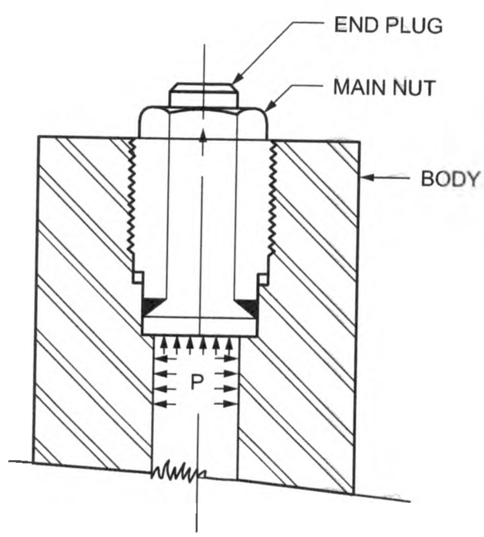
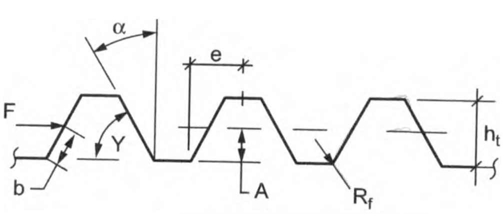



**Figure 8-23.** Simple threaded type closure with a Bridgman closure.




ACME THREADS - DIMENSIONS

$h_t = .5\ p_t = .5\ (1) = 0.5$ in
$A = .25\ p_t = .25\ (1) = 0.25$ in
$e = (.5\ p_t \text{ Tan } \alpha) + .1853\ p_t = 0.3146$ in
$b = A / \text{Cos } \alpha = .25 / \text{Cos } 14.5^o = 0.258$ in

2.0 STRESS IN SHELL

- Average longitudinal stress across vessel wall, $\sigma_X$

$$\sigma_X = (D_i^2\ P)/(D_o^2 - D_r^2)$$
$$= [17^2\ (25{,}000)]/[36^2 - 18^2] = 7{,}433 \text{ PSI}$$

- Damping factor, $\beta$

$$B = 1.285/(\ R_r\ t)^{1/2} = 1.285/(9\ (9))^{1/2} = .143$$

- Load diameter, $D_L$

$$D_L = .5\ (D_r + D_i) = .5\ (18 + 17) = 17.5 \text{ in}$$
$$R_L = .5\ D_L = .5(17.5) = 8.75 \text{ in}$$

- Neutral radius, $N_R$

$$N_R = [.5(R_o^2 + R_r^2)]^{1/2} = [.5(18^2 + 9^2)]^{1/2}$$
$$= 14.23 \text{ in}$$

- End load, F, Lbs/in at pitch diameter

$$F = (D_i^2\ P)/(4\ D_L) = (17^2(25{,}000))/(4(17.5))$$
$$= 103{,}214 \text{ Lbs/in}$$

- Longitudinal bending moment, $M_X$

$$X = \left[\frac{1}{1 + \frac{\beta h}{2} + \frac{1 - v^2}{2R_f\beta}\left(\frac{h}{1}\right)^3 \ln\left(\frac{R_o}{R_1}\right)}\right]$$

$$\left[\frac{1}{1 + \frac{.143(12)}{2} + \frac{91}{2(9).143}\left(\frac{12}{9}\right)^3 \ln\left(\frac{18}{9}\right)}\right]$$

$$\left[\frac{1}{1 + .858 + .353(2.37)\log 2}\right] = .474$$

$$M_X = F(N_R - R_L)\,X$$
$$= 103,214\,(14.23 - 8.75)\,.474$$
$$= 268,000 \text{ in-Lbs}$$

- Bending stress, $f_b$

$$f_b = (6\ M_X)/t^2 = (6\,(268,100))/9^2$$
$$= 19,860 \text{ PSI}$$

- Combined bending and longitudinal stress at root of thread, $S_{bL}$

$$S_{bL} = K_T(\sigma_X + f_b) = 1.25\ (7433 + 19,859)$$
$$= 34,115 \text{ PSI}$$

3.0 Thread Load Distribution

- Area between OD and pitch diameter, $A_m$

$$A_m = .25\,\pi\,(D_o{}^2 - D_L{}^2) = .25\,\pi\,(36^2 - 17.5^2)$$
$$= 777.34 \text{ in}^2$$

- Equivalent chamber area at pitch diameter, $A_P$

$$A_P = .25\,\pi\,D_L{}^2 = .25\,\pi\,(17.5^2) = 240.52 \text{ in}^2$$

- Body flexibility factor, $C_f$

$$C_f = p_t(1/A_m + 1/A_P)$$
$$= 1\ (1/777.34 + 1/240.52) = .00544$$

- Thread flexibility factor, $C_t$

$$C_t = 2/D_L = 2/17.5 = .11429$$

- Ratio $C_f$ / $C_t$

$$C_f/C_t = .005/.114 = .0476$$

| Thread Load Distribution | | | |
|---|---|---|---|
| Thread | $F_i$ | $F_{sum}$ | % |
| 1 | 1.00 | 1.00 | 3.24 |
| 2 | 1.0476 | 2.0476 | 3.39 |
| 3 | 1.1451 | 3.1927 | 3.71 |
| 4 | 1.2971 | 4.4898 | 4.2 |
| 5 | 1.5108 | 6.00 | 4.89 |
| 6 | 1.7964 | 7.7964 | 5.81 |
| 7 | 2.1675 | 9.9639 | 7.01 |
| 8 | 2.6418 | 12.6057 | 8.55 |
| 9 | 3.2418 | 15.8475 | 10.49 |
| 10 | 3.9961 | 19.8436 | 12.93 |
| 11 | 4.9407 | 24.7843 | 15.99 |
| 12 | 6.1204 | 30.9047 | 19.8 |
| Σ | 30.9053 | NA | 100 |

Where;

$$F_i = F_{i-1} + C_f/C_t(F_{sum})$$

And % = $F_i$ / $F_{sum}$ (Innermost thread)

- For the example shown, $F_n$ =

| Thread | Equation | Result |
|---|---|---|
| $F_2$ | $F_1$ +.0476 (1.00) | 1.0476 |
| $F_3$ | $F_2$ +.0476 (2.0476) | 1.1451 |
| $F_4$ | $F_3$ +.0476 (3.1927) | 1.2971 |
| $F_5$ | $F_4$ +.0476 (4.4898) | 1.5108 |

- For example shown, %

| Thread | Equation | Result |
|---|---|---|
| 1 | 1.00 / 30.9053 | .0324 |
| 2 | 1.0476 / 30.9053 | .0339 |
| 3 | 1.1451/ 30.9053 | .0371 |
| 4 | 1.2971 / 30.9053 | .0420 |

