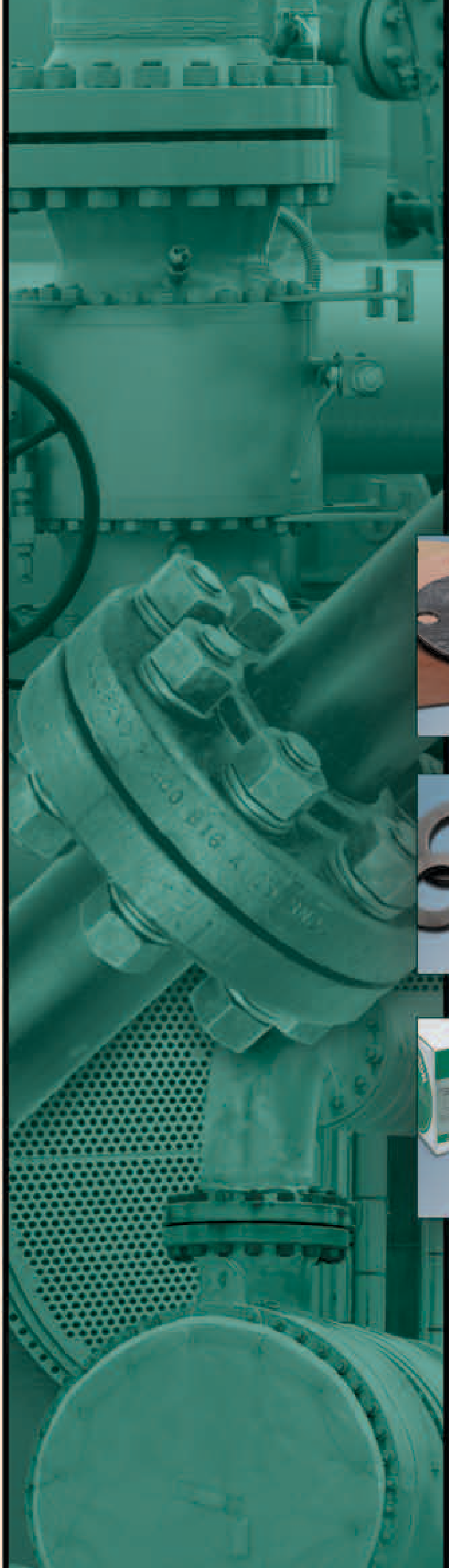

FLANGE SEALING GUIDE

GASKETS AND BOLTED CONNECTIONS

CHESTERTON®

Global Solutions, Local Service.

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SECTION 1

PIPES AND FLANGES

Pipes

In industry, the term "pipe" is simply defined as a tube used to transport fluid/medium between two points. "Piping" is the broader term that refers to the pipes and fittings that make up a system used to transport fluids—a fluid being defined as a gas, liquid, or solid.[1] "Tubing" is another term for pipe, but this is usually used to refer to specific components of boilers, heat exchangers, and other equipment. The US Navy differentiates the two by defining tubing as any tubular product whose specified size is actual measured outside diameter and actual wall thickness; pipe is any tubular product whose size is specified as a nominal size and wall thickness. Piping systems come in a wide variety of materials and designs that depend on, among other things, the fluid being transported, the temperature and pressure of the fluid, safety requirements specific to the system, and cost.

Pipe is manufactured using a number of different methods, each specific to the material and service requirements. In order to maintain consistency and ensure reliability of a piping system, organized standardization groups such as those shown in Table A have established well defined guidelines for piping systems. Note that the ISO is a worldwide organization that is made up of members of many different national standards groups; consolidation of many national standards into a more universal collection is intended to create a global standard rather than many stand-alone national standards. The standards established by these groups cover everything from methods of testing material properties, to specified dimensions for pipes, flanges, and fittings of pressure classes, to maximum temperatures and pressure capability of each pipe size, to support methods for piping systems; the list goes on and on. It is important to be aware of these specifications for they provide a great deal of information useful to engineers and maintenance personnel alike.

TABLE A: COMMON STANDARD GROUPS

Standards Group (**Abbriviation**/Full Name)	**Country**
ASTM/American Society for Testing and Materials	USA
ANSI/American National Standards Institute	USA
ASME/American Society of Mechanical Engineers	USA
API/American Petroleum Institute	USA
DIN/Deutsche Institut für Normung	Germany
ISO/International Standards Organization	Worldwide
JIS/Japanese Industrial Standard	Japan
BS/British Standard	United Kingdom

Piping systems can be extremely complicated, and a plant can have several types of piping systems to handle a variety of applications. A classic example is a shipboard engine room. Fuel oil, lube oil, high and low pressure steam, refrigerant, and seawater are a number of fluids required for normal operation. To accomodate pressure and temperature factors, the main steam system requires high strength, high temperature piping material, and must have valves and fittings designed to handle this application. A seawater cooling system doesn't require the high strength, high temperature piping, but must be resistant to the corrosion often encountered in salt water applications. The requirements of each system vary widely, as does the equipment designed for each system.

Pipes are manufactured in nominal sizes and schedules. The schedule number of a pipe refers to its wall thickness (higher schedule number = thicker pipe), and is calculated based on the system design pressure (maximum) and the allowable stresses of the pipe material. Typically, wall thicknesses increase with the maximum internal pressure the pipe can withstand. It is important to identify a pipe's schedule and to use that same exact schedule in any repairs; otherwise, the piping system's integrity may be compromised. Tables detailing the dimensions of various pipe schedules can be found in various machinists, handbooks or in literature supplied by pipe manufacturers. Pipe is normally supplied in lengths between 3,6 and 6+ m (12 and 20+ feet).

Materials

Given the wide variety of applications in any plant, there are dozens of different piping materials used. The choice of material is made by the design engineer and depends on a number of factors. Temperature, pressure, and the fluid being sealed are the obvious ones. Other considerations include a material's ability to tolerate vibration, thermal expansion, and contraction; expense of the material is not left out of the formula. As previously mentioned, there are usually specifications that dictate recommended materials for use in piping systems under various operating conditions. The following are brief, thumbnail sketches of the general uses of the more common piping materials.

One of the most common piping materials is steel. Steel has excellent temperature capability and chemical resistance. There is a wide variety of steel grades available that allow its use in almost any application. As a general class of material, steel has a higher tensile strength than most other piping materials, it is relatively easy to form into various shapes, and it is widely available commercially. One of the chief concerns with steel is its corrosion resistance under various conditions. Carbon steel, carbon-molybdenum steel, and chromium-molybdenum steel are commonly used materials; the first two are recommended for services below 412°C (775°F) and the third is used in applications up to 593°C (1100°F). Stainless steels are used in high temperature (and some grades in low temperature) applications, and where increased corrosion resistance is required.

Ductile iron is another commonly used piping material, but its use is typically limited to lower temperature applications, less than 232°C (450°F). Cast iron has been superseded by ductile iron as a piping material since the 1960s[2]. It has more corrosion resistance and is cheaper than many steels, but does not tolerate vibration and strain due to expansion and contraction. The chief use of ductile iron pipe is sewer, water (drainage and supply), and gas lines. Its corrosion resistance makes it an excellent material for use in underground or underwater applications.

Lined pipes are used in applications where the fluids being sealed are corrosive to metallic pipes. Lining materials include fiberglass, PTFE, and other plastics as well as lead, tin, and rubber; the lining provides the chemical resistance while the steel pipe provides structural strength. Lined pipes can be susceptible to exterior corrosion like regular steel pipe. PTFE-lined pipes are resistant to most every chemical, including strong acids.

FRP, fiberglass reinforced pipe, is used in chemical services. This material is a thermoset plastic resin reinforced with fiberglass filaments, and it can withstand a wide variety of chemicals that attack steel pipe. FRP is resistant to corrosion internally and externally because the whole pipe is made of FRP, not bonded to another material, as found in lined pipes. One concern with this material is its coefficient of thermal expansion. FRP tends to expand and contract much more than steel pipe during thermal cycling. This sometimes creates difficulty in maintaining load on flange seals as temperature changes occur. Hydrofluoric acid is one chemical FRP is not suited for.

Non-ferrous materials used for piping include copper, brass, and aluminum. Copper is the most commonly used, particularly as piping for domestic and commercial plumbing systems as well as in compressed air control systems and HVAC applications. Copper is easily bent, making it easy to work with. It has good corrosion resistance, but is not suited for high temperature applications (greater than 204°C [400°F]). Brass is a copper alloy used for basically the same applications as copper and also as threaded, flared, and compression fittings used with copper, brass, and iron pipe and tubing. Aluminum alloys have excellent corrosion resistance and are also easily manipulated. Their high strength and light weight make them preferred materials for use in a number of hydraulic systems on aircraft. Various alloys are available for food, beverage, and chemical applications.

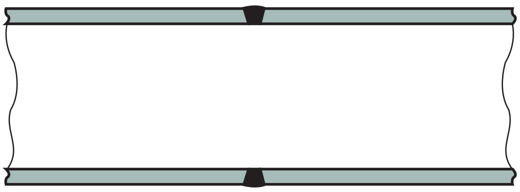
A number of different plastics are used for various piping applications. Polyvinyl Chloride (PVC) is the most well known. This material is primarily used in water, sewer, and drainage systems; its relative low cost, high strength, and chemical (and corrosion) resistance make it one of the most commonly used materials in this category. Polyethylene, polypropylene, and acrylonitrile butadiene are just a few of the plastics used for piping; each has specific chemical resistance and strength properties that make them useful for those applications where metallic piping is not suitable. Note that plastic piping materials are usually thermoplastics, meaning they soften when heated and harden when cooled. For this reason, these materials have relatively low allowable operating temperatures.

Jointing Methods

There are dozens of ways to connect pipes to one another. More than a few of these methods are specific to particular materials and the application in which the pipe is to be used. Cost, tightness, reliability, and flexibility are specific jointing concerns. These methods can be categorized in several general classes: welded, threaded, brazed or soldered, glued, and mechanical.

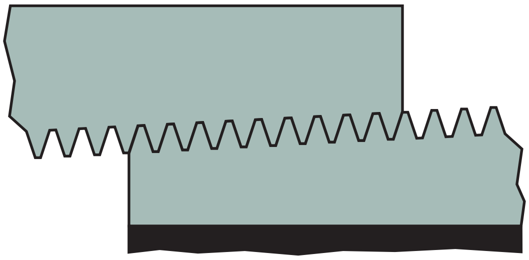
Welding of piping systems has distinct advantages over other types of jointing. It is cost-effective, there is less chance of leaks caused by vibration and creep because the system is essentially one piece, the joints have high strength, service life is the longest of any jointing method, and there is no obstruction of fluid flow (turbulence caused by flanges). Also, since welded pipe has no flanges, there are fewer potential leakage points. One disadvantage is difficulty in replacement of system components. In order to change a valve out of welded pipeline, the valve has to be cut out with a torch and a new one has to be welded back in; this can be time-consuming, depending on the equipment in question (and its location). Welding is often used in higher pressure or more critical applications because of the factors listed above. Note several types of plastic pipe can be fused together with heat.

BUTT WELDED STEEL PIPE



Threaded connections are common with many piping materials. Threading is relatively easy to do and can be accomplished with a minimum of tools and equipment. Threaded connections make it easier to replace sections of pipe, and also are safer to install in areas where fire and explosions are possible (no open flames allowed). The pipe has male threads cut on each end and the required fittings (i.e., flanges, unions, elbows, tees, and couplings) are female threaded. Sections of pipe are attached by screwing them together with the required fittings. Generally, threaded connections are used on lower pressure applications; actual maximum pressure depends on the material used and the size of the pipe. Metals and plastics can be threaded but the pressure rating of the piping system may be less than it would be if another fastening method was used.

ANSI STANDARD PIPE THREAD



Brazing or soldering are frequently used on the non-ferrous metals copper, brass, and aluminum. Temperature and the joint strength of the application determines whether solder, brazing, or a mechanical joint is used. Brazing produces a stronger joint with a higher allowable service temperature limit than soldering. Special fittings are manufactured to facilitate this method of jointing, and they are widely available.

Adhesives are used with plastics, aluminum, and FRP. Aluminum can be bonded to itself or various other metals. Adhesives used on PVC actually chemically weld the two pieces of pipe together. The cement melts the PVC and when the two pieces are pushed together, the PVC fuses. Several special epoxy adhesives are available for FRP materials; the exact material chosen depends on the fluid being sealed.

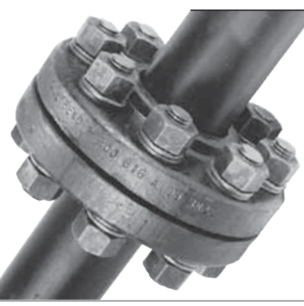
There are many different types of mechanical joints used to connect pipe together. Again, the method used depends on the pipe material and the application. Compression fittings (using molded rubber gaskets), O-ring joints, caulking, and mechanical couplings are used in a wide variety of applications. Each has specific characteristics to offer such as cost-effectiveness, long-term sealing, flexibility, ease of installation, and tightness. Flanged joints are the most common type of mechanical joints used in industry.

Flanges

A flange is a metal plate with bolt holes cut through it towards the outer edge. They are usually circular, but rectangular and oval shapes are sometimes used (heat exchangers and manhole/handhole covers). The flanges are attached to the ends of the pipe; sections of pipe and the required fittings are fastened together by bolting the flanges to each other, in most cases, with a gasket of some sort between them.

Flanges provide a relatively easy means of connecting and disconnecting piping system components, and they form a strong, tight joint. Just like the pipe itself, the flanges are

TYPICAL BOLTED PIPE FLANGE



made to written specifications (e.g., DIN, ANSI, JIS, and BS) that detail materials, dimensions, and tolerances. The specification also indicates how the flanges are to be marked so that they can easily be identified. Nominal size and pressure class, as well as the design specification are typically stamped on the OD edge of the flanges. These markings are extremely important. For a given nominal size, flanges made for different pressure classes or specifications may look similar; the markings make it clear that they are made differently, avoiding any problems caused by installing the wrong flange. All the details of a particular flange, including gasket dimensions, can be looked up in the specification once these identifying marks are found. Without knowing this information, it is difficult to answer specific questions on the flanges.

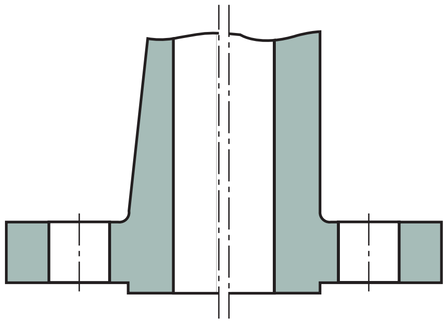
ANSI B16.5 is the most common flange standard in the US. Flanges are designated by their nominal pipe size in inches, the pressure class in psi, and ANSI B16.5 (e.g., ANSI B16.5, 2", 600#). DIN flanges are set up a little differently; the nominal size of the flange is given with the designation DN and the nominal pipe size in mm. The pressure class is designated with PN and is given in kg/cm^2. The DIN standard flanges also have individual specification numbers for each pressure class (e.g., DIN 2636 DN 40, PN64). The JIS flanges are designated in a similar fashion as the DIN flanges in similar metric units, except the nominal pressure classes differ slightly.

Just as the pipe thickness increases for higher pressure applications, flange dimensions increase with higher pressure classes. Flange thickness and diameter, as well as the quantity and diameter of the bolts increases to safely withstand the higher loads required to keep the flange closed and sealed at higher system pressures. This is why flanges of the same nominal size and different pressure classes cannot be used interchangeably. The nominal pressure classes given for the flanges are not the actual maximum allowable working pressure they can withstand; they are simply a number that labels and groups the flanges. The actual pressures that flanges can withstand are significantly higher than their nominal rating. Maximum allowable pressures and temperatures are spelled out in the flange specification.

Connecting the Pipe to the Flange

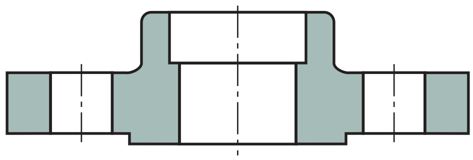
Flanges are affixed to the pipe using a number of methods. Integral flanges are forged or cast as part of the pipe, and the faces are then machined to get the proper finish. Threading is a method where the female threaded flange is screwed on to the male threaded pipe; this method is used in lower temperature applications and also for applications assembled in hazardous environments where welding or open flames are not allowed. In some cases, once the flange is screwed on, there may be a fillet weld applied between the pipe OD and end of the flange, providing a stronger, tighter joint than threading alone.

INTEGRAL FLANGE

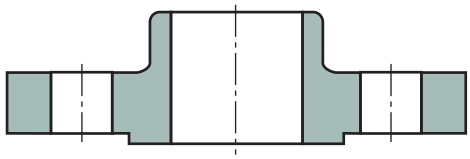


Welding a cast flange with a machined face to the end of the pipe is quite common. This provides a tight, strong joint with a decreased possibility for problems caused by thermal cycling that may affect threaded flanges. There are four basic designs for welding flanges to pipe: the socket weld flange, the slip-on flange, the weld neck flange, and the lapped flange. Socket weld flanges have a counterbore where the pipe is inserted; then the pipe is welded in place. The ANSI B16.5 standard says that socket welds (and threaded flanges) are not recommended in applications where the temperature is above 260°C (500°F) or below -45°C (-50°F) with heavy thermal cycling. Slip-on flanges have a smooth bore and slip over the end of the pipe, then a weld is made where the OD of the pipe meets the end of the flange. Weld neck flanges are one-piece components that are butt welded onto the end of the pipe. Lapped flanges are free-floating collars that fit over pipes with a flared end. A flared nipple is fitted with a lapped flange collar and then butt welded to the end of the pipe. These last two types are most often used in high pressure applications.

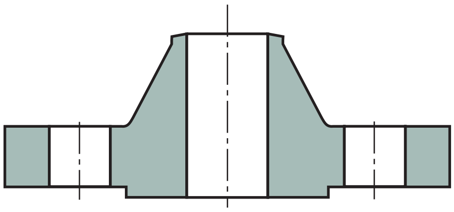
SOCKET WELD FLANGE



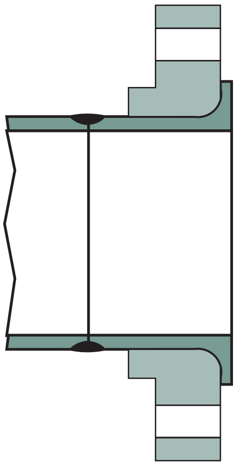
SLIP-ON FLANGE



WELD NECK FLANGE



LAPPED FLANGE



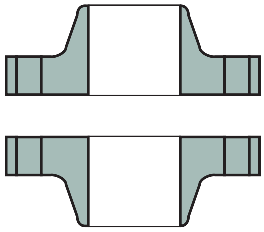
The use of these various flanging methods depends on the system design and specification requirements. The piping code for the system in question dictates what parameters are acceptable for use with each type of flange.

Flange Facings

The faces are the effective sealing areas of the flange. They are mated together when the flange is bolted up, and the gasket is compressed between them. Basically, friction between the gasket and the flange face is the force that holds the gasket in place. Tightening the flange bolts increases compression on the gasket causing increased friction between the gasket and flange face. There are several flange face designs used in bolted flanges. These are plain face, male and female, tongue and groove, raised face, and ring joint. All are designed for use with a gasket of some type, typically a cut piece of sheet gasket or preformed soft metal, filled metal, or spiral wound gasket.

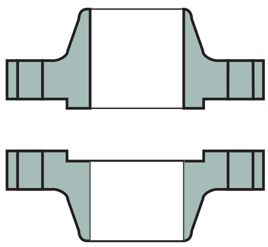
Plain face or flat face flanges use a full face gasket that covers the entire face, or a ring gasket that fits inside the bolt holes. Since the flange is completely flat, the area inside the bolt serves as the flange face; this is why plain face flanges have a large sealing surface area. While they may require higher loading to achieve a required gasket load, they do not tend to rotate like some other designs. The ANSI B16.5 standard specifies that plain, flat face flanges should be used in applications where cast iron flanges are bolted up to steel flanges, particularly where medium or high strength bolting is to be used. This ensures the bolt load is distributed across the entire gasket surface area, and that the flange does not rotate (possibly cracking the relatively brittle cast iron flange).

FLAT FACES

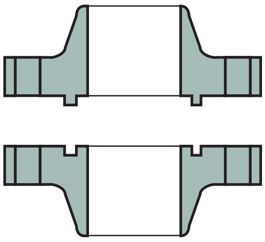


The male and female faces as well as the tongue and groove facings are similar; only the placement of the groove differs. With the male and female design, one flange has a raised face that extends from the ID of the pipe to somewhere inside the bolt holes, and the mating flange has a recessed face with an OD slightly smaller than the ID of the male flange. The tongue and groove design has a similar setup with a raised face and female groove, except that the ID of the machined groove sits between the inside of the bolt circle and the pipe ID. Both these designs have inherent advantages over flat face flanges. The recessed area restricts the gasket, improving blow-out resistance, and the smaller sealing surface area concentrates gasket load, creating a tighter seal at a given bolt load. These facing designs are used in higher pressure applications, especially hydraulic applications, and also on heat exchangers. Both the male and female and the tongue and groove designs are difficult to maintain; they require complete disassembly to remachine the faces if this is necessary. They also must be machined to tight tolerances to ensure there is no interference fit between the faces. Maintenance is also more difficult because the faces have to be pried further apart to remove and reinstall gaskets.

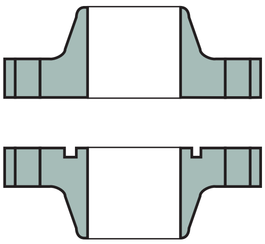
MALE AND FEMALE FACES



TONGUE AND GROOVE FACES

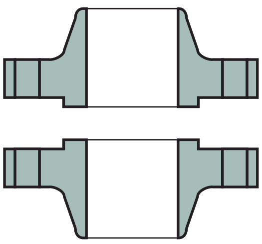


FLAT FACE AND GROVE FACES



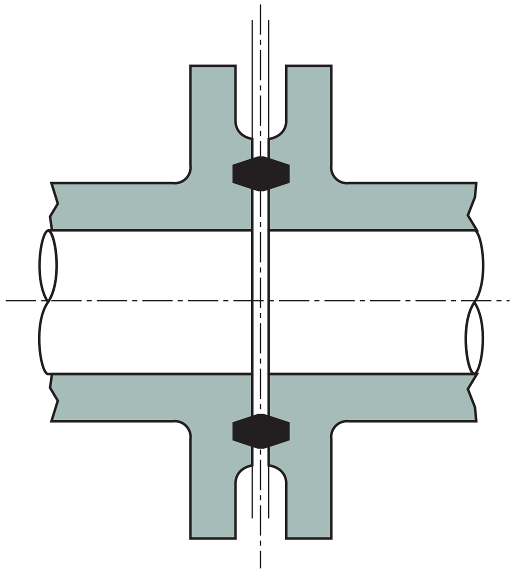
The raised face flange is the most common of the designs. Each flange has a narrow, raised area where the gaskets are seated. These flanges concentrate gasket load like the male and female and the tongue and groove designs, but are easier to maintain, and installation and removal of the gaskets are easier. The joint is easier to inspect, dimensional tolerances are not as critical because there are no mating parts, and the flanges need to be opened only slightly to remove a gasket.

