OAK RIDGE NATIONAL LABORATORY
operated by
UNION CARBIDE CORPORATION
NUCLEAR DIVISION
for the
U.S. ATOMIC ENERGY COMMISSION

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Volume 4

STRUCTURAL ANALYSIS OF SHIPPING CASKS
VOL. 4. EQUATIONS FOR DESIGNING TOP CLOSURES OF CASKS

A.E. Spaller

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OAK RIDGE NATIONAL LABORATORY
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CONTENTS

LIST OF FIGURES

STRUCTURAL ANALYSIS OF SHIPPING CASKS
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Abstract

Equations for designing top closures of shipping casks that will meet regulations governing the design and performance of casks in which radioactive material is shipped are developed in this report. The three main areas of the closure that require careful design are (1) the gasket or seal, (2) the retaining devices or bolts, and (3) the flanges on the lid and cask. Design data for different types of gaskets are given, equations for determining the bolting arrangement for cask lids are presented, and equations for determining the thicknesses of both round and rectangular flanges for the lid and cask are developed.

1. INTRODUCTION

The general public must be protected from exposure to radiation or the accidental release of activity when radioactive material is transported from one location to another. Spent radioactive fuels are generally shipped in steel-jacketed lead-shielded casks with either a cylindrical or prismatic configuration that have a shielded top closure or lid. The lid serves two main purposes, and these are (1) to provide access to the cavity within the cask and (2) to confine the radioactive material within the cask cavity during all conditions of transport. The work reported here was done to develop equations for use in designing cask lids that will meet the regulations formulated by the U. S. Atomic Energy Commission governing the design and performance of casks in which radioactive material is shipped.

The regulation governing the packaging of radioactive material appears as Part 71 of Title 10 of the Code of Federal Regulations,[1] and the

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[1] Code of Federal Regulations, Title 10, Part 71, Transport of Licensed Materials; see also Federal Register, Vol. 31, pp. 9941-9949, July 22, 1966.

requirements governing the performance of the cask are set forth for both normal and accident conditions. The paragraph related to cask performance under accident conditions is given below.[1]

> (a) A package used for the shipment of a large quantity of licensed material, ... shall be so designed and constructed and its contents so limited that if subjected to the hypothetical accident conditions specified ... as the Free Drop, Puncture, Thermal, and Water Immersion conditions, in the sequence listed ... , it will meet the following conditions:
>
> (1) The reduction of shielding would not be sufficient to increase the external radiation dose rate to more than 1,000 milliroentgens per hour or equivalent at 3 feet from the external surface of the package.
>
> (2) No radioactive material would be released from the package except for gases and contaminated coolant containing total radioactivity exceeding neither:
>
> (i) 0.1 percent of the total radioactivity of the package contents; nor
>
> (ii) 0.01 curie of Group I radionuclides, 0.5 curie of Group II radionuclides, and 10 curies of Group III and Group IV radionuclides, except that for inert gases, the limit is 1,000 curies.

The "Hypothetical Accident Conditions"[1] are given below.

> A free drop through a distance of 30 feet onto a flat essentially unyielding horizontal surface, striking the surface in a position from which maximum damage is expected.
>
> A free drop through a distance of 40 inches striking, in a position from which maximum damage is expected, the top end of a vertical cylindrical mild steel bar mounted on an essentially unyielding horizontal surface. The bar shall be 6 inches in diameter, with the top horizontal and its edge rounded to a radius of not more than 1/4 inch, and of such a length as to cause maximum change to the package, but not less than 8 inches long. The long axis of the bar shall be normal to the package surface.
>
> Exposure for thirty minutes within a source of radiant heat having a temperature of 1,475 degrees F. and an emissivity coefficient of nine-tenths, or equivalent. For calculational purposes, it shall be assumed that the package has an absorption coefficient of eight-tenths. The package shall not be cooled artificially until after the thirty-minute test period has expired and the temperature at the center of the package has begun to fall.
>
> Immersion in water for twenty-four hours to a depth of at least 3 feet.

In addition to the loads that may result from the above accident conditions, the cask lid must also contain the internal pressure of the cask.

The most widely used cask lid is a bolted flange type, as shown in Fig. 1. The closure consists of a lid flange, shield plug (usually stepped) attached to the lid flange, gasket, and retaining bolts or studs. This type closure has proved satisfactory in that it is economical to fabricate and easy to assemble, and it has performed well.

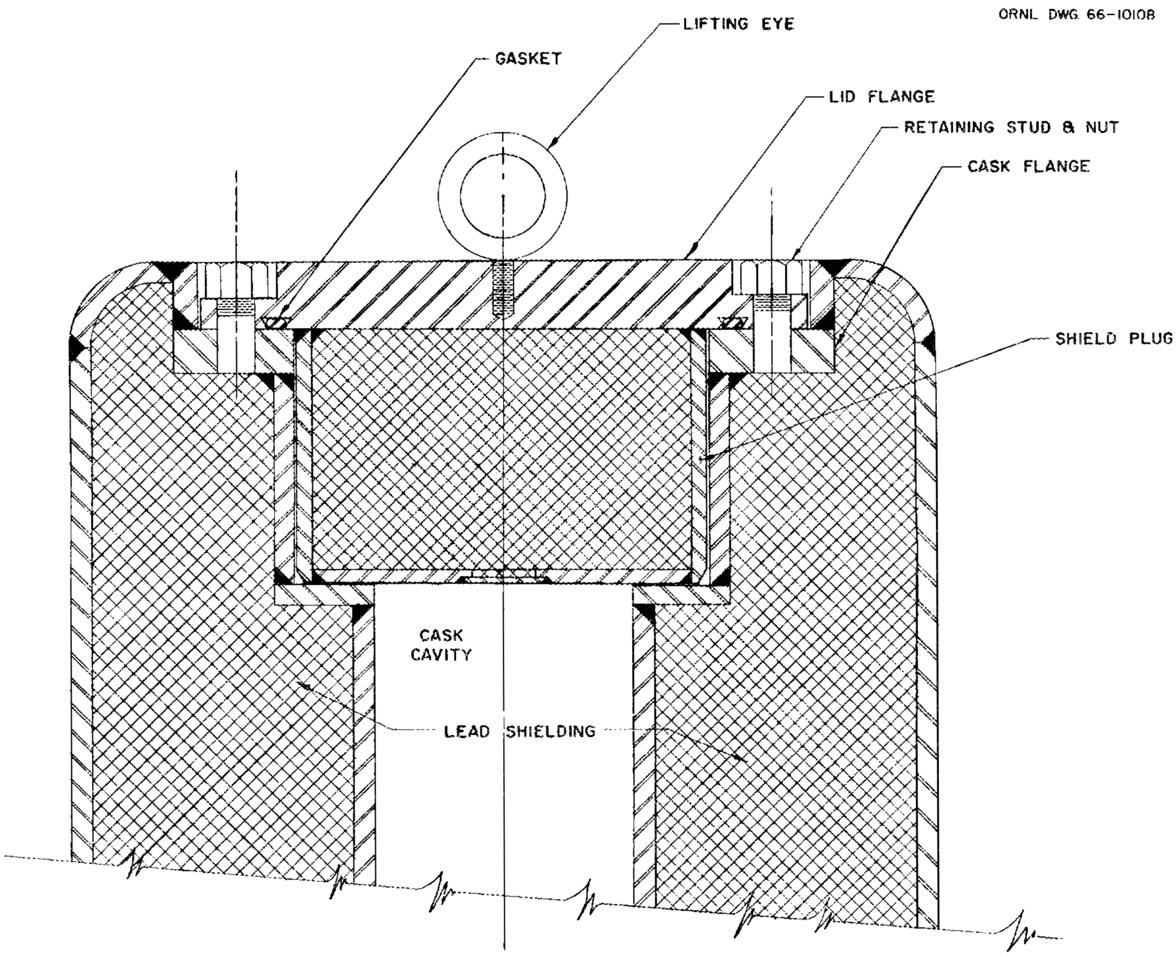


Fig. 1. Typical Top Closure for Shipping Casks.

There are three areas of the top closure that require strength calculations to establish a design for a typical lid that will meet the conditions of the Atomic Energy Commission regulation 10 CFR 71. These areas are (1) the gasket, (2) the retaining studs or bolts, and (3) the flanges on the lid and the cask. The analyses for these various areas are given in the following chapters, and the results are summarized in Chapter 5.

