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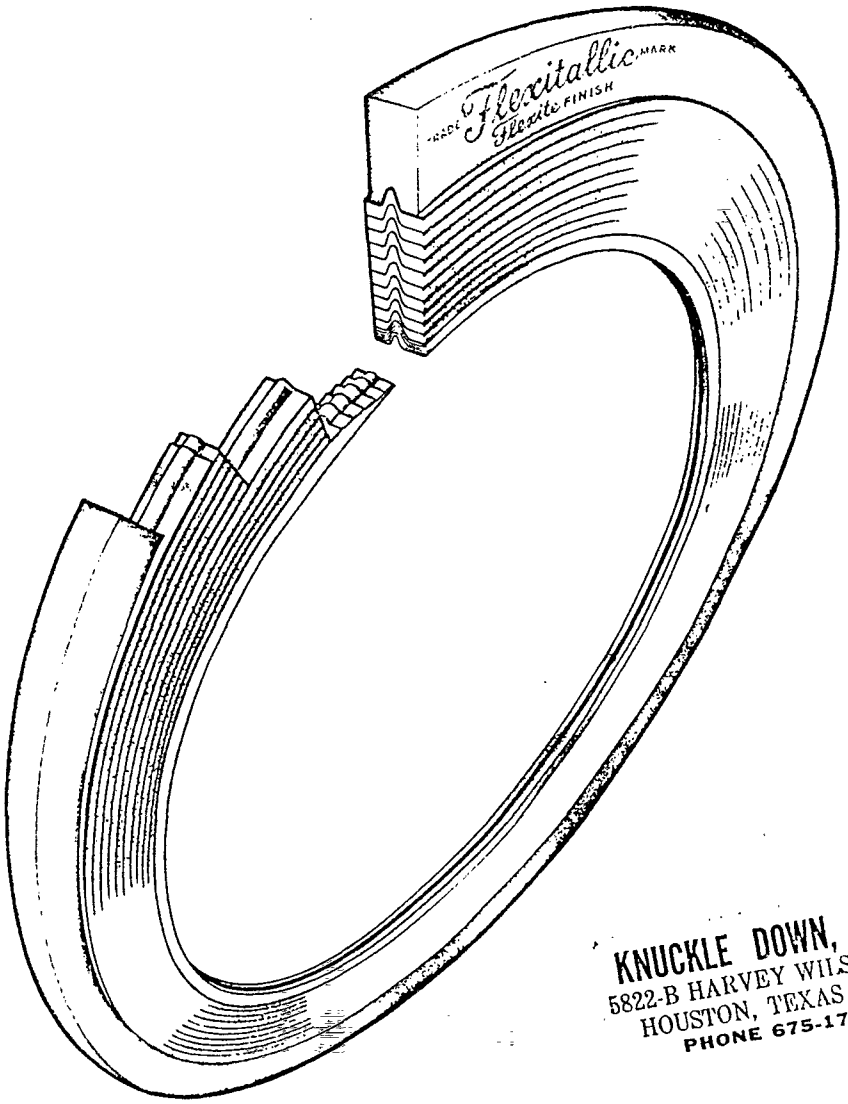
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GASKETS AND RACING SUPPLIES
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# DESIGN CRITERIA




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FLEXITALLIC GASKET COMPANY INC.
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STATION LANE, HECKMONDWIKE
YORKSHIRE, ENGLAND

AFFILIATED COMPANIES IN AUSTRALIA, HOLLAND AND SOUTH AFRICA

MILBURN
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Flexitallic®

# SPIRAL WOUND GASKETS
# DESIGN CRITERIA

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## INTRODUCTION

This paper has been prepared as the result of many requests received from design engineers, users and others for more detailed information on the design and use of FLEXITALLIC spiral-wound gaskets.

The concept of spiral-wound gasket construction was originated in 1912 by the FLEXITALLIC GASKET COMPANY. With the introduction of FLEXITALLIC gaskets, a new era in safe, effective sealing of joints was inaugurated. The versatility of this design concept is now attested by the wide range of successful applications, ranging from extremely high vacuums to pressures in excess of present flange standards of 2500 psi and from cryogenic temperatures to beyond 1500°F and against virtually every known corrosive media. The FLEXITALLIC design has not only kept pace with modern trends, but has been constantly in advance of such progress that combines materials and construction techniques which result in effective sealing that could not be achieved by any other known type of gasket.

Since 1912, the FLEXITALLIC GASKET COMPANY has had a record of steady growth and today is the largest organization in the world that exists solely to produce a spiral-wound gasket. Our entire efforts are directed toward producing for our customers, the highest quality and the most reliable gasket at competitive prices.

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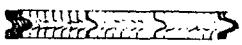
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Gaskets are essentially a packing designed for inclusion between rigid parts of a fluid container in an essentially stationary relationship. A FLEXITALLIC gasket consists of laminated, preformed metal strips and filler materials selected to meet the conditions of service required. Its construction provides internal resilience enabling it to follow flange movement within reasonable limits. Unfortunately, gaskets are all too often simply accepted per se, that is, "just another gasket." The irony of this is that one of the most costly and serious industrial problems today is leaky joints resulting in explosions, fires, loss of production and soaring maintenance costs. Much of this is due to insufficient attention to the proper application and design of closures and the gaskets that are used to effect the seal.

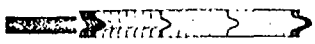
The FLEXITALLIC gasket is not "just another gasket." It is a specifically engineered product in the FLEXITALLIC program of continuous research and development in gasket design and production control to give you assurance of quality and safety.

It is our objective in the preparation of this bulletin to help prevent, through greater knowledge of the FLEXITALLIC gasket, the serious and costly waste of time, money and resources resulting from joint leakage. We trust the information contained herein will prove worthwhile and will benefit the equipment manufacturer, design engineer, and the user in sealing closures, whether high or low pressure, high or low temperature, and regardless of the nature of the corrosive media to be sealed.

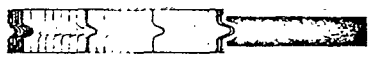
A FLEXITALLIC gasket is manufactured by spirally winding a preformed metal strip and a filler on the outer periphery of metal winding mandrels. The winding mandrel outside diameter forms the inner diameter of the gasket and the laminations are continually wound until the required outer diameter is attained. Normal practice is to reinforce the inner and outer diameters with several plies of metal with no soft fillers being introduced. Our method of manufacture includes custom designed devices which provide control of gasket density that permit compression to the operating thickness under a specified load. This engineered product is thus "tailor made" to be compatible with the flanged closure in which it is to be used. For example, a closure designed for vacuum service may require a gasket of exactly the same dimensions as a closure designed for 1500 psi service. The closure designed for the vacuum service would have relatively light bolting indicating the necessity for a soft gasket, while the 1500 psi application would have heavy bolting indicating a relatively dense gasket. It is usually within our capability to satisfy both requirements. Refer to paragraph on "Flange Design for Flexitallic Gaskets."



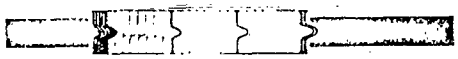
1. Style R --- This designates a round spiral-wound gasket with no accessory devices added. The notations, R1, R3, and R4 in our catalog, apply to our standard gaskets only, so indicated for use on standard flanges. If a Style R gasket is to be used on a special flange design (other than ANSI or BS flange specification), the gasket is usually termed a Special Style R.



2. Style RIR --- This is a round spiral-wound gasket fitted with an inner metal ring used to provide inner confinement to the gasket, to act as a compression stop if of the proper thickness, and to be used to fill the annular space between the flange bore and the gasket I.D. to minimize turbulence of process fluids and erosion of flange faces.



3. Style CG --- A Style CG gasket is a round, spiral-wound gasket with a solid metal outer ring forming a complete assembly. The outer ring serves as a compression stop, anti-blowout device, and to properly center the gasket on the flange.