RAISED FACES



Lastly, ring joints are specified in refinery applications or other critical applications where high gasket loading is required to maintain a tight seal. These flanges have a trapezoidal-shaped groove cut on both faces between the inside of the bolt circle and the ID of the pipe. A specially-shaped metal gasket is placed in the groove and the flanges are bolted together. This design allows higher unit loading of the gasket than the other designs. Care has to be taken to ensure the proper ring gasket material is chosen; the required loading of the gasket must be achievable with a bolt load that doesn't cause excessive deformation of the flange, which could cause the gasket to fail.

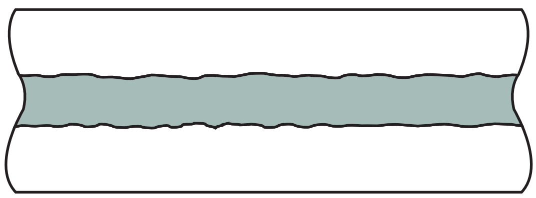
RING JOINT FACES



Surface Finish

"Finish" is a term used to describe roughness of a material's surface, the degree of which depends on the base material and the method of producing the material (e.g., machining and molded). Flange surface finishes may range from a rough casting to a smooth, mirror-like surface. Even flanges with an apparently smooth finish have microscopic peaks, valleys, and surface irregularities that require the use of some type of gasket material to create an effective seal. Finishes do affect the ability of a gasket to create a tight seal. Too rough a finish is difficult to seal because the gasket can't completely fill the deeper imperfections on the flange. On the other side of the coin, too fine a finish decreases the surface friction between the gasket and flange, making the gasket more likely to slip out of the flange while operating. In extreme pressure applications, the flanges may be specially designed with mirror finish, perfectly flat faces that are bolted together with no gasket, but this design is costly, labor intensive, and rarely used.

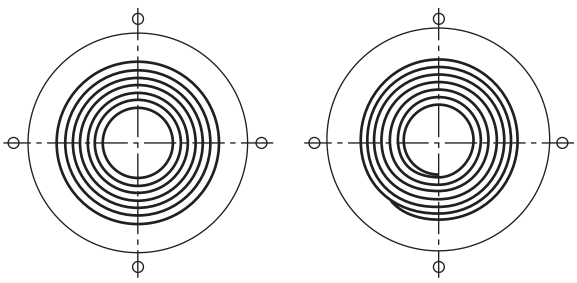
EXAGGERATED VIEW OF FLANGE FACE SURFACES



Finish is expressed as Ra, or Roughness Average, and is sometimes called AA, Arithmetic Average, in the US. When looking at the profile of a surface, the Ra is an average of the variation in peaks and valleys from a centerline over a predetermined length of the surface. The unit of measure can be in micro meters (µm) or micro inches (µin). 3.2 to 12.5 micro meters (125 to 500 micro inches) is typical for flange faces.

Many pipe flanges have a serrated concentric or serrated spiral groove cut into the face. Both types of grooves consist of a single cut or a series of cuts whose width and depth varies with the type of metal and tooling used for the flange. The spiral groove is like a phonograph record; it spirals from the ID to the OD of the flange face. A concentric grooved flange face has definite hills and valleys, and the grooves are not connected. Both these types of surfaces are used to increase the contact area between the gasket and the flange face; increasing this contact area results in increased surface friction and better blow-out resistance. The spiral groove arrangement is less desirable than the concentric type. If the compressed gasket does not completely fill the spiral groove, the fluid can leak along the groove and out of the flange.

CONCENTRIC GROOVES, SPIRAL GROOVES



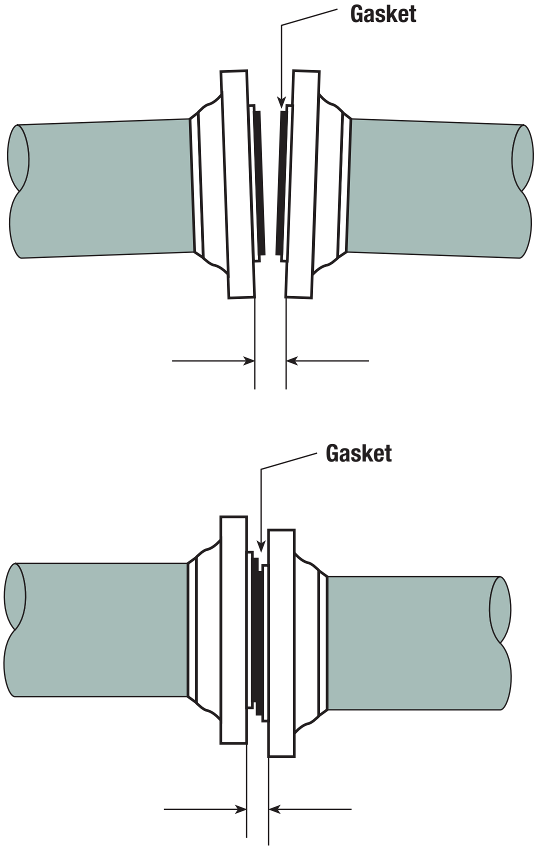
Any time flanges are disassembled, the finish should be inspected before installing the gasket. In-service wear and careless maintenance practices can cause damage to the faces. Minor isolated pits caused by galvanic corrosion usually can still be sealed properly with standard sheet gaskets. If the flange face has deep or widespread pitting or any radial cuts or marks, it should be repaired or the flange should be replaced. Gaskets do not fix the problem, they sometimes function as a bandage that can keep the application going. The area of the gasket compressed between the damaged portion of the faces is subject to lower loads, and since a fluid follows the path of least resistance, it will tend to move through or around that area. These radial marks or cuts can be caused by leakage of media (most often steam) past the previously used gasket, or by the tools used to remove the gasket from the flange. Proper installation techniques can help reduce damage due to careless maintenance practices.

One other consideration related to flange finish is scale and residual gasket material. Scale can be deposited on the flange face if there is some leakage past the gasket, or if some media was trapped between the flange face and gasket during installation. Before reinstalling the gasket, the flange face should be cleaned to remove any scale and debris. Loss of gasket load can occur if the scale or debris volatilizes; loss of gasket load can result in leakage unless the load is reapplied by retorquing. Scale and debris also form a layer between the gasket and flange, reducing the surface friction which is the force that holds the gasket in the flange. A suitable solvent and wire brush should be all that is required to clean the flange. Scrapers are often used to get the bulk of the gasket off the flange, but the wire brush should be used to get the metal surface completely clean. Make sure that the wire brush or scraper is made with a material softer than the flange face so as not to damage the surface finish.

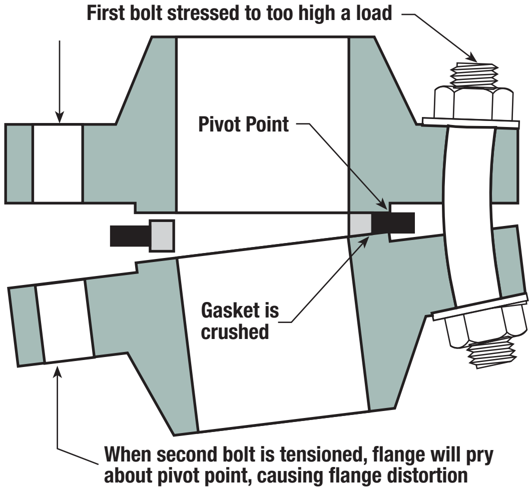
Flange Parallelism

Proper alignment of the flanges is a key to achieving a good seal. In order to provide even compression on the gasket, the flanges have to be parallel. Improper alignment could be caused by improper design, faulty installation, or by distortion due to heat. The gasket is a much more compressible, conformable material than the flange. Tightening the bolts on two flanges that aren't parallel may appear to bring them into alignment, but the gaskets distort first and will not be evenly loaded. At a uniform bolt torque, the section of gasket between the initial contact point of the flanges is under the heaviest load; the area directly opposite is under the least load. If the misalignment is bad enough, the gasket will not seal. Applying additional bolt load may get the gasket to seal, but then the potential for damaging the bolts also increases. Chesterton specifies that the flanges must be

MISALIGNED FLANGES



"BIRDMOUTHED" FLANGE



parallel within .015" for most gaskets, and metallic gaskets have even tighter requirements.

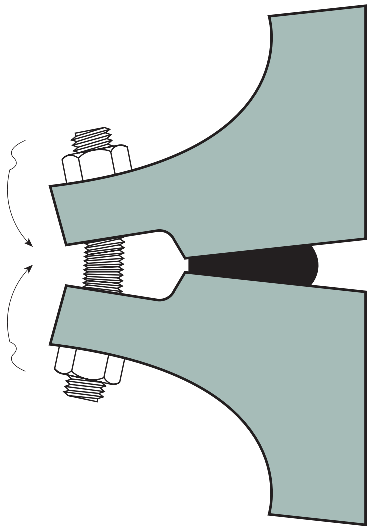
Even if flanges are parallel before assembly, they can be bent out of alignment by poor installation techniques. When tightening the flange bolts using a torque wrench or other device that loads one bolt at a time, the full bolt load should never be applied right away. Bolt load should be applied in increments of the recommended value. If the first bolt in a flange is torqued to the full value, the flange will "birdmouth" or close on one side only. When the opposite bolt is loaded to the same value, the flange will not come back to its original parallel alignment. In effect, a properly aligned flange will have the same sealing problems as a badly aligned one if improper installation techniques are used. Applying bolt load in increments helps prevent "birdmouthing" and ensures that the parallel flanges stay aligned, resulting in even gasket compression. A typical installation procedure would call for the bolts to be all tightened finger tight, then to 30% of the recommended torque value, then 60%, 90% and 100%.

Flange faces also have to be concentric to each other. The centerline of the flange faces must be the same. It would appear that this problem could be corrected simply by putting a pin through the bolt holes and lining up the flanges. However, this may make the flange faces concentric, but then the flanges takes them out of parallel; they would "birdmouth," causing the uneven gasket loading described above.

Flange Rotation

As load is applied to the bolts on a flange, the outer edges of the flange are pulled toward each other. This bending of the flange OD causes distortion of the entire flange and is called flange rotation. As the outside edges of the flanges are drawn together, the inside edges of the flange faces move away from each other. Instead of a uniform compression across the entire gasket face, there is a reduction in gasket load on the ID and an increase on the OD. The effective sealing width of the gasket is reduced, concentrating the stress on the outer diameter of the gasket. This has a beneficial effect in some applications, provided the rotation is not too severe. It can also cause premature leakage and failure of some gaskets like spiral wounds.

FLANGE ROTATION

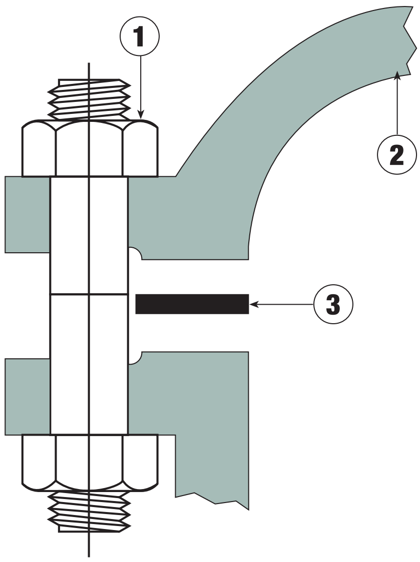


Up to a certain point, the flange will return to its original shape after the load is removed, but the deformation may become permanent if the flange is exposed to high temperatures for long periods of time, or the load put on the flanges is extreme (caused by excessively high torque). Most flanges (except flat face types) will rotate, the degree to which depends on the design and the load applied to the bolts. DIN flanges have been historically more susceptible to flange rotation, while ANSI flanges are more resistant to it. Excessive rotation can be avoided by using gaskets that can seal with less compressive stress (this means less torque has to be applied to the bolts), and by using caution when calculating and applying the bolt torque.

Hydrostatic End Thrust

Not only does the applied bolt load cause flange rotation, so does internal pressure in the flange. Pressure in the flange pushes in all directions. Fluid between the ID of the flange faces pushes outward on the gasket (friction between the gasket and flange resists this and keeps the gasket in place). It also pushes on each flange face, separating them. This force separating the flange faces unloads the gasket, at the same time it applies load to the bolts. Though there is a loss of gasket load, the bolts themselves don't undergo a load loss. The effect of hydrostatic end thrust is more severe on blank flanges and the covers of some heat exchangers because there is a large surface area exposed to the internal pressure, and subsequently a larger force pushing the flange faces apart. Flange design and calculations for bolt torque to achieve specific gasket stresses must account for this phenomenon, or a proper seal may not be achieved.

HYDROSTATIC END THRUST



1. Assembly load produced by bolts or clamps
2. Hydrostatic end thrust
3. Radial force produced by internal pressure

SECTION 2

GASKETING

Gaskets

An ideal flanged joint consists of two mirror-finish surfaces bolted directly together to create a perfect, leak-free seal. In the real world, this is not practical because of a number of factors that can affect the flange such as pressure and temperature variations as well as the costs of manufacturing and the difficulty of maintaining these types of flanges in the required condition. The most common flange face designs in industry today have minute surface irregularities that cannot be sealed without the use of some compressible, resilient material between them to create a tight barrier seal. This is the sole purpose of gasketing. Tightening the flange bolts applies a compressive load on the gasket resulting in high surface friction between the gasket and flange; this is the only force that holds the gasket in place. Ultimately, all gaskets leak, but their individual properties and the amount of compression applied controls the degree to which the media leaks through or around the gasket. Many factors must be considered when choosing the proper gasket for an application, including safety, temperature, pressure, the media to be sealed, equipment condition, versatility of the product, and cost.

Sheet Gaskets

Compressed fiber sheet gasketing products were first introduced in the 1890s. Since that time, compressed fiber gasketing has been the single most widely-used product for sealing flanges. Other varieties of sheet gasket materials include expanded graphite as well as PTFE. Sheet gasketing is a cost-effective and relatively easy-to-handle sealing material that can be used in a wide variety of applications. The bulk sheet format is the key to this type of gasketing's versatility. It gives the end user the capability of cutting a gasket to any required shape, on site, with a minimum of tooling and a relatively small quantity of stock material.

Chesterton has a full line of sheet gaskets that provide excellent sealability and superior reliability for a broad range of applications.

Compressed Fiber Sheets

Compressed fiber sheet gasketing consists of a fiber base that gives the sheet structural integrity, and this is combined with an inert filler such as clay, silica, or some other material that is used to decrease the porosity and take up space in the gasket. An elastomer is used as a binder, holding the gasket in sheet form while giving it flexibility; this is also what determines a gasket's compatibility with the media to be sealed. The remaining ingredients are primarily agents to cure the binder, color additives, and, in some cases, a non-stick surface coating such as graphite or silicone to ease removal from a flange.

The percentage of each of the components depends on the fiber properties, desired properties of the gasket, cost of materials, and the target applications for the product.

There are many different compressed gasket types available that are applicable for an extensive range of fluids, temperatures, and pressures. To ensure the optimal gasket for each application complete service conditions should be provided to the supplier to make a recommendation.

Elastomer Binders

The chemical and physical properties of elastomer binders directly affect which applications the finished compressed fiber gasket sheet can be used. If the binder and media to be sealed are not compatible, the binder could be destroyed, resulting in gasket degradation, and possibly leading to a catastrophic failure. In some cases, the binder may not be compatible with the media, so the gasket swells as the binder reacts with the media, resulting in increased gasket stress, whick actually tightens the seal. Since chemical compatibility charts typically do not show how a binder will react with a media, the best practice is to choose a gasket with a binder that is known to be compatible with the media to be sealed.

The elastomer binders used in compressed fiber sheet gasketing have some common properties. They are incompressible—applied stress changes the shape, but the volume remains constant. Heat gradually and irreversibly destroys their properties and integrity. Cooling causes them to become brittle and rigid; this effect is reversible. Prolonged deformation leads to some degree of compression set. See the chart below for common elastomer binders used and their general properties.

Manufacturing Compressed Fiber Sheets

Fibrous sheet gasketing is made using a machine called a calendar, which is basically a set of rollers that rotate towards each other. There is a larger, heated roll where the gasket is formed and a smaller, cooled roll that presses the raw gasket material onto the hot roller and maintains pressure on the sheet as it is formed.

The elastomer is prepared by immersing solid pieces of rubber in a solvent, liquifying them. At the same time, the body compound is mixed; this contains the fibers, fillers, and other ingredients for the sheet. A starter compound of high rubber/low fiber is applied to the hot roll first. Then the body of the gasket is applied to the rolls in increments until the required thickness is achieved. A finish compound is then applied, and the sheet is removed from the roll and cut into the required sheet sizes. Note that most calendars have rollers slightly over 1,5 m (60") wide, which limits the width of the sheets to 1,5 m (60") after the edges are trimmed; the sheet is typically 4,5 m (180") long and can be supplied in that length, but standard practice is to cut it in three pieces, yielding three sheets of 1,5 m x 1,5 m (60" x 60"). The percentage and type of each component used, process and mixing times and temperatures, as well as roll speeds are carefully controlled. Constant load applied by the calendar rolls on the dough during the combined compression/curing cycle is a critical factor that impacts the sealing characteristics of the gasket.

ELASTOMER BINDERS

Material	Abbreviation	Properties
Natural Rubber	NR	Good resistance to mild acids, alkalis, salts, and chlorine solutions. It has poor resistance to oils and solvents and is not recommended for use with ozone. Its temperature range is very limited and is suitable only for use up to 95°C (200°F).
Styrene-Butadiene Rubber	SBR	A synthetic rubber with excellent abrasion resistance and good resistance to weak organic acids, alcohols, moderate chemicals, and ketones. It is not compatible with ozone, strong acids, fats, oils, greases, and most hydrocarbons.
Buna-N Rubber Nitrile	NBR	A synthetic rubber that has good resistance to acids and solvents, aromatic and aliphatic hydrocarbons, petroleum oils, and gasoline over a wide range of temperatures. It also has a good resistance to caustics and salts but only fair resistance to acids. It is poor in strong oxidizing agents, chlorinated hydrocarbons, ketones, and esters.
Chlorosulfonated Polyethylene	CSM	Good resistance to acids, alkalis, and salts. It resists weathering, sunlight, ozone, oils, and commercial fuels such as diesel and kerosene. It is not compatible with aromatic or chlorinated hydrocarbons and has poor resistance against chromic and nitric acids.
Ethylene-propylene Monomer	EPDM	A synthetic material with good resistance to strong acids, alkalis, salts, and chlorine solutions. It is not suitable for use in oils, solvents, or aromatic hydrocarbons.

PTFE Sheet Gaskets

Fluoropolymer materials are indispensable in chemical applications. PTFE is an inert material that withstands just about any chemical except elemental fluorine and molten alkali metals. No other gasket material has this kind of resistance to chemicals. It has a fairly good temperature range from cryogenic to 260°C (500°F). Many of these PTFE sheet products are made by sintering PTFE powder and then forming the resulting solid into a sheet. Sintering is a heating process that softens the powder which is then pressed together to produce a micro-porous material. The sheets are cut off the billet of material in a process that works somewhat like a cheese cutter.

Virgin PTFE sheets, made from pure PTFE powder with no filler ingredients, are rarely used as sheet gasket today. The big problem with these products is creep, which is defined as a loss of tightness in a gasket measurable by torque loss; when the gasket is compressed, it tends to change shape and cold flow resulting in a loss of bolt load, a loss of gasket compression, and, ultimately, a leak. Because of this cold flowing characteristic, virgin PTFE sheets are only suitable for use in low pressures, approximately 7-14 bar g (100-200 psig), without some type of reinforcement to give them more structural strength and creep resistance.

To give PTFE sheets more physical strength and creep resistance, fillers are added to the PTFE powder before sintering. The fillers act like rebar in concrete; they help provide a more rigid structure, helping to keep the PTFE from moving under load. Glass is the most common filler material for PTFE sheets. Carbon, graphite, and bronze are also sometimes used, but more often they are used to improve wear resistance and frictional properties, and

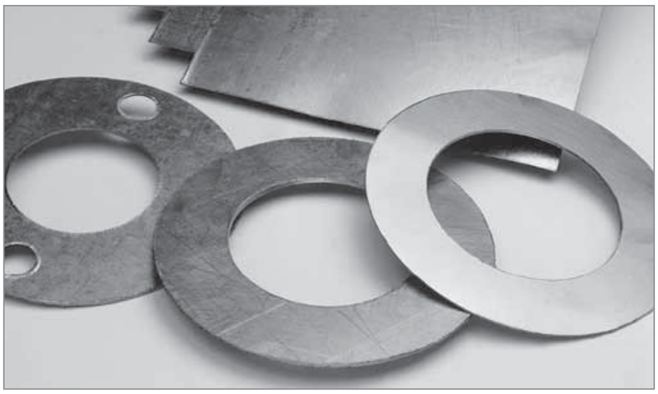
Chesterton Expanded PTFE seals worn or uneven flange surfaces reliably. Available in tape and sheet gaskets.

to harden the PTFE for use in bushings and load bearing components. Note that the fillers do decrease the chemical compatibility in some instances.

A more recent process has been developed to produce "expanded" PTFE products. These materials have no added fillers to give them increased creep resistance; the improved properties are achieved using a special processing method. These products are made by extruding the raw material from a hydraulic press, then stretching it and heating it to a high temperature. This yields a relatively soft, yet strong material that has fibers of PTFE embedded throughout the product. The fibers give the product much better creep resistance, with no decrease in chemical resistance. Expanded PTFE products have excellent conformability, creep resistance, excellent chemical resistance, good sealability, and better strength than virgin or filled PTFE products.

Flexible Graphite Sheet Gaskets

Flexible graphite sheets are made from naturally occurring graphite flakes. After processing to purify and expand the flakes, they are formed into pure graphite sheet gaskets with no fillers or binders. The lack of fillers or binders is important because, at high temperatures, there is no significant volume loss of sealing material, and the gasket does not harden like elastomer-bound compressed fiber sheets. Combine this with graphite's outstanding thermal stability, superior chemical resistance, and excellent compressibility, and the result is one of the best sheet gasket materials on the market today. Graphite sheets are more fragile than compressed fiber sheet gaskets and must be handled with care during cutting and installation. To make the graphite sheets easier to handle and cut without damaging them, metal foil inserts are sometimes added. The reinforcement decreases the maximum temperature limit of the graphite sheet somewhat, but that limit is still higher than that of any fibrous sheet gasket.