2. GASKET SELECTION AND DESIGN

The gasket or seal must be designed to satisfy both normal and accident conditions of operation. Maintaining this seal during the accident conditions postulated in the Atomic Energy Commission regulation 10 CFR 71 is very difficult because of (1) the 30-ft impact of the cask onto an unyielding surface and (2) the 30-minute fire test.

Generally, large forces and hence large distortion occur as a result of the 30-ft impact. These large forces tend to separate the lid from the body of the cask. Therefore, a great deal of the integrity of the seal depends upon the design of the flanges and retaining bolts or studs. If the lid and cask flanges are very rigid and if the bolting holds these flanges together so that there are no displacements between the two and if the gasket is located near the retaining studs, in all likelihood the seal will be maintained. In addition, the gasket must be capable of withstanding the high temperatures resulting from a 30-minute fire test.

Gaskets most applicable to use as lid seals fall into several basic groups. These are

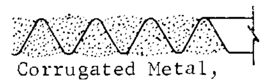
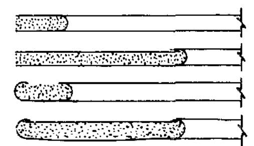
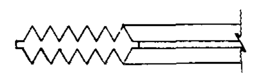
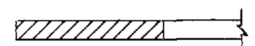
1. flat asbestos gaskets,
2. jacketed types (steel over asbestos, etc.),
3. corrugated metal with or without soft filler,
4. spiral-wound gaskets,
5. plain or machined flat metal,
6. O-ring type metallic gaskets,
7. solid metal with round or special cross section.

The general characteristics of each of these basic groups are discussed in the following paragraphs, and design information pertaining to some of the more common types of gaskets is given in Tables 1 and 2.

To maintain a fluid-tight joint, it is necessary that the parts be tightly bolted together. The initial or bolting-up pressure must be enough to cause local yielding of the gasket where it is in contact with the asperities of the metal flange surfaces. This minimum contact pressure, necessary to secure a tight joint even for low values of the internal

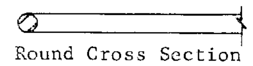
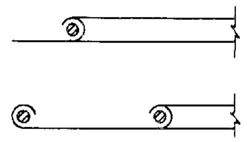
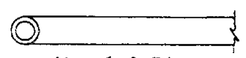
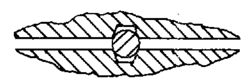
ORNL Dwg 66-10109

Table 1. Design Data For Different Types of Gaskets[a]

Type	Material	Gasket Factor M	Design Seating Stress Minimum Y	Facing Limitations From Table 2
 Flat	Asbestos			
	1/8 in. thick	2.00	1600	1a, 1b, 4, 6 only Column II
	1/16 in. thick	2.75	3700	
	1/32 in. thick	3.50	6500	
 Spiral Wound Metal Asbestos Filled	Carbon steel	2.50	2900	Column II 12 only
	Stainless or monel	3.00	4500	
 Corrugated Jacketed, Asbestos Filled  Corrugated Metal, Asbestos Cord Cemented in Corrugations	Soft aluminum	2.5	2900	Column II 12 only
	Soft copper or brass	2.75	3700	
	Iron or soft steel	3.00	4500	
	Monel	3.25	5500	
	Stainless steels	3.5	6500	
 Corrugated	Soft aluminum	2.75	3700	Column II 12 only
	Soft copper or brass	3.00	4500	
	Iron or soft steel	3.25	5500	
	Monel	3.50	6500	
	Stainless steels	3.75	7600	
 Metal Jacketed, Asbestos Filled	Soft aluminum	3.25	5500	1a, 2 only Column II
	Soft copper or brass	3.50	6500	
	Iron or soft steel	3.75	7600	
	Monel	3.50	8000	
	Stainless steels	3.75	9000	
 Flat Metal, Serrated or Grooved	Soft aluminum	3.25	5500	1a, 1b, 2, 3 only Column II
	Copper	3.5	6500	
	Soft steel	3.75	7600	
	Monel	3.75	9000	
	Stainless steels	4.25	10,000	
 Flat Metal	Lead	2.00	1400	Column I
	Soft aluminum	4.00	8800	
	Soft copper or brass	4.75	13,000	
	Iron or soft steel	5.50	18,000	
	Monel	6.00	21,800	
	Stainless steels	6.50	26,000	

ORNL Dwg 66-10109

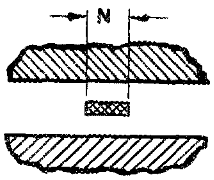
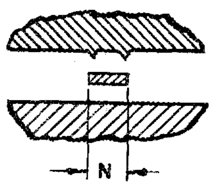
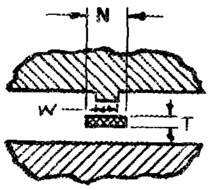
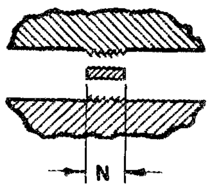
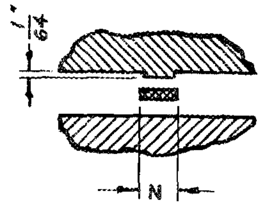
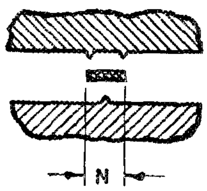
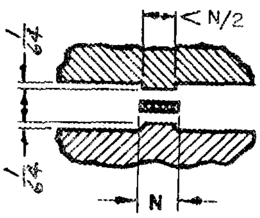
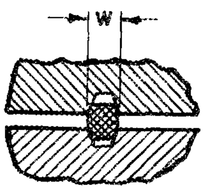
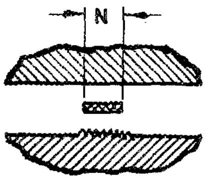
(Table 1 continued)

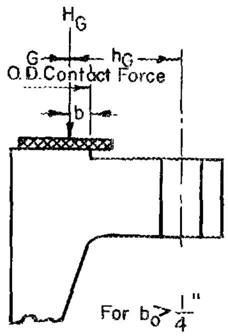
	Material	Gasket Factor M	Design Seating Stress Minimum Y	Facing Limitations From Table 2
 Round Cross Section	Aluminum		1300 lb/circular in.	
	Soft steel (iron)		4500 lb/circular in.	
	Stainless steel		6000 lb/circular in.	
 Wrapped Wire Core	Aluminum jacket		1500 lb/circular in.	
	Aluminum cores			
	Aluminum jacket		1500 lb/circular in.	
	Stainless steel cores			
	Stainless steel jacket		6000 lb/circular in.	
	Stainless steel cores			
 Metal O-Ring	1/16-in.-OD tube x 0.014-in.-thick wall			
	Aluminum		350 lb/circular in.	
	Mild steel		850 lb/circular in.	
	Inconel		1100 lb/circular in.	
	Stainless steel		1300 lb/circular in.	
	1/8-in.-OD tube x 0.012-in.-thick wall			
	Aluminum		100	
	Inconel		300	
	Stainless steel		320	
	1/4-in.-OD tube x 0.012-in.-thick wall			
	Stainless steel		90	
 Ring Joint	Iron or soft steel	5.50	18,000	
	Monel	6.00	21,800	8 only Column I
	Stainless steels	6.50	26,000	

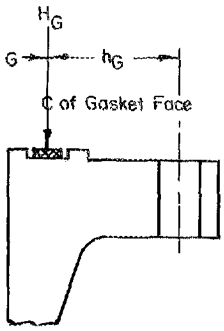
[a] Data taken from the "ASME Boiler and Pressure Vessel Code Section VIII, Unfired Pressure Vessels"; from Machine Design, Seals Reference Issue, 36(6):19 (June 1964); and from M. F. Spotts, p. 453 in Design of Machine Elements, 3rd. ed., Prentice Hall, 1962.