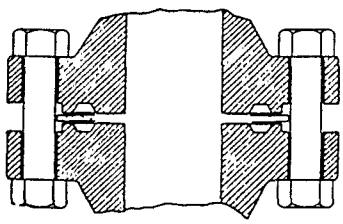


4. Style CGI Gasket --- This designates a CG gasket as above with the addition of an inner metal ring. The inner ring thickness is normally the same as the outer ring and serves to prevent material build-up between the flange bore and the gasket I.D., as a protection against excessive heat, to reduce process fluid turbulence and to minimize erosion of flange facings. Style CGI gaskets are frequently

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used on vacuum service. This style eliminates costly machining in one of the flange faces if a totally contained gasket is required. A Style CGI gasket will effectively provide a totally contained gasket without the additional costly machining.

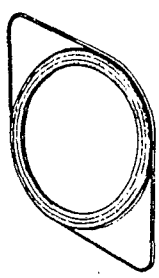
5. Style CG-RJ --- This style designates a Special CG gasket sized to be used on standard ring joint flanges as noted in our catalog. The outer ring is dimensioned to cover the ring joint grooves and to prevent the spiral-wound portion from entering the groove. The spiral-wound portion of the gasket is sized to fit between the flange bore and the ring joint groove. This type of gasket should be used only as a maintenance repair item and should not be used on new construction. If FLEXITALLIC gaskets are to be used in new construction, it is advisable to use standard raised face flanges and our Style CG gasket.



6. Style D --- This consists of a spiral-wound gasket with loops (usually 2) added to the O.D. of the gasket and dimensioned to fit over two diametrically opposite flange bolts for centering purposes. Style D gaskets are not generally recommended for use in applications where pressures are in excess of 600 psi. It is not as widely used or as popular as the Style CG gasket, with its outer metal ring functioning as a centering, anti-blowout device as well as a compression stop. The Style D gasket does lend itself to ease of assembly in areas of congested piping plus the fact that the standard Style D's are less expensive than the CG for the equivalent size and pressure rating.

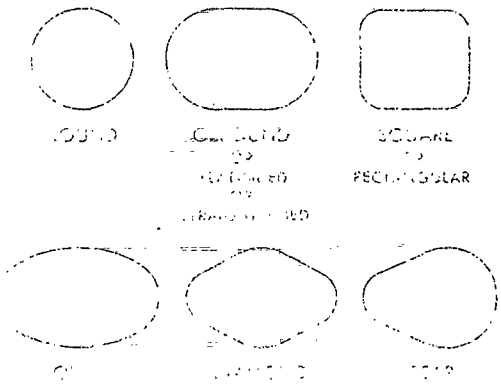
When Style D gaskets are required and are not standard catalog items, special winding mandrels must be purchased.

It should be noted, the spiral-wound portion of the Style D gasket is identical to the spiral-wound portion of a Style CG gasket in the same size and pressure series.



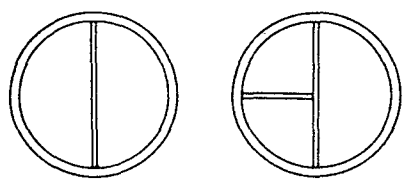
7. Style T --- This designates gaskets for boiler handhole and tube cap assemblies. They are available in round, oval, obround, square, pear and diamond shapes. Refer to our general catalog for standard Style T gaskets.

Please note Style T gaskets rely on internal pressure in the boiler to properly seat the gasket. This means, when a hydrostatic test is performed on the gasket, the pressure exerted against the plate will further compress the gasket -- and it is necessary to tighten each nut to compensate for the additional compression of the gasket under load.



8. Style M & MC --- These styles are designed for boiler manhole cover assemblies. They are usually of round, obround or oval shape, depending, of course, upon the manhole plate configuration. Style MC gaskets have a preformed spiral-wound centering ring snapped into the inner groove of the gasket proper. This centering guide permits the gasket to assume its correct position and to compensate for inequalities in plate contours and fillets in cold-pressed plates as well as to prevent shouldering and pinching caused by radial misplacement.

9. Style HE --- Style HE gaskets are for heat exchangers where pass ribs are required. The outer portion is of standard spiral-wound construction, whereas the rib portion is normally of single or double-jacketed style, securely fastened to the I.D. of the spiral-wound portion.



10. Style HE-CG --- This style is identical to the Style HE above, except that it is fitted with an outer ring on the O.D.

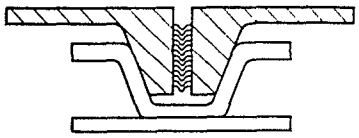
Note --- Style HE and Style HE-CG gaskets have a primary seal of spiral-wound construction with its inherent resiliency and excellent sealing quality. It is necessary that dimensional drawings locating the pass ribs and the configurations be submitted for all inquiries and orders for these style gaskets.

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11. Style HX*--- Style HX gaskets are suitable for standard heat exchanger flanges designed in accordance with BS 3274 and TEMA standards. The primary seal is a spiral wound gasket construction which is fitted with a stainless steel outer wound guide so as to correctly locate the gasket in the flange recess. An inner compression stop ring is fitted and for the tubeplate to channel connection, a compressed asbestos fibre insert with pass partition bars is incorporated. Where working conditions demand it, the compressed asbestos fibre insert can be dispensed with and the pass partition bars can be supplied in metal asbestos or solid metal as part of the inner compression stop ring.

*Note --- Not available in U. S. A.

12. Style 625 --- Similiar to Style R gaskets, except



original thickness is .0625". Limited to small diameters and narrow flange widths. Most frequently used on clamp-type closures where an extremely thin gasket is required.

13. Miscellaneous --- On rare occasions, applications for FLEXITALLIC gaskets arise where it is necessary to utilize a spiral-wound inner and/or outer ring. The spiral-wound inner or outer rings are used primarily as centering devices and are chosen in lieu of a solid metal ring, because of lower cost or because of limited space when the use of a solid metal ring would be prohibited because of the difficulty in fabrication.



Refer to our General Catalog for FLEXITALLIC gaskets available for standard flanges and common boiler handhole and manhole fittings.

The selection of materials of construction for FLEXITALLIC gaskets requires consideration of the following aspects:

1. The corrosive nature and concentration of the fluid to be confined.
2. The operating temperature.
3. The expected life of the installation.
4. The relative cost of alternative materials.

Specific recommendations for materials of construction are beyond the scope of this paper. The resistance to corrosive attack by the various materials used in FLEXITALLIC gaskets fluctuates widely, depending upon the concentration of the corrosive agent, presence of other contaminates and the operating variables of temperature and pressure. Lacking specific experience with the corrosive nature of any particular agent and those materials that would have sufficient corrosion resistance to the media, designers are recommended to contact the manufacturers of alloyed materials, who have available extensive information on the chemical resistivity of their products to various corrosive media. Another excellent source of corrosion resistance is contained in "Corrosion Data Survey" published by the Association of Corrosion Engineers, Houston, Texas. It is frequently necessary to conduct laboratory corrosive tests and/or pilot plant operations in order to obtain factual and reliable information.

When considering the choice of materials for FLEXITALLIC gaskets, designers should be guided by the following general comments:

1. Stress Corrosion -- FLEXITALLIC gaskets, when installed, are highly stressed, particularly in the area of the engineered wire formation, and adjacent to the flange seating surfaces. The 18-8 stainless steels are particularly subject to stress corrosion or stress corrosion cracking when exposed to certain media. Corrosive materials that induce stress corrosion cracking in metals are included in Appendix A. In such cases, alternative materials must be selected that are less susceptible to stress corrosion cracking.

2. Integranular Corrosion -- When austenitic stainless steels are subjected to temperatures in the range of 800° to 1500°F, carbides are precipitated along the grain boundaries. When exposed to certain chemicals, intergranular corrosion will occur. A list of corrosives which induce intergranular corrosion are included in Appendix B. When handling these media, special attention to material selection is necessary.