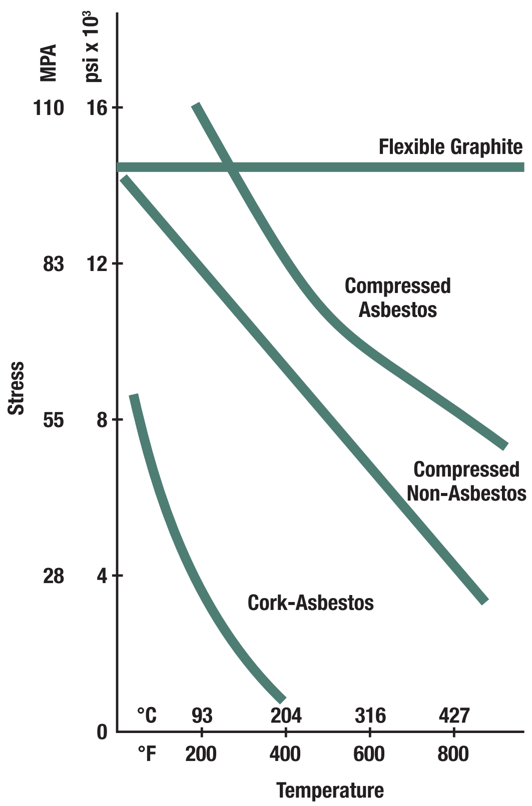


Chesterton graphite sheets for high pressure, high temperature applications.

To manufacture graphite, the graphite flakes first have to be extracted from ore. The ore is pulverized and then run through wet flotation purification to remove the water soluble materials from the graphite flake. Wet flotation produces a material that contains 92-94% graphite. The remaining materials, called ash, are primarily minerals. The raw graphite material is then chemically washed by dissolving the ash with acid. The next step involves treating the flake with high concentrations of chemicals such as sulfuric or nitric acid to weaken the bonds between the carbon atoms. After the reaction, the excess oxidizer is removed and the flake is exposed to high temperatures. This heating process expands the graphite flake particles in a lengthwise direction up to 200 times their original size. These worm-like particles are collected and calendared into a sheet form; for thicker materials or materials with foil inserts, several of these sheets may be laminated on top of each other. The maximum width of the sheet is 1,5 m (60") for the same reasons as the fibrous material: the calendar roll width limits the maximum sheet size that can be made.

Flexible graphite has many characteristics that make it different from any other gasket material. It is extremely resistant to almost all chemicals (except strong oxidizers). This makes it suitable for almost any application from a chemical compatibility standpoint. Graphite does not harden or become brittle with age, which means shelf life is not a concern. It has negligible cold flow and the lowest creep relaxation of any sheet gasket. This means the gasket holds its shape under compression and doesn't need to be retightened after system startup. Graphite has low gas permeability, which translates to extremely low leakage of media through the gasket. It conducts heat extremely well and expands very little at high temperature. It is extremely compressible, which means it can conform to irregularities on a flange face much better than some of the harder compressed fiber sheet materials. It is resilient, meaning that even if a load is applied and then released, it tends to bounce back towards its original thickness, even after being heated; resiliency means the gasket actually exerts force on the flanges, helping to maintain a tight seal.

LOAD BEARING ABILITY OF VARIOUS GASKET MATERIALS



Note: Each curve represents the maximum temperature pressure point for the gasket material.

Oxidation of Graphite

High purity flexible graphite has a maximum temperature limit of 2760°C (5000°F) in non-oxidizing media and a maximum limit of 450°C (850°F) in an oxidizing media. An oxidizing media is one that accepts electrons from another; oxygen is one of the more common oxidizing agents, but it is not the only one. Sulfuric, concentrated nitric, and chloric acids are strong oxidizers that will attack graphite; other chemicals not compatible with graphite are permanganates, chromium solutions and molten alkali and alkaline earth metals. Several factors affect the rate of oxidation of a graphite sheet in a flanged joint.

Any oxidation will take place on the exposed surface area of the gasket. Only the edges of a gasket clamped in a flange are exposed to media; a majority of the gasket's surface area is clamped between the flange faces and is not exposed to media. The OD of the gasket can also begin to oxidize if the air around the flange gets hot enough. Thicker gaskets exhibit higher oxidation rates because of the increased edge area. The gasket surface in contact with the flange surface tends to oxidize, and oxidation within the gasket body occurs if there is permeation of oxygen into the material. Foil inserted graphite sheets tend to oxidize faster than those without foil. Joint tightness has a direct effect on oxidation rate. The density of the graphite increases with higher clamping force; the higher the density, the less permeable the gasket.

Another factor is the purity of the graphite; lower purity, high ash content graphite oxidizes more quickly. Ash is primarily oxides that remain in the product even after high temperature exposure; lower grade graphite products have higher ash content. Oxidation testing is highly dependent on the sample thickness, and the test fixture. Testing done on a free standing gasket (not clamped in a flange) in a hot air furnace would show drastic, rapid oxidation, while a gasket bolted in a flange with media being piped through the ID would undergo oxidation at a much more reduced rate.

There are passive oxidation inhibitors in some graphite sheets that reduce oxidation at high temperatures (but nothing completely stops it). Inhibitors raise the threshold of oxidation by about 100°C (212°F) and reduce oxidation by a factor of ten or more. These inhibitors convert to a glass-like structure at elevated temperatures, creating a physical barrier that covers the "active sites" on the graphite crystal, the starting points of oxidation.

Metallic Gaskets

Spiral Wound Gaskets

Spiral wound gaskets are preformed rings made with windings of a chevron-shaped metal strip and a layer of filler material sandwiched between them. These gaskets are made according to a number of possible specifications, depending on the country, which specify details of the manufacturer to ensure uniformity and quality of product. Gaskets made to these specifications should be of consistent quality because they are all made the same. The specification dictates gasket dimensions, materials, marking, and design. The dimensions of the gaskets are preset for a given nominal pipe size and pressure class. The customer only has to know what the pipe standard is (e.g., ANSI B16.5), the pressure class, and the pipe size to find the gasket dimensions from the specification; the winding and filler materials options are chosen by the customer and depend on the chemical to be sealed and the maximum temperature of the application.

The windings have a special chevron design that provides resiliency and acts as containment for the softer filler material. The API 601 specification calls for a minimum of three windings on the ID before the filler is added, with a minimum of three spot welds on the ID and OD. The filler material conforms to the imperfections on the flange face, creating the seal. The windings can be made with a variety of stainless steels or exotic metals to handle a wide range of chemicals and temperatures. The most common filler materials are flexible graphite and PTFE, each with specific chemical resistance and temperature capabilities.

Spiral wound gaskets are available in several configurations. For raised or flat faced flanges, the "CG" type has a steel ring on the OD that serves as a compression gauge, preventing overcompression of the gasket and also as a guide that fits inside the bolt circle, centering the gasket on the flange face. The ring also adds strength to the gasket and helps prevent blowout. API 601 indicates that gaskets above specific nominal pipe sizes in higher pressure classes should also be fitted with an inner ring as well, to provide additional overcompression protection (these are designated CGR). For male/female, tongue and groove, or plate and groove flanges, there is an "R" type available. "R" stands for ring, and there are no centering guides on the ID or OD. These joint facings contain the gasket and serve as the blow out preventer; the groove dimensions are designed to also prevent overcompression of the gasket.

Spiral wound gaskets are color coded to identify the winding and filler material at a glance. On gaskets with centering guides, there is additional information stamped on the guide showing pressure class; nominal pipe size; the winder, filler, and centering ring material; a mark indicating the gasket specification; and the manufacturer's mark.

Chesterton offers a full line of Spiral Wound gaskets.

Spiral wound gaskets are typically used in higher pressure applications. They are capable of withstanding very high pressures and are relatively cheap, but there are also some drawbacks. With sheet gasketing, a bulk material is available for the mechanic to make a gasket for any size and shape of flange as required. Spiral wound gaskets are pre-made and, in order to cover all the applications in a plant, the stockroom has to have all sizes and pressure classes of spiral wounds available. Spiral wounds are also very sensitive to uneven bolt loading as well as proper alignment in the flange. If they are off center, in some cases, the windings can break, resulting in catastrophic failure. Also, spiral wounds require substantial loads to achieve a tight seal. They are relatively thick and, therefore, have more surface area exposed to pressure; this results in higher forces trying to force the gasket out of the flange.

Camprofile Gaskets

Camprofile or "grooved" gaskets offer a high quality, low emissions sealing alternative. Camprofile-type gaskets consist of a metal core with concentric grooves and normally have sealing layers of either flexible graphite or PTFE. Metal cores are usually selected based on the metallurgy of the piping system.

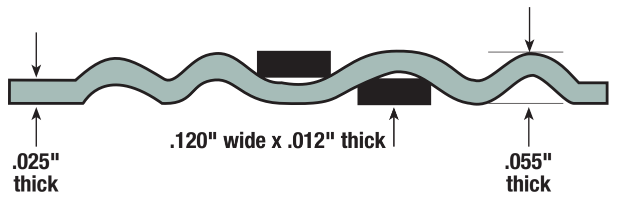
Camprofile gaskets are often used in applications where high pressures and temperatures are maintained and, consequently, high bolt loads need to be controlled. Camprofile gaskets offer outstanding flexibility and recovery, assuring seal integrity under pressure, temperature fluctuations, flange rotation, bolt stress relaxation, and creep.

Chesterton offers a complete line of Camprofile Gaskets.

Steel Trap™ Gaskets

The Steel Trap gasket is a preformed ring gasket that has some improvements over spiral wound gaskets. The Steel Trap can be used in extreme pressure, extreme temperature, and extremely aggressive media. It can be configured for service in virtually any application. The two available sealing element materials, graphite and PTFE, can seal almost every chemical media, and the range of carrier materials is extensive, allowing the gasket to be tailored to specific application requirements. The standard configuration is graphite and 316 stainless steel, with a single sealing strip on the top and bottom of the gasket; larger sized gaskets are supplied with double sealing rings (two on the top and bottom) to offset problems with flange deformation at higher pressures.

PROFILE OF STEEL TRAP CARRIER AND SEALING STRIPS



As the flanges are closed with tightening of the bolts, the containment grooves are flattened until the sealing element fills the groove entirely. The containment grooves trap the sealing elements, preventing extrusion and reducing creep to almost nil; graphite has low creep characteristics already, but problems of creep with PTFE are eliminated in the Steel Trap. In the fully-compressed state, the carrier also isolates or “traps” the sealing element from full exposure to the media or atmosphere outside the flange, thereby extending its life.

The narrow sealing strips on the Steel Trap concentrate the applied bolt load over a smaller area than a full face or spiral wound gasket, and the load required to achieve a seal is lower than other metallic gaskets. At a given bolt torque, the load is distributed across the narrow sealing strips, not the entire surface of the flange sealing area using a full face gasket. This means higher gasket stresses, which result in a tighter seal. Also, because the Steel Trap design concentrates the applied bolt load over a smaller sealing area, lower bolt torque can be used to achieve the required gasket stress. This helps reduce problems with high bolt loads causing deformation of the flanges. The Steel Trap is an inherently thin gasket, 0.8 mm (1/32") thick, compared to a spiral wound, 3 mm (1/8") thick. In an application under normal operating conditions, there is a much smaller edge profile exposed to pressure and, therefore, there are much lower forces trying to push the Steel Trap out of the flange.

Steel Trap is the ideal replacement gasket for heat exchangers and other critical applications.

Steel Traps are offered in several configurations: ring, self-locating, and specialty (heat exchanger) types. The standard ring type is made to the same dimensions as spiral wound gaskets; the exact dimensions depend on the flange specification (ANSI, DIN, or JIS). If the flange is not a standard specification design, a Steel Trap ring gasket can be custom made to fit the application. The OD of the Steel Trap ring gasket fits just inside the bolt circle of the flange, centering the Steel Trap gasket like a “CG” spiral wound.

The self-locators are a step up from the ring gaskets. This design incorporates specially designed tines that center the gasket in the flange, serving the same function as the outer ring on “CG” spiral wound gaskets. The difference is in the design of the self-locating tines. For any nominal pipe size, the gasket dimensions, bolt size and quantity,

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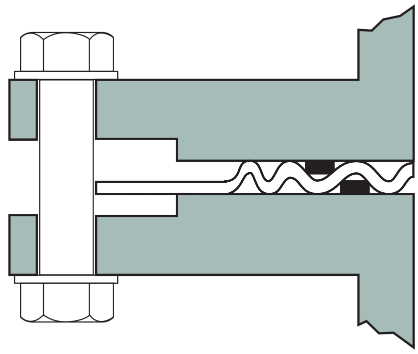

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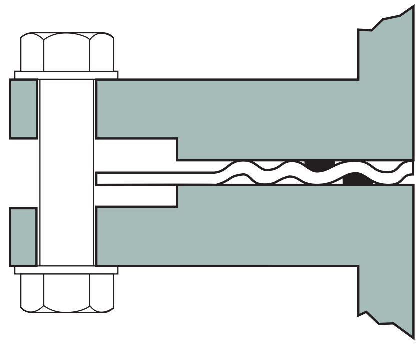
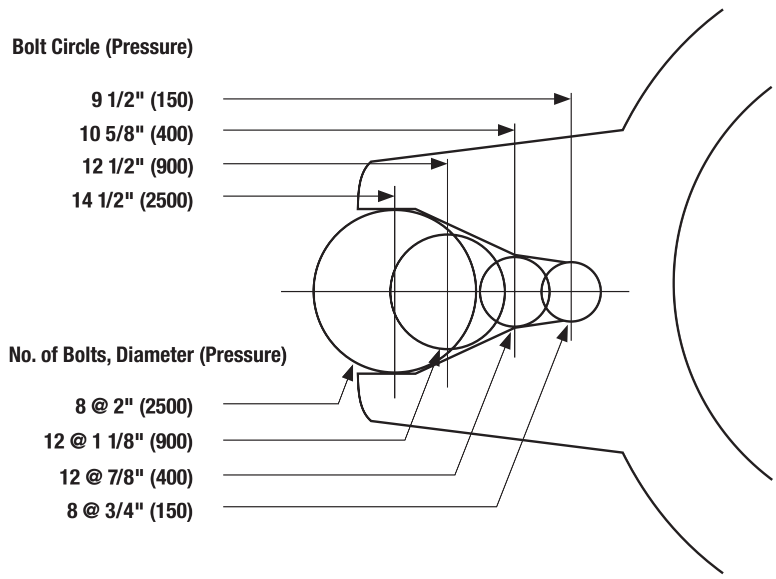


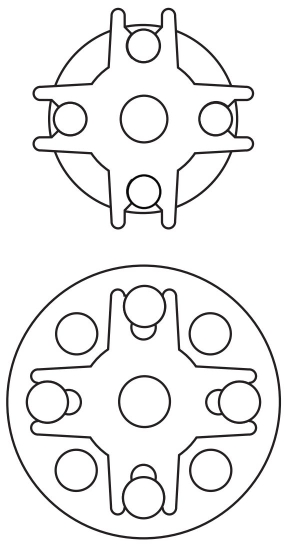
and the bolt circle increase with the pressure class. The self-locator is designed to center the gasket on all pressure classes of a given nominal size of pipe, and it is available for ANSI, DIN, and JIS standard flanges for nominal sizes up to 300 mm (12"). For example, on a 2" ANSI B16.5 flange, one Steel Trap can be used to seal a class 150 or a class 2500 flange (and any class in between); it will fit both flanges and properly center the gasket on the faces. For a 2" pipe, class 150, class 600, and class 2500, a plant would have to purchase three different spiral wound gaskets, or they can get one Steel Trap that will fit all three.

The specialty type Steel Trap is for heat exchangers. These gaskets are custom-made and can be configured to fit all varieties of heat exchangers, from a simple ring to a Z-bar or quad-cross bar type of virtually any diameter. Steel Trap gaskets, in conjunction with flange live-loading, are capable of providing long-term, leak-free performance in heat exchangers, one of the more difficult sealing applications in any plant.

SELF-LOCATING MULTI FIT DESIGN



SELF-LOCATOR STEEL TRAP



Gasket Test Standards

Most literature for gasket materials makes reference to various properties, the most common being sealability, creep relaxation, compressibility, and recovery. These properties are determined using standards that dictate how the test is to be conducted. Most of the tests are done according to ASTM standards, as well as DIN standards. The point of these tests is to indicate what physical features the gaskets have; knowing these helps the end user determine which material is best for their applications. It is important to know what these tests reveal, and just how useful this data is.

Compressibility and Recovery

Compressibility and Recovery of Gasket Materials, ASTM standard F-36, is used to determine "the short-time compressibility and recovery at room temperature of sheet gasket materials.[3]" "DIN 28090-2, *Static gaskets for flanged connections – Part 2 Gaskets made from sheets; special test procedures for quality assurance"* is another standard used to determine these characteristics, but it is done both at room temperature and elevated temperatures (dependent on the material type). Compressibility is a measure of the reduction in gasket thickness under a specified load. Recovery is the increase in thickness, indicated as a percentage of the original compression, after that load is removed. Higher compressibility means a gasket is more able to conform to the irregularities of a flange. A more intimate fit with the flange increases the friction holding the gasket in place, and it leaves fewer leak paths between the gasket and flange surface. Recovery essentially indicates how much "spring" there is to a gasket. Higher recovery means the gasket will exert more force on the flange, helping to maintain a tight seal.

The information obtained using these two test methods is of limited use for several reasons. Room temperature testing is not a good indicator of performance of materials in actual applications which are most often at elevated temperatures. Compressed fiber sheets tend to harden at high temperatures as the binders cure and any moisture in the filler material boils out. At actual operating temperatures and pressures, these types of sheets will not have the same kind of compressibility and recovery as shown by this ASTM procedure. The other drawback is the short-term nature of the test. Prolonged tests give more realistic data on what happens to gaskets which are not typically used for only short periods of time.

Creep Relaxation

"ASTM F38-00(2006) Standard Test Methods for Creep Relaxation of a Gasket Material" is used to measure "the amount of creep relaxation of a gasket material at a predetermined time after a compressive stress is applied.[4]" "DIN 28090-2, Static gaskets for flanged connections – Part 2 Gaskets made from sheets; special test procedures for quality assurance" is another standard which is used to determine these characteristics. These are basic tests that simply measure the amount of permanent deformation of a gasket after a load is applied, and the result is typically expressed in percent loss in initial applied bolt load. The gasket is compressed to a specific load between two bolted plates, it is then heated in an oven, removed, and cooled, and the loss in bolt load is measured. There is also DIN 52913 which looks not at the load lost but the gasket stress retained. In this case, the gasket is loaded to a high stress (similar to an actual application) and heated to elevated temperatures for 16 hours, and the gasket stress remaining after the test is determined. Creep is an undesirable characteristic in gasket materials. The only thing holding the gasket in place is the surface friction between the gasket and flange, and that friction is directly related to the surface finish and the load applied by the flanges. If that load is somehow lost, the surface friction decreases, and the gasket is more likely to leak or blow out of the flange. If a gasket has high creep, the load holding it in the flange will have to be periodically reapplied, a maintenance headache.

There are a few notable factors when looking at creep relaxation properties obtained using the ASTM standard. According to the standard, gasket thickness for the test is specified as 0,8 mm (1/32"), unless otherwise noted. 1/32" gasket is typically the thinnest sheet gasket available. For comparison, DIN 28090-2 utilizes 2 mm thick material for testing. Thinner gaskets will creep less than thicker ones because there is less material to flow. In a majority of applications in the field, 1,5 mm (1/16"), 2 mm, or 3 mm (1/8") gaskets are used, so the actual creep relaxation in service will be more than shown in the product literature (unless the literature specifically indicates the thickness tested is 1/16"). When comparing test data on creep relaxation of sheet materials, ensure that the test data was obtained using the same thickness of gasket material. If 1,5 mm (1/16"), 2 mm, or 3 mm (1/8") thick gaskets are standard in a plant, one other consideration is the test method. There are two test methods listed in the standard. Method "A" is done at room temperature and Method "B" is done at room or elevated temperatures. Since gaskets tend to undergo more creep relaxation at higher temperatures, the values obtained by this test, if done at room temperature, may be even higher in actual service. When comparing values for two different gasket materials, make sure they were tested using the same method and under the same conditions, otherwise the comparison will be inaccurate.

Sealability

Sealability testing is used to define leakage rates of either liquids or gases through and around the gasket, under controlled conditions. In any application, higher sealability, which translates into less product loss, better safety, and increased cost savings is preferred. Zero leakage is an inaccurate term when describing leak rates and gasketing. All gaskets leak, but some leak less than others; near-zero leakage is a more accurate term. "Sealability of Gasket Materials, ASTM F-37" gives "a means of evaluating sealing properties of sheet and solid form in place gasket materials at room temperature.[5]" Another test method is "DIN 28090-2, Static gaskets for flanged connections – Part 2 Gaskets made from sheets; special test procedures for quality assurance" specifically the "Leakage Test."

The F-37 test is used to determine the sealability of a gasket at room temperature with Method "A" being done in liquids at room temperature and Method "B" being done with liquids or gases at room or elevated temperatures. It is a starting point to determine the applicability of a gasket; if the gasket leaks excessively at room temperature, it will probably fare worse at higher temperatures. Here again, the test is done at room temperature which does not accurately simulate most real-world applications. Temperature affects the sealability of gaskets. As in the creep relaxation test, the gasket thickness specified by the standard is 0,8 mm (1/32"). Thinner gaskets will have better sealability, because there is less area for the fluid to leak through. Since the standard thickness used in most applications is 1,5 mm (1/16") or 3 mm (1/8"), the sealability of the gaskets in actual applications would most likely be higher than shown by this test without even considering the effects of temperature. Note that the leakage rates of gaskets tested with liquids will be lower than gaskets tested with gases. Gases are typically harder to seal because the smaller molecule size allows them to penetrate or pass by the gasket more easily. As with other tests described earlier, when comparing values for two different gasket materials, make sure they were tested using the same method and under the same conditions, otherwise the comparison will be inaccurate.

Gas Permeability, DIN 3535, is a test that can provide more accurate data on the leakage of a fluid through a gasket. In this test, the test fluid is a gas, nitrogen, and the test pressure, 40 bar g (580 psig), is higher than in the ASTM sealability test. The key difference between the DIN and ASTM standards is the test temperature. The DIN standard can be elevated to give leakage data on a gasket operating under more realistic conditions, whereas the

ASTM test is done at room temperature. Overall, the DIN 3535 test is a more versatile test that can provide somewhat more realistic data on gasket sealability. Again, make sure when looking at comparative test data that the test parameters used for each sample are the same, otherwise the comparison will not be accurate.