ORNL Dwg 66-10110

Table 2. Effective Gasket Width[a]

Facing Sketch (Exaggerated)	Basic Seating Width b_o Column I	Basic Seating Width b_o Column II	Facing Sketch (Exaggerated)	Basic Seating Width b_o Column I	Basic Seating Width b_o Column II
	$\frac{N}{2}$	$\frac{N}{2}$		$\frac{3N}{8}$	$\frac{7N}{16}$
	$\frac{w+T}{2}$ $\frac{w+N}{4}$ max.	$\frac{w+T}{2}$ $\frac{w+N}{4}$ max.		$\frac{N}{4}$	$\frac{3N}{8}$
	$\frac{w+N}{4}$	$\frac{w+3N}{8}$		$\frac{N}{4}$	$\frac{3N}{8}$
	$\frac{w}{2}$ $\frac{N}{4}$ min.	$\frac{w+N}{4}$ $\frac{3N}{8}$ min.		$\frac{w}{8}$	
	$\frac{3N}{8}$	$\frac{7N}{16}$			





EFFECTIVE GASKET SEATING WIDTH

$b = b_o$, when $b_o \leq 1/4$"

$b = \frac{\sqrt{b_o}}{2}$ when $b_o > \frac{1}{4}$

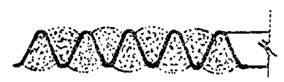
[a]Data taken from the "ASME Boiler and Pressure Vessel Code Section VIII, Unfired Pressure Vessels".

ORNL Dwg 66-10111

Corrugated Gaskets

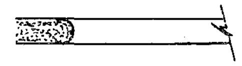
Plain metal gasket with corrugations or embossed interruptions. Corrugations are concentric with ID. Best suited for smooth-faced, complex or noncircular, low-pressure (500 psi) applications such as valve bonnets, aircraft gas turbine fuel and combustion lines, etc. Available in metal thicknesses 0.010 to 0.031 in., with corrugation pitches 0.045 to 0.250 in. Overall gasket thickness is 40-50 per cent of corrugation pitch.

Same as plain corrugated, only a coating of sealing compound is applied. Compound extends pressure limit to 1000 psi. Flange surface finishes can be somewhat rougher.



Corrugated metal gasket with asbestos cord cemented in corrugations. Generally best suited for low pressure (600 psi) on relatively large uneven surfaces such as machined flanges, steam chests, low pressure high temperature exhaust-gas ducts, etc. Available with 5/32, 3/16, and 1/4 in. corrugation pitch only. Thickness is 65-75 per cent of pitch.

Metal Jacketed, Soft-Filler Gaskets

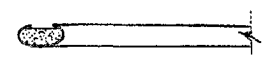


One-Piece French Type

Used for narrow circular applications requiring positive unbroken metal gasket face across full width. Requires flange surface finish of 80 rms or better in sizes less than 1/4 in. wide. Over 1/4 in. wide requires concentric serrated flange face. Minimum gasket width is gasket thickness times 1.5.

Two-Piece French Type

Used for wide or irregular shapes not requiring protection of filler material or additional flange support at outer edge. Tooling less costly than for one-piece type. Interchangeable with one-piece type.



Single Jacket

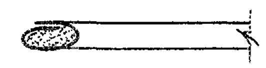
Used for relatively narrow applications similar to French type, but width-diameter limitations do not apply. Generally less costly than French type. Noncircular as well as circular shapes can be furnished. If over 1/4 in. wide, use double-jacketed type. Requires flange surface finish 80 rms or better.

Double Jacket

Used when complete protection of the filler material is required. Also provides additional support of flange at outer edge by addition of lapped over jacket. Available in irregular noncircular shapes but tooling more costly than other types. For widths less than 5/32 in., use French or single-jacket type. Requires flange surface finish 80 rms or better.

Corrugated Jacketed

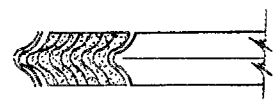
Jacket is corrugated to increase its resilience. Used for circular and moderately noncircular shapes in widths 1/2 in. and wider. Sealability better than other types because of corrugations. Sealability can be further improved by use of gasket compound. With corrugated metal filler instead of asbestos, temperature limited only by metal selected.



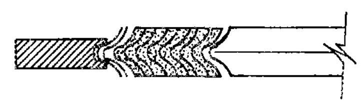
Modified Double Jacket

Used when completely enclosed gasket is required in widths less than those available in plain double jacket. Generally not available smaller than 1-in. ID. Also available with filler made from meshed metal wire which imparts more resilience than nonmetallic filler.

Spiral Wound Gaskets

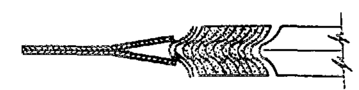


General purpose, spiral-wound gasket. Consists of preformed, V-shaped strip of metal which is wound into a spiral. Metal layers are separated by a filler, usually asbestos. Has good resilience and sealability. This type used where no centering or compression limiting device is required. Also used in metal-to-metal joints.



Spiral-wound gasket provided with a solid metal centering and compression limiting ring around the outer edge. Used where gasket must be located remote from the bolts or other centering means and when it is important to limit compression because of possible over-bolting, control of stack-height, etc. For closures with circular or moderately noncircular bores and noncircular outer perimeter, the solid metal ring can be made to the required configuration and bolt holes drilled through it if necessary.

Spiral-wound gasket with inner and outer compression limiting rings of solid metal for the most extreme operating conditions. Rings fill space between flange faces which might otherwise allow excessive turbulence or erosion of facings.



Spiral-wound gasket with lightweight metal devices to center the gasket without restricting compression. Used in non-critical or lightly bolted assemblies where there is no possibility of over bolting or over compression. Many different centering device configurations are available.

Fig. 2. Jacketed, Corrugated-Metal, and Spiral-Wound Gaskets (from Machine Design, Seals Reference Issue, 36(6) June 1964).

service, and the actual temperature limit depends upon the temperature limits of the filler and metal of the gasket.

The asbestos fillers are generally not considered suitable for applications where the actual temperature of the gasket exceeds 850°F and where the assembly is subjected to cyclical temperature and pressure variations. However, there are many satisfactory installations where line temperatures have been as high as 1300°F for sustained periods of operation. For gasket temperatures above the 850°F range, so-called "ceramic fiber" fillers are available, and when used with a suitable metal, they expand the temperature limits to about 1600°F.[1]

Normally, these two types of gaskets require a moderately high seating stress, and therefore the flanges must be very rigid and have sufficient bolting capacity.

Flat, Round, and Special-Cross-Section Solid Metal Gaskets

The flat, round, and special-cross-section types of solid metal gaskets are illustrated in Fig. 3. All of these types seal by virtue of the flow of the gasket surface caused by brute-force compressive loads. The loads must actually exceed the yield strength of the gasket metal on the gasket-contact area. Therefore, the surface finish of both the flanges and the gasket is very important.

For the best performance, flat plain metal gaskets should be used between flange faces with concentric serrated surfaces. If this is not practical, a very light-cut spiral tool finish of 80 microinch rms may be used. For solid metal gaskets with a round cross section and flat-faced flanges, the surface finish of the flanges should be 80 microinch rms or better. If one flange has a V-shaped groove, the other should be flat. The volume of the V-groove should be less than the volume of the gasket so that there is no possibility of metal-to-metal contact of the flanges. The surface finish of the flanges should be a maximum of 150 to 200 microinch rms.[1]

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[1] Machine Design, Seals Reference Issue, 36(6): 19(June 1964).

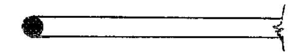
ORNL Dwg 66-10112

Flat Metal Gaskets

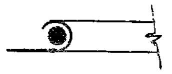
Plain, solid, flat gasket. Probably more widely used than any other type where compressibility is not required to compensate for flange surface finish, warpage or misalignment, and sufficient clamping force is available for the metal selected. Can be fabricated in any desired configuration. Unless the gasket is machine formed, width should be at least metal thickness plus 50 per cent. There are no limitations on flat gasket dimensions. However, width of metal sheet commercially available may limit size and require welding to obtain gaskets beyond certain sizes.

Serrated or grooved, flat metal gasket. Used when a solid metal gasket is required because of pressure (radial strength), temperature, or the highly corrosive attack of the confined fluid, and bolting force is not sufficient to seal a flat gasket. Another application is in screwed closures where the relatively small contact area of the thin serration peaks keeps friction down to a level low enough to seal the joint. Available in some simple noncircular shapes. Generally 3/64 in. or thicker.

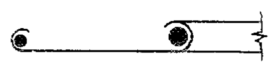
Round Cross-Section Solid Metal Gaskets



Round cross-section, wire ring. Used on valve bonnets, air or gas compressors, vacuum pumps, and accessory connections. Flanges are usually grooved to locate gasket during assembly.

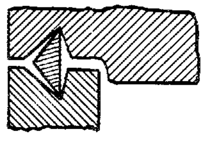


Wire core partially wrapped with a jacket of the same or softer metal. For use between nongrooved flanges, outer edge of jacket provides means of centering on flange bolts. Also allows use of soft metal jackets for sealability and hard metal core for radial reinforcement.