3. Expected life of the Installation -- The trend in industry today is to lengthen the time between overhauls and consequently, the best possible material, regardless of initial cost, is frequently the most economical. Gaskets are relatively low cost items when compared to labor costs to install, downtime on equipment and loss of productivity, should premature failure occur. Hence, for "permanent" installations, the cost of special gasketing materials should be equated against increased productivity and reliability. The FLEXITALLIC GASKET COMPANY maintains a substantial inventory of a wide variety of metals and filler materials to meet specific operating conditions and requirements.

The following materials are normally inventoried to insure prompt delivery:

Type 304 Stainless Steel -- This 18-8 Chromium-Nickel Steel is the most common metal used in the fabrication of FLEXITALLIC gaskets. It has excellent corrosion resistance to a wide variety of chemicals. It is, however, subject to stress corrosion cracking when exposed to certain media, and to intergranular corrosion at temperatures between 800°F and 1500°F in the presence of certain media. Under these conditions, alternative materials should be selected. Due to the precipitation of carbides along grain boundaries, its use is normally limited to a maximum continuous temperature of 800°F if any danger of the materials that would cause intergranular corrosion are expected to be present.

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Type 347 Stainless Steel -- The addition of columbium to this 18-10 Chromium-Nickel Steel serves to keep carbides in solution, and hence, inhibits carbide precipitation along the grain boundaries. Type 347 Stainless Steel has corrosion resistant qualities comparable to Type 304 Stainless Steel. It is not as subject to intergranular corrosion as is Type 304 Stainless Steel, and can be used at continuous temperatures up to the 1400--1500°F range. Type 347 is subject to stress corrosion cracking, however, as is Type 304.

Type 316L Stainless Steel -- The 2% molybdenum added to this 18-12 Chromium-Nickel alloy increases its creep strength at elevated temperatures. Carbon content is held at a maximum of .03% which inhibits the tendency toward carbide precipitation. Type 316L is subject to stress corrosion cracking and also to intergranular corrosion, but to a lesser degree than Type 304 Stainless Steel. Its continuous maximum temperature exposure would be in the range of 1400--1500°F.

Type 309SCb Stainless Steel -- This 25-12 Chromium-Nickel steel has excellent corrosion resistance due to the high Chromium-Nickel content. It is not as subject to stress corrosion cracking as are the 18-8 stainless steels. The presence of Columbium inhibits carbide precipitation. Maximum continuous temperatures would be in the range of 1600°F--1700°F.

Type 304L Stainless Steel -- This 18-8 stainless steel alloy has the same excellent corrosion resistance as does Type 304, but its carbon content is maintained at a maximum of .03%, which tends to reduce the precipitation of carbides along grain boundaries, and as a consequence, would be less subject to intergranular corrosion than is the Type 304 stainless steel. It is, however, subject to stress corrosion cracking.

Plated Low Carbon Steel -- In FLEXITALLIC gasket construction, the use of this metal is normally limited to low pressure steam applications at maximum temperatures in the range of 500°F.

Monel Metal -- This Nickel base alloy contains 67% Nickel and 30% Copper. It is widely used as a gasketing material due to its excellent resistance to most acids and alkalis, except the strong oxidizing acids. In combination with PTFE, it is widely used in FLEXITALLIC gaskets for hydrofluoric acid service. Maximum upper temperature limit for Monel metal is in the range of 1500°F. Monel metal is subject to stress corrosion cracking when exposed to fluorosilic acid, mercuric-chloride and mercury, and should not be used with these media.

Inconel 600 -- This metal is a Nickel base alloy containing 77% Nickel, 15% chromium and 7% iron. It has excellent high temperature strength and can be used at temperatures up to the 2000°F range. Inconel 600 has little tendency toward stress corrosion cracking and is frequently used as a gasketing material to overcome this problem.

| MATERIAL | MAX/TEMPERATURE* |
|---|---|
| 321 S. S. | 1400/1500°F |
| 430 S. S. | 1400/1500°F |
| Carpenter 20 | 1400/1500°F |
| Phosphor Bronze | 500°F |
| Nickel | 1400°F |
| Titanium | 2000°F |
| Hastelloy C & B | 2000°F |
| Inconel X | 2000°F |
| Copper | 500°F |
| 310 S. S. | 1900°F |

On special order, gaskets can be fabricated from gold, zirconium, platinum, aluminum and tantalum.

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

1. Canadian Asbestos Paper (Chrysotile) -- Canadian Asbestos Paper is the most common filler material used in the fabrication of FLEXITALLIC gaskets. Canadian Asbestos is a hydrated magnesium-silicate. The form, as we use it, is composed of approximately 90% Canadian Chrysotile Asbestos, 7% vegetable rubber latex binder and 3% water-proofing binder. This material has poor acid resistance and strong mineral acids dissolve out all magnesia, leaving a residue of nearly pure insoluble silica. It does, however, show excellent resistance against alkaline solutions, such as sodium-hydroxide or caustic soda and can be used against solvents, aqueous and salt solutions, and gases (except oxygen). Canadian Asbestos has a very high melting point fusing at approximately 2770°F. However, it loses 11% by weight of water at 1100°F and about 13.5% at 1400°F. From this point to the fusion temperature, it has turned to a powder with very little tensile strength left. In spite of this, if it is still confined, it can continue to produce a satisfactory seal since it is completely confined between the metal windings of the gasket and the seating surfaces of the flange.

Our standard Canadian asbestos paper contains .1% blue vegetable dye for identification purposes. This material is available without the dye added if the possibility of color contamination is a problem.

Note: All asbestos paper used by FLEXITALLIC GASKET COMPANY is tested for leachable chlorides and meets the 200 parts per million requirements for total soluble chlorides.

2. Blue African Asbestos (Crocidolite) -- Blue African Asbestos is a complex silicate of iron and sodium. Blue African Asbestos has excellent acid and alkaline resistance. It was originally developed particularly for use against strong mineral acids such as sulphuric, nitric and hydrochloric. For this reason, it is frequently used as a filler material for hot strong mineral acids in FLEXITALLIC gasket construction.

Blue African asbestos has a fusion point of approximately 2180°F. However, because of the very small loss in weight due to water loss, it may perform excellently as a seal up to 1900 to 2000°F.

Note: 1 - Blue African Asbestos is not available in U. K.
2 - When Blue African Asbestos is required, always specify "Blue African" to eliminate any confusion with our "blue-dyed" Canadian Asbestos.

3. Canadian Asbestos paper with an Inorganic Binder - This is a Canadian Asbestos paper that contains an inorganic binder for applications where Canadian Asbestos is satisfactory but where any organic materials cannot



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be tolerated.

4. Compressed Asbestos Fibre Type Special -- This is a grade of compressed asbestos fibre specially developed for use in spiral wound gaskets. The homogeneous structure of the material allows it to more readily conform to the spiral wound steel profile than conventional compressed asbestos fibre materials. The material contains an asbestos content of 70% and a binder content of 12%, this being a blend of NR and SBR. The material shows good performance in steam and hydrocarbon service at high temperatures.

5. Polytetrafluoroethylene (PTFE) -- PTFE is used as a filler material in FLEXITALLIC gaskets where extreme chemical inertness is required for temperatures ranging from cryogenic to 500°F. PTFE is unaffected by any known chemicals except molten alkali metals and fluorine precursors. Because of its low permeability, PTFE is also frequently used as a filler material on FLEXITALLIC gaskets in vacuum applications.

6. FLEXICARB -- This material is a pure pyrolitic graphite tape and does not melt, rather it sublimates at 6600°F. This material shows promise as being as excellent filler material for extremely high temperature in reducing atmospheres.

7. Other Materials -- FLEXITALLIC gaskets are also available with various compressed asbestos sheet packings utilizing Buna S., Neoprene, or Buna N as a binder or a PTFE Impregnation for special applications.

Note: Refer to Appendix C for properties of Canadian and Blue African Asbestos.