4.0 THREAD ROOT TOOTH BENDING STRESS

- Thread tooth bending stress, $\sigma_{tt}$

$$\sigma_{tt} = \frac{F}{n}\left[\frac{1.5\ A}{e^2} + \frac{\cos\gamma}{2e} + \frac{.45}{\sqrt{be}}\right]$$
$$= 103,214/12(3.789 + .3979 + 1.579)$$
$$= 49,593 \text{ PSI}$$

- Stress concentration factor, $K_n$

$$K_n = \left[1 + .26\ (e/R_f)^{.7}\right]$$
$$= \left[1 + .26(.3146/.125)^{.7}\right] = 1.496$$

- Highest load distribution factor, $K_f$

$$K_f = n(\%\,\text{Max}/100) = 19.8/100 \times 12 = 2.376$$

- Thread tooth intensity, $S_{tt}$

$$S_{tt} = K_f\,K_T\,\sigma_{tt} = 2.376(1.496)\ (49,593)$$
$$= 176,278$$

- Combined stress, $S_T$

$$S_T = S_{bL} + S_{tt}$$
$$= 33,054 + 176,278 = 209,332 \text{ PSI}$$
$$< 4\ F_y = 216,000 \text{ PSI}$$

### 3.9. Studs and Nuts

**General.** As pressures get higher and higher, the studs and nuts that secure the end closures get larger and larger. This results in a corresponding increase in the bolt circle to accommodate the increased number and size of studs given the spacing of the studs required. As the bolt circle increases, the flange gets thicker to resist the bending moment. Eventually the designer reaches the point of diminishing returns.

The typical bolt spacing for standard flange design is based on wrench clearances. And the wrench clearances end up governing the design.

One of the principles of good flange design is to reduce the bolt circle to a minimum. This will minimize the size of all the components and result in the most efficient design.

Designers faced with this dilemma in the past have come up with special nuts for high pressure applications. They are called sleeve nuts.

Sleeve nuts can be made much smaller in diameter than nuts for the same diameter. In addition the spacing between studs will not be based on wrench clearances, but on whatever tightening device is utilized.

The spacing of sleeve type nuts should be 1.5 times the stud diameter, as opposed to 2 to 2-1/2 times the stud diameter for conventional bolting.

**Studs.** Studs used for high pressure applications are integral to the overall design, the performance of the joint, and functioning of the vessel. For these reasons, it is imperative that every design detail receive the appropriate attention.

Typically the threaded shank of the stud creates a stress concentration. Numerous tests have shown that this is the most likely area of failure. To preclude this, and minimize stress concentrations, a generous radius is machined into the shank of the stud at the location of the first thread.

This is critical for cyclic service or high temperature operation. Any time high temperature or fatigue are involved, the studs should have a generous radius machined at the root of the first thread.

**Sleeve Nuts.** Sleeve type nuts are utilized whenever normal spacing of nuts based on wrench clearances cannot be met or is not desirable.

Sleeve nuts are a smooth sleeve of metal that is internally threaded. The nuts may have a nut machined to the top of the cylinder to accommodate hand or wrench tightening. However, they are not designed for manual tightening.

Most high pressure applications do not utilize wrench tightening anyway. So why base the bolt spacing based on wrench clearances?

In this case, the nut is not there to create the stud tension, but only used to secure the elongation of the stud. Sleeve nuts achieve this goal with a minimum of metal. Sleeved nuts may have holes drilled in them to facilitate the turning of the nut during stud elongation.

Sleeve nuts have longer threads to develop full strength as opposed to nuts. Sleeve nuts should have a minimum of 10 threads to ensure full strength.

**Washers.** Washers also become instrumental in good joint design for critical applications. Ordinary hardened washers will transmit any eccentric loading due to unlevelness of the bearing surface. Any out of line loading will cause a bending moment in the stud. In order to negate the effects of an unlevel or non-perpendicular surface, spherical washers are used.

We are all aware of the compound effect that bending will have when combined with tension. However, in the design of the bolting, perfect alignment and pure tension are assumed. Our designs are based on the stud being stressed perfectly in tension without any bending. Without spherical washers, this may not be the case.

Typically spot-facing is done to preclude this condition but may not be 100% effective.

The eccentric surface can be caused by several conditions;

a. The back of the flange is not spotfaced
b. The spotfacing was not done properly
c. The studs are not mounted precisely perpendicular to the flange face.

Spherical washers are used in pairs and carefully machined to nest together. In this manner, whatever misalignment occurs, the spherical washers automatically account for the misalignment and keep the stud in pure tension.

#### Stud Pretightening Methods

a. Mechanical pre-tightening
b. Thermal pre-tightening
c. Hydraulic pre-tightening

### Achieving the optimum tightening

For high temperature connections, there are concerns with over-tightening. Over-tightening can cause the bolts to exceed yield after heat up, thus reducing the gasket load. The required torque range should be between the following limits;

a. Above the value to seat the gasket at ambient condition
b. Below the value where the bolts will be overstressed due to high temperature

**Mechanical Pretightening.** Mechanical pretightening is accomplished with wrenches. This method may result in torquing the stud rather than tightening the nut. The accuracy of torque is based on the lead angle of the threads and the frictional characteristics of the overall assembly. These coefficients are variable and depend on a number of factors.

Because of all the factors involved, this method is not considered appropriate for large, critical flange joints.

**Thermal Pretightening.** In this method, the required elongation of the bolt is achieved by thermal expansion of the heated stud. It is essential that the temperature required does not alter the mechanical properties of the stud.