Alternative Gasket Tests

In an effort to better quantify the performance characteristics of gaskets, there has been a fairly recent movement, spearheaded in the United States by the Pressure Vessel Research Committee (PVRC), to come up with more accurate test methods that better describe how a gasket will perform at higher temperatures. The Tightness Testing and Research Laboratory at the École Polytechnique of Montréal has been heavily involved in the development of the test procedures and equipment required to obtain more accurate data on gasket performance characteristics.

As noted above, most of the ASTM (and some of the DIN) tests are done at room temperature and for relatively short durations. These newer tests are designed to provide more useful, realistic data on gasket properties such as creep relaxation, tensile strength, blowout resistance, leakage rate, and other factors, both at ambient and elevated temperatures. This data gives a much better picture of a gasket's performance under actual operating conditions. The tests can be broken down into four categories: mechanical tests, screening tests, leakage tests, and fire tests. Mechanical tests are used to determine the short term, room temperature characteristics; the screening tests determine the long term gasket properties over a wide range of temperatures; the leakage tests measure the tightness of gaskets under actual operating conditions of a plant; and the fire tests determine how well a gasket can maintain a seal after exposure to a fire. While these tests are not yet included in ASTM or DIN standards, some of them are in the process of being approved and should eventually be included in the standards.

Selecting the Proper Gasket

In order to ensure that an application is sealed satisfactorily, the first step is to choose the right gasket for the job. When engineers or mechanics are looking for the correct gasket, they must consider several general factors: chemical compatibility, the temperature and pressure of the application, condition of the sealing area, and the thickness of the gasket to be used. These factors are all related to each other; a change in one parameter affects the other parameters, and they cannot be considered separately.

Chemical Compatibility

The gasket for any application must be compatible with the media to be sealed. Chemical attack can cause a change in the gasket's properties or cause physical degradation, resulting in excess leakage or a complete catastrophic failure. Only the ID of the gasket is exposed to media in a flange application, so the surface area that undergoes chemical attack is quite small. This is why gaskets usually don't fail immediately upon startup; it takes some time for the media to attack the gasket across its entire surface area. Some gaskets swell when exposed to certain media. In these cases, even though the media is not compatible with the gasket, the application may possibly be sealed successfully. The safest choice for a gasket is always one that is completely unaffected by the media in the application. Most manufacturers can provide compatibility information for their gasket materials with the more common chemicals. The compatibility of fibrous sheet gaskets is based on the elastomer binder. If the binder is chemically attacked, the sheet begins to lose its physical integrity and starts to break down (see Elastomer Binders Chart, page 16). Graphite sheets are more universally compatible with just about everything except strong oxidizing medias. PTFE can handle even more chemicals than graphite, including the oxidizing agents, but can't withstand the extreme temperatures that graphite can. For more information on the compatibility of Chesterton's sheet products, consult engineering.

Temperature

Temperature is a key consideration when choosing the right gasket material for an application. Heat affects types of gaskets differently, and each gasket has a published maximum temperature limit that should not be exceeded. Elastomer-bound gaskets have some degree of filler materials in them; both the elastomer binder and filler are affected by high temperatures. These types of gaskets will harden and lose resilience at high temperatures as the binder fully cures and any moisture in the filler evaporates. PTFE materials are not affected in the same way, meaning they don't harden, but they do tend to creep more at higher temperatures. Graphite sheets are capable of withstanding much higher temperatures and they do not harden due to their lack of any fillers or binders, but oxidation is a concern at temperatures above 450°C (850°F) in certain environments.

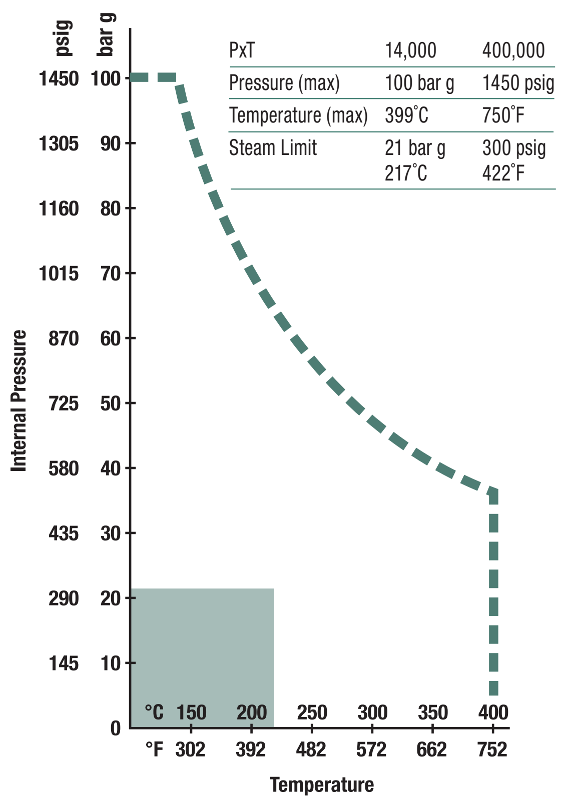
Low temperatures have different effects on gasket materials. As gaskets get colder, they start to lose resiliency and become more brittle. Retightening the gasket at cryogenic temperatures will more likely result in fracturing of the gasket rather than achieving a tighter seal. Gaskets to be used in extremely cold services should be kept clean and dry. The fibrous types of gaskets absorb moisture more readily than the graphite or PTFE-based materials, and this moisture can expand as the temperature drops, changing the loads on the various flange components.

Pressure

The maximum pressure that gaskets can withstand is closely tied to their maximum temperature capability. Most gaskets cannot be used in applications where they are exposed to their maximum temperature and pressure limits at the same time; there is a compromise between the two. The gaskets have reduced pressure capability at their maximum temperature and a reduced temperature limit at maximum pressure. This relationship is expressed as a PxT value where P = listed maximum internal pressure (bar g or psig), and T = listed maximum operating temperature (°C or °F). For a given application, if the temperature multiplied by the pressure of the application is below the PxT value for the gasket in question, the gasket can be used at those parameters, assuming chemical compatibility has been verified. If the PxT value for the gasket is less than the PxT of the application, another gasket should be chosen. Note that most non-asbestos fiber sheet gaskets have even lower limits and PxT value in saturated steam. Steam is one of the more aggressive media to these kinds of gaskets; the steam weakens the base fibers used for reinforcement, decreasing the gasket's operating capabilities.

(Refer to PxT Curve Example.) A gasket has a maximum pressure limit of 100 bar g (1450 psig), a maximum temperature rating of 399°C (750°F), and a PxT rating of 14,400 bar g °C (400,000 psig/°F). At the maximum pressure limit of 100 bar g (1450 psig), the maximum allowable temperature limit is equal to the PxT value divided by the maximum pressure limit: T = 14,400 bar g °C/100 bar g = 144°C, or T = 400,000 psig °F/ 1000 psig = 400°F. The maximum allowable pressure limit at the maximum temperature limit can be determined in the same way: at the maximum temperature limit of 370°C (750°F), the maximum pressure limit is: P = 12,500 bar g °C/370°C = 33.8 bar g, or P = 350,000 psig °F/700°F = 500 psig.

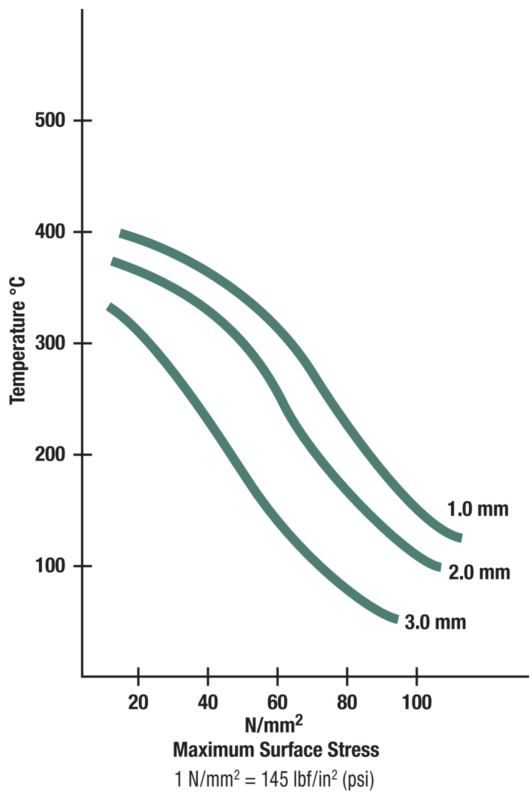
PxT CURVE EXAMPLE



Thickness

Once a gasket is found that can withstand the chemical media, the temperature and pressure of the application, there is still one consideration: the maximum thickness required for the application. The general rule of thumb is to use the thinnest gasket possible for several reasons. A thinner gasket has less surface area exposed to the system pressure and, therefore, less force working to push the gasket out of the flange; decreased exposure to the media means less material leakage through the gasket. The minimum stress required to achieve a seal decreases with gasket thickness. Thinner gaskets undergo less stress relaxation than thicker ones, and they have superior load bearing resistance. Not only do thinner gaskets perform better, they cost less. There are only a few reasons a thicker gasket should be used. In cases where the flange surface is fairly rough, a thicker gasket will be required to conform to the irregularities of the surfaces, particularly if the gasket is not very compressible. There are also some applications where the designer has set up a flanged

MAXIMUN GASKET STRESS VS THICKNESS



joint that depends on the gasket to maintain specific clearances between two parts, as well as maintain a seal. For the most part, 1,5 mm (1/16") or thinner sheet gasket material should be fine for a standard flange sealing application. A 3 mm (1/8") thick sheet gasket should only be used on flanges where there is excessive corrosion or flange damage that cannot be repaired at the time of reassembly.

Plant Specifications

Many plants may have a set of specifications that dictate applications that each type of sealing material is suitable for. Having structured selection criteria is one way a plant can work to ensure that gasket failures caused by misapplication are reduced. Typically these criteria are conservative to account for the variations in flange conditions typically found in the field, and to provide some measure of safety by not using a product at its maximum service parameters. Chesterton believes that safe conservative guidelines like these are an excellent framework to safely cover all the gasketing applications in a plant and can help plants develop gasket recommendations.

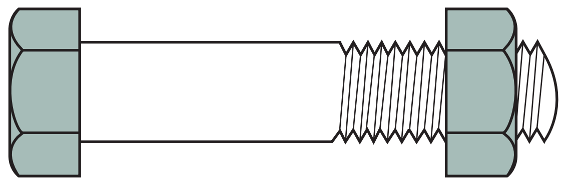
SECTION 3

BOLTED FLANGE JOINTS

Bolts and Threads

A bolt can be simply defined as a threaded fastener where the thread is "an inclined plane wrapped around a cylinder." [6] There are many factors that contribute to the success or failure of a bolt in an application, some of which will be touched on later. Bolts are used for two basic purposes: keeping two or more components aligned with one another or to store elastic energy so as to clamp components together, the latter being the more common of the two.[7]

STANDARD BOLT



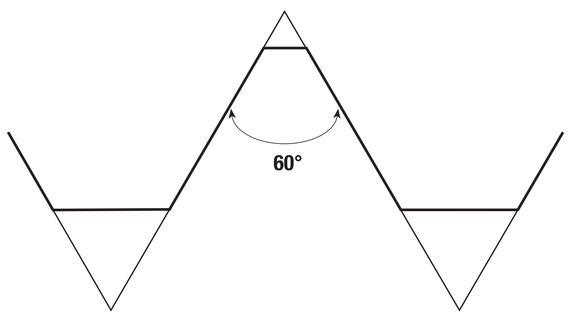
Bolts (and screws) are a common method of fastening components together and have been for quite a long time. Like pipe flanges, bolts and their base materials are made to specific standards (e.g., ASTM, DIN, JIS, and BS) that dictate physical properties, materials, tolerances, and thread design. With standardization of threading systems, bolts made to a given specification are interchangeable, regardless of the manufacturer. A bolt made by any manufacturer will fit a mating threaded component made to the appropriate corresponding specification. The use of published standards makes it possible for an engineer to use off-the-shelf components in the design of a system because the characteristics of the bolt are known and consistent. Bolts are relatively cost-effective compared to other clamping methods. Standardization helps keep costs down as manufacturers compete for business. Standardization also usually means a broader availability of fasteners. Bolts are easy to use, and there is an enormous variety of standard designs that cover a wide spectrum of applications. Various industries have established standards that help define bolting materials to suit their particular needs, for example SAE for the automotive industry and ASTM for systems design and large structural projects.[8]

Thread Design

The design of the threads used on bolts and nuts (or tapped holes) is the key to these types of fasteners. It allows for the repeated installation and removal of the bolt in an application, and provides an easy means of fastening components together. Just as the bolt as a whole is made to a standard, the design of the threads is specified in a separate, written standard. The bolt specification makes reference to the thread standard which contains the details of the thread geometry.

All common threaded fasteners used today are based on a 60° angle form, whether they are metric or English. In the United States, Canada, and the United Kingdom, the standard inch series fasteners are made to the Unified Thread Form, designated "UN" or "UNR." The difference between UN and UNR is a slight variation in the shape of the thread's root. In addition, the US military uses a modified UN/UNR specification called "UNJ" which has a yet another slightly different thread root profile.

BASIC 60° THREAD PROFILE



The Unified Threads are classified into groups called "thread-series." "Thread series are groups of diameter-pitch combinations distinguished from each other by the number of threads per inch applied to a specific diameter."[9] The thread series designated as simply "UN" or "UNR" is sometimes called constant, or uniform, pitch. Pitch is the term that describes the number of threads per inch. On constant pitch threads, the number of threads per inch remains constant as the diameter increases. For example, in the UN16 series, a 1/2" and a 2" bolt have the same pitch, 16 threads per inch. There are eight constant pitch series specified for the UN/UNR threads: 4, 6, 8, 12, 16, 20, 28, and 32. The standard 8-thread series (UN) was originally intended for bolts used in gasketed joints containing high pressure. It is not uncommon to see the UN8 series used instead of the UNC series for bolts above 1" diameter in flange applications.

The UNC, or coarse, series is set up so that as the bolt diameter increases, the number of threads per inch decreases. A 1"- UNC bolt has a pitch of 8 threads per inch, whereas a 2"- UNC bolt has a pitch of 4.5 threads per inch. Coarse threads are for general purpose applications and are widely used, they are easy to start (meaning to get the bolt engaged with the nut) and remove, and they are typically used for larger-diameter bolts.

UNF, or fine, series threads are different from the coarse series in that for a given diameter, they have more threads per inch than the UNC thread series. Fine threads are used in certain applications where greater strength for a given nominal size and bolt material is required, or they can be used to withstand the same loads as a coarse thread but with a shorter length. They are also used where better control of adjustments is required. Fine threads have a larger surface contact area and, therefore, more surface friction than coarse threads, and they are easier to cross-thread and damage during installation. The same goes for the UNEF, or extra-fine, series threads; they have even more threads per inch than the UNF series.

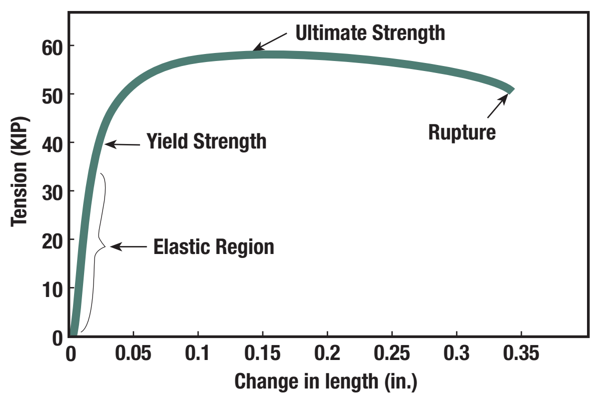
Metric threads are designated with an "M," and typically categorized as either coarse or fine pitch. ISO has widely used specifications for various types of metric threads; most of the ANSI standards have English unit specifications, and equivalent metric standards as well. The pitch for metric threads is expressed differently than it is for English standard threads.

In the metric system, the pitch is given as mm per thread, not threads per inch. An "M48 x 5" bolt has a nominal diameter of 48 mm and a pitch of 5 mm per thread. The geometry of the threads is the same for both the English and metric, just the units of measure differ.

How Bolts Work

The basic theory of operation of a standard bolt as a clamping device is relatively simple. The bolt is installed through the holes in two or more flanges; a nut is attached to one end, or the bolt is fitted into a tapped hole. As the bolt is turned, the threads cause the applied rotational force to be transferred to tensile stress which causes the bolt to stretch. Bolts can be viewed as strong (and complex) rubber bands or springs; they store energy when stretched and apply some of that energy to the joint in which they are used. Not all the energy applied by turning the nut is used to stretch the bolts. Friction between the threads, torsional stresses, and elastic interactions all absorb some of this applied energy.

TYPICAL ELASTIC CURVE FOR A BOLT



Key concepts in understanding a how a bolt works as an elastic element are tensile strength and yield strength. As a tensile stress is applied to a bolt, it begins to stretch. Up to a certain point, if the load is removed from the bolt, it will return to its original length with no permanent deformation. That point is the yield strength of the bolt. Once the bolt is loaded to a point beyond its yield strength, it begins to undergo some permanent deformation as it starts to go from an elastic to a plastic condition. Eventually the bolt reaches a point where it will not return to its original length when the load is removed. Yield strength is defined as the load at which a bolt undergoes a specific amount of permanent deformation (a percentage of its original length often specified as .2% or .5%).

The maximum load a bolt can withstand is its ultimate strength, but the bolt is plastic at this point and stores little or no elastic energy. When the bolt is loaded to its ultimate strength (and constant load is maintained), it continues to lengthen until it eventually breaks. If the load is removed, the bolt will not return to its original condition. Though the ultimate strength is higher than the yield strength, bolts are only useful, at least in flange applications, at loads below their yield strength, i.e., in the elastic condition. Once it becomes plastic, a bolt cannot provide clamping force on the flange and maintain load on the gasket.

Note that it is possible to load some bolt materials somewhat beyond their yield strength, remove the load, and retighten the bolt with no ill effects. The bolt undergoes some permanent deformation, but it's yield strength actually increases; this is called work hardening. A work-hardened bolt can be retightened and used at a stress somewhat higher than its original limit, provided it is not loaded beyond that point. The procedure can't be continually repeated to achieve a slightly higher yield strength each time; the bolt will eventually begin to yield and lose elasticity. Some plant procedures allow for the bolts to be loaded slightly over their yield strength during the installation to work harden them, but this is usually only done in situations where the joint is static and won't see a lot of thermal or load cycling.

Ideally, a bolt would be used at a point as close to its tensile strength as possible, 100% with zero permanent deformation. This would make full use of the bolt's elasticity, but unfortunately, it is not practical in the real world. Design loads on a bolt are calculated as some percentage of the bolt's tensile strength, typically between 30% to 70%. This percentage depends in part on the effect of a number of factors such as tightening methods, friction, thermal expansion and contraction, creep, vibration, external loads (applied by the piping system), and the characteristics of the bolt design and base material. All these factors can cause the actual load on the bolt under operating conditions to vary from the calculated load specified by the engineer. Take, as an example, the accuracy of the bolt tightening method. A mechanic uses a calibrated torque wrench to load a bolt to 90% of its yield strength. If a particular calibrated torque wrench has a ± 20% accuracy, it is possible the wrench would allow the bolt to be stretched beyond its yield strength (90% + 20% = 110%) without the user knowing it. This doesn't even account for the additional effects of the other factors mentioned above.

There are a wide variety of available bolt materials available, each with specific characteristics that make it desirable for certain applications. Corrosion resistance, strength, high and low temperature capability are all primary concerns. The bolt has to be able to resist attack from the atmosphere surrounding it; corrosion can cause a reduction in strength and size resulting in failure. Most materials have decreased strength at higher temperatures; the bolt has to be able to withstand the required load, with a safety factor, at the worst-case temperature of the application. Often the engineer has to pick a material with a satisfactory compromise of several of these properties.

Strength of Materials

In the English system, tensile and ultimate strength are expressed in psi (lbs/in^2); since these values are typically quite high, they may be abbreviated by expressing them in ksi (lbs/1000 in^2). There is no set system for classifying bolts between the various standards (i.e., ASTM, and SAE) in the English system (note there are publications that cross reference similar materials between standards), nor does the designation indicate anything about the bolt's properties. In order for the engineer to determine the strength of the bolt, they have to know what standard the bolt is made to and look up the information in that specification. It is extremely important to know both the exact standard and grade the bolt is made to. Often there are several grades of bolt under one specification, and it is possible that the same grade designation can be used under two different specifications, each with completely different properties. An SAE J429, Grade 4 bolt has a tensile/ultimate strength of 690/792 MPa (100/115 ksi) while an ASTM F593, Grade 4 bolt has a tensile/ultimate strength of 241/483 MPa (35/70 ksi). Both are designated Grade 4 materials, but they have completely different strength characteristics.

In the metric system, tensile and ultimate strength are expressed in MPa (Mega Pascals). There are many countries standardized in the English system, but the metric system is more universal. Unlike the English system, there is a classification system universal to most metric specifications for bolts. This system has two numbers separated by a decimal point. The first number indicates the minimum ultimate strength of the bolt, in Mpa divided by 100, and the second number indicates what percentage of the ultimate strength the tensile strength is equal to. For example, a Grade 8.9 bolt has a minimum ultimate strength of 800 MPa and a tensile strength equal to 90% of 800 MPa or 720 MPa. This system makes it possible to universally identify a bolt's strength characteristics without knowing the actual standard it is made to.

Note in some specifications, for a given grade of material, there may be separate strength properties listed for smaller and larger bolts; larger bolts have a lower ultimate and tensile strength than smaller bolts made of the same material. This is because larger diameter bolts may not be fully hardened in the center, so the core of the bolt is weaker than the outer diameter. Subsequently, though the larger bolts can withstand higher loads, their actual tensile and ultimate strength in psi, or MPa, is lower. To illustrate this point, the tensile strength of an ASTM A193 Grade B7 bolt, 25,4 mm (1") in diameter is 719 MPa (105 ksi) while an ASTM A193 Grade B7 bolt, 76,2 mm (3") in diameter is 650 Mpa (95 ksi).