**TABLE I**

| Gasket Thickness | Max. I.D. | Max. Flange Width | Recommended Compressed Thkns.* |
|---|---|---|---|
| .0625 | Up to 6" | 3/8" | .050/.055 |
| .0625 | 6 to 9" | 1/4" | .050/.055 |
| .100 | 10" | 3/4" | .075/.080 |
| .125 | Up to 20" | 1" | .090/.100 |
| .125 | 20" to 40" | 3/4" | .090/.100 |
| .125 | Over 40" Convert to .175" thickness | | |
| .175 | Up to 12" | 1-1/2" | .125/.135 |
| .175 | 12" to 20" | 1-3/8" | .125/.135 |
| .175 | 20" to 40" | 1-1/8" | .125/.135 |
| .175** | 40" to 60" | 1" | .125/.135 |
| .175** | 60" to 70" | 7/8" | .125/.135 |
| .175** | 70" to 75" | 3/4" | .125/.135 |
| .175 | Over 75" Convert to .250 or .285" thickness | | |
| .250 | Up to 90" | 1-1/4" | .180/.200 |
| .250 | 90" to 150" | 1" | .180/.200 |
| .285 | Up to 90" | 1-1/4" | .200/.220 |
| .285 | 90" to 150" | 1" | .200/.220 |

Notes: *The recommended compressed thickness is what experience has indicated to be the optimum range in order to achieve maximum resiliency of the gasket. An additional spread of .010 in either direction can be tolerated on all gasket thicknesses with the exception of the .0625 and the .100" thick gasket. This is on the assumption that the flange surface finishes are relatively smooth. Refer to "Flange Surface Finish" on page 9.

**PTFE filled FLEXITALLIC gaskets in this size range are unstable and are subject to "springing apart" in shipping and handling. A .285" thick gasket should be specified in order to minimize this problem.

FLEXITALLIC gaskets are available in sizes from 1/2" I.D. to approximately 150" I.D. depending upon gasket thickness. Table I indicates size ranges available in various thicknesses, maximum flange widths and the recommended compressed thickness.

FLEXITALLIC gaskets are subject to standard manufacturing tolerances listed in Table II. If tighter tolerances are required, consult our Engineering Department.

**TABLE II**

| Gasket Diameter | I.D. | O.D. |
|---|---|---|
| Up to 24" | ± 1/64 | ± 1/32 |
| 24" to 36" | ± 1/32 | ± 1/16 |
| 36" to 60" | ± 3/64 | ± 1/16 |
| 60" & Above | ± 1/16 | ± 1/16 |

Tolerance on gasket thickness is ± .005" on all thicknesses except as follows where standard tolerance becomes plus .010", minus .005".

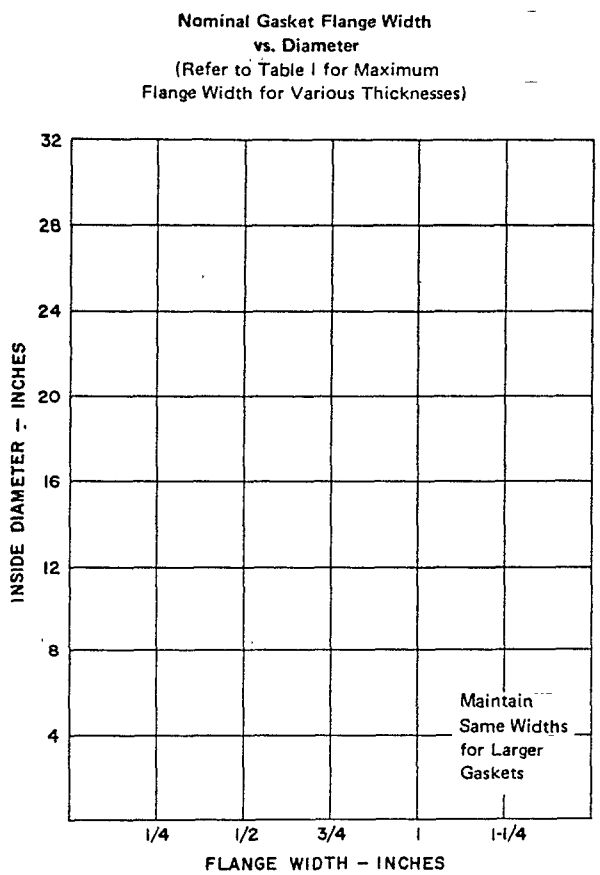
Gaskets with less than 1" I.D.
Gaskets using PTFE fillers
Gaskets using compressed asbestos sheet filler.
Gaskets with flange widths greater than 1".

Regardless of the type of flange facing in use, FLEXITALLIC gaskets must be sized to insure the spiral-wound element is seated against a flat surface. This is of utmost importance. If the spiral-wound element protrudes into the flange bore or extends beyond a raised face, mechanical damage will occur to the gasket during initial compression, and ultimate failure will result. In addition, should the gasket protrude into the flange bore, the windings can possibly enter the process stream with severe damage to other equipment resulting. With recessed flange facings, limiting dimensions of the gasket are established by dimensions of the groove. On flat or raised face flanges, considerable leeway is available. The following rules will be generally applicable for limiting dimensions of spiral-wound components. (See Chart I for nominal flange widths.)

1. Gasket confined on both I.D. & O.D. -- This is the type facing encountered in tongue and groove joints, and groove to smooth joints. Standard practice is to allow a 1/16" diametrical clearance between the I.D. of the groove and the I.D. of the gasket and 1/16" diametrical clearance between the O.D. of the gasket and O.D. of the groove.

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2. Gasket confined on O.D. only -- This is the type of facing encountered with male and female facings and female to flat facings. Standard practice is to allow a 1/16" diametrical clearance between the O.D. of the gasket and the O.D. of the groove. If possible, allow a minimum 1/4" diametrical clearance between the I.D. of the seating surface and the I.D. of the gasket.




3. Gasket unconfined on both the I.D. & O.D. -- Allow a minimum 1/4" diametrical clearance between the gasket I.D. and the I.D. of the seating surface. The O.D. should be kept as close as possible to the bolt circle to minimize flange bending moments. If the gasket is used with raised face flanges, allow a minimum 1/4" diametrical clearance between the gasket O.D. and the raised face O.D. and determine the I.D. on the basis of the desired flange width. Important -- Please note the above rules establish general limits for sizing FLEXITALLIC gaskets. It is frequently necessary to adjust dimensions in order to achieve a proper balance between gasket area and bolt area to maintain a reasonable compressive force on the gasket and a minimum gasket factor "m" of three. Please refer to section covering ASME Boiler and Pressure Vessel Code.

4. Metal Gauge Rings -- When FLEXITALLIC gaskets are required to be equipped with outer metal rings or with inner metal rings, limitations on the minimum flange widths of the solid metal ring are necessary due to the availability of machining facilities and rigidity of completed assemblies. Table III indicates the minimum flange width for solid metal rings based on the ring I.D.

**TABLE III**

| Diameter of Ring | Minimum Flange Width* |
|---|---|
| UP to 10" I.D. | 3/8" |
| 10" to 24" I.D. | 7/16" |
| 24" to 50" I.D. | 1/2" |
| 50" to 70" I.D. | 5/8" |
| 70" and Larger | 3/4" |

*Note: Where space is limited and narrower flange widths are necessary, it may be possible to supply inner and outer spacer rings of solid metal spiral-wound construction. Consult our Engineering Department for advice.

Standard practice is to size outer rings with the outside diameter equal to diameter of the bolt circle less the diameter of one bolt. Inner rings are normally sized with an inside diameter equal to the flange bore.

5. Non-Circular Gaskets -- FLEXITALLIC gaskets can be fabricated in non-circular shapes within limitations. As a general rule, if the ratio of the long I.D. to the short I.D. exceeds 3 to 1 and should any of these sides approach a straight line, it may not be possible to manufacture a FLEXITALLIC gasket that would be suitable. Our product requires a definite radius or curvature to give it inherent strength and stability and to prevent it from springing apart. Any application requiring a non-circular gasket should be submitted to our Engineering Department for review to determine the feasibility of producing a satisfactory gasket as early as possible in the design stage.