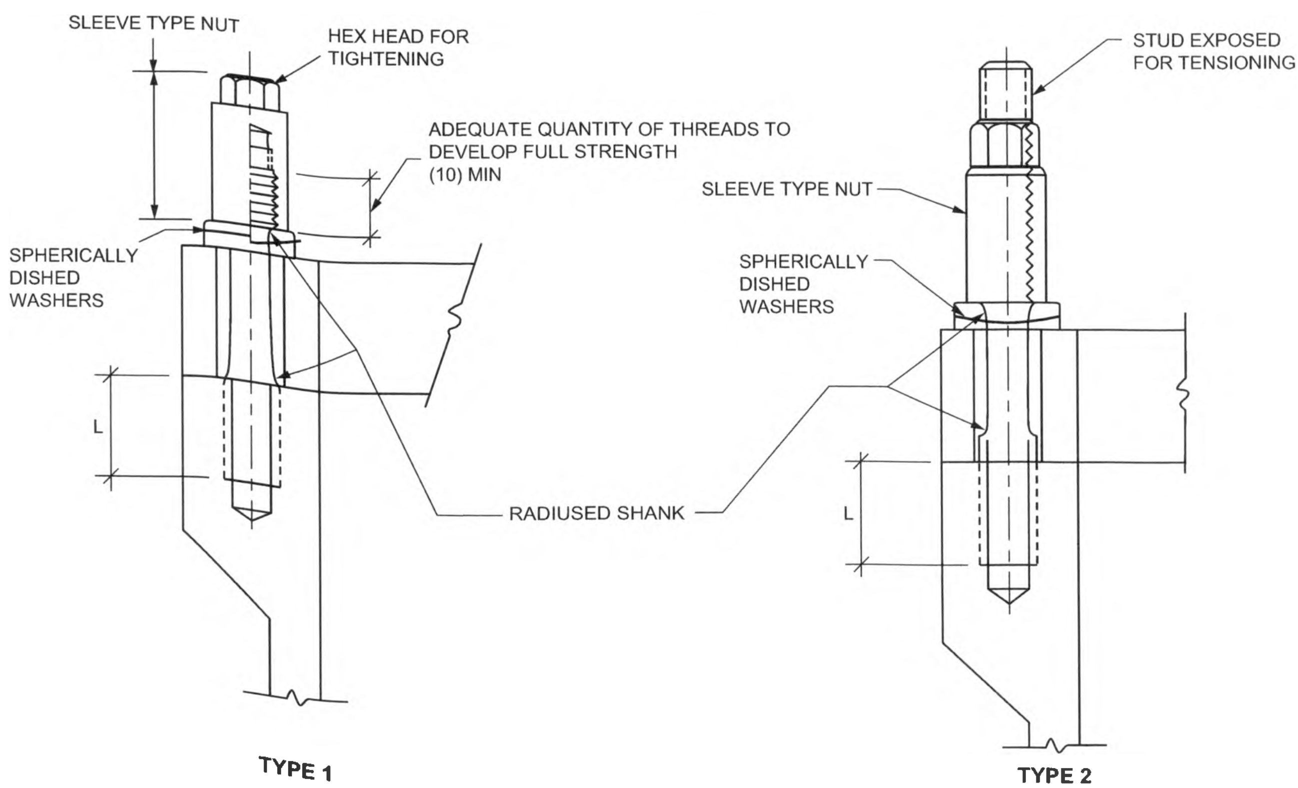
Thermal pretightening is achieved by putting electric resistance wires down a pilot hole in the stud. As the stud is heated, it elongates. Since the thermal coefficient and length are known, it is a simple task to determine the temperature necessary to accomplish a precise elongation.

**Hydraulic Pretightening.** In this case, the elongation of the stud is accomplished by means of a hydraulic jack. The stud is elongated, and then the nut is hand tightened to secure the degree of elongation desired.

Usually four jacks are used simultaneously so that the joint experiences uniform pressure. All four jacks are rotated in turn to the next set of studs.

Tightening is normally done in stages, typically in three increments of 1/3 total elongation required.

SLEEVE TYPE NUTS



## NOMENCLATURE

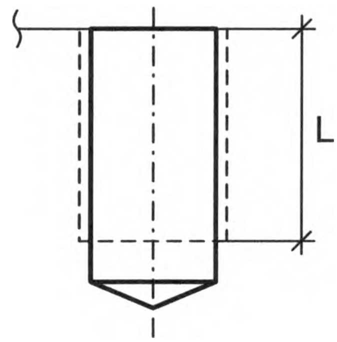
$F_{ys}$ = Yield strength, stud, at design temperature, PSI

$F_{yf}$ = Yield strength, flange, at design temperature, PSI

$d_s$ = Root diameter of stud, in

$R_a$ = Root area, $in^2$

$S_b$ = Average primary stress in stud, PSI



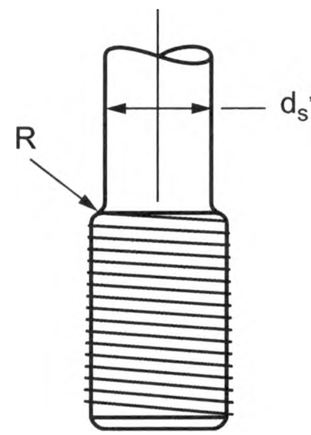

**TYPICAL TAPPED HOLE**

## CALCULATIONS

- Maximum stress in stud, $S_b$
  $S_b = F_{ys} / 1.8$

- Maximum stress in reduced Shank stud, $S_b$
  $S_b = F_{ys} / 1.5$

- Minimum length of thread Engagement, L
  $L > .75\ d_s\ (F_{ys}\ /\ F_{yf}) > d_s$