Used in full-faced applications where flange dishing would be excessive with other round section types. Inner wire cross section normally about 0.020 in. heavier than outer wire. Assures full initial seating load at gasket ID, but outer bulb restricts the amount of flange bending. Bolt holes are in connecting web or jacket. Especially useful in large diameter vacuum equipment costly to provide with special flange facings.

Solid Metal, Heavy Cross-Section Gaskets

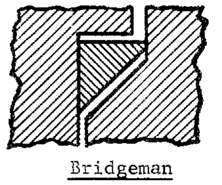


Delta

This gasket operates on the unsupported area principle. It is mainly a pressure vessel or valve-bonnet gasket. Groove and gasket dimensional control are too precise and costly to allow its use as a piping gasket. It is useful in the pressure range 5000 psi and up.

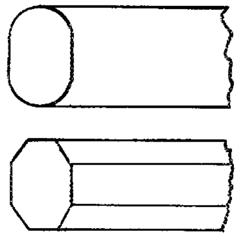
The gasket is mounted in a triangular-shaped groove, with the apex of the triangle outward. The groove is shaped so that only two points of the trianglar gasket contact the groove during initial seating. The two points deform under seating pressure, causing the initial seal.

When pressure is applied, the exposed inner periphery of the gasket reacts, and the gasket flexes about the two contact points. The inner periphery tends to become curved, and the sides of the triangle wedge into the groove, increasing the effectiveness of the seal.



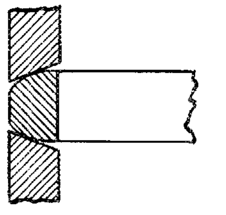
Bridgeman

A type of pressure activated gasket for pressure-vessel heads and valve bonnets. Used for pressures 1500 psi and up. Also has been adapted to piping joints subject to extreme temperature shock conditions. A Bridgeman gasket may be one of several designs, the name referring to the type of closure. The gasket is supported against the periphery of the head and the vessel wall, in such a way that increasing pressure magnifies the force holding the gasket in place. Therefore, the force tending to secure the seal is always greater by some fixed amount than the internal pressure in the vessel. Requires fine tolerances and careful handling to effect satisfactory seal.



Oval or Octagonal

Used in high pressure piping systems and pressure vessels. Range 1000 to 10,000 psi. Excellent mechanical joint. Very high gasket pressures can be obtained with moderate bolt loads. Used only in ring-gasket joints. The standard gaskets are not pressure actuated. The BX modification of the octagonal cross section was developed for oilfield drilling and production equipment at pressures to 15,000 psi. The BX design is pressure actuated.



Lens Ring

The lens gasket is a line contact seal for high-pressure piping systems, with some application in pressure-vessel heads. There are many modifications of the basic lens ring. The most popular lens ring has spherical faces as shown above, and is used between flanges with straight tapered (20 deg.) faces. The line of contact between gasket and flange faces is at a point approximately 1/3 of the way across the gasket face.

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.

Hardness of the conventional lens ring varies with the metal required for the service conditions. Generally the ring should be softer than the flanges.

Fig. 3. Types of Solid Metal Gaskets (from Machine Design, Seals Reference Issue, 36(6) June 1964).

Each of the other special cross section types of gaskets shown in Fig. 3 requires a fine surface finish and close tolerance control along with careful assembly. The maximum temperature limits recommended for continuous service operation of some of the materials used for solid metal gaskets are given in Table 3.

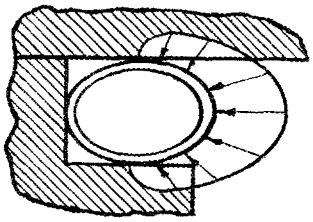
Table 3. Temperature Limits of Metallic Gasket Materials (from Ref. 1)

Material	Maximum Temperature (°F)
Lead	212
Common brasses	500
Copper	600
Aluminum	800
Stainless steel type 304	800
Stainless steel type 316	800
Soft iron, low carbon steel	1000
Stainless steel type 502	1150
Stainless steel type 410	1200
Silver	1200
Nickel	1400
Monel	1500
Stainless steel type 309	1600
Stainless steel type 321	1600
Stainless steel type 347	1600
Inconel	2000

O-Ring Type Metallic Gaskets

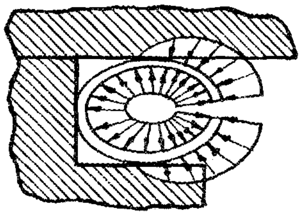
Three types of metallic O-ring gaskets are illustrated in Fig. 4. Under operating conditions, hollow metallic O-ring gaskets possess certain characteristics that static rubber O-rings do not possess. Under very high pressures, clearances for rubber O-rings must be prohibitively small to prevent rubber extrusions, and use of the rubber O-ring is limited to temperatures below the decomposition temperature of the rubber.[1]

ORNL Dwg 66-10113



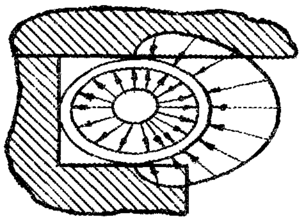
Plain, Sealed Metallic O-Ring

Used for fully-confined or semi-confined designs. When used in fully-confined ring joints, standard metallic O-rings and pressure filled O-rings are useful at temperatures from -420°F to +3000°F. They will seal vacuum, pressure, corrosive liquids, and gases. Standard metallic O-rings will not seal in semi-confined designs to the same high temperature and pressure as a pressure filled O-ring, but they are more economical.



Self-Energizing Metallic O-Rings

Used exclusively for semi-confined designs. Inner periphery is vented by small holes, therefore, pressure inside the ring is the same as the pressure in the system. Since sealing occurs at two upper and lower points, increasing the internal pressure causes ring to be crammed into groove, increasing sealing effectiveness.



Pressure-Filled Metallic O-Ring

Used for fully-confined or semi-confined design. Ring is filled with an inert gas at usually about 600 psi. At elevated temperatures the gas pressure increases, offsetting the inherent loss of strength in tubing at high temperatures, and actually increasing the resilience. This ring cannot support the pressure that the self-energizing ring can endure, but it is useful in the temperature range of 800°F to 1500°F.

Fig. 4. Three Types of Hollow Metallic O-Rings (from Machine Design, Reference Issue, 36(6), June 1964).

The hollow doughnut-shaped metallic O-ring is made by forming a metal tube, usually in a hoop shape, welding the ends together, and then grinding the weld flush. These O-rings have a natural resiliency somewhat similar to that of rubber but without the critical temperature limitation, and they are used like gaskets in static joints. There are two types of static ring joints for metallic O-rings: fully confined gasket joints and semi-confined joints. O-rings used in both types of joints are dependent upon good compressive forces in the flange faces to create cold flow of the metal. This cold flow creates the seal just as in an ordinary gasket.

Semi-confined joints are the most often used with hollow metal O-rings. In these joints, the ring is squeezed in only one plane. The groove or gland is shaped so that the ring is distorted out of round when the joint is tightened, and sealing occurs on the two flattened faces of the ring. It is possible to use designs that seal on the outside or inside diameter of the O-ring, but these applications are more difficult.

A typical fully confined joint for a hollow metal O-ring is formed by machining two V-grooves in matching flange faces, as illustrated in Fig. 5. This joint resembles the ring joint used with ordinary solid metal rings. The volume of the V-groove is about equal to the volume of the O-ring. When the joint is bolted down, the O-ring is distorted to a round-cornered square shape, and when the clamping forces are released, the ring never fully recovers its round shape. Although this joint provides a good seal, the concentricity requirements are critical, which makes the joint expensive.[1]

ORNL Dwg 66-10114

(a) (b)

Fig. 5. Typical Fully Confined Joint for a Hollow Metal O-Ring. (a) Matching V-Grooves are Machined in the Flange Faces, and (b) When Bolted Down, the O-Ring is Distorted to a Round-Cornered Square. (From Ref. 1)

Metallic O-rings are usually made from one of the types of stainless steels, with Type 321 being the most commonly used. Type 321 stainless is titanium-stabilized, which prevents carbide precipitation during welding. Other metals, such as aluminum and copper are available.

