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# CHAPTER 25

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# GASKETS

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In the field of gaskets and seals, the former are generally associated with sealing mating flanges while the latter are generally associated with sealing reciprocating shafts or moving parts. Some designers refer to gaskets as static seals and consider seals to be dynamic sealing components. This chapter covers gaskets, and Chap. 21 discusses seals.

## **25.1 DEFINITION**

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A *gasket* is a material or combination of materials clamped between two separable members of a mechanical joint. Its function is to effect a seal between the members (flanges) and maintain the seal for a prolonged period. The gasket must be capable of sealing mating surfaces, must be impervious and resistant to the medium being sealed, and must be able to withstand the application temperature. Figure 25.1 depicts the nomenclature associated with a gasketed joint.

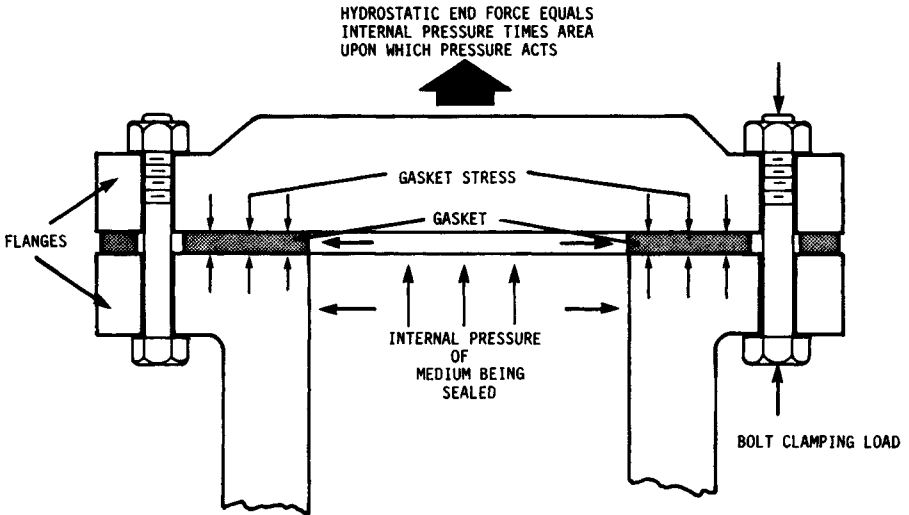
## **25.2 STANDARD CLASSIFICATION SYSTEM FOR NONMETALLIC GASKET MATERIALS<sup>†</sup>**

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This classification system provides a means for specifying or describing pertinent properties of commercial nonmetallic gasket materials. Materials composed of

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<sup>†</sup> Ref. [25.1] (ANSI/ASTM F104).



**FIGURE 25.1** Nomenclature of a gasketed joint.

asbestos, cork, cellulose, and other organic or inorganic materials in combination with various binders or impregnants are included. Materials normally classified as rubber compounds are not included, since they are covered in ASTM Method D 2000 (SAE J200). Gasket coatings are not covered, since details are intended to be given on engineering drawings or in separate specifications.

This classification is based on the principle that nonmetallic gasket materials can be described in terms of specific physical and mechanical characteristics. Thus, users of gasket materials can, by selecting different combinations of statements, specify different combinations of properties desired in various parts. Suppliers, likewise, can report properties available in their products.

In specifying or describing gasket materials, each *line call-out* shall include the number of this system (minus the date symbol) followed by the letter F and six numerals, for example, ASTM F104 (F125400). Since each numeral of the call-out represents a characteristic (as shown in Table 25.1), six numerals are always required. The numeral 0 is used when the description of any characteristic is not desired. The numeral 9 is used when the description of any characteristic (or related test) is specified by some supplement to this classification system, such as notes on engineering drawings.

### **25.3 GASKET PROPERTIES, TEST METHODS, AND THEIR SIGNIFICANCE IN GASKETED JOINTS**

Table 25.2 lists some of the most significant gasket properties which are associated with creating and maintaining a seal. This table also shows the test method and the significance of each property in a gasket application.