**REDUCED SHANK STUD**
$d_s' < .9\ d_s$

**REF: ASME VIII-3, ARTICLE KD-6**

**Figure 8-24.** Tapped hole and stud requirements.

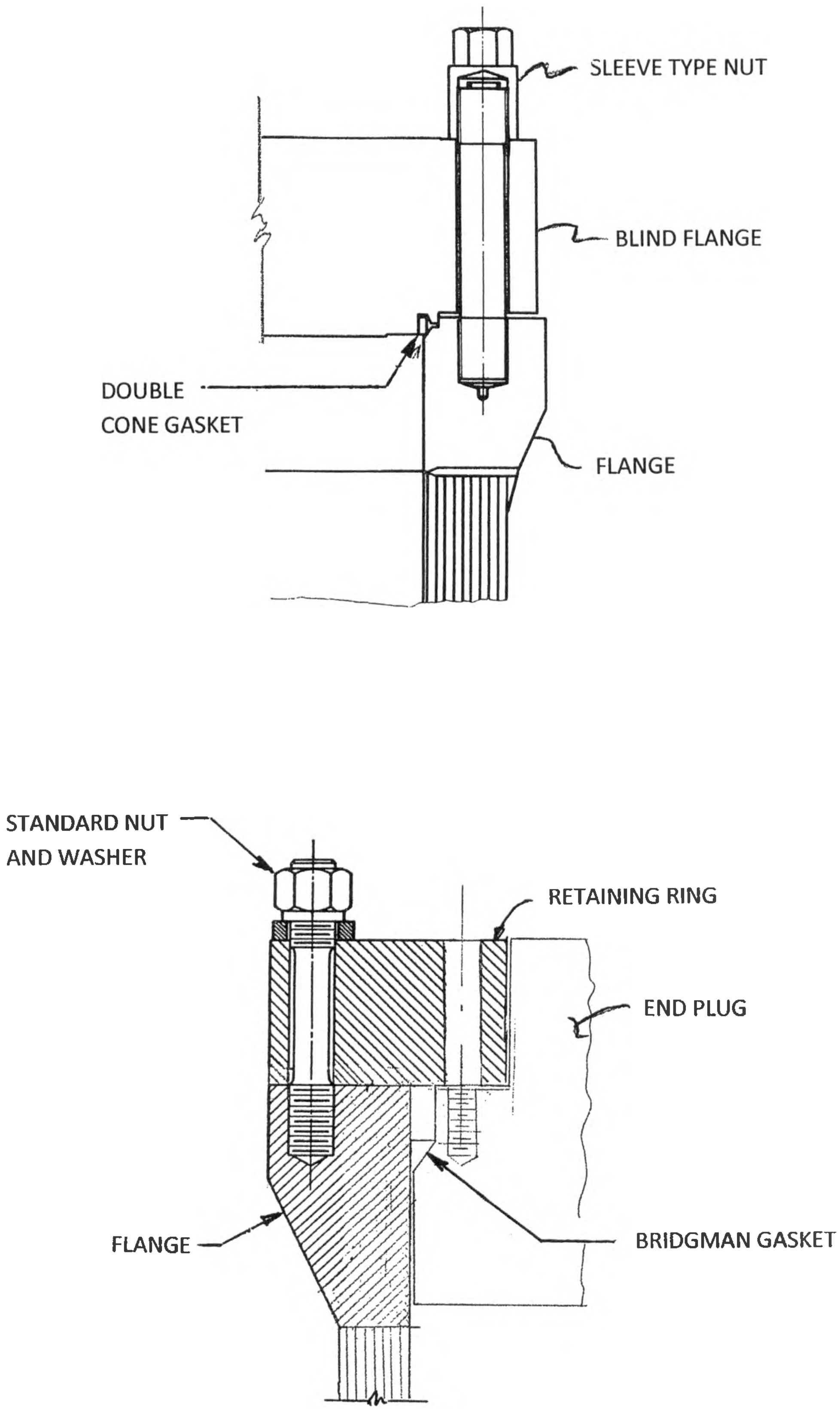



**Figure 8-25.** Examples of studded flanges.

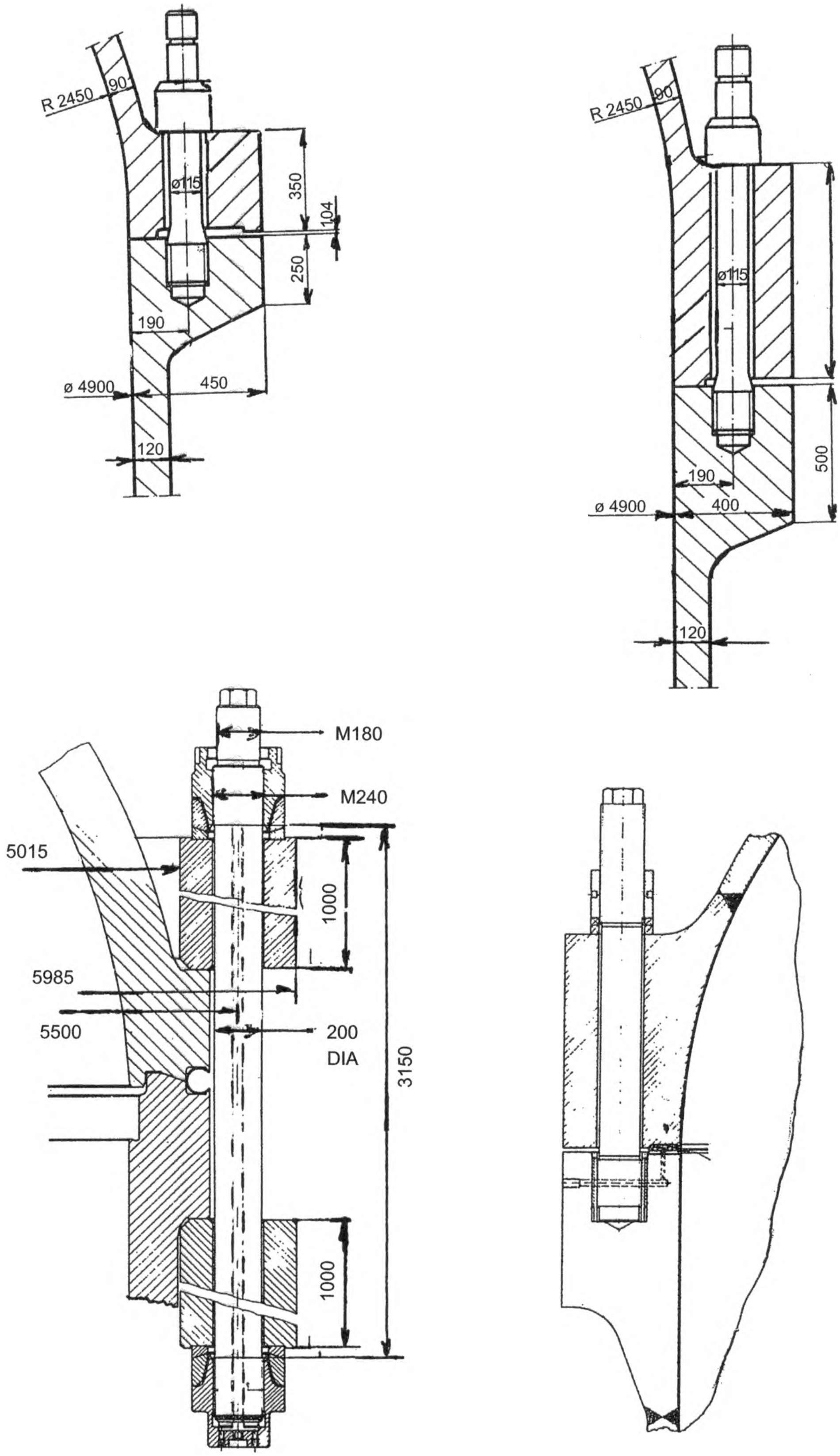



**Figure 8-26.** Examples of studded flanges & head closures.

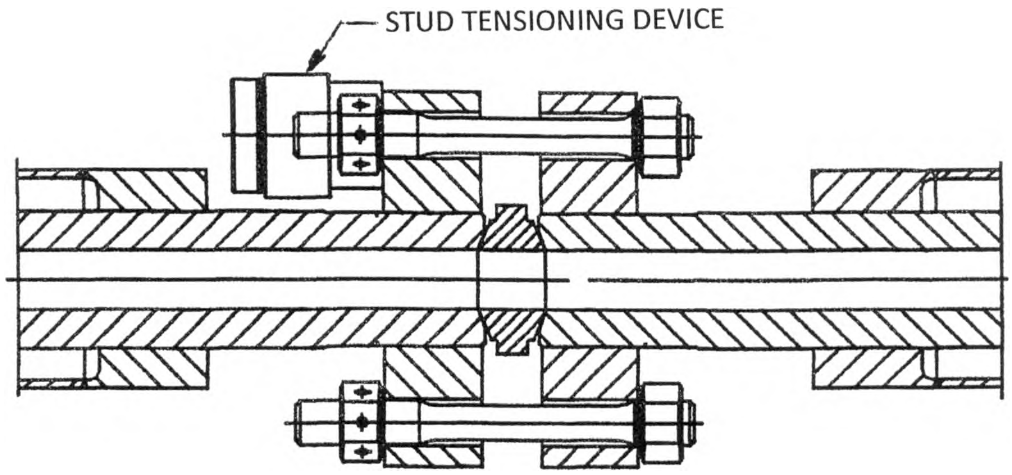



**Figure 8-27.** Piping sub-assy showing, tapered studs, lens ring gasket and clearances for stud tensioning equipment.

## 4.0. Nozzles

Nozzle attachments to high pressure vessels are almost always made through the heads, or end closures, rather than through the shell as is done in common pressure vessels. This is primarily due to the thickness of the shells and the fact that multilayered vessels are common in high pressure systems. Nozzles through multilayer vessels, while not impossible, are discouraged.