Hollow metallic O-rings are often used with a coating to increase sealing effectiveness in grooves with poor finish. Coatings of silver and other metals are sometimes used. These coatings help to prevent seizing or galling when the rings are used in screwed closures and are mandatory to provide an effective seal against gases.

For vacuums, use a plain stainless steel ring with a silver coating. For pressures up to 100 psi use a plain ring with medium wall thickness. For pressure 100 to 40,000 psi, and above, use a self-energizing ring with a strong wall. Coatings are necessary on rings sealing gases or volatile liquids. The fluid sealed determines the necessity for coating or plating O-rings as follows.

1. Lubricating oils and heavy liquids can be sealed with an unplated ring.

2. Gases, vacuums, and light liquids such as water require a coating or plating as follows:

   a. -430 to 1300°F - silver plating,

   b. above 1300°F - consult O-ring manufacturer,

   c. if silver is not compatible with fluid - consult O-ring manufacturer.

At very cold temperatures, flanges, bolts, and O-rings should be made of the same material. Temperature determines the basic O-ring material as follows:

a. -430 to 450°F - Type 321 stainless steel,

b. 450 to 800°F - Inconel,

c. 800 to 1300°F - Inconel X,

d. above 1300°F - consult O-ring manufacturer.

The thickness of the wall of the tubing used to form these O-rings provides the necessary resistance to compression that creates the initial seal. The wall thickness required depends a great deal upon the nature of the material to be confined. Heavy liquids are the easiest to seal and can be confined with thin-walled rings. Gases require a coated, strong, or heavy-wall ring. The strong-wall rings can support heavier flange loads, and consequently, they provide tighter seals.[1]

The amount of cross-sectional squeeze or compression required to assure a proper seal is governed by the pressure and temperature to which the ring will be exposed. The actual amount of cross-section compression required for sealing varies between 20 and 30% of the tube diameter, and the amount of compression force required to achieve this squeeze varies with the O-ring material and wall thickness of the tube. The average natural resiliency and seating force required for tubing of different materials and sizes with varying wall thicknesses are given in Table 4. The cross section of tubing required is determined by the outside diameter of the ring and the space available.[1] The best way of limiting the squeeze is by housing the ring in a recess or groove in one of the flange faces. Another method is to house the O-ring in a metal back-up ring, as illustrated in Fig. 6.

ORNL Dwg 66-10115

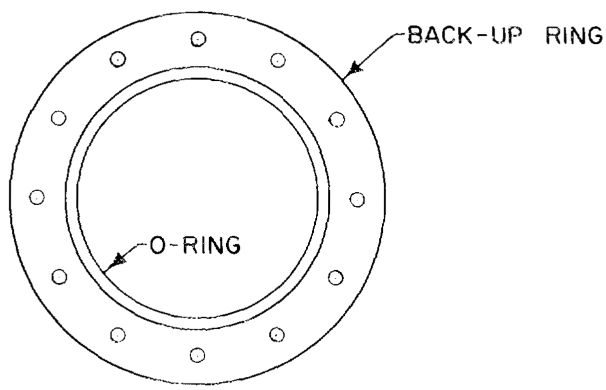


Fig. 6. Hollow Metallic O-Ring With Back-Up Ring Used to Limit the Amount of Squeeze Experienced By the O-Ring. The Back-Up Ring Provides Mechanical Support, and When the O-Ring is Compressed, It Flows Outward Against the Inside Diameter of the Back-Up Ring.

Flange and groove surfaces that contact the O-ring must have a surface finish of 32 microinches rms or better. Tool or grinding marks must be concentric with the ring. Care should be exercised during assembly to assure that the finished surfaces and the O-ring are not marred or scratched.

Table 4. Average Natural Resiliency and Seating Force of Tubing[a]

Basic Tubing OD (in.)	Wall Thickness (in.)	Tubing Material	Elastic Spring Back (in.)	Compression Force of O-ring (lb per linear in.)
1/32	0.005	Stainless Steel	0.002	300
	0.010	Stainless Steel	0.002	400
	0.012	Stainless Steel	0.002	800
	0.005	Inconel	0.0015	200
	0.005	Inconel X	0.001	300
1/16	0.005	Stainless Steel	0.003	200
	0.010	Stainless Steel	0.002	500
	0.012	Stainless Steel	0.002	800
	0.014	Stainless Steel	0.0025	1300
	0.016	Stainless Steel	0.003	1500
	0.006	Inconel	0.002	300
	0.010	Inconel	0.002	420
	0.012	Inconel	0.002	550
	0.014	Inconel	0.002	1100
	0.010	Inconel X	0.002	550
	0.012	Inconel X	0.002	700
	0.010	Mild Steel	0.002	400
	0.014	Mild Steel	0.002	850
	0.010	Aluminum	0.0015	200
	0.014	Aluminum	0.0015	350
	0.010	Monel	0.002	450
	0.010	Copper	0.001	150
	0.012	Copper	0.001	250
	0.014	Copper	0.001	350
3/32	0.007	Stainless Steel	0.002	200
	0.010	Stainless Steel	0.002	350
	0.012	Stainless Steel	0.002	425
	0.018	Stainless Steel	0.0035	1100
	0.007	Inconel	0.0025	150
	0.010	Inconel	0.0025	250
	0.012	Inconel	0.002	350
	0.018	Inconel	0.0025	1000
	0.010	Inconel X	0.0025	300
	0.010	Mild Steel	0.002	200
	0.012	Mild Steel	0.002	250
	0.018	Mild Steel	0.002	950
	0.010	Aluminum	0.002	200
	0.012	Aluminum	0.002	175
	0.018	Aluminum	0.002	350
	0.010	Monel	0.002	200
	0.012	Copper Annealed	0.001	200
	0.018	Copper Hard	0.004	500
	0.012	Tantalum Annealed	0.002	650

Table 4 continued

1/8	0.010	Stainless Steel	0.003	210
	0.012	Stainless Steel	0.003	320
	0.020	Stainless Steel	0.004	1000
	0.010	Inconel	0.004	250
	0.012	Inconel	0.002	300
	0.020	Inconel	0.0035	1000
	0.025	Inconel	0.004	1400
	0.020	Inconel X	0.004	800
	0.025	Inconel X	0.004	1600
	0.010	Mild Steel	0.002	250
	0.020	Mild Steel	0.002	700
	0.010	Aluminum	0.002	75
	0.012	Aluminum	0.002	100
	0.020	Aluminum	0.002	220
	0.025	Aluminum	0.002	280
	0.010	Monel	0.003	250
	0.018	Copper	0.002	500
	0.030	Copper	0.002	800
5/32	0.010	Stainless Steel	0.004	150
	0.012	Stainless Steel	0.004	160
	0.016	Stainless Steel	0.003	300
	0.025	Stainless Steel	0.003	1000
	0.010	Inconel	0.003	150
	0.025	Inconel X	0.002	950
3/16	0.010	Stainless Steel	0.005	150
	0.012	Stainless Steel	0.004	175
	0.020	Stainless Steel	0.004	450
	0.032	Stainless Steel	0.005	2300
	0.020	Inconel	0.004	600
1/4	0.010	Stainless Steel	0.006	75
	0.012	Stainless Steel	0.006	90
	0.020	Stainless Steel	0.006	350
	0.035	Stainless Steel	0.006	1100
	0.049	Stainless Steel	0.007	2500
	0.035	Aluminum	0.003	250
5/16	0.050	Stainless Steel	0.005	2000
3/8	0.035	Stainless Steel	0.005	500
	0.049	Stainless Steel	0.005	1750
1/2	0.080	Stainless Steel	0.008	3300
	0.120	Stainless Steel	0.007	7600

[a]Data taken from Machine Design, Seals Reference Issue, 36(6), June 1964.

3. DESIGN FOR BOLTS OR STUDS TO RETAIN CASK LID

There are four loads that must be considered when developing a design for bolting the cask lid. They are (1) the load due to internal pressure, F_p, (2) the load due to the apparent weight of the lid and the contents of the cask under impact conditions, F_w, (3) the force required to seat the gasket, F_{SG}, and (4) the force on the gasket required to maintain a tight joint under service conditions, F_{oc}.

The load on a cylindrical cask lid due to the internal pressure of the cask,

$$F_p = p\left(\frac{\pi d_g^2}{4}\right) , \tag{1a}$$

where

p = the design pressure, psi,

d_g = the mean diameter of the gasket, in.