The comments above relating to availability of sizes and recommended clearances for proper sizing of FLEXITALLIC gaskets are general in nature. Many applications will arise where the recommended clearances are impractical due to space limitation on the flange. Frequently, clearances between gasket sealing member and grooves must be reduced in order to effectively maintain a seal under operating conditions, particularly when the higher pressures are encountered. Under such circumstances, FLEXITALLIC GASKET COMPANY engineers should be consulted prior to finalizing designs.

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## "FLANGE SURFACE FINISH"

FLEXITALLIC gaskets rely on the combined reaction of the metal and filler material to effect a seal. When the gasket is compressed, the filler material flows into minute imperfections and the metal serves to completely trap the filler material and provide the necessary strength and resilience to the gasket assembly. Obviously, the deeper the serrations, or the rougher the flange surface finish, the higher the force required to flow the gasket into the imperfections. Although FLEXITALLIC gaskets will seal against virtually any commercial flange surface finish, abnormally high bolt loads may be required to obtain a seal against rough flange finishes. For this reason, flanges to be used with FLEXITALLIC gaskets should be specified with surface finishes specified in Table IV.

**TABLE IV**

| Service | Flange Surface Finish in Micro Inches |
|---|---|
| General Purpose | 125 to 200 RMS |
| Hazardous Service and Hard to Hold Fluids | 125 RMS |
| Vacuum Service | 80 RMS |

Important — Under no circumstances, should flange sealing surfaces be machined in a manner that tool marks would extend radially across the seating surface. Such tool marks are practically impossible to seal, regardless of the type of gasketing material used. A phonographic or concentric serrated finish is preferred for FLEXITALLIC gasket applications.

## "ASME BOILER AND PRESSURE VESSEL CODE REQUIREMENTS"

Section VIII, of the ASME Boiler & Pressure Vessel Code, establishes criteria for flange design and suggests values of "m" (gasket factor) and "y" (minimum gasket seating stress) as applied to spiral-wound gaskets. For the most part, the defined values have proven successful in actual applications. However, much confusion exists regarding these values, primarily due to a misunderstanding of the definitions of the terms and their significance in practical applications.

Mandatory Appendix II. in Section VIII of the Boiler Code, requires in the design of a bolted flange connection, complete calculations shall be made for two separate and independent sets of conditions.

1. Operating Conditions

Condition one (1) requires a minimum load be determined in accordance with the following equation:

$$(1)\quad W_{m1} = \frac{3.14G^2P}{4} + 2b\ 3.14GmP$$

This equation states the minimum required bolt load for operating conditions is the sum of the hydrostatic end force plus a residual gasket load on the contact area of the gasket times a factor times internal pressure. Stated another way, this equation requires the minimum bolt load be such that it will maintain a residual unit compressive load on the gasket area that is greater than internal pressure when the total load is reduced by the hydrostatic end force.

It should be noted that Table UA-49.1 suggests a gasket factor "m" for a spiral-wound gasket of 2.5 for carbon steel and 3.0 for stainless or monel. It is important to note the gasket factor "m" is suggested and is not mandatory. (Note: See paragraph on gasket factor "m" on page 10 for a more detailed discussion.)

2. Gasket Seating

Condition two (2) requires a minimum bolt load be determined to seat the gasket regardless of internal pressure and utilizes a formula:

$$(2)\quad W_{m2} = 3.14bGy$$

The "b" in this formula is defined as the effective gasket width and "y" is defined as the minimum seating stress in psi that depends upon the type of gasket material. Table UA-49.1 suggests a minimum "y" value for a spiral-wound gasket of 2900 psi for carbon steel and 4500 psi for stainless or monel. These design values are suggested and are not mandatory.

The term "b" is defined as:

$$b = b_o \text{ when } b_o \leq 1/4"$$

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

In the case of a spiral-wound gasket, $b_o = \frac{N}{2}$ in all cases where N is the radial flange width of the spiral-wound portion of the gasket.

Our experience has been that the suggested values for "y" using the effective seating width as defined in the Code is too low. The minimum design seating stress for a FLEXITALLIC gasket is subject to variables.

(Note: See paragraph on Minimum Seating Stress — "y" on page 10.)

After $W_{m1}$ and $W_{m2}$ are determined, the minimum required bolt area, $A_m$ is determined as follows:

$A_{m1} = \frac{W_{m1}}{S_b}$ where $S_b$ is the allowable bolt stress at operating temperature, and $A_{m2} = \frac{W_{m2}}{S_a}$ where $S_a$ is the allowable bolt stress at atmospheric temperature.

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

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At this point, it is important to realize the gasket must be capable of carrying the entire compressive force applied by the bolts when prestressed unless provisions are made to utilize a compression stop in the flange design or by the use of a compression gauge ring. For this reason, FLEXITALLIC'S standard practice is to assume W is equal to $A_b S_a$.*

If we equate $W_{m2} = W$ on the assumption we can develop in the gasket sufficient density to carry the entire bolt load and solve this equation for $D_g$ we have,

(3) $D_g = \frac{A_b S_a}{3.14\, bG}$. Where $D_g$ is the actual load carrying capacity or density of the gasket.

This is the gasket density used by FLEXITALLIC in assigning construction details for fabrication in our shops. Following this procedure, once $A_b$ has been determined, we are no longer concerned with which is greater, $W_{m1}$ or $W_{m2}$. We are concerned with actual values.

If the values determined for $D_g$ are within the density range indicated in Table V, a suitable gasket can be supplied. If the values determined are greater than those listed, either a wider gasket is required or a compression stop must be utilized.

As indicated previously, our experience has indicated the minimum "y" value as defined in Table UA-49.1, Section VIII, coupled with the effective gasket seating width "b" as defined in Table UA-49.2 is unrealistic. The "y" value should be in the 9000 psi range rather than the 4500 psi range suggested in the Code. (It is possible to reduce the minimum "y" value for a FLEXITALLIC gasket to the Code figure if an extremely narrow gasket is furnished in relationship to its I.D. However, to achieve such a low figure would result in a radial width of the gasket we believe to be inadequate for long term satisfactory service and would be below what we would recommend as a minimum gasket width.) If gasket proportions as indicated on Chart I are followed, Table V can be used as a guide to "y" (minimum seating stress) and $D_g$ (actual density.)

The minimum seating stress is a function of flange surface finish as well as gasket density. The rougher the surface finish, the higher the required forces to flow the gasket into the imperfections. Dg and "y" are normally considered to be the unit load required to compress the gasket to its optimum operating thickness.

TABLE V

GASKET SEATING STRESSES

| | "y" (minimum seating stress) psi | $D_g$ (actual density range) psi |
|---|---|---|
| Canadian or Blue African asbestos filled | 9000 | 9000 to 30000 |
| TFE filled | 10000 | 10000 to 13000 |
| Flexicarb filled | 10000 | 10000 to 20000 |

Appendix II, Section VIII, of the Boiler Code under paragraph VA-49 makes the statement "the 'm' factor is a function of the gasket material and construction." We do not agree entirely with this interpretation of "m". Actually, the gasket does not create any forces and can only react to external forces. We believe a more realistic interpretation of "m" would be "the residual compressive force exerted against the gasket contact area must be greater than internal pressure when the compressive force has been relieved by the hydrostatic end force." It is the ratio of residual gasket contact pressure to internal pressure and must be greater than unity otherwise leakage would occur. It follows then, the use of a higher value for "m" would result in a closure design with a greater factor of safety. Experience has indicated a value of 3 for "m" is satisfactory for flanged designs utilizing FLEXITALLIC gaskets regardless of the materials of construction.

In order to maintain a satisfactory ratio of gasket contact pressure to internal pressure, two points must be considered. First, the flanges must be sufficiently rigid to prevent unloading the gasket due to flange rotation when internal pressure is introduced. Secondly, the bolts must be adequately prestressed.

The Boiler Code recognizes the importance of prestressing bolts sufficiently to withstand hydrostatic test pressure. Appendix S, in the Code, discusses this problem in detail.