Openings through the end closures are typically bolted in place with the flange facing machined onto the outside face of the closure. End closures can be very thick (2-3 feet thick) so welding through the head thickness is not an option. Nozzles can be of the set on type but welding can be difficult or create problems such as lamellar tearing. Bolted on connections typically have tapped holes in the closure for attaching the external flanges. Any facing configuration can be machined into the closure.

Nozzle penetrations through the end closures can also be screwed into tapped connections in the head. These type of connections should be avoided if the vessel is in cyclic service.

Nozzles are attached by one of the following methods;

1. Screwed
2. Bolted
3. Welded
4. Socket Welded

Of these, the first two are the most common. Screwed connections can easily be machined into the closure and are strong enough for applications up to about 60,000 PSI. Flanged connections to the end closure are also used up to about 20,000 PSI. The problem with flanged connections at high pressure is the bending moments created by the extreme end force as well as the large bolting forces and local discontinuities.

Due to the undesirability of leaks in high pressure systems, connections are deliberately kept to small sizes, probably less than 4" NPS. Most connections are smaller than that. Screwed connections up to 1¼ inch may be used up to 20,000 PSI and smaller connections up to 60,000 PSI.

It is not uncommon to machine flats for branch connections to main nozzles on a distribution forging. A distribution forging can be designed to accommodate multiple nozzles. In this manner a heavier forged nozzle neck can be machined to accommodate the size and shape of the attached connection. These attachments should be provided by the vessel fabricator and the connections tested with the vessel during the standard shop hydrotest.

High pressure connections should avoid gasket configurations which are not contained. Ring type joint, male-female, and tongue and groove should be utilized wherever possible.

Nozzle reinforcement:

Nozzle reinforcement is not an issue for smaller nozzles through the end plug or closure. However when the size of the opening exceeds ½ of the nominal vessel diameter, special analysis is required. Utilize standard ASME Code reinforcement rules for all other openings. All nozzles shall be integrally reinforced. No reinforcing pads are allowed.

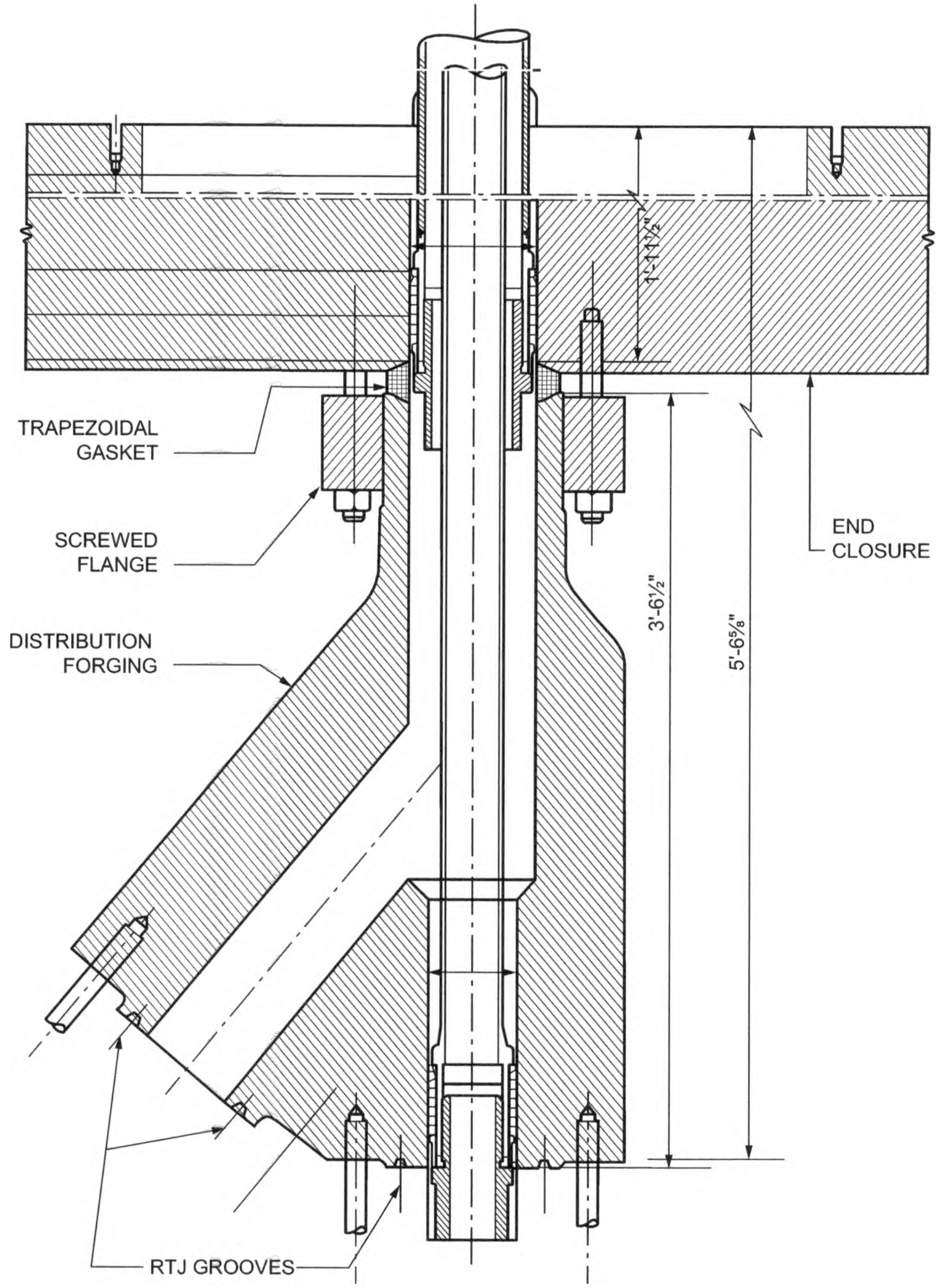
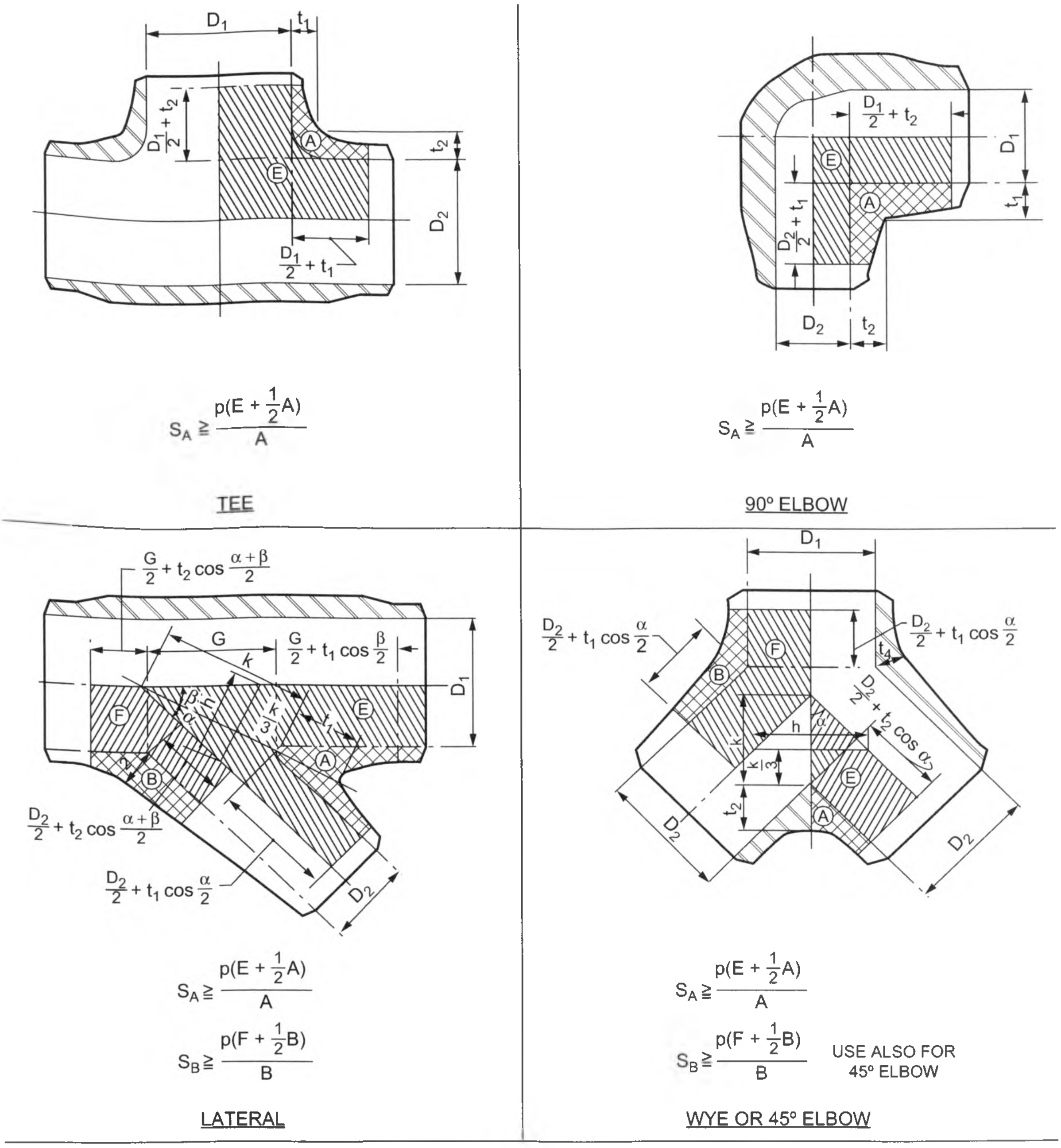