Tightening Methods

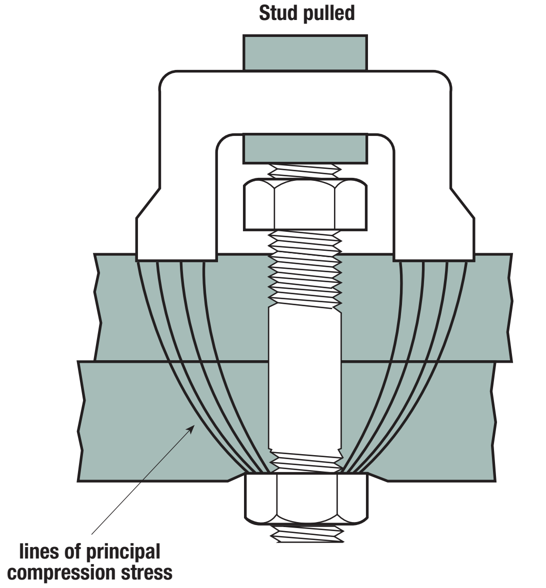
There are a number of different methods to apply load to a bolted joint, each with specific drawbacks and benefits. Installation and loading of the bolts are critical to the sealability of a bolted flange application. The gasket requires compressive stress to create a seal, and the bolts are used to apply and maintain that stress. Besides the bolt's base material and design itself, strong consideration has to be given to how the bolts are tightened. Required accuracy and the design safety factor, the costs associated with each method (usually the expense of renting or buying equipment), as well as sufficient time and available working space are all considerations when choosing a preloading method. In many cases, a combination of two or more of these methods can be used together to increase the accuracy and control of bolt loading.

The oldest method of tightening a bolted joint is the standard box or slug wrench. This is also the most inaccurate method of tightening a joint and the cheapest in the short run. The mechanic in the field puts the wrench on the bolts and tightens the joint until it feels right. This method can be ± 100 to 200% inaccurate on bolts less than 1" (25,4 mm) in diameter . With this wide variation in preload, there can be huge inconsistencies in the performance of bolted joints, depending on who performed the installation. This sometimes explains mysterious failures that occur in applications that "always worked before" and where the installation has "been done the same way for years." In today's industrial applications, demand for safety, reliability, and long-term service require more accurate methods of applying loads to bolts. With an ever-widening array of systems being developed for this purpose, the cost of achieving consistency in the loading on a bolted joint is decreasing. The cost of tightening a joint by "feel" can seem cheap to start, but quickly increases when lost production time, equipment damage, or personal injury results from a bolt breaking.

The use of torque wrenches is probably the most common method of applying preload to a bolt today, though other, more accurate technologies are now available and becoming more widespread. Torquing of bolts is relatively easy and is also cost-effective, hence its continued use as a means of tightening joints. Manual torque wrenches are used for lower values on smaller bolts. These can be digital, click type, dial, or gauge types. Accuracy can vary depending on the manufacturer, design, type of joint, and whether or not the wrench is calibrated; ± 30% or more can be encountered on gasketed joints, less on metal-to-metal joints. Higher torque (close to 7,375 mm or 10,000 ft. lbs.) can be achieved with multipliers in conjunction with regular manual wrenches (with somewhat less accuracy) or with hydraulic wrenches which have a ± 10% accuracy. Air-powered tools and impact wrenches have less accuracy than manual or hydraulic torque wrenches, but they are considerably faster; speed is a factor in manufacturing environments. Controlling variables such as thread friction and the condition of the bolts themselves is critical in keeping this method as accurate as possible. Loss of bolt strength due to torsional stresses is a factor to keep in mind.

Bolt tensioning works a little differently than torquing. Two different methods, hydraulic tensioning and bolt heaters, will be briefly discussed here. Both methods are typically used on applications with larger bolts. Basically, the hydraulic tensioning device has a threaded collar fitted onto the end of the stud, with the nut already threaded on the bolt; the device has supports that rest on the surface of the joint to be tensioned. Load is applied by pumping hydraulic fluid into the unit; the fluid applies force to the collar, pushing it away from the flange and stretching the bolt. Once the bolt is stretched to the proper limit, the nut is run down the bolt until it sits on the flange face, the tensioner is shut off and vented, and the nut assumes the load. The nut is run down onto the flange with a torque wrench, so there are some of the same concerns with thread friction and bolt and flange conditions as there are with straight torquing.

HYDRAULIC TENSIONER



Bolt heaters achieve basically the same thing in a completely different manner. Heating rods are inserted in holes drilled down the center of the bolts. Heat is applied to the bolts, causing them to grow in length. Once the length increases the desired amount, the nut is tightened to a specific torque value and the heaters are shut off. As the bolt cools, it shrinks, resulting in tension. Heaters are relatively cheap to operate, but require skilled operators and time. Both methods make it possible to tighten a number of bolts at the same time to the same load. This helps ensure even compression on gaskets and flange faces, speeds up the process of bolting up larger pieces of equipment with a number of bolts, and reduces problems with elastic interactions between bolts.

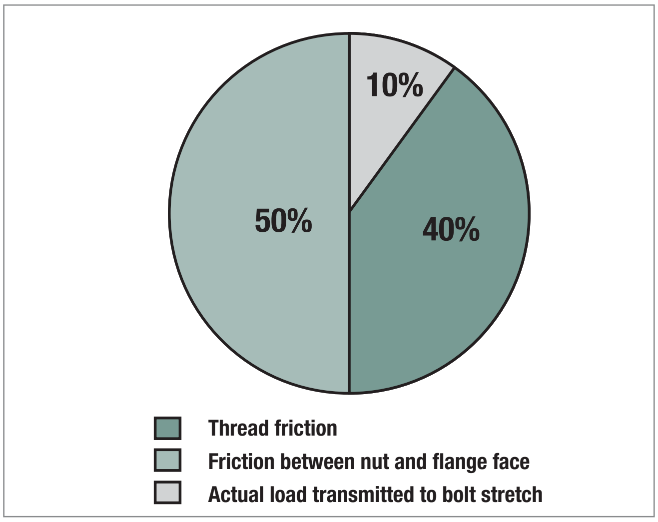
Another method of loading a bolted joint is to use a wrench to apply load to the bolt, but instead of measuring torque, the actual change in length of the bolt is measured. By knowing the properties of the bolt material to be used and its dimensions, the change in length from unloaded to the loaded condition can be calculated. Once this figure is known, the bolt is tightened until the calculated length is achieved. Measurements can be done using depth micrometers on bolts with predrilled holes down the center or, on short bolts, a C-micrometer can be used. Mechanical measurement is less accurate on shorter bolts; shorter bolts stretch less than long ones at a given load so the measuring device has to be more capable of accurately measuring the smaller changes in length. A more precise method is to use ultrasonic measuring devices. These devices electronically measure the change in bolt length, and this is one of the more accurate ways to determine load on a bolt. It is used in critical applications like nuclear reactors, heat exchangers, and other applications where high accuracy is essential. The increased accuracy of these methods allows a bolt to be tightened to a higher percentage of its yield strength, thereby using the bolt to its fullest potential. Note that special attention must be paid to temperature and material quality and condition when using this method.

Factors Affecting Bolted Joints

When considering a bolted joint, there are three components to consider: the flange, the gasket, and the bolts. The characteristics of each of these components can affect the overall performance of the application; flanges and gaskets are discussed in separate sections. The following are some of the factors related primarily to the bolts.

Friction

Friction is one of the biggest concerns in a bolted joint, specifically those applications where a torquing method of some type is used to stretch the bolt (see Tightening Methods). Since torquing methods are the most common means of loading a bolt, friction is one of the critical factors in a majority of bolted joints. There are three general ways friction can cause problems in bolted joints: it prevents a good portion of the applied force from being used to stretch the bolt, it decreases the accuracy with which all the bolts in an application can be loaded, and it causes galling of the threads.



A commonly accepted general statement says that only 10% of applied torque is transferred to force that actually stretches the bolt. Various friction losses account for the other 90% of the applied force. The friction between the threads of the bolt and nut accounts for roughly 40% of the losses, and the other 50% occurs between the nut and the mating surface on the flange. The word “roughly” is used here because, in actuality, some of the friction forces are transferred to heat and other forms of energy. That figure of 10% can be drastically reduced by any number of factors such as poor surface finishes, lack of a proper lubricant, etc. This narrow margin of useful load has to be protected so the bolt can do its job, and to do this, care has to be taken to keep the effects of friction to a minimum.

Not only does friction reduce the amount of load that can be applied to the bolt, it also reduces the consistency with which a load can be applied. A mechanic using a torque wrench cannot tell which bolts are subjected to more friction; the wrench indicates the torque applied to the bolt, not the actual tensile load the bolt is subjected to. Even if all the bolts on a single flange are initially torqued to the same value, friction can cause a wide variation in loads from bolt to bolt. The result is uneven compression on a gasket, damaged flanges, or the compression on the flange and gasket is nowhere near the minimum required to achieve a seal.

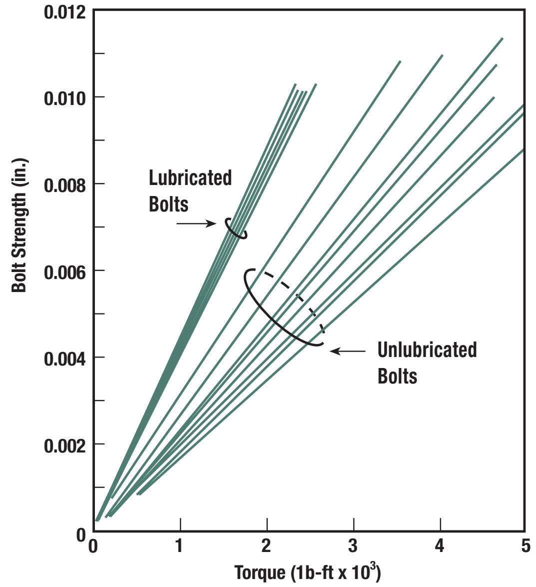
If enough pressure is applied, it is possible to actually fuse the threads of a nut and bolt assembly and also the contact area between the bolt head (or nut) and flange. This is called galling; the load between the surfaces increases to a point where the parts actually weld themselves together. If galling occurs during assembly, a dramatic drop occurs in the amount of applied torque that is actually transferred to tensile stress. Most, if not all, of the applied energy is absorbed by friction as the metal parts fuse together. If severe galling occurs, the threads can no longer move relative to each other and the bolt can no longer be stretched via torquing. Increasing the torque applied to a galled bolt increases torsional stresses but doesn't create any increase in tensile stress in the bolt. Galling can be reduced by decreasing the thread pitch, by changing the bolt or nut material, and by using a lubricant.

These friction losses are caused by a number of things. Surface finish is one; rough finishes can create higher friction in both the threads and the contact area between the bolt head (or nut) and flange. This is one reason flat washers are sometimes recommended under the head of the bolt and the nut; they provide a flatter, smoother surface resulting in lower surface friction.

The base material of the nut and bolt is another consideration. Given the same size and thread design, there can be considerable differences in the coefficient of friction of a bolt depending on its base material. Coating or plating a bolt can dramatically change its coefficient of friction, even if the base material is the same. Also consider that the nut is not necessarily made from the same material as the bolt. To decrease the coefficient of friction and to reduce the chance of galling, the engineer may specify a different nut material (with suitable strength characteristics) instead of one made from the same material as the bolt. For example, an ASTM A193, grade B7 bolt a high-strength, low alloy steel, is sometimes used with a 300 series stainless steel nut to reduce galling.

A proper lubricant is one of the primary tools to reduce the problems caused by friction. In most instances, the use of a lubricant on a bolt lowers the coefficient friction when compared to the same bolt assembled dry. As coefficient of friction decreases, more of the torque applied is converted to tensile stress. If a mechanic uses a torque wrench to tighten two bolts, one with lube and one without, the unlubricated bolt will have a lower load than the lubricated bolt, even though the torque wrench indicates both are torqued to the same value. On the other side of the coin, the lower coefficient of friction makes it possible to apply the same load to the lubricated bolt with a lower torque than the unlubricated bolt would require.

Lubricants increase the accuracy of the torquing method used. The engineer calculates the torque required to achieve a certain load on a bolt, assuming a certain coefficient of friction which can vary widely on a dry bolt. The use of a lubricant increases the consistency in the coefficient of friction, and reduces the chances of galling. This consistency makes it more likely that the desired preload is achieved at the calculated torque value. On a flange with 12 bolts coated with a proper lubricant,



it is more likely that the applied load will be closer to the calculated value, ensuring that the gasket stress is sufficient to maintain a tight seal. Not only is the achieved load closer to what is anticipated, but the variation in load between the bolts in the flange is reduced, ensuring uniform gasket compression. Without the accuracy provided by lubricants, the engineer and mechanic would be shooting in the dark trying to achieve bolt loads.

The choice of lubricant is based on, among other things, the operating temperature, media the bolt could be exposed to, and coefficient of friction of the lube. If a lube is applied beyond its recommended temperature, the coefficient of friction and its lubrication properties will change. Note that the lubricant may start to break down at higher temperatures, so the coefficient of friction is usually only accurate at or around room temperature. Some applications require that only certain lubricants can be used because of safety or operational concerns. For example, nickel anti-seizes and other similarly made types of lube cannot be used if they are going to be exposed to chlorine, because of the risk of chemical attack or a severe reaction. The coefficient of friction for a lubricant can vary, the degree of which depends on its design. A product with minimal variation between the minimum and maximum coefficient offers the best consistency. Better efficiency, meaning lower friction losses, is achieved with lubricants having the lowest coefficient of friction.

Nut Factor and Coefficient of Friction

When describing a bolted joint, it is common to hear both of these terms used interchangeably, but in fact they are two different things. The coefficient of friction of a material describes the properties of just that specific material, regardless of the application. For a lubricant, the coefficient is usually determined by a test where the lube is spread between two horizontal plates; a special apparatus is used to measure how much force is required to move the plate, and the coefficient of friction is then calculated. The coefficient of friction for a material considers only its lubricity, nothing else.

A nut factor, sometimes denoted with a "K" in calculations, describes the properties of a threaded fastener, not just the material. It is an experimentally-derived constant that encompasses all the factors that affect how much of the applied load is transferred to actual stretch. It is basically a "fudge factor" that considers the effects of not only the coefficient of friction between the threads and other contact surfaces, but also the fastener condition, thread design, contact area, and any other hidden factors. Given that fact, the nut factor is going to be higher than just the coefficient of friction for the lubricant or bolt material. A nut factor is determined via experiments on the fastener and, in order to be truly accurate, it should theoretically be determined for each application by testing in the lab. Note that the values given in product literature, whether a coefficient of friction or a nut factor, are usually the mean tested value; there can be variations above or below that printed value.

Relaxation of Material

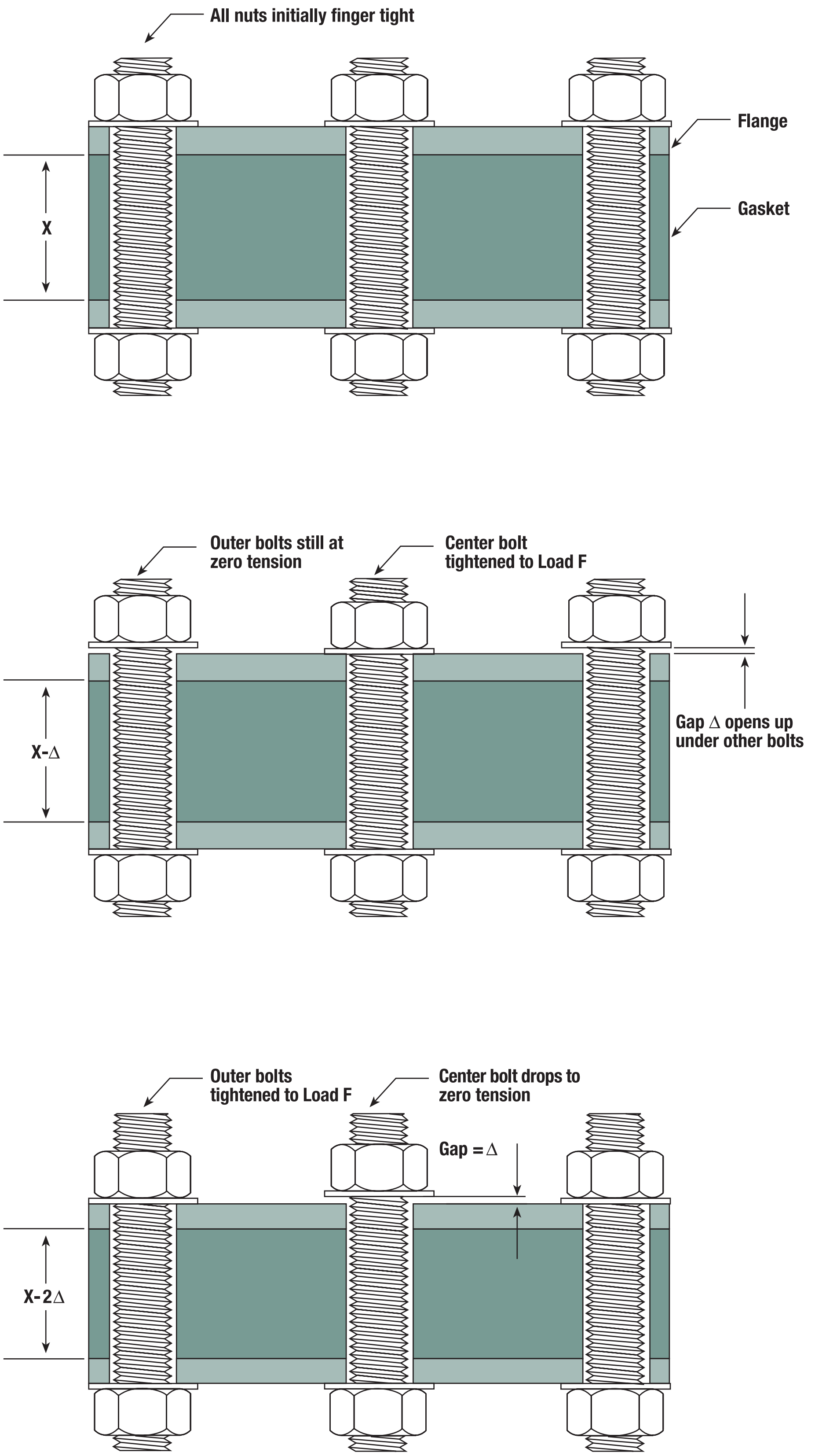
The surface of any manufactured component, even ones with fine finishes, is never perfectly flat. Take a bolt as an example. If you look closely at the threads with some type of magnification, the surface will have peaks and valleys. When the nut is turned down the threads and torqued, the load is first carried by the peak points of the thread surface. At some point, as load increases on the small area of the peak points, those points begin to deform and flatten in the same way a gasket material creeps, resulting in more thread surface area carrying the applied load. This relaxation of the threads is called embedment and is a normal process for bolts. Some installation techniques such as ultrasonic measurement can show the effects of relaxation, allowing the installer to account for them; torquing and tensioning techniques do not show the effects. In most cases, the load lost as the bolt creeps can be accounted for by modifying the installation procedure. Allowing the bolts to sit for a period of time and reapplying the load will help account for embedment. Note that after a new bolt relaxes under load and the thread area is flattened, it will undergo much less embedment the next time it is tightened. Embedment occurs on not only the bolt threads but also on the nuts and washers.

Elastic Interactions

Not only do fasteners undergo embedment, they also interact with each other as they are tightened. In any bolted joint, as one bolt is tightened, the ones next to it relax. Picture three bolts in a row on a rectangular flange. Tighten the first bolt to a specific torque, then the second bolt to the same torque, and then the third. After tightening the last bolt, a check on the other two bolts will show a reduction in load. Essentially, the first bolt is loaded to a particular point, but the second bolt picks up some of that load, loosening the first bolt and, subsequently, when the third bolt is tightened, the first and second bolt lose some preload as the third bolt picks some of it up.

It is important to realize that elastic interactions are a factor in bolted joints, and if not properly accounted for, they can cause great difficulty in achieving a proper seal on a bolted joint. Using a good installation procedure goes a long way to alleviating the affects of these interactions. Using specific tightening patterns and tightening the bolts in increments of the recommended load are useful techniques, especially when using a torque wrench. Tightening all the bolts to the recommended torque at the same time will also decrease the effects of these interactions. Some tensioning equipment can be used to load a number of bolts at once.

ELASTIC INTERACTIONS



Tolerance, Component Fit, and Condition

If a bolt hole through a flange is machined with too small a clearance, there can be an interference fit between the bolt and the hole. An interference fit means additional friction and a reduction in the percentage of applied force being transmitted to bolt stretch. Specifications for pipe flanges dictate the bolt hole diameter and the bolt size for every flange, so problems caused by improper dimensions are more likely to occur in custom-made flanges. An interference fit can also occur if a mechanic inadvertently installs the wrong sized bolt in the flange, or if the flanges are not properly aligned or parallel. An interference fit can also occur on certain flanges with tongue and groove, or male and female faces. Again, pipe flange specifications usually specify clearances and tolerances, but a bad fit is possible if the faces are deformed or the flanges are custom-designed with improper clearances or machining tolerances.

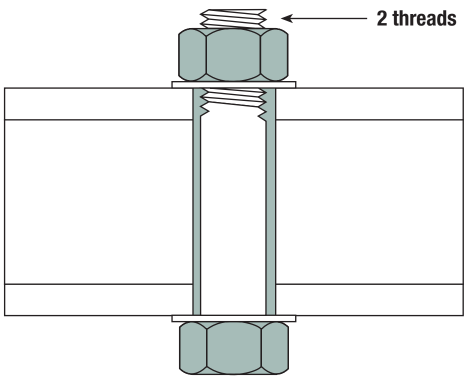
Bolts are made with certain thread allowances and tolerances. The thread allowance controls the range of dimensions for both male and female threads. It is what controls the fit between the two, whether it be tight or loose. On top of this, there is a tolerance which allows dimensional variations due to machine inaccuracies. The tolerance is always allowed in the direction of less material only; this ensures that there is not an interference fit between the threads, especially on those with an extremely tight fit. Too loose a fit on the bolts results in a smaller thread area actually supporting the load, so care has to be taken to install bolts with the proper allowances, or the bolt may not carry the fully intended load.

The condition of the threads, regardless of the design, is a big concern and should be checked any time a joint is assembled. If the bolt threads have been subjected to stresses beyond the yield point and have undergone too much plastic deformation, there will not be a proper fit between the threads of the bolt and nut or tapped hole. If the threads are deformed, the allowance between them is likely to be nonexistent; an interference fit means higher thread friction which decreases the percentage of applied torque being converted to bolt stretch. A quick, general check to see if the bolt threads are deformed can be done by running a new nut down the bolt, or checking them with a proper thread gauge.

Thread Engagement

Just like bolts, the nuts are made to specifications that dictate their dimensions, materials used, and tolerances. The bolting standard should indicate the appropriate matching nut designs, referencing them by specification. A general rule of thumb says that there should be at least 2-3 threads of a bolt showing past the nut before an actual load is applied. This ensures that there is the full engagement between the threads of the nut and bolt. In most standard designs, the nut design is set up so that it withstands more stress than the mating bolt. This is so that if excessive load is applied to the bolt and nut assembly, the bolt body will break before the threads strip. It is difficult to see if threads have stripped because they will still support load after failing, and they aren't visible to the naked eye. If the body of the bolt breaks, there is no difficulty in detecting failure. Having less than full thread engagement means the bolt load is distributed over a smaller surface area, possibly resulting in stripping of the threads and failure of the nut. Typically, standard hex nuts have a height equal to .8 times the bolt diameter; heavy hex nuts have a height equal to the bolt diameter. For engagement of a bolt in a tapped hole, the general rule is 1 to 1.5 times the bolt diameter.