NOTATIONS

- $A_b$ = Actual total cross-sectional root area of bolts or section of least diameter under stress; square inches.
- $A_m$ = Total required cross-sectional area of bolts, taken as greater of $A_{m1}$, or $A_{m2}$; square inches.
- $A_{m1}$ = Total required cross-sectional area of bolts required for operating conditions; square inches.
- $A_{m2}$ = Total required cross-sectional area of bolts required for gasket seating; square inches.
- b = Effective seating width; inches.
- 2b = Joint-contact-surface pressure width; inches.
- $b_o$ = Basic gasket seating width; inches.
- G = Diameter of location of gasket load reaction; inches.
- m = Gasket factor.
- N = Radial flange width of spiral-wound component; inches.
- P = Design pressure; psi.
- $S_a$ = Allowable bolt stress at atmospheric temperature; psi.
- $S_b$ = Allowable bolt stress at design temperature; psi.
- W = Flange design bolt load; pounds.
- $W_{m1}$ = Minimum required bolt load for operating conditions; pounds.
- $W_{m2}$ = Minimum required bolt load for gasket seating; pounds.
- y = Minimum gasket seating stress; psi.
- $D_g$ = Actual gasket density, psi.

(*Note: If a prestress greater than $S_a$ is to be used (as is normal practice in order to successfully pass a hydrostatic test) the actual prestress value will be used in lieu of $S_a$ in equation (3).

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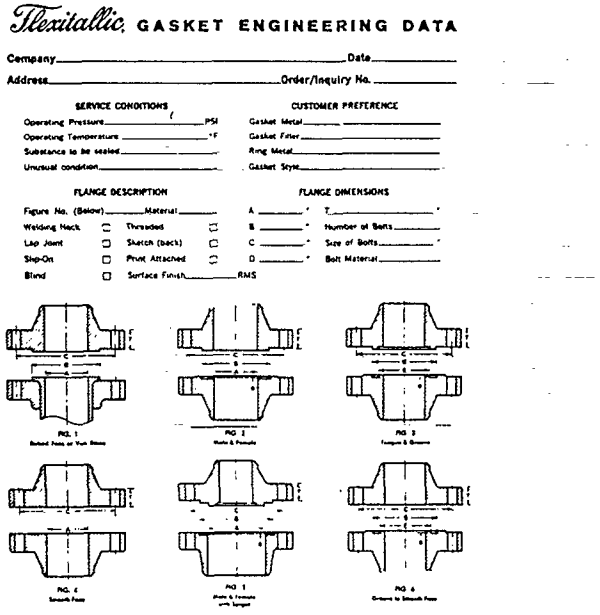
1. Determine the required bolt load for operating conditions $W_{m1}$, (Equation (1).) (Approximate gasket size by referring to Chart I and Table I, and use an "m" value equal to three.)
2. Determine the required bolt load to effect a seal ($W_{m2}$) by solving equation (2). Use a minimum seating stress "y" per Table V.
3. Determine the minimum bolt area by dividing $W_{m1}$ by the allowable bolt stress at operating temperature and by dividing $W_{m2}$ by the allowable bolt stress at atmospheric temperature. The greater of these two values becomes $A_m$, the minimum bolt area. Select the number and size of bolts that will give an actual total cross sectional area equal to or greater than $A_m$.
4. If $W_{m1}$ is greater than $W_{m2}$, equate $W_{m1} = 3.14bG\ D_g$ and solve for "$D_g$". If "$D_g$" falls within the ranges shown on Table V, a FLEXITALLIC gasket can be designed to carry the load. If the resulting value of $D_g$ is greater than the maximum indicated in Table V, it is necessary to provide a compression stop either by utilizing a ring on the inside or the outside of the gasket or by designing into the flange, a compression stop that would bring the flanges metal to metal.
5. If $W_{m2}$ is substantially greater than $W_{m1}$ resulting in a greater bolt load than required for operating conditions, it is possible to reduce $W_{m2}$ by reducing the flange width of the gasket. If this is the case, we suggest you contact FLEXITALLIC engineers for recommendations as to minimum flange widths for the diameter and pressure/temperature conditions.
6. Proceed strictly to the flange design procedures in Appendix II of the Boiler Code.
7. After all flange dimensions have been determined, determine flange rotation at the gasket interface at test conditions. If the flange rotation is excessive, extraordinary precautions must be taken to insure a satisfactory hydrostatic test. (Refer to section covering "Bolting Up Procedures.")

In order for the FLEXITALLIC GASKET COMPANY to design a gasket suitable for the application, it is imperative that complete details be submitted for review. The information we require is the following:

1. Type of flange facing
2. Dimensions of the gasket seating surfaces
3. Number, size and material of bolts
4. Bolt circle diameter
5. Operating pressure & temperature (Process material if known)
6. Hydrostatic Test pressure
7. Initial bolt pre-stress
8. Customer preference on gasket materials

FLEXITALLIC supplies engineering data sheets at no cost on which this information may be submitted.

As a gasket manufacturer, it is impossible for us to review every flange design to make certain that flange rotation and flange stresses are within allowable limits defined in the Code. We proceed on the assumption the design engineer has followed the design criteria established by the ASME Boiler Code and that the flanges are sufficiently rigid under the most severe condition to preclude the possibility the gasket could become unloaded either during operating conditions or hydrostatic test conditions. We are aware that most flange designers do not take into consideration, flange rotation at test conditions. This is a very important aspect that is not normally considered, and we urge every flange designer to check out flange rotation at test conditions prior to finalizing his design. We also, of a practical necessity, must assume the bolt material being used is adequate for all conditions including operating pressure at operating temperature and hydrostatic test pressure at ambient temperature. The use of the optimum material for bolts is a very complex subject and we suggest ASME paper, number 52-BET-7 entitled "Modern Steel Bolting for Piping & Pressure Vessels", authored by C. M. Vogrin, Frank S. G. Williams and John S. Worth, be consulted for guidance in the proper selection of bolting material for piping and pressure vessel applications.


Flexitallic GASKET ENGINEERING DATA

Company _____ Date _____

Address _____ Order/Inquiry No. _____

SERVICE CONDITIONS

Operating Pressure _____ PSI
Operating Temperature _____ °F
Substance to be sealed _____
Unusual condition _____

CUSTOMER PREFERENCE

Gasket Metal _____
Gasket Filler _____
Ring Metal _____
Gasket Style _____

FLANGE DESCRIPTION

Figure No. (Below) _____ Material _____
Welding Neck ☐ Threaded ☐
Lap Joint ☐ Sketch (back) ☐
Slip-On ☐ Print Attached ☐
Blind ☐ Surface Finish _____ RMS

FLANGE DIMENSIONS

A _____ " T _____ "
B _____ " Number of Bolts _____
C _____ " Size of Bolts _____ "
D _____ " Bolt Material _____

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## BOLTING-UP PROCEDURES

FLEXITALLIC GASKET COMPANY supplies thousands of different gaskets for special flange designs that must successfully pass a hydrostatic test and maintain a satisfactory seal under operating conditions. Our experience has indicated that virtually every time a leaky joint is encountered, the actual cause of failure relates to something other than the gasket design. These causes are itemized below:

1. On low pressure applications, flange designers have followed the Code suggestions for a minimum seating stress (y value) that we know from experience are impractical and hence, neither the bolting nor the rigidity of the flanges, are adequate to initially seat the gasket to obtain a seal.
2. Flange designers do not take into consideration, the rotation of the flanges and the necessity for the flanges and the bolting to maintain a sufficiently high residual unit load on the gasket contact surface to contain internal pressure.

These two conditions contribute to a vast majority of complaints on joint leakage.