**Figure 8-28.** Example of a distribution forging.



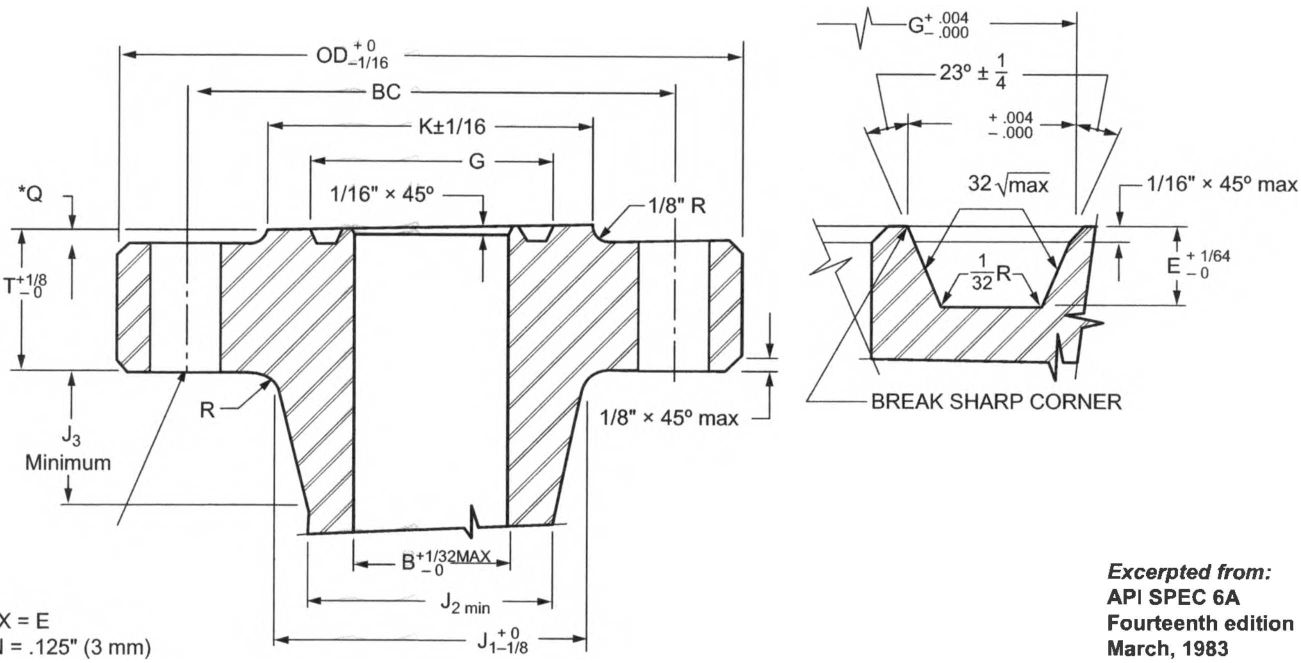

NOMENCLATURE

A, B - METAL AREA, (SQ.IN.)
$D_1$, $D_2$ - INSIDE DIAMETER OF FITTINGS, (IN.)
E, F - INDICATED PRESSURE AREA, (SQ.IN.)
G,h,k - INDICATED LENGTHS, (IN)
p - DESIGN PRESSURE, AT DESIGN TEMPERATURE, (PSIG)
$S_A$, $S_B$ - ALLOWABLE STRESS AT DESIGN TEMPERATURE, (PSI)
$t_1$, $t_2$ - INDICATED METAL THICKNESS, (IN.)
$t_3$ - AVERAGE METAL,THICKNESS OF FLAT SURFACE, (IN.)
$\alpha$, $\beta$ - INDICATED ANGLES.