## 25.4 PERMEABILITY PROPERTIES

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For a material to be impervious to a fluid, a sufficient density to eliminate voids which might allow capillary flow of the fluid through the construction must be achieved. This requirement may be met in two ways: by compressing the material to fill the voids and/or by partially or completely filling them during fabrication by means of binders and fillers. Also, for the material to maintain its impermeability for a prolonged time, its constituents must be able to resist degradation and disintegration resulting from chemical attack and temperature of the application [25.2].

Most gasket materials are composed of a fibrous or granular base material, forming a basic matrix or foundation, which is held together or strengthened with a binder. The choice of combinations of binder and base material depends on the compatibility of the components, the conditions of the sealing environment, and the load-bearing properties required for the application.

Some of the major constituents and the properties which are related to impermeability are listed here.

### 25.4.1 Base Materials—Nonmetallic

**Cork and Cork-Rubber.** High compressibility allows easy density increase of the material, thus enabling an effective seal at low flange pressures. The temperature limit is approximately 250°F (121°C) for cork and 300°F (149°C) for cork-rubber compositions. Chemical resistance to water, oil, and solvents is good, but resistance to inorganic acids, alkalis, and oxidizing environments is poor. These materials conform well to distorted flanges.

**Cellulose Fiber.** Cellulose has good chemical resistance to most fluids except strong acids and bases. The temperature limitation is approximately 300°F (149°C). Changes in humidity may result in dimensional changes and/or hardening.

**Asbestos Fiber.** This material has good heat resistance to 800°F (427°C) and is noncombustible. It is almost chemically inert (crocidolite fibers, commonly known as blue asbestos, resist even inorganic acids) and has very low compressibility. The binder dictates the resistance to temperature and the medium to be sealed.

**Nonasbestos Fibers.** A number of nonasbestos fibers are being used in gaskets. Some of these are glass, carbon, aramid, and ceramic. These fibers are expensive and are normally used only in small amounts. Temperature limits from 750 to 2400°F (399 to 1316°C) are obtainable. Use of these fillers is an emerging field today, and suppliers should be contacted before these fibers are specified for use.

### 25.4.2 Binders and Fillers

**Rubber.** Rubber binders provide varying temperature and chemical resistance depending on the type of rubber used. These rubber and rubberlike materials are used as binders and, in some cases, gaskets:

1. *Natural* This rubber has good mechanical properties and is impervious to water and air. It has uncontrolled swell in petroleum oil and fuel and chlorinated solvents. The temperature limit is 250°F (121°C).