MINIMUM THREAD ENGAGEMENT



Thermal Expansion and Contraction

The effects of thermal expansion and contraction are often overlooked in bolted joints. Temperature has several effects on bolts; an increase in temperature causes them to physically grow and also causes a drop in their overall strength. Because of this, the actual load on a bolt will fluctuate as the temperature varies, making it possible to yield a bolt without applying any additional load to it. If a bolt is torqued to 90% of its yield strength at room temperature and then heated to its maximum allowable service temperature, the yield strength of the material drops, possibly below the load that has been applied, resulting in failure. The engineer has to consider the properties of the bolt at its actual operating temperature and calculate the required torque value based on the bolt's strength at that temperature. Prolonged exposure to high temperatures also can cause creep relaxation of the bolt. The effects of high temperature creep on the bolted joint as a whole can be kept to a minimum simply by carefully choosing a bolt material that is designed for high temperature service.

The effects of thermal expansion and contraction become even more complex when looking at the flange joint as a whole. Not only does the bolt expand and contract as the temperature changes, but the flange does as well (the gasket also undergoes this expansion and contraction, but for the sake of simplicity, the focus will be on the bolt and flange). Each component is made from a different material, each with a different rate of expansion and contraction. As the temperature increases, the bolt grows in length and the flange increases in thickness, but the rate at which each grows is different. Depending on the material characteristics, one grows faster or slower than the other. Also note that the dimensions of each component are different (the bolt length is more than the flange thickness), so the amount that each expands is going to vary.

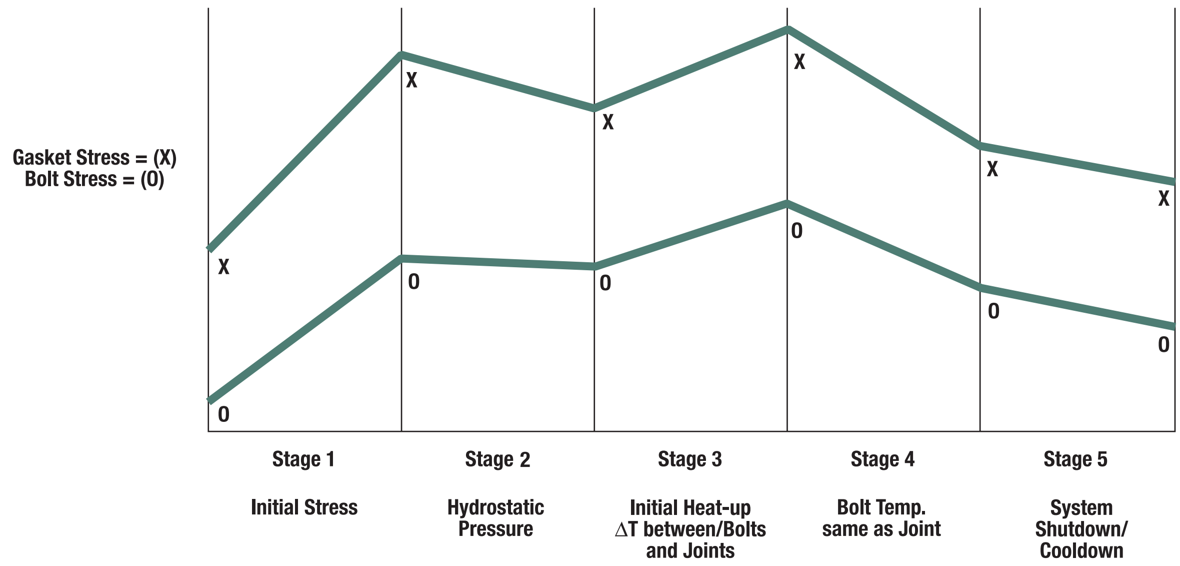
The drawing below depicts the effects of thermal expansion and contraction in a gasketed flange in a heat exchanger; however, the events shown are common to all gasketed flanges in applications where thermal cycling takes place. Stage 1 shows the bolt and gasket stresses increase as the flange is assembled and the bolts is torqued to the required value. Stage 2 shows the effects of hydrostatic end thrust. At startup of the system, the internal pressure increases, pushing the flange faces apart and partially unloading the gasket; note that the bolt load remains relatively constant and does not reveal any decrease in gasket load.

For the purpose of this discussion, assume the bolt grows at a slower rate than the flange. In Stage 3, heat is applied to the joint. When the temperature begins to increase, the flange thickness increases faster than the bolt length. The flange starts to exert more force on the rest of the joint as it grows; this causes a rise in the stress on the gasket and the bolt. As the flange reaches full temperature, it stops increasing in thickness; this is the point where the peak stress on the gasket occurs.

Stage 4 shows the bolt continuing to grow in length while the flange has finished growing; the result is a gradual decrease in the load on the gasket and bolt. When the bolt stops growing, the stress on the gasket reaches a steady state. The relative decrease in gasket stress from the beginning of Stage 3 to the end of Stage 4 is a result of gasket creep at elevated temperature and the fact that no gasket recovers 100%. This effect becomes even more pronounced as the gasket thickness increases.

Remove the heat source, and the opposite of the heating cycle takes place. This is shown in Stage 5. The flange and bolts begin to contract, with the flange shrinking faster than the bolts. Remember the bolts were stretched when the initial load was applied. As the flange contracts, the

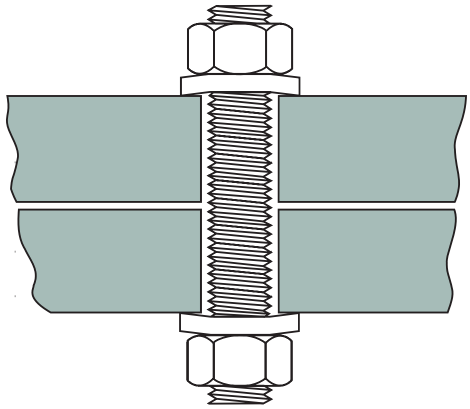
TYPICAL EFFECTS OF THERMAL EXPANSION AND CONTRACTION



energy stored in the stretched bolt causes it to contract with the flange, but as it does, the bolt starts to lose that stored energy. As the bolt load decreases, the stress on the gasket also drops. This is typically the point where a leak develops. At some point, the gasket stress may drop below the minimum required to seal and the media makes its way through or around the gasket. The gasket may recover somewhat, meaning it grows in thickness as the stress applied to it decreases, but the amount the gasket recovers is usually negligible compared to the change in the flange and bolt dimensions. With the drop in pressure, it is likely that the leak that has developed may not be discovered until the system is restarted.

One means of decreasing the problems associated with thermal cycling in flange applications is to spring load the flange bolts. This is sometimes called Live-Loading and uses flange discs to provide additional elastic energy to maintain load on the bolts as thermal cycling takes place. The flange discs are used because they can provide high loads in a small, compact package. Flanges typically don't have much radial distance between the center of the bolt hole and the outer edge, so it's necessary to have a spring that fits in a small space. The flange discs are compressed during the torquing procedure and at any point where the bolt contracts or load starts to decrease, the discs begin to uncompress, applying their stored energy to the flange. This maintains load on the gasket as the various flange components undergo thermal expansion and contraction, preventing a leak from developing. The actual arrangement of the flange discs on the bolt depends on the load and amount of additional travel (simulated bolt stretch) they are required to maintain.

FLANGE DISCS ON A FLANGE BOLT



Other Factors

Also consider the effects of weight on a joint. If a vertical heat exchanger has a heavy end cover, some of the preload on the bolt will be used just to support the cover and draw it into place. A lower percentage of the applied load will be transferred to force actually compressing the gasket. In other words, the torque wrench will indicate that the correct torque is applied to the bolt, but the compression on the gasket will be less than the engineer calculated. This could, in some cases, lead to leakage from a gasketed joint where the installation was done correctly and the application appears to have no complicating factors.

SECTION 4

HEAT EXCHANGERS

Overview

Heat exchangers can be the most difficult plant gasketing application to seal. They are often perceived as just another static flange application when, in fact, they are much more complicated than that. By design, heat exchangers operate in such a way that there can be a fairly wide range of operating temperatures occurring at the same time at various points on the equipment; this causes varying degrees of expansion and contraction of individual equipment components, making it a dynamic rather than static application. Heat exchangers are often subject to severe thermal shocks and sometimes several thermal cycles a day, particularly if they are installed outdoors and are exposed to the weather. These tough operating conditions make maintaining a seal on the flange joints difficult. A joint that starts leaking causes process inefficiency, product loss, and system downtime which ultimately means higher costs. To prevent catastrophic failures from occurring, a better understanding of the heat exchanger system is required, with a focus on proper bolting practices, gasket selection, applied gasket stress, and gasket load distribution during thermal cycling.

A service may be single-phase (such as the cooling or heating of a liquid or gas) or two-phase (such as condensing or vaporizing). Since there are two sides to a STHE (Shell and Tube Heat Exchanger), this can lead to several combinations of services.

Broadly, services can be classified as follows:

- Single-phase (both shell side and tube side)
- Condensing (one side condensing and the other single-phase)
- Vaporizing (one side vaporizing and the other side single-phase)
- Condensing/vaporizing (one side condensing and the other side vaporizing)

The following nomenclature is usually used:

Heat exchanger: both sides single-phase and process streams (that is, not a utility).

Cooler: one stream a process fluid and the other cooling water or air.

Heater: one stream a process fluid and the other a hot utility, such as steam or hot oil.

Condenser: one stream a condensing vapor and the other cooling water or air.

Chiller: one stream a process fluid being condensed at sub-atmospheric temperatures and the other a boiling refrigerant or process stream.

Re-boiler: one stream a bottoms stream from a distillation column and the other a hot utility (steam or hot oil) or a process stream.

Heat exchangers are crucial for the energy efficiency of a process. The average refinery nowadays uses only 5-6% of the fuel throughput for its own energy use and this is largely thanks to the use of heat exchangers that allow excess heat to be used to pre-heat other products.

Manufacturers of exchangers are found in all locations and are typically relatively local businesses.

There are a large number of different heat exchanger designs on the market today. The two most popular are tubular types and plate types. Plate type heat exchangers are relatively inexpensive and easy to manufacture, and they are compact. They have specially designed corrugated plates which are stacked on top of each other and compressed sandwich-like between two thick end plates; fluid flows between the stacked plates, and they are isolated from each other by special gaskets applied to the plates. These gaskets are specially molded gaskets specifically for the heat exchanger. Therefore, this type of heat exchanger falls out of the scope of the Chesterton program.

The tubular designs are most common, with the most popular type in this category being the shell and tube design. The shell and tube heat exchanger has the advantage over other designs in that it can be made to fit in relatively small areas (though typically larger spaces than plate types), and it can withstand higher pressures and temperatures than other designs. The Tubular Exchanger Manufacturers Association, known as TEMA, provides minimum standards and data relating to fabrication, tolerances, installation, operation, maintenance, and mechanical design of shell and tube exchangers. The mechanical design information in the TEMA standards provide information specific to heat exchangers and supplements the major pressure vessel design codes which are American Society of Mechanical Engineers, (ASME), Section VIII, Division 1: "Rules for Construction of Pressure Vessels"; British Standards Institute, BS 5500, "Specification for Unfired Fusion Welded Pressure Vessels"; and the European Pressure Equipment Directive.

Essentially, a shell and tube heat exchanger consists of two separate pressure vessels contained in one package that allows heat transfer between two fluids without any physical contact between them. The basic design has a bundle of tubes contained within a cylinder called a shell. One fluid flows through the tube bundle, transferring heat through the tube wall to the other fluid which flows around the outside of the bundle. A tube bundle is simply a group of metal pipes, the ends of which pass through holes in a metal plate called a tubesheet. The ends of the tubes are fastened to the tubesheet, either by welding or by flaring them with special tools. This forms the tube bundle into a rigid body, and it seals the tubes tightly into the sheet, preventing fluid leakage through it.

The portion of the heat exchanger that contains the fluid moving through the inside of the tube bundle is referred to as the "tube-side." The tube-side fluid comes into a header, often called the stationary header, at the front end of the heat exchanger and passes through the tubes into the rear header, (except on a U-tube type) that either redirects the fluid back through the tubes or out of the heat exchanger. The number of passes and the path of the fluid through the tubes is controlled by partitions built into the header(s); the fluid may be directed one or more times through separate sections of the tube bundle to achieve the required amount of heat transfer. Headers can be made in two basic ways. A one-piece bonnet header with the partitions cast into it is the cheaper design, but the entire header must be removed to service the tubes. To do so requires that the inlet and outlet connections on the header be disconnected first, a time-consuming procedure. A pillbox or bobbin-type of header is a cylinder with the partitions built in; one end is bolted or welded to the tubesheet and the other end has a bolt on the cover. This design allows inspection, servicing, and cleaning of the tube bundle by simply removing the end plate; the inlet and outlet flanges do not have to be removed.

The portion of the heat exchanger that contains the fluid moving outside the tube bundle is called the "shell-side." The shell-side fluid flows into the shell, passes over the tube bundle, and out of the heat exchanger. Several inlet and outlet flanges may be used to create a specific flow pattern of the fluid through the shell. In many applications, baffles are installed to control the velocity of the shell-side fluid across the tubes; they also serve to support the tubes, keeping them rigidly grouped together in a bundle. Gasketed flanges are most often used to separate the tube-side and shell-side of the heat exchanger, though in some cases, a packed stuffing box may be used to help account for thermal expansion and contraction.

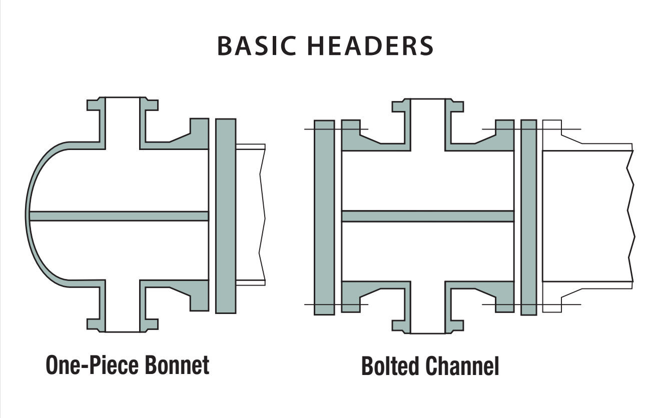
Heat Exchanger Types

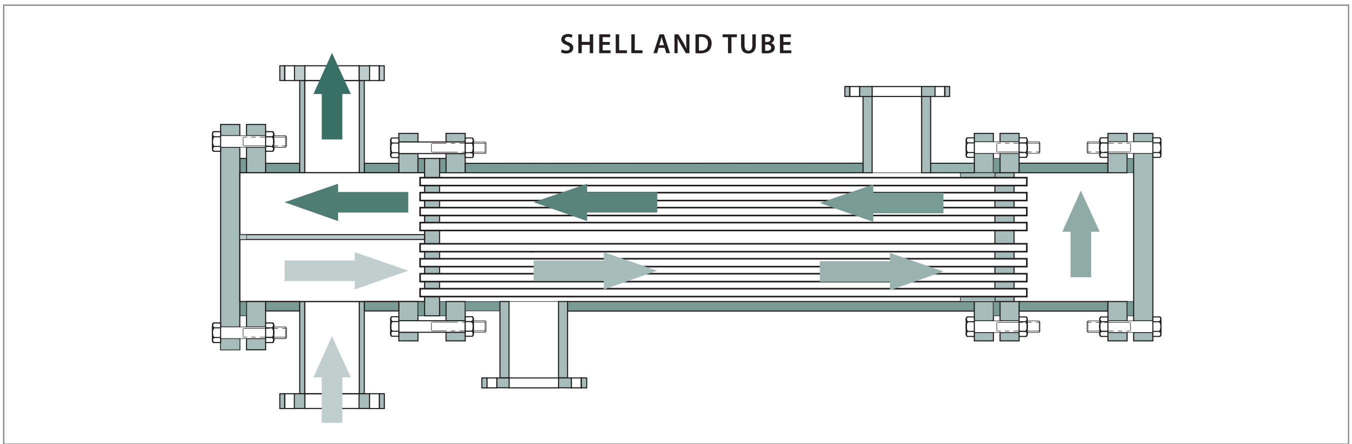
Shell and tube type heat exchangers have several basic designs, the choice of which is dependent on factors such as the media being sealed, ease of maintenance, material and manufacturing costs, and the safety factor required.

There are 4 basic types:

- Fixed Tubesheet (box type)
- U-tube
- Split Backing Floating Head
- Pull-Through Floating Head

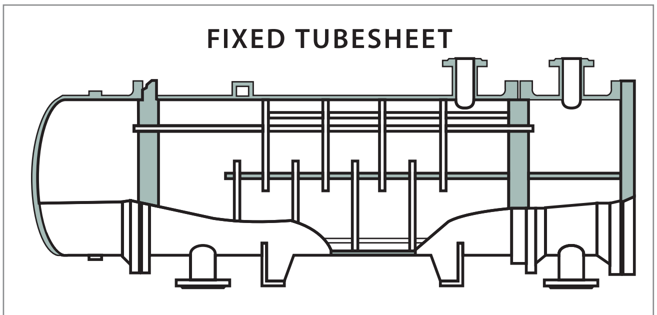
There are other special designs for certain applications, but the focus here will be on the more common types.





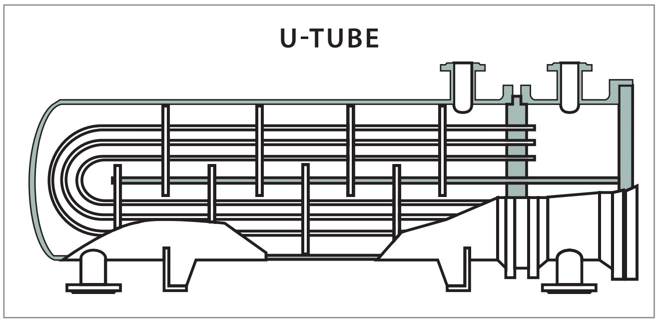
Fixed Tubesheet

A fixed tubesheet heat exchanger, sometimes called a box type, has both tubesheets welded directly to the shell. This is a simple design that has few gasketed joints, but it has some disadvantages. The OD of the tube bundle can only be cleaned chemically because the tubesheets can't be removed; the ID of the tubes can be mechanically cleaned. Note that this type of heat exchanger doesn't have a means to allow for the difference in thermal expansion between the tube bundle and the shell as the exchanger undergoes temperature changes.



U-tube

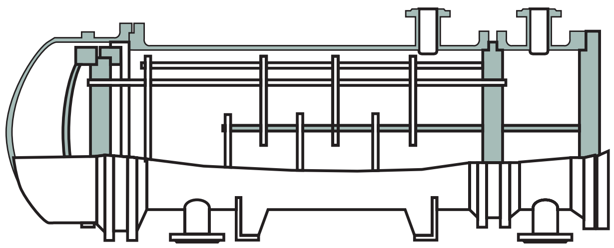
The U-tube heat exchanger has only one tubesheet, compared to two in the fixed tubesheet design. The tubes have a "U" shape and, because both ends are connected to one tubesheet, they can expand and contract independently of the shell. This heat exchanger only has one or two gasketed joints, depending on whether the header is a manifold with the partitions cast into it or a bolt-on channel with an end-cover. Fewer gasketed joints mean fewer possible failure (leakage) zones. A drawback to this design is the inherent difficulty in removing the tube bundle. Because of their U-shape, the inside of the tubes can only be cleaned using a chemical process, so certain coagulating materials or fluids that can precipitate scale are not used on the tube-side of the exchanger.



Split Back Floating Head

The split back floating head heat exchanger has one end of the tubesheet fixed with respect to the shell, while the other tubesheet end "floats," so the differential movement due to thermal expansion and contraction of the shell and tubes is not an issue. Removal of both the stationary, rear and floating heads allows removal of the tube bundle from the shell, providing access to the inside and outside of the tube bundle. This allows the shell- and tube-side of the exchanger to be mechanically cleaned, a much easier, less costly method than chemical cleaning. Unlike the U-tube type of exchanger, the floating head type exchanger can be used in fouling fluids because it can be inspected and cleaned more easily. There are two drawbacks: both ends of the heat exchanger have to be opened to remove the tube bundle and there is an internal joint which requires shutdown and disassembly if the gasket fails. Because of this, in applications where any possible mixing of the two fluids could cause a hazardous condition or reaction, this type of exchanger is not recommended.

SPLIT BACK FLOATING HEAD

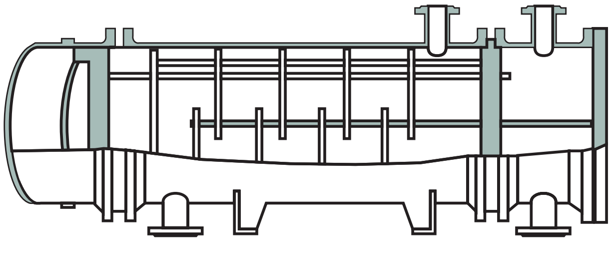


Pull-through Floating Head

The pull-through floating head type is a slightly different version of the split backing ring types.

The pull-through has a different joint setup on the floating head that allows the entire tube bundle to be pulled out of the shell by removing the stationary header only, just like a U-tube type. This provides an added measure of convenience and allows for more rapid disassembly and cleaning of the shell-side without complete disassembly of the tube-side flange joints.

PULL-THROUGH FLOATING HEAD



TEMA groups heat exchangers into three classes, Class R, Class C, and Class B. Class R covers petroleum refining and related heavy-duty applications. Class C encompasses general process and commercial applications of moderate operating parameters. Class B is for chemical service applications. TEMA also has a standard means of identifying the size of a heat exchanger according to the inside diameter and length of the shell. A designation of 1220/4064 has an inside shell diameter of 1220 mm and a length of 4064 mm; 48/160 indicates the heat exchanger shell has a 48" ID and is 160" long.

TEMA also provides a three-letter type designation system that defines the exchanger construction completely and avoids confusion. The first letter defines the stationary head type and how it is attached to the tubesheet and shell. The middle letter defines the shell type, or shell-side nozzle arrangement; it details how many nozzles are on the shell side and how the fluid generally flows. The last letter defines the rear head type; these are similar to the stationary head types, but have several extra different configurations for floating head exchangers.

There are pipes, called nozzles, welded to the header and the shell of the heat exchanger that serve as the inlet and outlet for the two working fluids. The nozzles are connected to the piping system either by welding them or by using bolted flanges; the most common flange types are weld neck or slip-on (see Flanges). These nozzle flanges are made to the same design standard (ANSI, DIN, JIS, or BS) as the flanges of the piping system that serve the heat exchanger.