**Figure 8-29.** Special heavy wall fittings: check of reinforcement for internal pressure.

## High Pressure Flanges

**5,000 to 20,000 PSI**




Q MAX = E
Q MIN = .125" (3 mm)

*Excerpted from:*
**API SPEC 6A**
**Fourteenth edition**
**March, 1983**

| API Type 6BX High Pressure Flanges | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| NOM SIZE | B | OD | T | $J_1$ | $J_2$ | $J_3$ | R | BC | N | $d_b$ | $d_{bh}$ | $L_s$ | K | G | W | E | Ring No. |
| | | | | | | | | **5000 PSI** | | | | | | | | | |
| 1-1/2" | 1.69 | 7 | 1.5 | | | | | 4.875 | 4 | 1 | 1.12 | 5.5 | | | | | 20 |
| 2" | 2.06 | 8.5 | 1.813 | | | | | 6.5 | 8 | 0.875 | 1 | 6 | | | | | 24 |
| 2-1/2" | 2.56 | 9.63 | 1.938 | | | | | 7.5 | 8 | 1 | 1.12 | 6.5 | | | | | 27 |
| 3" | 3.12 | 10.5 | 2.188 | | | | | 8 | 8 | 1.125 | 1.25 | 7.25 | | | | | 35 |
| 4" | 4.06 | 12.25 | 2.438 | | | | | 9.5 | 8 | 1.25 | 1.38 | 6 | | | | | 39 |
| 5" | 5.12 | 14.75 | 3.188 | | | | | 11.5 | 8 | 1.5 | 1.62 | 10 | | | | | 44 |
| 6" | 7.06 | 15.5 | 3.63 | | | | | 12.5 | 12 | 1.375 | 1.5 | 10.75 | | | | | 46 |
| 8" | 9 | 19 | 4.06 | | | | | 15.5 | 12 | 1.625 | 1.75 | 12 | | | | | 50 |
| 10" | 11 | 23 | 4.688 | | | | | 19 | 12 | 1.875 | 2 | 13.75 | | | | | 54 |
| 13" | 13.63 | 26.5 | 4.438 | 18.938 | 16.68 | 4.5 | 0.625 | 23.25 | 16 | 1.625 | 1.75 | 12.5 | 18 | 16.063 | 0.786 | 563 | |
| 16" | 16.38 | 30.38 | 5.5 | 21.875 | 20.75 | 3 | 0.75 | 28.63 | 16 | 1.875 | 2 | 14.5 | 21.06 | 18.832 | 0.705 | 0.328 | |
| 18" | 18.75 | 35.63 | 6.53 | 26.56 | 23.56 | 6 | 0.625 | 31.63 | 20 | 2 | 2.12 | 17.5 | 24.68 | 22.185 | 1.006 | 0.718 | |
| 21" | 21.25 | 39 | 7.12 | 29.875 | 26.75 | 6.5 | 0.68 | 34.88 | 24 | 2 | 2.12 | 18.75 | 27.63 | 24.904 | 1.071 | 0.75 | |
| | | | | | | | | **10,000 PSI** | | | | | | | | | |
| 1-1/2" | 1.69 | 7.188 | 1.656 | 3.31 | 2.4 | 1.85 | 0.375 | 5.56 | 8 | 0.75 | 0.88 | 5 | 4 | 2.893 | 0.45 | 0.218 | BX-150 |
| 2" | 2.06 | 7.875 | 1.73 | 3.94 | 2.938 | 2.03 | 3.75 | 6.25 | 8 | 0.75 | 0.88 | 5.25 | 4.375 | 3.395 | 0.498 | 0.234 | BX-152 |
| 2-1/2" | 2.56 | 9.125 | 2.01 | 4.75 | 3.625 | 2.25 | 0.375 | 7.25 | 8 | 0.875 | 1 | 6 | 5.188 | 4.046 | 0.554 | 0.265 | BX-153 |
| 3" | 3.12 | 10.625 | 2.21 | 5.59 | 4.34 | 2.5 | 0.375 | 8.5 | 8 | 1 | 1.125 | 6.75 | 6 | 4.685 | 0.606 | 0.297 | BX-154 |
| 4" | 4.06 | 12.438 | 2.76 | 7.188 | 5.75 | 2.875 | 0.375 | 10.188 | 8 | 1.125 | 1.25 | 8 | 7.25 | 5.93 | 0.698 | 0.328 | BX-155 |
| 5" | 5.12 | 14.063 | 3.12 | 881 | 7.188 | 3.188 | 0.375 | 11.813 | 12 | 1.125 | 1.25 | 8.75 | 8.7 | 6.955 | 0.666 | 0.375 | BX-169 |
| 6" | 7.06 | 18.875 | 4.06 | 11.938 | 10 | 3.75 | 0.625 | 15.875 | 12 | 1.5 | 1.625 | 11.25 | 11.875 | 9.521 | 0.921 | 0.438 | BX-156 |
| 8" | 9 | 21.75 | 4.875 | 14.75 | 12.875 | 3.688 | 0.625 | 18.75 | 16 | 1.5 | 1.625 | 13 | 14.125 | 11.774 | 1.039 | 0.5 | BX-157 |
| 10" | 11 | 25.75 | 5.563 | 17.75 | 15.75 | 4.06 | 0.625 | 22.25 | 16 | 1.75 | 1.875 | 15 | 16.438 | 14.064 | 1.149 | 0.563 | BX-158 |
| 13" | 13.63 | 30.25 | 6.63 | 21.75 | 19.5 | 4.5 | 0.625 | 26.5 | 20 | 1.875 | 2 | 17.25 | 20.38 | 17.033 | 1.279 | 0.625 | BX-159 |
| 16" | 16.38 | 34.312 | 6.63 | 25.813 | 23.813 | 3 | 0.75 | 30.563 | 24 | 1.875 | 2 | 17.5 | 22.688 | 18.832 | 0.705 | 0.328 | BX-162 |
| 18" | 18.75 | 40.938 | 8.78 | 29.63 | 26.583 | 6.125 | 0.625 | 36.438 | 24 | 2.25 | 2.375 | 22.5 | 27.438 | 22.752 | 1.29 | 0.719 | BX-164 |
| 21" | 21.25 | 45 | 9.5 | 33.38 | 30 | 6.5 | 0.613 | 40.25 | 24 | 2.5 | 2.625 | 24.5 | 30.75 | 25.507 | 1.373 | 0.75 | BX-166 |
| | | | | | | | | **15,000 PSI** | | | | | | | | | |
| 1-1/2" | 1.69 | 7.625 | 1.75 | 3.688 | 2.688 | 1.875 | 0.375 | 6 | 8 | 0.75 | 0.875 | 5.25 | 3.813 | 2.893 | 0.45 | 0.218 | BX-150 |
| 2" | 2.06 | 8.188 | 1.78 | 3.84 | 2.813 | 1.875 | 0.375 | 6.313 | 8 | 0.875 | 1 | 5.5 | 4.188 | 3.062 | 0.466 | 0.218 | BX-151 |
| 2-1/2" | 2.56 | 8.75 | 2 | 4.375 | 3.25 | 2.125 | 0.375 | 6.875 | 8 | 0.875 | 1 | 6 | 4.5 | 3.395 | 0.498 | 0.234 | BX-152 |
| 3" | 3.12 | 10 | 2.25 | 5.0625 | 3.94 | 2.25 | 0.375 | 7.875 | 8 | 1 | 1.125 | 6.75 | 5.25 | 4.046 | 0.554 | 0.266 | BX-153 |
| 4" | 4.06 | 11.313 | 2.53 | 6.063 | 4.813 | 2.5 | 0.375 | 9.06 | 8 | 1.125 | 1.25 | 7.5 | 6.06 | 4.685 | 0.606 | 0.297 | BX-154 |
| 5" | 5.12 | 14.188 | 3.09 | 7.69 | 6.25 | 2.875 | 0.375 | 11.438 | 8 | 1.375 | 1.5 | 9.25 | 7.625 | 5.93 | 0.698 | 0.328 | BX-155 |
| 6" | 7.06 | 19.875 | 0.468 | 12.81 | 10.875 | 3.625 | 0.625 | 16.875 | 16 | 1.5 | 1.625 | 12.75 | 12 | 9.521 | 0.921 | 0.438 | BX-156 |
| 8" | 9 | 25.5 | 5.75 | 17 | 13.75 | 4.875 | 0.625 | 21.75 | 16 | 1.875 | 2 | 15.75 | 15 | 11.774 | 1.039 | 0.5 | BX-157 |
| 10" | 11 | 32 | 7.375 | 23 | 16.81 | 9.29 | 0.625 | 28 | 20 | 2 | 2.125 | 19.25 | 17.875 | 14.064 | 1.149 | 0.563 | BX-158 |
| 13" | 13.63 | 34.875 | 8.06 | 23.438 | 20.813 | 4.5 | 1 | 30.375 | 20 | 2.25 | 2.375 | 21.25 | 21.313 | 17.033 | 1.279 | 0.625 | BX-159 |
| 18" | 18.75 | 45.75 | 10.06 | 32 | 28.75 | 6.125 | 1 | 40 | 20 | 3 | 3.125 | 26.75 | 28.438 | 22.752 | 1.29 | 0.719 | BX-164 |
| | | | | | | | | **20,000 PSI** | | | | | | | | | |
| 1-1/2" | 1.813 | 10.125 | 2.5 | 5.25 | 4.313 | 1.938 | 0.375 | 8 | 8 | 1 | 1.125 | 7.5 | 4.625 | 3.062 | 0.466 | 0.218 | BX-151 |
| 2" | 2.06 | 11.313 | 2.813 | 6.063 | 5 | 2.063 | 0.375 | 9.063 | 8 | 1.125 | 1.25 | 8.25 | 5.188 | 3.395 | 0.498 | 0.234 | BX-152 |
| 2-1/2" | 2.56 | 12.813 | 3.125 | 6.813 | 5.688 | 2.3125 | 0.375 | 10.313 | 8 | 1.25 | 1.375 | 9.25 | 5.938 | 4.046 | 0.554 | 0.266 | BX-153 |
| 3" | 3.12 | 14.063 | 3.375 | 7.563 | 6.313 | 2.5 | 0.375 | 11.313 | 8 | 1.375 | 1.5 | 10 | 6.75 | 4.685 | 0.606 | 0.297 | BX-154 |
| 4" | 4.06 | 17.563 | 4.1875 | 9.563 | 8.125 | 2.875 | 0.375 | 14.063 | 8 | 1.75 | 1.625 | 12.25 | 8.625 | 5.93 | 0.698 | 0.328 | BX-155 |
| 7" | 7.06 | 25.813 | 6.5 | 15.188 | 13.313 | 3.813 | 625 | 21.813 | 16 | 2 | 2.125 | 17.5 | 13.875 | 9.521 | 0.921 | 0.438 | BX-156 |
| 8" | 9 | 31.688 | 8.06 | 18.938 | 196.88 | 4.25 | 1 | 27 | 16 | 2.5 | 2.625 | 22.375 | 17.375 | 11.774 | 1.039 | 0.5 | BX-157 |
| 10" | 11 | 34.75 | 8.813 | 22.313 | 20 | 4.063 | 1 | 29.5 | 16 | 2.75 | 2.875 | 23.75 | 19.875 | 14.064 | 1.149 | 0.563 | BX-158 |
| 13" | 13.63 | 45.75 | 11.5 | 27.313 | 24.75 | 5.25 | 1 | 40 | 20 | 3 | 3.125 | 30 | 24.188 | 17.003 | 1.279 | 0.625 | BX-159 |