For rectangular cask lids, the load due to internal pressure,

$$F_p = pLB , \tag{1b}$$

where

L = the inside length of a rectangular cask (measured center to center of gasket), in.,

B = the inside width of a rectangular cask (measured center to center of gasket), in.

The load due to the apparent weight of the cask lid and the contents of the cask,

$$F_w = N_g(W_L + W_c) , \tag{2}$$

where

N_g = the number of "g's" to which the lid and contents of the cask are subjected upon impact,

W_L = the weight of the cask lid, lb,

W_c = the weight of the contents of the cask, lb.

For a cylindrical lid, the force required to seat the gasket[1] or make the gasket material flow and deform to seal itself into the irregularities of the flange faces,

$$F_{SG} = b\pi Gy \ , \tag{3a}$$

where

b = the effective gasket seating width, in. (see Table 2),

G = the diameter at the location of the gasket load reaction, in.,

y = the gasket or joint constant for surface unit seating load, psi (see Table 1).

For rectangular lids, the force required to seat the gasket,[1]

$$F_{SG} = bCy \ , \tag{3b}$$

where

C = the length of the gasket, in.,

$= 2(L + B)$.

For cylindrical cask lids, the force required on the gasket to maintain a tight joint under service conditions,[1]

$$F_{oc} \quad 2\pi bGmp \ , \tag{4a}$$

where

m = the gasket factor (see Table 1).

For rectangular lids, the force on the gasket required to maintain a tight joint under service conditions,[1]

$$F_{oc} = 2bCmp \ . \tag{4b}$$

After the gasket has been selected and the loads given by Eqs. 1, 2, 3, and 4 have been determined, the minimum bolt area may be determined. The minimum bolt area,

$$A_m = \frac{F_{SG}}{S_a} \ , \tag{5}$$

where S_a = the allowable bolt stress at atmospheric temperature, psi, or

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[1] "ASME Boiler and Pressure Vessel Code Section VIII, Unfired Pressure Vessels," American Society of Mechanical Engineers, New York, 1965.

$$A_m = \frac{(F_p + F_w + F_{oc})}{S_b} , \tag{6}$$

where S_b = the allowable stress at operating temperature, psi, whichever is larger.

After the minimum bolt area required has been determined, the actual bolting pattern may be established. A simple procedure for cylindrical lids is as follows.

1. To obtain an approximation of the number of bolts required, allow one bolt for each inch of inside diameter of the lid flange. If the resulting number is not already a multiple of four, use the next larger number that is a multiple of four.

2. Divide the minimum bolt area, A_m, by the number of bolts required; this gives the required area per bolt.

3. Refer to Table 5 and select the bolt size. Because of the danger of overstressing smaller sized bolts, a 1/2-in.-diameter bolt is considered the minimum size to be used.

4. Check to see that the resulting bolt spacing is close enough to maintain adequate unit pressure on the gasket between the bolt holes. Application of the following equation gives excellent results.[2]

$$\text{Maximum bolt spacing} = \frac{2a + 6t}{m + 0.5} , \tag{7}$$

where

a = the diameter of the bolts, in.,

t = the thickness of the lid flange, in.,

m = the gasket factor.

5. Make any adjustments that may be suggested by practical considerations subject only to the requirements for the minimum total bolt area and spacing.

A similar technique may be used to establish the bolting design for rectangular cask lids.

Care must be exercised to avoid overloading the gasket, particularly in designing large-diameter flanges or even relatively small ones for

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[2] Taylor Forge and Pipe Works Catalog 571, Products for Piping and Pressure Vessel Construction, 2nd ed., 1961.

Table 5. Design Data for U. S. Standard Bolts[a]

Diameter (in.)	Number of Threads per Inch	Diameter at Root of Thread (in.)	Diameter of Tap Drill (in.)	Stress Area $(in.)^2$
1/2	13	0.400	27/64	0.1416
9/16	12	0.454	15/32	0.1816
5/8	11	0.507	17/32	0.2256
3/4	10	0.620	41/64	0.3340
7/8	9	0.731	3/4	0.4612
1	8	0.838	55/64	0.6051
1 1/8	7	0.939	31/32	0.7627
1 1/4	7	1.064	1 3/32	0.9684
1 3/8	6	1.158	1 7/32	1.1538
1 1/2	6	1.283	1 11/32	1.4041
1 3/4	5	1.490	1 17/32	1.8983
2	4.5	1.711	1 49/64	2.4971
2 1/4	4.5	1.961	2 1/64	3.2464
2 1/2	4	2.175	2 15/64	3.9976

[a]Data taken from E. Oberg and F. D. Jones, Machinery's Handbook, The Industrial Press, New York, 16th ed., 1959, and M. F. Spotts, p. 194 in Design of Machine Elements, Prentice Hall, 3rd ed., 1962.

high-pressure service. Overloading can result in crushing of the gasket or yielding of the flange or both. To avoid this, the unit load on the gasket should not exceed twice the gasket sealing stress, y, given in Table 1, nor should this unit stress exceed the yield point of the flange material.

A torque wrench should be used in bolting the cask lid. An approximate relationship between the torque applied to the bolt or nut and the force induced in the bolt or stud for unlubricated conditions is given by Eq. 8.[3]

$$T = 0.2aF \ , \tag{8}$$

where

T = torque, in.-lb,

a = the diameter of the bolt, in.,

F = the induced force, lb.

Therefore, the torque required to seal the gasket for a lid,

$$T_{SG} = \frac{0.2aF}{N} \ SG \ , \tag{9}$$

where

N = the number of bolts required.

The torque for operating conditions,

$$T_{oc} = \frac{0.2a(F_p + F_w + F_{oc})}{N} \ . \tag{10}$$

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[3] M. F. Spotts, p. 204 in Design of Machine Elements, Prentice Hall, 3rd ed., 1962.

4. FLANGE DESIGN

The flanges that must be designed by performing strength analyses are the lid flange and the flange on the cask to which the lid flange is bolted.

Lid Flange

The following assumptions were made to develop a simplified but conservative procedure for calculating the thickness of the lid flange.

1. The flange is subject to pressure loads, F_p, gasket loads, F_{oc}, and impact loads caused by the weight of the shield plug attached to the lid flange and the weight of the contents of the cask, F_w.

2. The impact loads due to the weight of the shield plug and the contents of the cask are distributed uniformly over the flange.

3. The lid flange is fixed at the bolt line so that no rotation will occur.

A schematic of the forces on the lid flange and a load diagram are shown in Fig. 7.

ORNL Dwg 66-10116

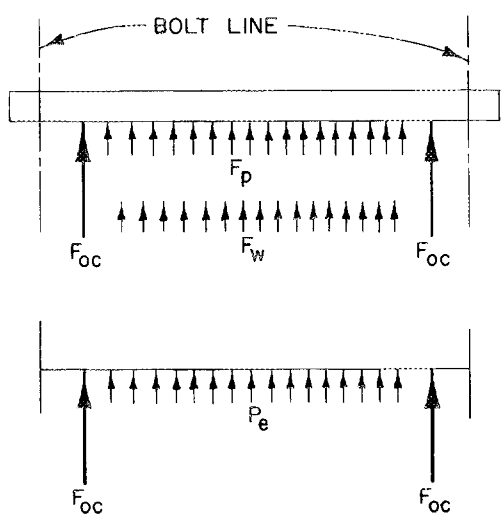

Fig. 7. Schematic and Load Diagrams of Forces Sustained by the Lid Flange.

In Fig. 7, the pressure loads,

$$F_p = pA \ , \tag{11}$$

where

p = the internal design pressure, psi,

A = the area of the lid flange exposed to this pressure, in.^{2}

The impact loads,

$$F_w = N_g(W_L + W_c) \ , \tag{2}$$

where N_g, W_L, and W_c are as previously defined. F_{oc} is the total gasket force under operating conditions, and P_e is the equivalent pressure load resulting from the actual internal pressure and the impact load.

Separate procedures were developed for designing round flanges and rectangular flanges.

Design Procedure for Round Flanges

The method of superposition was used to determine the total stress in the flange, and the maximum normal stress theory[1] was used as the criterion for establishing the required thickness of the flange. Schematic diagrams of the two separate loads are shown in Fig. 8.

[1] R. J. Roark, p. 218 in Formulas for Stress and Strain, McGraw-Hill Book Co., 4th ed., 1965.