**TABLE 25.1** Basic Physical and Mechanical Characteristics

Basic six-digit number	Basic characteristic										
First numeral	<p>Type of material (the principal fibrous or particulate reinforcement material from which the gasket is made) shall conform to the first numeral of the basic six-digit number as follows:</p> <ul style="list-style-type: none"> <li>0 = not specified</li> <li>1 = asbestos or other inorganic fibers (type 1)</li> <li>2 = cork (type 2)</li> <li>3 = cellulose or other organic fibers (type 3)</li> <li>4 = fluorocarbon polymer</li> <li>9 = as specified†</li> </ul>										
Second numeral	<p>Class of material (method of manufacture or common trade designation) shall conform to the second numeral of the basic six-digit number as follows:</p> <p>When first numeral is 1, for second numeral</p> <ul style="list-style-type: none"> <li>0 = not specified</li> <li>1 = compressed asbestos (class 1)</li> <li>2 = beater addition asbestos (class 2)</li> <li>3 = asbestos paper and millboard (class 3)</li> <li>9 = as specified†</li> </ul> <p>When first numeral is 2, for second numeral</p> <ul style="list-style-type: none"> <li>0 = not specified</li> <li>1 = cork composition (class 1)</li> <li>2 = cork and elastomeric (class 2)</li> <li>3 = cork and cellular rubber (class 3)</li> <li>9 = as specified†</li> </ul> <p>When first numeral is 3, for second numeral</p> <ul style="list-style-type: none"> <li>0 = not specified</li> <li>1 = untreated fiber—tag, chipboard, vulcanized fiber, etc. (class 1)</li> <li>2 = protein treated (class 2)</li> <li>3 = elastomeric treated (class 3)</li> <li>4 = thermosetting resin treated (class 4)</li> <li>9 = as specified†</li> </ul> <p>When first numeral is 4, for second numeral</p> <ul style="list-style-type: none"> <li>0 = not specified</li> <li>1 = sheet PTFE</li> <li>2 = PTFE of expanded structure</li> <li>3 = PTFE filaments, braided or woven</li> <li>4 = PTFE felts</li> <li>5 = filled PTFE</li> <li>9 = as specified†</li> </ul>										
Third numeral	<p>Compressibility characteristics, determined in accordance with 8.2, shall conform to the percentage indicated by the third numeral of the basic six-digit number (example: 4 = 15 to 25%):</p> <table border="0"> <tr> <td>0 = not specified</td> <td>5 = 20 to 30%</td> </tr> <tr> <td>1 = 0 to 10%</td> <td>6 = 25 to 40%</td> </tr> <tr> <td>2 = 5 to 15%†</td> <td>7 = 30 to 50%</td> </tr> <tr> <td>3 = 10 to 20%</td> <td>8 = 40 to 60%</td> </tr> <tr> <td>4 = 15 to 25%</td> <td>9 = as specified†</td> </tr> </table>	0 = not specified	5 = 20 to 30%	1 = 0 to 10%	6 = 25 to 40%	2 = 5 to 15%†	7 = 30 to 50%	3 = 10 to 20%	8 = 40 to 60%	4 = 15 to 25%	9 = as specified†
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2 = 5 to 15%†	7 = 30 to 50%										
3 = 10 to 20%	8 = 40 to 60%										
4 = 15 to 25%	9 = as specified†										