## 5.0. References

1. Fryer DM, Harvey JF. High Pressure Vessels. Chapman and Hall; 1998.
2. Comings EW. High Pressure Technology. McGraw Hill Book Co; 1956.
3. Brownell LE, Young EH. Process Equipment Design. John Wiley & Sons; 1959. Chapters 14 & 15.
4. Crum ASD. The distribution of load over threads of a threaded closure, including temperature effects and local imperfections. High Pressure Technology, Material Design Stress Analysis and Applications, ASME PVP-Vol. 148; 1988.
5. Mraz GJ. Development of design criteria for a high pressure vessel construction code. ASME Journal of Pressure Vessel Technology 1987;Vol. 109:256–9.
6. Boyer CB. The development of a high pressure idea. High Pressure Technology, Codes, Analysis and Applications, PVP-Vol. 263; 1993.
7. Coopey W. Holding High Pressure Joints. Petroleum Refiner May, 1956.
8. Antalffy LP, Chaku PN. The New Division 3 High Pressure Vessel Code. High Pressure Technology, ASME 1997.
9. Kendall DP. A short history of high pressure technology from Bridgman to Division 3. Journal of Pressure Vessel technology August, 2000;Vol. 122. ASME.
10. Ludwig EE. How to specify high pressure vessels. Petroleum Refiner October, 1956;Vol. 35(No 10).
11. Faupel JH. Proven pressure vessel design methods. Petroleum Refiner Sept, 1959;Vol. 38(No. 9).
12. Meyer R. Sky high process pressure demand massive vessels. The Refining Engineer January, 1957.
13. Kalnins A, Updike DP. Limit pressures of cylindrical and spherical shells. Journal of Pressure Vessel Technology, Transactions of the ASME August, 2001; Vol. 123.
14. Motoihaga H, Tahara T. Manufacturing monobloc ultra high pressure vessels. High Pressure Technology, PVP-Vol. 344, ASME; 1997.
15. Korner P, Vardag T. A new design method for ultra high pressure vessels. High Pressure Technology, PVP-Vol. 297, ASME; 1995.
16. Barake N, Chaaban A. Effect of cracks on load distribution in threaded end closures and fracture detection. High Pressure Technology, PVP-Vol. 281, ASME; 1994.
17. Chaaban A, Muzzo U. On the design of threaded-end closures for high pressure vessels. High Pressure – Codes, Analysis and Applications, PVP-Vol. 263, ASME; 1993.
18. Grewal AS, Sabbaghian M. Load distribution between threads in threaded connections. Journal of Pressure Vessel Technology February, 1997;Vol. 119.
19. Jawad MH, Farr JR. Structural Analysis & Design of Process Equipment. Second Edition. John Wiley & Sons; 1989. Chapter 15.