3. The insistence of some inspection personnel that hydrostatic test conditions must be carried out at stress values for initial pretensioning of bolts at the allowable design stresses specified in the Code. Appendix S, in the Code, specifically covers this area and must be taken into cognizance anytime a hydrostatic test is to be performed. From a practical standpoint, when a flange is designed for pressure conditions of 600 psi and the hydrostatic test pressure is to be performed at 900 psi, it is obvious that a higher prestress must be applied to the bolts is a satisfactory test is to be applied.
4. The use of low yield bolting material, such as the austenitic stainless steels or ordinary carbon steel machine bolts. With both of these materials, it is relatively easy to stress the bolts beyond their yield point with eventual failure occurring using a standard wrench for the nominal bolt diameter. In order to successfully pass a hydrostatic test, it is often desirable and permissible to utilize a high strength alloy bolt for hydrostatic testing purposes. When this procedure is followed, the following steps are recommended:

   a. For hydrostatic testing, use ASTM B 193, Grade B 7 bolting material, or equivalent, to initially seat the gasket and perform the hydrostatic test.
   b. After achieving a successful hydrostatic test, relieve the tension on the bolts to approximately 50% of the allowable bolt stress and replace the bolts one at a time with the required bolting material.
   c. When replacement is made, the bolts should be stressed to the allowable stress for operating conditions.

It should be pointed out a bolt initially prestressed to 20,000 psi will very rapidly relax to approximately 50% of the initial applied stress due to creep and relaxation in the bolt material. Bolting that is stressed to 30,000 psi initially will relax to approximately 75% of its initial prestress over a short period of time. This indicates it is essential to prestress the bolt to a level that will guarantee maintenance of a stress level at operating conditions that will insure a safe joint.

### Thermal Expansion

When bolts are initially prestressed to compensate for relaxation plus the usual hydrostatic end load, plus a residual gasket load, consideration must be given to stresses induced by thermal expansion under operating conditions. This can be caused by differential expansion of the flanges and the bolt material due to different coefficients of expansion or a temperature gradient that is present between the flange and the bolt material. In many installations, stresses developed by thermal expansion can adversely affect both the ability of the gasket to carry the applied forces and/or the ability of the bolt material to remain in an elastic state without being over-stressed. When thermal expansion is a serious problem and excessive bolt stresses or gasket loading can result, it is recommended the gasket be compressed only to a point that will permit further compression of the gasket as the loading due to thermal expansion is applied. In the case of a gasket with a compression stop, it is recommended initial compression be limited to within 8 to 10 thousandths of the final compressed range that is controlled by the thickness of the compression gauge ring or by the depth of the groove. By doing so, the build up of excessive stresses in the bolts can be eliminated. In the case where a compression stop is not provided, the additional stresses can be absorbed by the gasket providing the stresses resulting from thermal expansion will not crush the gasket beyond its elastic limit.

### Developing Prestresses in Bolting

The only completely satisfactory method for closing the flange and developing uniform stresses in each bolt at a controlled bolt stress is by measuring bolt elongation. This is an impractical approach, since the cost of bolting up using this procedure becomes excessive. An alternate means must be considered. The most frequent method is by utilizing torque wrenches. The use of torque wrenches, however, introduces many variables, and are not normally reliable methods of determining the actual bolt stress developed. Some of the factors that enter into the actual developed bolt stress are:

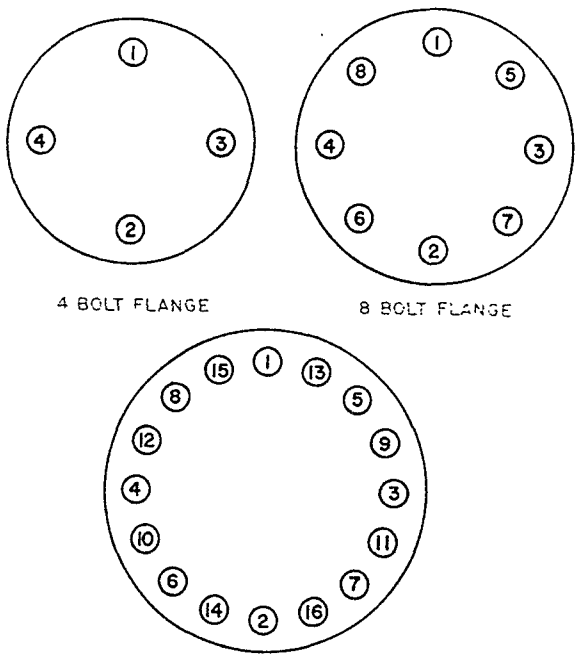
1. The class of fit of the bolt and nut
2. Presence of burrs
3. The degree of lubrication achieved
4. The presence of grit, chips and dirt in the threads in the bolt and nut
5. Nicks
6. The relative condition of the seating surface on the flange against which the nut is rotated

All of these variables have a marked effect on the amount of torque required to produce a given stress. Where

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conditions are fairly constant and reasonably controlled, it is possible to give nuts a certain torque value and obtain stresses that are consistent within reasonable limits. The required torque value can be obtained by tightening a sample bolt to a desired elongation while measuring the torque or by calculating it by means of formula. If this procedure is not followed, the actual developed stress in a stud can be completely unrealistic. Field tests have indicated that the use of torque wrenches in attempting to develop required bolt stresses can vary as much as 100% due to the variables present that have been previously mentioned. It is a requirement, particularly in the use of a FLEXITALLIC gasket, that a reasonably, even compressive force, be applied. From a practical standpoint, in order to achieve this, a very definite bolt up procedure must be followed. This procedure is detailed as follows:

1. Install the gasket on the gasket seating surface and bring the cover flange in contact with the gasket.
2. Install all bolts, making sure they are free of dirt and grit, and are well lubricated.
3. Run up all nuts finger tight.
4. Develop the required bolt stress in a minimum of three steps, following a tightening up procedure as recommended in Sketch I. It is important to make certain that no more than 50% of the required bolt stress is achieved on the initial set. Should this occur, serious damage can be done to the FLEXITALLIC gasket and subsequent tightening cannot offset the damage. Once a complete cycle of stressing has been achieved, it is not necessary to follow the exact bolting sequence recommended. However, we cannot emphasize the importance for following the sequence in the first application of pre-tightening.




**BOLTING UP SEQUENCE**

**SKETCH I**

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In summary, the FLEXITALLIC gasket is a carefully engineered, precisely made, quality product capable of providing industry with the optimum in closure sealing when used judiciously in accordance with good engineering practice. Beyond the horizon of today, FLEXITALLIC will maintain its leadership by providing for industry's needs as we face tomorrow's technological advances.

The information contained in this bulletin is not to be taken as a warranty or representation for which we assume legal responsibility. It is offered solely for your consideration, investigation and verification.

Comments, criticism and discussion on its contents are cordially invited. Please write, Flexitallic Gasket Co., Inc., P.O. Box 680, Camden, New Jersey 08101 U.S.A. or Flexitallic Gaskets (Wood Bros.) Ltd. Station Lane Heckmondwike, Yorkshire, England.

The authors are appreciative of the assistance freely given by their co-workers and others outside the company in preparing this bulletin and the extremely helpful suggestions and criticisms offered.