**TABLE 25.1** Basic Physical and Mechanical Characteristics (*Continued*)

Fourth numeral	<p>Thickness increase when immersed in ASTM no. 3 oil, determined in accordance with 8.3, shall conform to the percentage indicated by the fourth numeral of the basic six-digit number (example: 4 = 15 to 30%):</p> <table> <tbody> <tr> <td>0 = not specified</td> <td>5 = 20 to 40%</td> </tr> <tr> <td>1 = 0 to 15%</td> <td>6 = 30 to 50 %</td> </tr> <tr> <td>2 = 5 to 20%</td> <td>7 = 40 to 60%</td> </tr> <tr> <td>3 = 10 to 25%</td> <td>8 = 50 to 70%</td> </tr> <tr> <td>4 = 15 to 30%</td> <td>9 = a specified†</td> </tr> </tbody> </table>	0 = not specified	5 = 20 to 40%	1 = 0 to 15%	6 = 30 to 50 %	2 = 5 to 20%	7 = 40 to 60%	3 = 10 to 25%	8 = 50 to 70%	4 = 15 to 30%	9 = a specified†
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2 = 5 to 20%	7 = 40 to 60%										
3 = 10 to 25%	8 = 50 to 70%										
4 = 15 to 30%	9 = a specified†										
Fifth numeral	<p>Weight increase when immersed in ASTM no. 3 oil, determined in accordance with 8.3, shall conform to the percentage indicated by the fifth numeral of the basic six-digit number (example: 4 = 30% maximum):</p> <table> <tbody> <tr> <td>0 = not specified</td> <td>5 = 40% max.</td> </tr> <tr> <td>1 = 10% max.</td> <td>6 = 60% max.</td> </tr> <tr> <td>2 = 15% max.</td> <td>7 = 80% max.</td> </tr> <tr> <td>3 = 20% max.</td> <td>8 = 100% max.</td> </tr> <tr> <td>4 = 30% max.</td> <td>9 = as specified‡</td> </tr> </tbody> </table>	0 = not specified	5 = 40% max.	1 = 10% max.	6 = 60% max.	2 = 15% max.	7 = 80% max.	3 = 20% max.	8 = 100% max.	4 = 30% max.	9 = as specified‡
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1 = 10% max.	6 = 60% max.										
2 = 15% max.	7 = 80% max.										
3 = 20% max.	8 = 100% max.										
4 = 30% max.	9 = as specified‡										
Sixth numeral	<p>Weight increase when immersed in water, determined in accordance with 8.3, shall conform to the percentage indicated by the sixth numeral of the basic six-digit number (example: 4 = 30% maximum):</p> <table> <tbody> <tr> <td>0 = not specified</td> <td>5 = 40% max.</td> </tr> <tr> <td>1 = 10% max.</td> <td>6 = 60% max.</td> </tr> <tr> <td>2 = 15% max.</td> <td>7 = 80% max.</td> </tr> <tr> <td>3 = 20% max.</td> <td>8 = 100% max.</td> </tr> <tr> <td>4 = 30% max.</td> <td>9 = as specified‡</td> </tr> </tbody> </table>	0 = not specified	5 = 40% max.	1 = 10% max.	6 = 60% max.	2 = 15% max.	7 = 80% max.	3 = 20% max.	8 = 100% max.	4 = 30% max.	9 = as specified‡
0 = not specified	5 = 40% max.										
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3 = 20% max.	8 = 100% max.										
4 = 30% max.	9 = as specified‡										

†On engineering drawings or other supplement to this classification system. Suppliers of gasket materials should be contacted to find out what line call-out materials are available. Refer to ANSI/ASTM F104 for further details (Ref. [25.1]).

‡From 7 to 17% for type 1, class 1 compressed asbestos sheet.

- Styrene/butadiene* This rubber is similar to natural rubber but has slightly improved properties. The temperature limit also is 250°F (121°C).
- Butyl* This rubber has excellent resistance to air and water, fair resistance to dilute acids, and poor resistance to oils and solvents. It has a temperature limit of 300°F (149°C).
- Nitrile* This rubber has excellent resistance to oils and dilute acids. It has good compression set characteristics and has a temperature limit of 300°F (149°C).
- Neoprene* This rubber has good resistance to water, alkalies, nonaromatic oils, and solvents. Its temperature limit is 250°F (121°C).
- Ethylene propylene rubber* This rubber has excellent resistance to hot air, water, coolants, and most dilute acids and bases. It swells in petroleum fuels and oils without severe degradation. The temperature limit is 300°F (149°C).
- Acrylic* This rubber has excellent resistance to oxidation, heat, and oils. It has poor resistance to low temperature, alkalies, and water. The temperature limit is 450°F (232°C).

**TABLE 25.2** Identification, Test Method, and Significance of Various Properties Associated with Gasket Materials

Property	Test method	Significance in gasket applications
Sealability	Fixtures per ASTM F37-62T	Resistance to fluid passage
Heat resistance	Exposure testing at elevated temperatures	Resistance to thermal degradation
Oil and water immersion characteristics	ASTM D-1170	Resistance to fluid attack
Antistick characteristics	Fixture testing at elevated temperatures	Ability to release from flanges after use
Stress vs. compression and spring rates	Various compression test machines	Sealing pressure at various compressions
Compressibility and recovery	ASTM F36-61T	Ability to follow deformation and deflection; indentation characteristics
Creep relaxation and compression set	ASTM F38-62T and D-395-59	Related to torque loss and subsequent loss of sealing pressure
Crush and extrusion characteristics	Compression test machines	Resistance to high loadings and extrusion characteristics at room and elevated temperatures

8. *Silicone* This rubber has good heat stability and low-temperature flexibility. It is not suitable for high mechanical pressure. Its temperature limit is 600°F (316°C).
9. *Viton* This rubber has good resistance to oils, fuel, and chlorinated solvents. It also has excellent low-temperature properties. Its temperature limit is 600°F (316°C).
10. *Fluorocarbon* This rubber has excellent resistance to most fluids, except synthetic lubricants. The temperature limit is 500°F (260°C).