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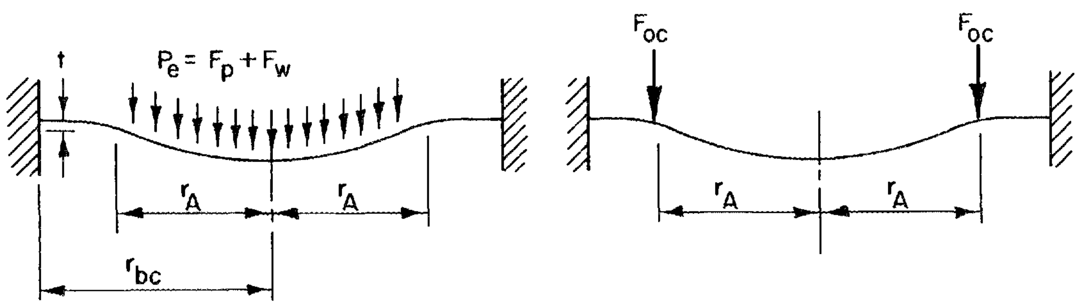

Fig. 8. Schematic Diagrams of Equivalent Pressure Load and Total Gasket Force on Round Flange.

In Fig. 8, r_A represents the radius of the area of the flange subjected to pressure, and r_{bc} represents the radius of the bolt circle. Assume that $r_A > 0.588r_{bc}$; then the maximum stress,[1]

$$S = \frac{3(F_p + F_w)}{2\pi t^2}\left(1 - \frac{r_A^2}{2r_{bc}^2}\right) + \frac{3F_{oc}}{2\pi t^2}\left(1 - \frac{r_A^2}{r_{bc}^2}\right) . \tag{12}$$

Solving Eq. 12 for t,

$$t = \sqrt{\frac{3}{2\pi S}\left[F_p + F_w + F_{oc} - \frac{r_A^2}{r_{bc}^2}\left(\frac{F_p + F_w}{2} + F_{oc}\right)\right]} . \tag{13}$$

Design Procedure for Rectangular Flanges

In order to determine the maximum normal stress in a rectangular flange, the maximum bending moment must be determined. Schematic load, shear, and bending moment diagrams for rectangular flanges are illustrated in Fig. 9. For rectangular flanges, r_A represents half the flange width that is subjected to the pressure load, and r_{bc} represents the half width of the shortest dimension of the bolt line. The long dimension of the flange subjected to pressure is represented as L. The gasket force per unit length of gasket $= \frac{F_{oc}}{2(L + 2r_A)}$ (lb/in.) and the equivalent pressure, P_e, resulting from the actual internal pressure and the impact load $= \frac{F_w + F_p}{2r_A L}$ psi.

In a rectangular flange the maximum bending moment will be assumed to occur in a strip of unit width whose length is $2r_{bc}$. For such a strip (beam) the loads are as shown in Fig. 9.

The maximum bending moment per unit length may be at the ends and therefore equal to M_o, or it may be at the center and equal to M_c.

$$M_c = R_L r_{bc} - M_o - \frac{F_{oc} r_A}{2(L + 2r_A)} - \frac{F_w + F_p}{2L}\frac{r_A}{2} , \tag{14}$$

where from the shear diagram in Fig. 9,

$$R_L = \frac{F_{oc}}{2(L + 2r_A)} + \frac{F_w + F_p}{2L} . \tag{15}$$

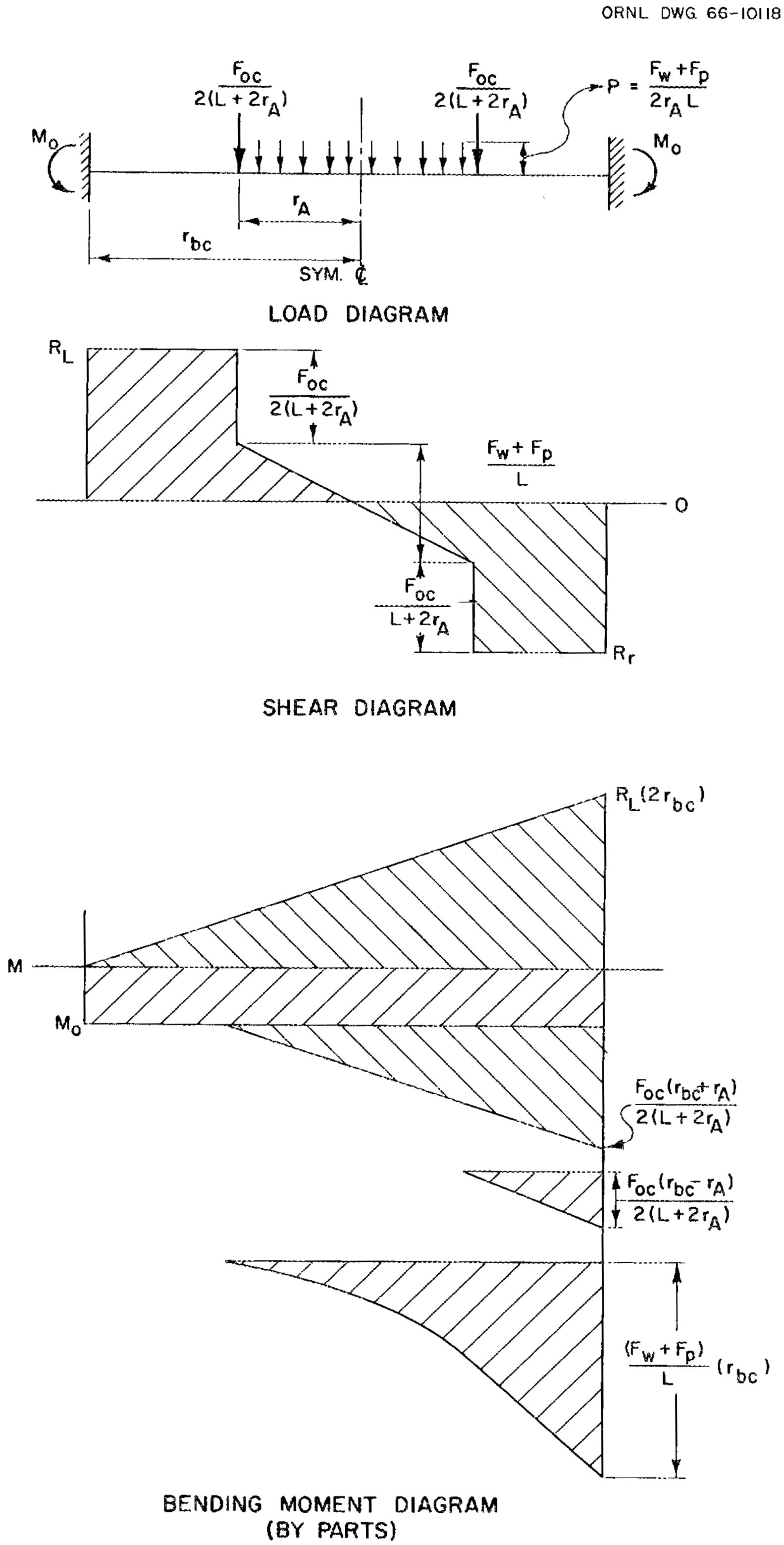

Fig. 9. Load, Shear, and Bending Moment Diagrams for Rectangular Lid Flanges.

From Roark,[2] the bending moment at the ends,

$$M_o = \frac{F_{oc}(r_{bc}^2 - r_A^2)}{4r_{bc}(L + 2r_A)} + \frac{(F_w + F_p)(3r_{bc}^2 - r_A^2)}{12Lr_{bc}} \quad . \tag{16}$$

Once the maximum moment has been determined, the thickness of the flange required to support this moment without exceeding the allowable stress of the flange material may be calculated. Since

$$S = \frac{M_c C}{I} = \frac{M \frac{t}{2}}{\frac{t^3}{12}} = \frac{6M}{t^2} \ ,$$

where C = the distance to the outermost fiber in in.; the thickness of the flange,

$$t = (6M/S)^{1/2} \ , \tag{17}$$

where

t = the thickness of the flange, in.,

M = the maximum moment from Eq. 14 or 16,

S = the allowable stress of the flange material, 2/3 yield or 1/3 ultimate, whichever is less.

Cask Flange

The design for the flange on the cask to which the lid flange is bolted is not quite as simple as that for the lid flange. Determination of the thickness of the cask flange on the basis of strength alone may not be practical from the standpoint of fabrication, bolting, and deflection. As a lower limit, the minimum thickness of the cask flange should be equal to or greater than 1.5 times the nominal diameter of the bolts or studs. This will allow ample room to attach the studs in a manner that will develop their full strength.