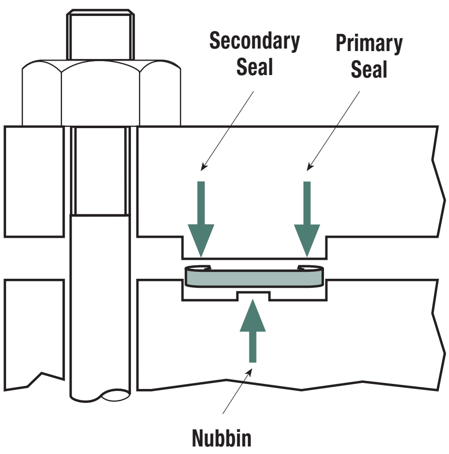
Except for the nozzles, there are no set dimensional specifications for the flanges on a heat exchanger, so identifying the sealing joint members for a specific heat exchanger is important. There can be flanged joints that connect and seal the cover to the stationary channel, the stationary channel or bonnet to the tubesheet, the tubesheet to the shell, and the shell to the rear channel; if there is a floating head, there is also a gasketed joint between the cover and the floating end of the tubesheet. The total number of these flanges can vary from one to six, depending on the design of the heat exchanger.

The flanges for the headers and shell are attached to the equipment using methods similar to pipe flanges. The flange face is prefabricated (and can have extremely large diameters on the bigger heat exchangers) and is then attached to the shell or header usually by welding. Slip-on, weld neck and lapped flanges are the most common (see Flanges). The flange faces designs are similar to the standard pipe flanges; the most common are either tongue and groove, flat face and groove, or male and female. Some of the tongue and groove, and flat face and groove facings also have a 0.4 mm (1/64") high ridge, called a nubbin, in the center of the groove theoretically to increase sealability of the gasket, but it is only for use with filled metallic-type gaskets. Spiral wound, Steel Trap™ and sheet gaskets cannot be used on flange faces with nubbins unless the nubbin is removed or the gaskets are specially made to fit inside or outside the nubbin.

PROPERTIES OF FOUR BASIC HEAT EXCHANGER TYPES

Exchanger Type	Fixed Tubesheet	U-Tube	Split Back Floating Head	Pull-Through Head
Differential movement between tubes and shell	No	Yes	Yes	Yes
Tube bundle removable	No	Yes	Yes	Yes
Individual tube replacement	Yes	Only outside tubes	Yes	Yes
Tube Cleaning- Inside	Any	Chemical	Any	Any
Tube Cleaning- Outside	Chemical	Any	Any	Any
Internal Gaskets	No	No	Yes	Yes

TONGUE AND GROOVE FACING WITH NUBBIN

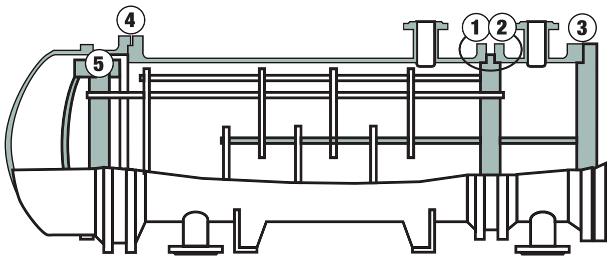


Heat exchanger gaskets come in a wide variety of shapes and styles. The gaskets can be simple rings, or more complicated crossbar designs, depending on their location. Ring gaskets can usually be found between the tubesheet and shell, and the rear header and shell. The special cross bar design gaskets are usually found between the header (stationary, floating, and rear) and tubesheet, and the channel and cover. The crossbars create the seal between the header's partitions and the tubesheet. These gaskets with cross bars prevent mixing of fluid between the partitions in the headers; even though the same fluid is present throughout the tube-side of the heat exchanger, the temperature of the fluid in each partitioned area varies depending on the number of passes that have been made through the tubes.

Because there is no written dimensional standard, heat exchanger gaskets can't be ordered by nominal size and pressure class like gaskets made to fit standard flanges (e.g., ANSI B16.5). Specific dimensions have to be measured (or taken from a drawing), and other factors such as bolting materials and quantity of bolts have to be determined. Each application has to be analyzed individually to help ensure proper sealing is achieved. To assist in gathering this required data, Chesterton has designed a Heat Exchanger Data Collection Sheet (see Appendix E). The form also shows the various configurations of gaskets made for heat exchangers and the required dimensions that must be known in order to ensure the best achievable heat exchanger sealing system is recommended.

Using the information gathered on the Heat Exchanger Data Collection Sheet, Chesterton can provide gasket recommendations, recommended torque values, and installation techniques. The information is useful in determining possible problem areas in the bolted joints on heat exchangers. The factors associated with the flanges, bolting, and gasketing mentioned in previous sections are considered to provide the end-user with a complete sealing solution on these difficult applications.

TYPICAL HEAT EXCHANGER SEALING LOCATIONS



1. Shell and Tubesheet
2. Tubesheet to Channel Box
3. Channel Box Cover
4. Shell Cover
5. Floating Head

SECTION 5

THE WHOLE JOINT: FLANGE, GASKET, AND BOLTS

Overview

The function of each component in a bolted, gasketed flange, with general descriptions of each and some of the basic factors affecting their performance, has been discussed in previous sections. The flange provides a means of connecting and disconnecting two components, and provides the proper surface to hold the gasket in place. The gasket creates the seal between the two flanges, and the bolts serve as the elastic element that clamps the flanges together in order to create and maintain compression on the gasket. Any problems with one component affects the performance of the entire joint.

When engineers look to determine the recommended bolt torque required for an application, they have to consider all three components, the flange, the gasket, and the bolts.

The gasket material chosen requires a certain amount of compression to achieve a seal. The required compression depends on the pressure and temperature of the application and the physical properties of the gasket.

Once the best gasket for the application is chosen, the engineer then has to determine what effect the application parameters have on the flange. Hydrostatic end thrust on the flanges causes changes in gasket load. Flange rotation can occur if the flange is not designed to withstand the loads applied by the bolts. Expansion and contraction of the flange can cause variations in gasket and bolt load. Large sealing surface areas on the flange faces require higher overall bolt load (and loads on the flange) to achieve the required gasket stress. In most cases, the flange is the most difficult component to modify, so changes are usually made to the bolt materials and the torque applied to them, as well as choice of gasket materials.

The load applied to the bolts has to be high enough to compress the flanges and the gasket to achieve a tight seal. On top of that, the engineer looks to use the bolts to the highest percentage of their yield strength possible while ensuring they won't fail under worst-case operating conditions. Using high-strength bolts at only 30% of their yield is inefficient; there is minimal stretch in the bolt, making it more likely that thermal expansion and contraction will cause the gasket stress to drop below the minimum required to keep the seal. In some applications, it is better to use lower-strength bolts at a higher percentage of their yield strength; the high strength isn't required to achieve the right compression on the gasket, and the bolts are used more efficiently, making it more likely that the seal will be maintained during thermal cycling.

The process of determining the bolt torque value for an application is a circular one. Once the gasket is chosen, the stress required to seal the application is determined. When the required gasket stress is determined, the bolt torque necessary to achieve that stress is calculated. The load on the bolts is checked to make sure they are used at an acceptable percentage of their yield strength and with an appropriate margin of safety. Then the engineer checks to make sure that pressure inside the flange doesn't unload the gasket too much.

If any one of the components is under too much or too little load, the engineer has to go back to the drawing board. As an example, if the bolts are used at 70% of yield and the stress on the gasket is insufficient to seal the application, the engineer has to decide whether to change the gasket material, increase the bolt torque, or use stronger bolts. Each time a component of the joint is modified in some way, there is a ripple effect. When a change is made to one component of the joint, the effect on the others has to be checked. Given the fact that each component significantly affects the sealing ability of the joint, it is important to carefully consider changes to any one component.

ASME Boiler and Pressure Vessel Code

Engineers in various industries often use the ASME Code as a framework for determining what gasket stress is required to achieve a leak-free seal. The ASME Code lists certain gasket factors for various materials; these factors are what the engineers tend to use to calculate the maximum stress to be applied to the gaskets and subsequently the flange bolts. The Code's intended purpose is to provide guidelines for the engineers designing flanges so that they do not fail; standard design criteria and procedures are necessary to ensure uniform and safe design of flanges. Gasket factors given in the Code aren't intended to calculate the gasket stresses required for a leak-free seal; they are intended to give the engineer an idea of how much stress the flange joint is going to have to withstand for a given gasket material.

Gasket factors in the Code do not consider criteria such as temperature, creep, sealability, or other factors, that we know to be critical to performance.

This is also true for maximum allowable bolt stresses. The Code specifies maximum allowable stresses to be considered when designing the flange; it does not dictate maximum bolt stresses when the flange is assembled. Certain sections of the Code pertaining to critical components in nuclear power plants do indicate rough guidelines on what acceptable operating bolt stresses are, not the maximum allowable assembly stress (remember that assembly stress can change dramatically when the application is put into service).

To sum up this point, "the ASME Code is intended to be a designer's document, not an assembler's document."[10]

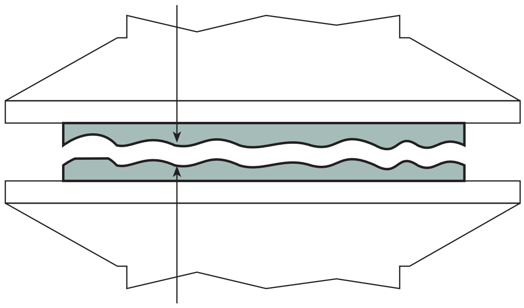
Installation Procedures

Having mentioned a few of the more common factors that affect a bolted joint in previous sections, the big question is, "What can I do to decrease the effects of these factors?" Some of the answers are specific to a particular component of the flanged joint and can be taken care of in the design and product selection stage. A big key to decreasing or eliminating the effects of some of these factors in the field is the use of a written procedure for the assembly of any bolted joint. Putting the required steps in writing makes it easier to go through the installation process without overlooking a critical step required to achieve a leak-free joint. It is important to emphasize to those personnel doing an installation just how critical it is to follow a step-by-step procedure, and how each step affects the end result. If every person involved on an installation is working from the same perspective, more consistency is achieved, resulting in better long-term sealing, less required maintenance for equipment, fewer premature failures, and less money lost because of down time and material costs. Given the complexity of bolted flange joint assemblies and the importance of assembly in ensuring that assemblies provide leak tight performance in-service, the American Society of Mechanical Engineers (ASME) has created a detailed document, "ASME PCC-1 Guidelines for Pressure Boundary Bolted Flange Joint Assembly." Not only does this standard offer guidelines covering flange joint assembly and disassembly, it recommends that personnel conducting flange joint assembly should undergo training and qualification testing. It is that important. The following are some of the important considerations a plant should incorporate in their installation procedures for bolted flanges with gaskets.

Disassembly

The first step once a joint is disassembled is to check the condition of the components. The flange sealing surface is important. Thoroughly clean the flange surface of old gasket material and scale. Use a wire brush with bristles or a suitable material that will not damage the faces. Surface finish should be verified. Check for pits, cuts, or grooves on the flange face; in some cases, light pitting or small grooves can be effectively sealed, depending on the gasket used, but the best practice is to repair any grooves, pitting, or cuts by either machining the flange face or replacing the flange. Radial marks that lead from ID to OD of the sealing surface are unacceptable; they are potential leak paths (probably made by leakage past the gasket in a previous installation). Check for waves on the flange; the maximum allowable variation from peak to valley on the waves is 0.38 mm (0.015").

WAVY FLANGE FACE



If the bolts are to be reused, inspect each and every part for galling, cracking, burrs, or other defects. Check the threads by running a nut down them to ensure that their dimensions are still within specification, and that they have not been damaged. Wire brush the bolts to remove any rust, dirt, or old lubricant, and chase the threads with a die to remove any burrs. Using damaged bolts can result in insufficient gasket stresses and chronic leakage problems with the flange. If the bolts need to be replaced, make sure that the new parts are made of the same material and to the same specification as the originals.

Discard the old gasket unless a failure analysis is required. Chesterton does not recommend reusing gaskets in any application. Before a gasket is installed in a flange, its physical characteristics are known. Once it is compressed in the flange, it conforms to the irregularities in the flange, decreases in thickness, and increases in density. After it is put into service, the gasket may harden, creep, and/or undergo some degree of chemical attack, and its sealing ability may decrease as the gasket ages. The used gasket is not the same material that was first installed in the flange and cannot be expected to provide consistent sealing if it is subsequently reused.

Assembly

Make sure the flange and gasket is clean and dry. Chesterton does not recommend the use of any lubricant, coating, or jointing compound on a gasket to be installed in a flange. These materials may decrease the coefficient of friction between the gasket and flange, resulting in an increase in the load required to prevent gasket blowout. They may also attack some gasket binders (if used), decreasing their physical integrity. A lubricant is likely to volatilize and burn out of the flange at higher temperatures, resulting in a sudden loss of volume between the flange faces; this causes a corresponding drop in bolt load, the gasket stress drops dramatically, and a leak develops.

The next step is to assemble the flange with the gasket in place. Check the fit of the gasket between the flange faces. Make sure the ID of the gasket does not extend into the pipe, otherwise premature erosion could occur. Lubricate all bolts and nuts with an appropriate lubricant; use a light, consistent coat of lube on the threads and any other mating surfaces of the fastener. Be sure to mix the lube well and use the same lubricant on all bolts in the flange. Align the flange, making sure the bolt holes line up properly, and that the faces are parallel to within 0.38 mm (0.015"). Snug all the nuts onto the flange finger tight, again ensuring the flanges are properly aligned.

This rough procedure assumes that a torque wrench is the standard means of tightening the joint. Given its relative accuracy, low cost, and ease of use, this is the most common means of tightening the bolts in a flange. Other methods of tightening the bolts such as heaters or hydraulic tensioners obviously require procedures different than simple torquing. Still, the basic concepts addressed by the steps in this procedure should be followed where applicable.

When using a torque wrench, always make sure that the target torque value is achieved with the wrench in motion. If the wrench sticks or chatters at the torque value, it is a good idea to back the bolt or nut off slightly and reapply the value again with the wrench in motion. When load is applied with a wrench, the nut remains stationary until the force applied exceeds the static friction force. Static friction opposes the applied force from the torque wrench and tries to keep the bolt from turning. Once the bolt goes from a static to a dynamic state, meaning it begins to move, the friction force trying to prevent the bolt from turning drops dramatically. This means that if the torque is applied in motion when there is less friction opposing the bolt, more of that applied force is converted to bolt stretch.

This ensures the gasket is compressed to a consistent stress across its entire surface. This technique helps keep the flanges parallel as the load is increased. Tightening the bolts in even smaller increments is a better, more effective technique, but it is more time consuming, and time becomes a big factor during an outage where dozens of flanges may need to be refitted with new gaskets.

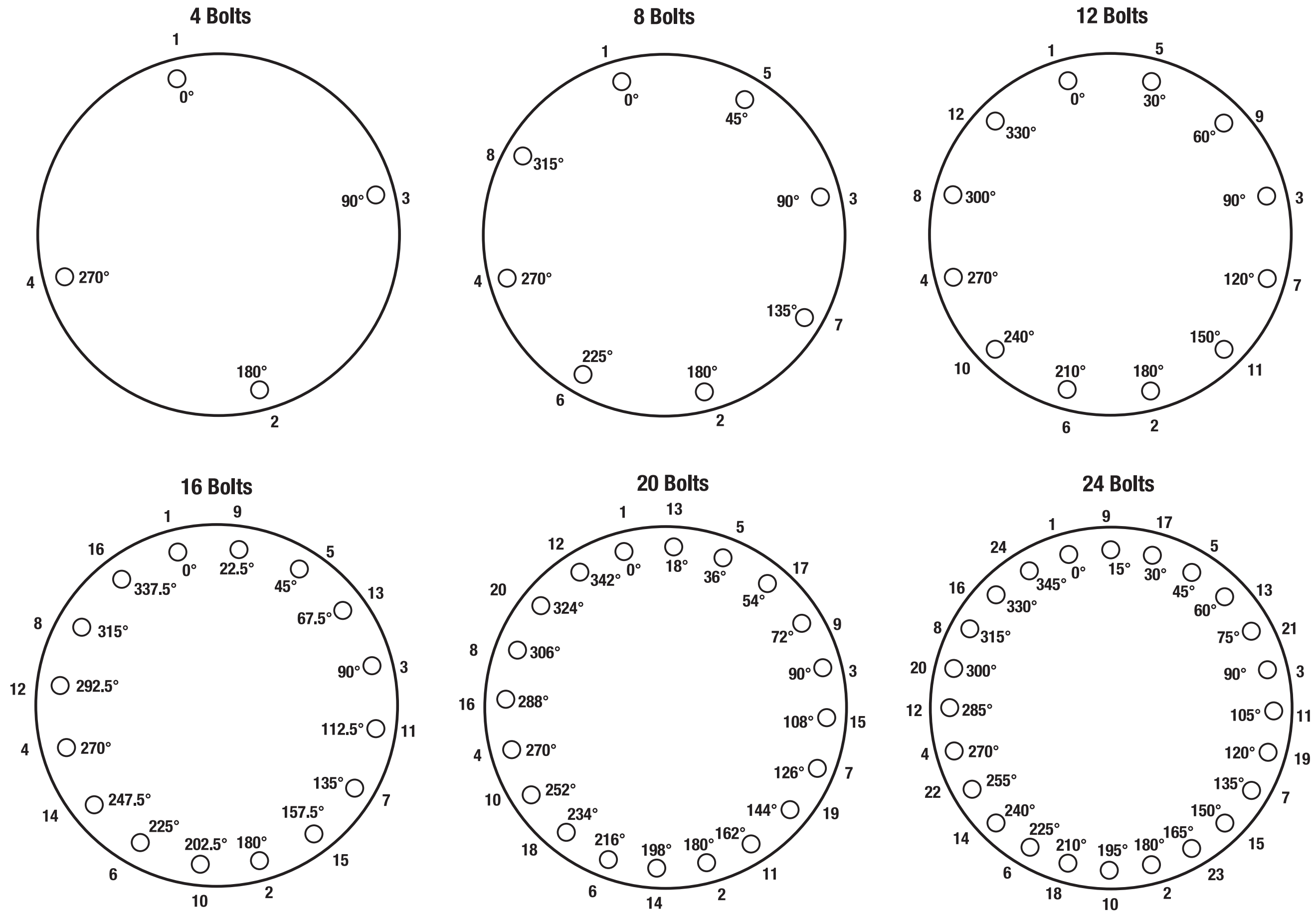
> The bolts should be tightened in increments of the recommended bolt torque. A commonly used procedure is the following:
>
> **Pass 1:** Bring all nuts up hand-tight; then snug-tight evenly.
>
> **Pass 2:** Torque to a maximum of 30% of the final torque. Check that the flange is bearing uniformly on the gasket.
>
> **Pass 3:** Torque to a maximum of 70% of the final torque.
>
> **Pass 4:** Torque to 100% torque.

As torque is applied during each pass, the bolts should be tightened in a crisscross pattern. This helps minimize the effects of elastic interactions between the flange bolts, and helps prevent wrinkling of the gasket. On larger flange applications, it is helpful to number and mark the bolts to ensure the tightening pattern is consistent. Once the 100% of the torque value is applied to all the bolts, another pass should be made, tightening the bolts in reverse order. It is not uncommon to find some of the bolts are no longer at the 100% torque value applied during the previous pass. Several passes through the pattern may be required before all the bolts are at the same torque value.

Depending on the material used, gaskets will creep to some degree. This creep causes a loss in load on the gasket and a corresponding decrease in bolt load. The effects of creep can be accounted for by letting the joint sit for one hour, before starting up the system, and then reapplying the recommended torque. This recompresses the gasket to its original load, and because the gasket thickness has been decreased, the gasket will undergo much less creep. Some procedures also recommend that the very last torque pass through the bolts should be done in a circular pattern; the idea is to alter the bolting pattern to check again for any relaxation caused by elastic interactions.

Per the ASME PCC-1, the basic "star" pattern described above "has been successfully applied through industry for all gasket styles and flange types. It is the standard 'Best Practices' assembly procedure for bolted, flanged connections."[11] With that said, there are some alternate assembly procedures described in ASME PCC-1 that have also been used in the field more recently. These alternate assembly procedures have been developed and tested for specific applications and materials; they can either save time and effort in assembly without altering performance of the assembly, or in some cases, they may improve the performance of the gasketed assembly. The "PCC-1 Guidelines for Pressure Boundary Bolted Flange Joint Assembly" do caution that using alternate patterns should be carefully evaluated prior to using in actual service.

VARIOUS BOLTING SEQUENCE



Refer to ASME PCC-1 for alternate pattern guidelines or consult Chesterton Application Engineering.

SECTION 6

REFERENCE

Appendix A: Glossary of Terms

In many design engineering manuals, familiar and unfamiliar terms will be encountered. This section is designed to provide and define many of those terms when referring to gaskets.

Accelerated Test Services—If test results are required in a short time, the materials under test will be subjected to exaggerated test conditions, usually involving temperature, speed, and other conditions.

Ambient Temperature—The temperature of the environment to which a material is subjected.

ANSI—Abbreviation for American National Standards Institute.

API—Abbreviation for American Petroleum Institute.

Apparent Density—The density of a material is the amount of material, or mass, per unit volume. It is ordinarily expressed in grams per cubic centimeter or pounds per cubic foot.

Aqueous Solutions—Any fluid solution containing water. (See further discussion under pH).

ASME—Abbreviation for American Society of Mechanical Engineers.

ASTM—Abbreviation for American Society for Testing and Materials.

Beater Add (Beater Addition)—A manufacturing process for sheet employing a paper making process using Fourdrinier- or cylinder-type paper machines which deposit a binder (SBR-Neoprene-Nitrile) uniformly over individual fibers (non-asbestos and cellulose) while suspended in water.

Burst—A rupture caused by internal pressure.

Calender—A machine equipped with two or more heavy, internally-heated or cooled rolls, which is used for: sheeting, plying, or densifying compound or fiber/rubber composite or frictioning or coating fabric with a rubber compound.

Cold Flow—Continued deformation under stress.

Elastomer—Any of various elastic substances resembling rubber. This man-made rubber (also called polymers) is produced by the combination of monomers. *Also see Rubber.*

Extrusion—Permanent displacement of part of a seal into a gap, under the action of fluid pressure, or flange loading.

Flange—The rigid portion of a gasketed joint that contacts the sides or edges of the gasket.

Flange Joint—A joint formed by two abutting flanges.

Flat Metal Gaskets—By definition, gaskets that are relatively thin compared to their width. They may be plain, solid, serrated, or grooved.

Flat Ring—A flange gasket lying wholly within the ring of bolts.