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# CORROSIVES WHICH CAN INDUCE STRESS CORROSION CRACKING IN METALS

Legend: I - Intergranular Cracks; T - Transgranular Cracks; IT - Intergranular and/or Transgranular Cracks

| | Steel(1) | 18-8 | Brass | Monel | Nickel Inconel | Titanium | Aluminum |
|---|---|---|---|---|---|---|---|
| Ammonium Chloride | IT | | | | | | |
| Amines | | | T | | | | |
| Ammonia (pure) | I | | | | | | |
| Ammonia (Dilute) | | | IT | | | | |
| Ammonium Nitrate | I | I | I | | | | |
| Butane + Sulfur Dioxide | | | T | | | | |
| Cadmium | | | | | | I | |
| Calcium Bromide | | T | | | | | |
| Chloride salts (see Inorganic and Organic Chlorides) | | | | | | | |
| Chromic Acid | T | | | | | | |
| Cresylic Acid (vapors) | I | | | | | | |
| Cyanogen | T | | | | | | |
| Fluosilicic Acid | | (7) | | IT | | | |
| Hydrogen Chloride (some water) | T | T | | | | | |
| Hydrogen Cyanide (some water) | T | | | | | | |
| Hydrogen sulfide (some water) | IT | | | | | | |
| Hydrofluoric Acid | | IT | | IT(3) | I(3) | | |
| $HNO_3$, HCl, HF Pickling Acids | | T | | | | | |
| Inorganic Chlorides (some water present) | | T | | | | | |
| Inorganic Nitrates | I | | | | | | |
| Mercurous Nitrate | | | IT | IT | | | |
| Mercury | | | IT | IT | | | |
| Metal chlorides (see Inorganic chlorides) | | | | | | | |
| Mixed Acids ($H_2SO_4$ + $HNO_3$) | I | | | | | | |
| Nitric Acid + Manganese Chloride | I | | | | | | |
| Nitric Acid (Red Fuming) | | | | | | I | |
| Nitric Acid (vapors) | | | T | | | | |
| Oleum | IT | | | | | | |
| Organic Chlorides (some water present) | | T | | | | I(4) | |
| Potassium Hydroxide | I | T | | I | | | |
| Potassium Permanganate | I | | | | | | |
| Salt Water + Oxygen | | T | | | | I(4) | I(4) |
| Silicofluoride Salts | | | | I | | | |
| Sodium hydroxide | I | IT | IT | I | I | | |
| Steam | | | (5) | (5) | (6) | | |
| Sulfate Liquor (white) | T | IT | | | | | |
| Sulfide Liquor | | IT | | | | | |
| Sulfur Compounds | | | | | I(2) | | |
| Uranyl Sulfate | | | | | | I | |

(1) Acid attack on steels containing martensite can cause cracking by hydrogen absorption in many media.

(2) Heating to temperatures of 475 C and above after exposure to sulfur compounds.

(3) (vapor)

(4) (alloyed)

(5) Stress Cracks Si-Bronze 400 F; Monel, 600 F.

(6) Stress Cracks Inconel, 1500 F.

(7) Stress Cracks Monel.



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## CORROSIVES WHICH INDUCE INTERGRANULAR CORROSION IN AUSTENITIC STAINLESS STEEL

Acetic Acid
Acetic Acid + Salicylic Acid
Ammonium Nitrate
Ammonium Sulfate
Ammonium Sulfate + $H_2SO_4$
Beet Juice
Calcium Nitrate
Chromic Acid
Chromium Chloride
Copper Sulfate
Crude Oil
Fatty Acids
Ferric Chloride
Ferric Sulfate
Formic Acid
Hydrocyanic Acid
Hydrocyanic Acid + Sulfur Dioxide
Hydrofluoric Acid + Ferric Sulfate
Lactic Acid
Lactic Acid + Nitric Acid
Maleic Acid
Nitric Acid
Nitric Acid + Hydrochloric Acid
Nitric Acid + Hydrofluoric Acid
Oxalic Acid
Phenol + Naphthenic Acid
Phosphoric Acid
Phthalic Acid
Salt Spray
Sea Water
Silver Nitrate + Acetic Acid
Sodium Bisulfate
Sodium Hydroxide + Sodium Sulfide
Sodium Hypochlorite
Sulfite Cooking Liquor
Sulfite Solution
Sulfite Digester Acid (Calcium Bisulfite + Sulfur Dioxide)
Sulfamic Acid
Sulfur Dioxide (Wet)
Sulfuric Acid
Sulfuric Acid + Acetic Acid
Sulfuric Acid + Copper Sulfate
Sulfuric Acid + Ferrous Sulfate
Sulfuric Acid + Methanol
Sulfuric Acid + Nitric Acid
Sulfurous Acid
Water + Starch + Sulfur Dioxide
Water + Aluminum Sulfate

## PROPERTIES OF ASBESTOS FIBRES

Chemical Composition of Asbestos Fibers

| % | Canadian Asb. (Chrysotile) | Blue African (Crocidolite) |
|---|---|---|
| $SiO_2$ | 37 – 44 | 49 – 53 |
| MgO | 39 – 44 | 0 – 3 |
| FeO | 0.0 – 6.0 | 13 – 20 |
| $Fe_2O_3$ | 0.1 – 5.0 | 17 – 20 |
| $Al_2O_3$ | 0.2 – 1.5 | |
| $H_2O$ | 12.0 – 15.0 | 2.5 – 4.5 |
| CaO | Tr – 5.0 | |
| $Na_2O$ | | 4.0 – 8.5 |

## EFFECT OF HEAT ON LOSS IN WEIGHT OF ASBESTOS FIBRES

It has been found that with the corresponding loss in weight, proportional reduction in tensile strength of the fibres takes place. When the loss in weight on exposure to heat of "blue" crocidolite asbestos fibres is compared with that of chrysotile it may be seen that at 800°F., crocidolite asbestos loses a great part of its combined water (and becomes brittle), whilst chrysotile loses 1/6th only. Further, crocidolite asbestos completely gives off its water content at 1,200°F., whilst chrysotile asbestos loses the last trace at 1,800°F.

Effect of temperature on percent loss in weight of asbestos fibers.

| Temp. F. | Time | Chrysotile | Crocidolite |
|---|---|---|---|
| 400 | 2 hrs. | 0.30 | 0.08 |
| 600 | 2 hrs. | 0.85 | 0.25 |
| 700 | 2 hrs. | 1.78 | 0.49 |
| 800 | 2 hrs. | 2.17 | 0.73 |
| 900 | 2 hrs. | 2.83 | 0.83 |
| 1,000 | 2 hrs. | 3.99 | 0.86 |
| 1,100 | 2 hrs. | 10.38 | 1.00 |
| 1,200 | 2 hrs. | 12.75 | 1.04 |
| 1,400 | 2 hrs. | 13.43 | 1.03 |
| 1,500 | 2 hrs. | ---- | ---- |
| 1,600 | 2 hrs. | 13.62 | 0.93 |
| 1,700 | 2 hrs. | ---- | ---- |
| 1,800 | 2 hrs. | 13.77 | 0.77* |
| Fusion Point F. | | 2,770 | 2,180 |

*Iron changing in weight caused by oxidation.

## CHEMICAL RESISTANCE

The varieties of asbestos vary considerably in their chemical resistance. Chrysotile asbestos is essentially a basic magnesium silicate, with a magnesia-silica ratio of approximately 1:1. This means that there is an excess of magnesia over the chemically equivalent quantity of silica.

As might be expected from the composition, the acid resistance of chrysotile is poor. Strong mineral acids dissolve out all the magnesia leaving an insoluble residue of nearly pure silica. The silica residue retains the fibrous form of the asbestos but practically all the original strength is lost.

In crocidolite asbestos, the basic constituents and the silica are in almost chemically equivalent proportions, and the acid resistance is good.

M.S. Badollet has published the following tables showing the effect of chemicals on asbestos: ---

Solubility of Asbestos
Percent loss in weight due to refluxing for two hours in 25 percent acid or Caustic Soda

| | HCl | $CH_3COOH$ | $H_3PO_4$ | $H_2SO_4$ | NaOH |
|---|---|---|---|---|---|
| Chrysotile . . . . . | 55.69 | 23.42 | 55.18 | 55.75 | 0.99 |
| Crocidolite . . . . . | 4.38 | 0.91 | 4.37 | 3.69 | 1.35 |

Percent loss in weight due to exposure at 26°C. for 528 hours in 25 percent Acid or Caustic Soda

| | | | | | |
|---|---|---|---|---|---|
| Chrysotile . . . . . | 56.00 | 24.04 | 56.45 | 56.00 | 1.03 |
| Crocidolite . . . . . | 3.14 | 1.02 | 3.91 | 3.48 | 1.20 |

HCL — Hydrochloric Acid. $CH_3COOH$ — Acetic Acid.
$H_3PO_4$ — Phosphoric Acid. $H_2SO_4$ — Sulfuric Acid.
NaOH — Sodium Hydroxide.

DUP 0827129



DU 015178