As in the design for lid flanges, separate procedures were developed for designing round cask flanges and rectangular cask flanges.

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[2] R. J. Roark, Cases 32 and 34, p. 112 in Formulas for Stress and Strain, McGraw-Hill Book Co., 4th ed., 1965.

Design Procedure for Round Flanges

The assumed load diagram for the round cask flange is shown in Fig. 10. In this diagram, the total load is represented as F_t.

$$F_t = F_{oc} + F_p + F_w , \tag{18}$$

where F_p and F_w are as defined on page 19 and F_{oc} is as defined on page 20.

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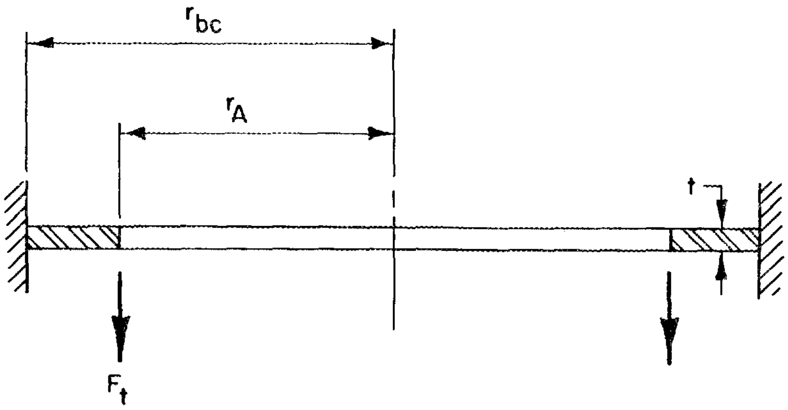

Fig. 10. Load Diagram for a Round Cask Flange.

From Roark,[3] the maximum stress in the cask flange when $r_A > 0.417\ r_{bc}$,

$$S_{max} = \frac{3F_t}{2\pi t^2}\left(1 - \frac{2\mu r_A^2 - 2r_A^2(\mu + 1)\ \ln \dfrac{r_{bc}}{r_A}}{r_{bc}^2(\mu - 1) + r_A^2(\mu + 1)}\right) , \tag{19}$$

where $\mu = 1/\upsilon$ and υ = Poisson's ratio. When $r_A < 0.417$, the maximum stress in the cask flange,

$$S_{max} = \frac{3F_t}{2\pi\mu t^2}\left(1 + \frac{\mu r_{bc}^2(\mu - 1) - \mu r_A^2(\mu + 1) - 2(\mu^2 - 1)r_{bc}^2\ \ln \dfrac{r_{bc}}{r_A}}{r_{bc}^2(\mu - 1) + r_A^2(\mu + 1)}\right). \tag{20}$$

---

[3]R. J. Roark, Case 18, p. 221 in Formulas for Stress and Strain, McGraw-Hill Book Co., 4th ed., 1965.

Solving Eq. 19 to determine the thickness of the cask flange when $r_A > 0.417\ r_{bc}$,

$$t = \left[\frac{3F_t}{2\pi S}\left(1 - \frac{2\mu r_A^2 - 2r_A^2(\mu + 1)\ \ln \frac{r_{bc}}{r_A}}{r_{bc}^2(\mu - 1) + r_A^2(\mu + 1)}\right)\right]^{1/2} . \tag{21}$$

Solving Eq. 20 to determine the thickness of the cask flange when $r_A < 0.417\ r_{bc}$,

$$t = \left[\frac{3F_t}{2\pi\mu S}\left(1 + \frac{\mu r_{bc}^2(\mu - 1) - \mu r_A^2(\mu + 1) - 2(\mu^2 - 1)r_{bc}^2\ \ln \frac{r_{bc}}{r_A}}{r_{bc}^2(\mu - 1) + r_A^2(\mu + 1)}\right)\right]^{1/2} . \tag{22}$$

Design Procedure for Rectangular Flanges

The rectangular cask flange acts as a cantilever beam of length g loaded at its unsupported end by F_t. The assumed load diagram for a rectangular cask flange is shown in Fig. 11.

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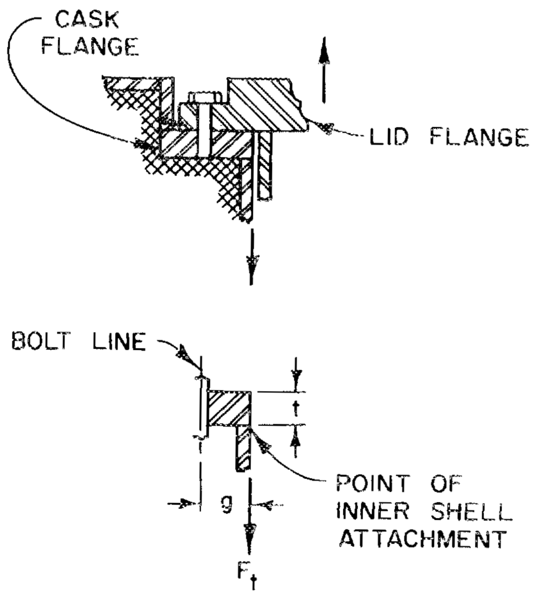

Fig. 11. Load Diagram for a Rectangular Cask Flange.

The maximum stress in the cask flange,

$$S_{max} = \frac{F_t g(t/2)(12)}{2(L + B)t^3} = \frac{3F_t h}{(L + B)t^2} , \qquad (23)$$

where

L = the inside length of a rectangular cask, in.,

B = the inside width of a rectangular cask, in.

Solving Eq. 23 to determine the thickness of the cask flange,

$$t = \left(\frac{3F_t g}{(L + B)S}\right)^{1/2} . \qquad (24)$$

5. SUMMARY

In designing a top closure or lid for a shipping cask that will meet the proposed AEC regulation 10 CFR 71, there are three main areas of the closure that require careful attention. These are (1) the gasket or seal, (2) the retaining devices or bolts, and (3) the flanges on the lid and the cask.

Gasket types and materials most applicable to use as lid seals are discussed in Chapter 2, and design data for the different types of gaskets are given in Tables 1 and 2.

Equations for determining the bolting arrangement for cask lids were developed in Chapter 3. The minimum bolt area required,

$$A_m = \frac{F_p + F_w + F_{oc}}{S_b} \;, \tag{6}$$

where

F_p = the load due to internal pressure,

$$= p\left(\frac{\pi d_g^2}{4}\right) \text{ for cylindrical casks} \tag{1a}$$

= pLB for rectangular casks,

F_w = the load due to the apparent weight of the cask lid and the contents of the cask

$$= N_g(W_L + W_c) \;, \tag{2}$$

F_{oc} = the force required on the gasket to maintain a tight joint under service conditions,

$$= 2\pi bGmp \text{ for cylindrical casks} \tag{4a}$$

$$= 2bCmp \text{ for rectangular casks,} \tag{4b}$$

S_b = the allowable stress at operating temperature.

The maximum bolt spacing $= \dfrac{2a + 6t}{m + 0.5}$, and the bolt torque for operating conditions,

$$T_{oc} = \frac{0.2a(F_p + F_w + F_{oc})}{N} \;. \tag{10}$$

Equations were developed in Chapter 4 for determining the thicknesses of the lid flange and the cask flange. The thickness of the lid flange,

$$t = \sqrt{\frac{3}{2\pi S}\left[F_p + F_w + F_{oc} - \frac{r_A^2}{r_{bc}^2}\left(\frac{F_p + F_w}{2} + F_{oc}\right)\right]} \tag{13}$$

for round flanges, and

$$t = \left(\frac{6M}{S}\right)^{1/2} \tag{17}$$

for rectangular flanges. The thickness of the cask flange,

$$t = \left[\frac{3F_t}{2\pi\mu S}\left(1 + \frac{\mu r_{bc}^2(\mu - 1) - \mu r_A^2(\mu + 1) - 2(\mu^2 - 1) r_{bc}^2 \ln \frac{r_{bc}}{r_A}}{r_{bc}^2(\mu - 1) + r_A^2({}^2 + 1)}\right)\right]^{1/2} \tag{22}$$

for round flanges, and

$$t = \left(\frac{3F_t g}{(L + B) S}\right)^{1/2} \tag{24}$$

for rectangular flanges.

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