Full Face Gaskets—Gasket with dimensions to match entire flange diameter; encompasses flange bolts.

Gasket—A deformable material, which when clamped between essentially stationary faces, prevents the passage of matter through an opening or joint.

Gasketing—Material bulk form from which gaskets may be cut.

Gasketing Sheet—Refers to specific form of gasketing material from which gaskets are cut and/or fabricated.

Gasketing Joint—The collective total of all members used to effect a gasketed seal between two separate members.

Homogeneous—Products that are of uniform composition throughout.

ID—Symbol for inside diameter.

ISO—Abbreviation for International Standards Institute.

Joint—An interstice (crevice) between rigid members of a fluid container.

Jointing—Common term in Europe for gasketing.

Appendix A: Glossary of Terms (continued)

Leak—The passage of matter through interfacial openings or passageways, or both in the gasket.

Leakage Rate—The quantity, either mass or volume, of fluid passing through and/or over the faces of gaskets in a given length of time.

"M" Maintenance Value—An empirical design constant of a flange gasket used in the ASME Boiler and Pressure Vessels Code. The Code equation defines this term as the ratio of residual gasket load to fluid pressure at leak. The definition of "M" has varied in successive editions of the Code, according to the method employed for computing residual gasket load.

OD—Symbol for outside diameter.

Oil Swell—The change in volume or weight of a gasketing product resulting from contact with oil.

Permanent Set—

1. Permanent set is the deformation remaining after a specimen has been stressed a prescribed amount for a definite period and released for a definite period.
2. In creep determinations, permanent set is the residual unrecoverable deformation after the load causing the creep has been removed for a substantial and definite period of time.

Permeability and Porosity—A permeable material is one that contains pores or small openings that permit liquids or gases to seep through. After manufacturing of graphite, definite interstices (pores) will exist between the crystal granules. Porosity is the percent by volume of pores in relation to the total volume of a piece. Permeability is the rate of flow of a gas through such a piece. The size and/or number of connected pores or channels that are continuous determine the amount of permeability. Therefore, materials with a high porosity do not necessarily have a high permeability. Permeability is usually given in terms of the cubic feet of cold air that will pass per hour through a one-square-foot panel, one inch in thickness, with a pressure drop of one inch of water.

Pressure—Force per unit area, usually expressed in pounds per square inch or N/mm^2.

Pressure, Absolute—The pressure above zero absolute, i.e., the sum of atmospheric and gage pressure. In vacuum related work it is usually expressed in millimeters of mercury (mm Hg).

Pressure, Atmospheric—Pressure exerted by the atmosphere at any specific location (seal level pressure is approximately 14.7 pounds per square inch absolute).

Pressure, Gage—Pressure differential above or below atmospheric pressure. Expressed as pounds per square inch gage (psig).

PT Value—A numerical value resulting from the multiplication of the internal pressure (psig) by the temperature of the fluid being sealed. Used only as rough safety guide for limiting gasket usage.

Raised Face Gaskets—Gaskets for use with raised face flanges. *Also see also flat ring or ring gasket.*

Recovery—(Sheet Gasketing) The percentage decrease in compressed deformation during a specified time interval and at a specified temperature, following release of load, as defined by ASTM F-36.

Reinforcement—The strength members, consisting of fabric, cord, metal, or other reinforcement, of a composite material.

Residual Load—Axial load developed against, or by, a flange gasket at some interval after initial tightening. Is less than preload by the amount of combined fluid relief and creep-relaxation.

Ring Gasket—A flange gasket lying wholly within ring of bolts. *Also see flat ring or raised face gasket.*

Rubber—An elastic substance, obtained naturally or synthetically, which is modified by chemical treatment to increase its useful properties, such as the binder for gasket material.

Sealability—The measure of fluid leakage through and/or across both faces of a gasket. Measured either by using ASTM F-37 or DIN 3535 standard equipment and procedures.

Strain—The deformation of a gasket specimen under the action of applied force of stress.

Stress—The intensity of the load at a point in the gasket specimen.

Stress-Relaxation—A transient stress-strain condition in which the gasket stress decays as the strain remains constant. (This condition is encountered in grooved-face gasketing joints in which metal-to-metal contact occurs. This condition is also approached in flat-face gasketing joints when the bolt is practically infinitely rigid).

Stress-Strain—The relationship of load and deformation in a gasket under stress. In most non-metallic gasketing, this is commonly the relationship of compressive load and compression (strain).

Surface Finish—The measure of surface smoothness (or "roughness").

Tensile Strength—Tensile strength is defined as the pure unit area required to rupture a material by pulling it apart. It is expressed in pounds per square inch.

Wicking—Leakage through a gasket (not around it).

Work Pressure—The maximum operating pressure encountered during normal service.

Yield "Y" Factor—The minimum design seating stress on the gasket in either psi or megapascals that is required to provide a sealed joint with no internal pressure in the joint.

Appendix B: Troubleshooting Guide

Fault	Cause	Remedy
Design		
Insufficient gasket stress	Insufficient bolt load	• Increase no. of bolts • Increase dia. of bolts • Change to higher tensile material
	• Gasket too wide • Wrong gasket type	• Reduce area of gasket • Fit gasket which requires lower seating stress
Excessive gasket stress	Excessive bolt load	• Reduce no. of bolts • Change to lower tensile material
	• Gasket too narrow • Wrong gasket type	• Increase gasket area • Fit gasket which requires lower seating stress
Assembly		
Lack of compression	• Bolts insufficiently tightened • Incorrect tightening procedure	• Apply additional torque • Bolts should be tightened in sequence—i.e., diametrically—and gradually increasing load in each bolt alternately.
	Gasket relaxed due to operating temperature	• Use gasket with lower creep relaxation at operating temperature.
	Bad threads	• Ensure nuts are a good running fit over the entire length of bolt thread.
	Insufficient thread length	• Ensure threads are sufficiently long to allow nuts to make contact with metal faces.
Metal Faces		
Uneven	Flanges too thin	Flanges should always be sufficiently rigid not to be distorted by the bolt load.
	Flanges not parallel	• Flange faces should always be parallel and bolt load should never be relied on to pull flanges together. • Bolts should be tightened in proper sequence to prevent "cocking."
Damage	Mechanical damage while faces exposed	Every attention should be given to ensure faces are clean, flat, and free from imperfections too deep for the gasket material to completely fill.
	Overzealous abrading during cleaning.	Care should be taken in removal of old gasket material.
Dirty or corroded	Previously used jointing compounds frequently harden and form uneven surface (old gasket not completely removed).	• Faces should be wire brushed down to clean metal. • Serrations should be perfectly clean and of sound contour.
Incorrect surface texture	• Delivered from manufacturer out of specification • Damaged in transit	• Concentric grooving is ideal for high pressure. • Where phonographic or continuous spiral groove is used, the depth must not be too deep for gasket to fill.

Appendix B: Troubleshooting Guide (cont.)

Fault	Cause	Remedy
Gasket Material		
Loss of resilience and interface contact	Re-use of old gasket	The re-use of gaskets is not recommended. The material will have hardened and taken up the contours of flange surface imperfection. It is unlikely that a gasket would be replaced in exactly the same position. Furthermore, downtime costs to replace faulty gasket are far in excess of initial gasket cost.
Material deteriorates rapidly	Material incompatibility with contained fluid/temperature	Check material recommendations and select material capable of withstanding condition.
Gasket extrudes from faces	Too high seating stress	See recommendations under design faults.
	Excessive use of jointing compounds	• Unless specified, the use of compounds and paste is not recommended; these act as lubricants which reduce friction between the gasket and the metal faces, thereby reducing the load bearing properties. • Where non-stick finish is required, it can be applied during production of the gasket material.
Incorrect dimensions	Design or manufacturing error	• Gaskets should always have clean cut edges slightly larger than that of the vessel or pipe, i.e., the gasket should not protrude into the flow path of the fluid. Protrusion could create turbulance in addition to restricting flow. The gasket could also suffer damage through erosion by the fluid. • Providing the gasket is compressed over the entire face there is little likelihood of absorption of the fluid. • Back holes should be sufficiently large to allow a clearance around the bolts.

Appendix C: Useful Conversions

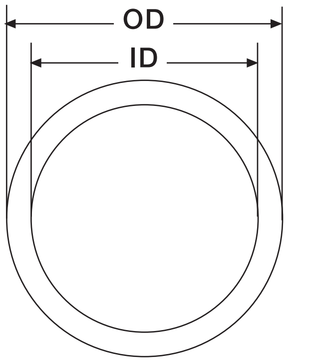
SURFACE FINISH/R_a OR RMS VALUES

Nominal R_a Values		Roughness N-Grade
R_a – µm	R_a – µinches	
50	2000	N12
25	1000	N11
12,5	500	N10
6,3	250	N9
3,2	125	N8
1,6	63	N7
0,8	32	N6
0,4	16	N5
0,2	8	N4
0,1	4	N3
0,05	2	N2
0,025	1	N1
0,0125	0.5	—

CONVERSION TABLE

(a) x	(b) =	(c)
Atmosphere	1.4696×10^{1}	Pounds/sq. in.
Bar	1.45×10^{1}	Pounds/sq. in.
Bar	1.0×10^{-1}	N/mm^2
Centimeter	3.281×10^{-2}	Feet
Centimeter	3.937×10^{-1}	Inch
Cubic Centimeter	3.531×10^{-5}	Cubic Feet
Cubic Centimeter	6.102×10^{-2}	Cubic Inch
Cubic Feet	2.8320×10^{4}	Cubic Centimeter
Cubic Feet	1.728×10^{3}	Cubic Inch
Cubic Feet	2.832×10^{-2}	Cubic Meter
Cubic inch	1.639×10^{1}	Cubic Centimeter
Cubic Inch	5.787×10^{-4}	Cubic Feet
Cubic Meter	3.531×10^{1}	Cubic Feet
Cubic Meter	6.1023×10^{4}	Cubic Inch
Feet	3.048×10^{1}	Centimeter
Gram	3.527×10^{-2}	Ounce (avdp)
Gram	2.205×10^{-3}	Pounds
Grams/Cubic Centimeter	6.243×10^{1}	Pounds/Cubic Foot In
Inch	2.54	Centimeter
Inch	2.54×10^{1}	Millimeter
Kilogram	2.2046	Pounds
Kilogram/Cubic Meter	6.243×10^{-2}	Pounds/Cubic Foot
Meter	3.281	Foot
Millimeter	3.281×10^{-3}	Foot
Millimeter	3.937×10^{-2}	Inch
Newton/Sq. Millimeter	10×10^{1}	megapascal (MPa)
Newton/Sq. Millimeter	145.035	Pounds/Sq. Inch
Ounce	2.8349×10^{1}	Grams
Ounce	6.25×10^{-2}	Pounds
Pounds	4.5359×10^{-2}	Grams
Pounds/Cubic Foot	1.602×10^{-2}	Gram/Cubic Centimeter
Pounds/Sq. Foot	6.944×10^{-3}	Pounds/Sq. Inch
Pounds/Sq. Foot	4.882	Kilogram/Sq. Meter
Pounds/Sq. Inch	144	Pounds/Sq. Foot
Pounds/Sq. Inch	6.896×10^{-3}	Newton/Sq. Millimeter
Square Feet	9.29×10^{2}	Square Centimeter
Square Feet	1.44×10^{2}	Square Inch
Square Inch	6.452	Square Centimeter
Square Inch	6.944×10^{-3}	Square Feet
Square Millimeter	1.55×10^{-3}	Square Inch
Square Meter	1.076×10^{1}	Square Feet
Temperature		
Centigrade	(°C x 1.8) + 32	Fahrenheit
Fahrenheit	0.555 (°F-32)	Centigrade
	9	
Torque		
Foot-Pound	1.356	Newton-Meter
Inch-Pound	1.13×10^{-1}	Newton-Meter

Appendix D: Heat Exchanger Gasket Configuration Drawings



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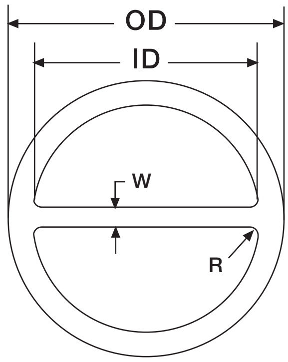

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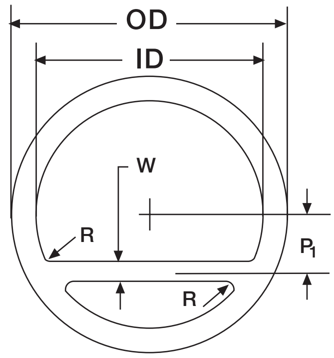

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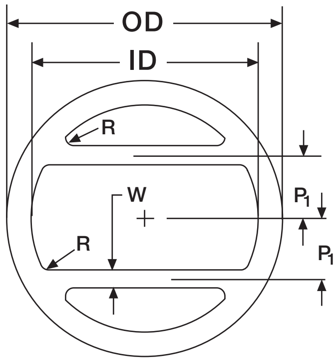

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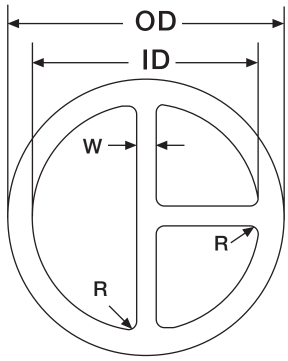

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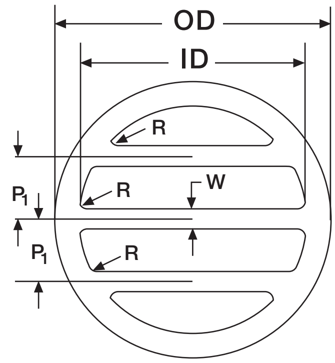

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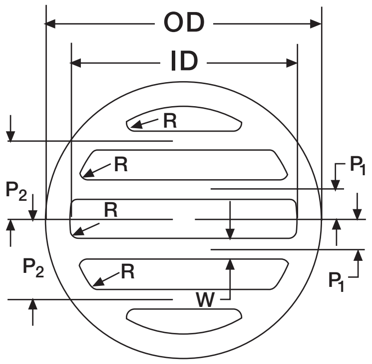

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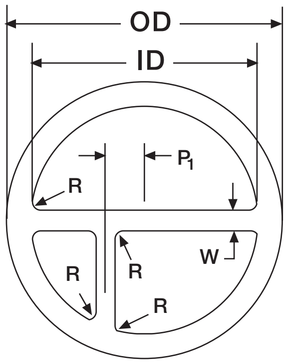

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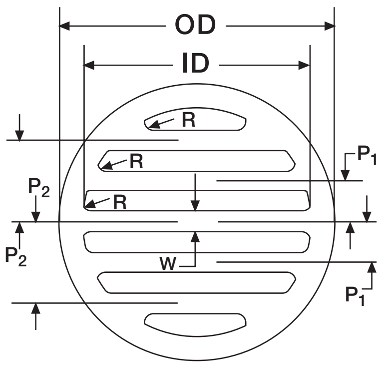

I

Dimensional Data		
Gasket drawing letter:		
ID:	R:	P_1:
OD:	W:	P_2:

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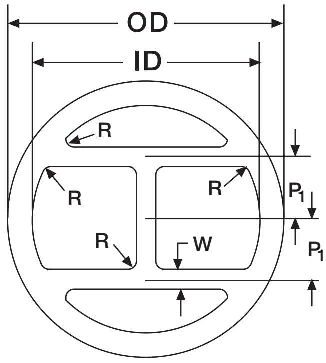

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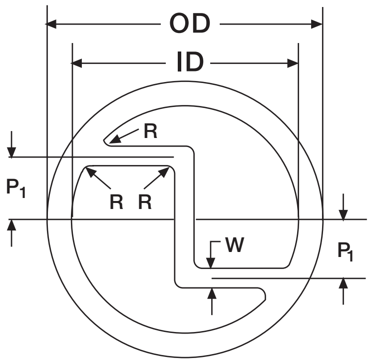

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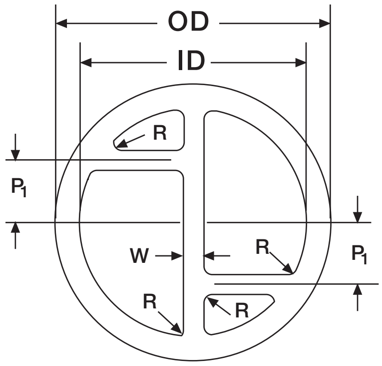

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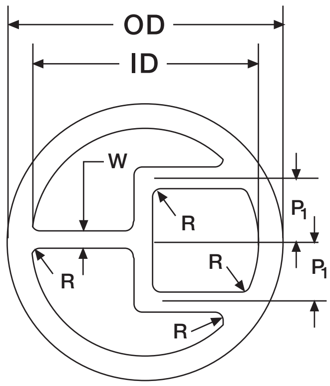

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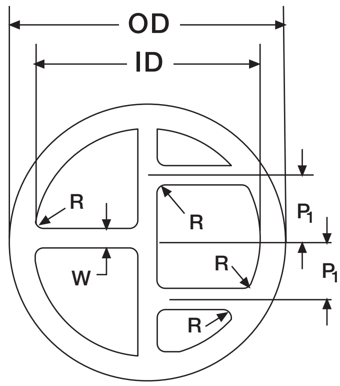

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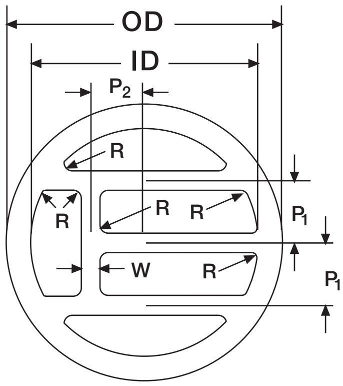

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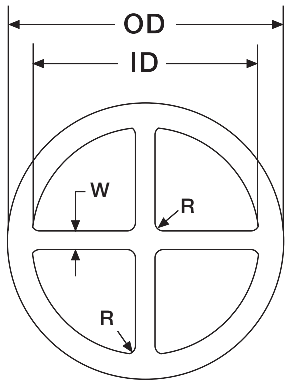

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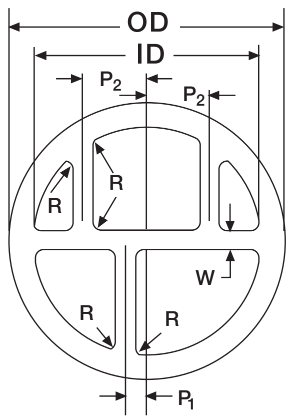

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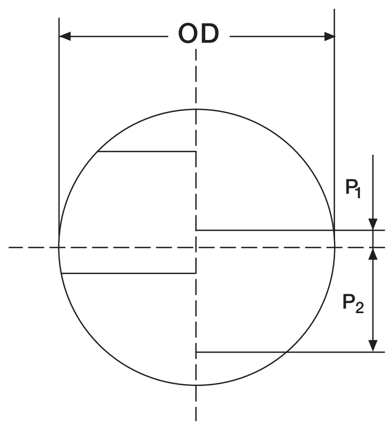


R

Dimensional Data		
Gasket drawing letter:		
ID:	R:	P_1:
OD:	W:	P_2:

Appendix E: Instructions for Heat Exchanger Data Collection Sheet

Instruction sheet for completing Heat Exchanger Data Collection Sheet

Note: please do not forward any drawings unless you can see the five following items:

1. Gasket ID
2. Quantity of bolts
3. Nominal bolt diameter
4. Bolt material/designation
5. Maximum operating temperature

Information required to request a quotation

1. All of the information above
2. Detailed gasket drawing(s) for all gaskets
3. Fluid/medium(s) sealed
4. The 252 Heat Exchange Steel Trap (Place an X in the appropriate boxes)

Thickness	Material	Metallurgy
☐ 0,8 mm (1/32")	☐ Graphite	☐ 316SS (1/32" thickness normally provided up to 36" OD)
☐ 1,6 mm (1/16")	☐ PTFE	☐ Carbon steel, 1524 mm (1/16" thickness above 36" and up to 60") OD **if chemically compatible**
		☐ 304SS 1524 mm (1/16" thickness above 36" and up to 60") OD
		☐ Above 1524 mm (60") OD, consult factory

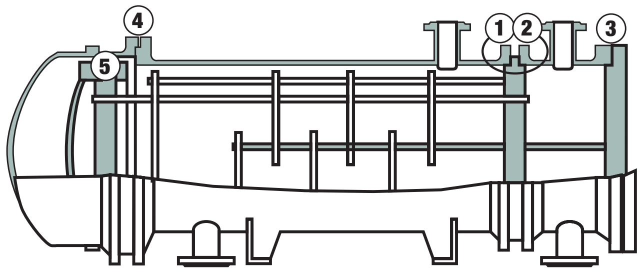
Information required to engineer application for an order

1. All information required to supply a quotation
2. A completed Heat Exchanger Data Collection Sheet with all required (bold print items must be provided) information completed
3. In the unlikely event that a gasket(s) drawing is unavailable, refer to gasket configuration drawings and complete all requested dimensional data. A plant representatives' signature is required on the configuration form.
4. The heat exchanger engineering form will be returned to the distributor with an assigned survey number. This number must be used when placing an order and when discussing this application with the engineering department.

Return all information to the attention of your local Chesterton Specialist or Application Engineering

Example Heat Exchanger Data Collection Sheet

Date:
Distributor:
Specialist:
Customer:
Customer Contact:
Heat Exchanger Equipment Item No:
Heat Exchanger Manufacturer's Name:



1. Shell and Tubesheet
2. Tubesheet to Channel Box
3. Channel Box Cover
4. Shell Cover
5. Floating Head

	Shell to Tubesheet	Tubesheet to Channel Box	Channel Box Cover	Shell Cover	Floating Head
	1 – Shell	2 – Tubes	3 – Tubes	4 – Shell	5 – Tubes/Shell
Fluid/Medium					
Temperature – Operating ˚C (˚F)					
Operating Pressure bar g (psig)					
Temperature – Design ˚C (˚F)					
Design Pressure bar g (psig)					
Test Pressure bar g (psig)					
Flange Face (Raised, Flat, Tongue and Groove)					
Flange Material (Metal)					
Flange Bolting Data					
Stud/Bolt Diameter (mm)					
Stud/Bolt Length (mm)					
Stud Type (Stretch, All Thread, Hex Bolt)					
Stud/Bolt Quantity					
Stud Specification and Material					
Yield Strength (Mpa)					
Previous Torque Applied (N•m)					
Gasket Data (Enter data below or provide heat exchanger manufacturer's gasket drawing)					
Gasket drawing letter (A-R) **See Appendix D**					
Gasket Type (Camprofile, Steel Trap, 198/199/459)					
Gasket thickness					
ID					
OD					
R					
W					
P_1					
P_2					

Endnotes and Bibliography

Endnotes

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Notes

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Notes

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