**Resins.** These usually possess better chemical resistance than rubber. Temperature limitations depend on whether the resin is thermosetting or thermoplastic.

**Tanned Glue and Glycerine.** This combination produces a continuous gel structure throughout the material, allowing sealing at low flange loading. It has good chemical resistance to most oils, fuels, and solvents. It swells in water but is not soluble. The temperature limit is 200°F (93°C). It is used as a saturant in cellulose paper.

**Fillers.** In some cases, inert fillers are added to the material composition to aid in filling voids. Some examples are barytes, asbestos, and cork dust.

### 25.4.3 Reinforcements

Some of the properties of nonmetallic gasket materials can be improved if the gaskets are reinforced with metal or fabric cores. Major improvements in torque retention and blowout resistance are normally seen. Traditionally, perforated or upset metal cores have been used to support gasket facings. A number of designs have been utilized for production. Size of the perforations and their frequency in a given area are the usual specified parameters.

Adhesives have been developed that permit the use of an unbroken metal core to render support to a gasket facing. Laminated composites of this type have certain characteristics that are desired in particular gaskets [25.3].

#### 25.4.4 Metallic Materials

**Aluminum.** This metal has good conformability and thermal conductivity. Depending on the alloy, aluminum suffers tensile strength loss as a function of temperature. Normally it is recommended up to 800°F (427°C). It is attacked by strong acids and alkalies.

**Copper.** This metal has good corrosion resistance and heat conductivity. It has ductility and excellent flange conformability. Normally 900°F (482°C) is considered the upper service temperature limit.

**Steel.** A wide variety of steels—from mild steel to stainless steel—have been used in gasketing. A high clamping load is required. Temperature limits range from 1000 to 2100°F (538 to 1149°C), depending on the alloy.

### 25.5 LOAD-BEARING PROPERTIES

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#### 25.5.1 Conformability and Pressure

Since sealing conditions vary widely depending on the application, it is necessary to vary the load-bearing properties of the gasket elements in accordance with these conditions. Figure 25.2 illustrates stress-compression curves for several gasket components and indicates the difference in the stress-compression properties used for different sealing locations.

Gasket thickness and compressibility must be matched to the rigidity, roughness, and unevenness of the mating flanges. An entire seal can be achieved only if the stress level imposed on the gasket at clampup is adequate for the specific material. Minimum seating stresses for various gasket materials are listed later in this chapter. In addition, the load remaining on the gasket during operation must be high enough to prevent blowout of the gasket. During operation, the hydrostatic end force, which is associated with the internal pressure, tends to unload the gasket. Figure 25.3 is a graphical representation of a gasketed joint depicting the effect of the hydrostatic end force [25.4].

The bolt should be capable of handling the maximum load imposed on it without yielding. The gasket should be capable of sealing at the minimum load resulting on it and should resist blowout at this load level.

Gaskets fabricated from compressible materials should be as thin as possible [25.5]. The gasket should be no thicker than is necessary if it is to conform to the unevenness of the mating flanges. The unevenness is associated with surface finish, flange flatness, and flange warpage during use. It is important to use the gasket's unload curve in considering its ability to conform. Figure 25.4 depicts typical load-compression and unload curves for nonmetallic gaskets.

The unload curve determines the recovery characteristics of the gasket which are required for conformance. Metallic gaskets will show no change in their load and unload curves unless yielding occurs. Load-compression curves are available from gasket suppliers.

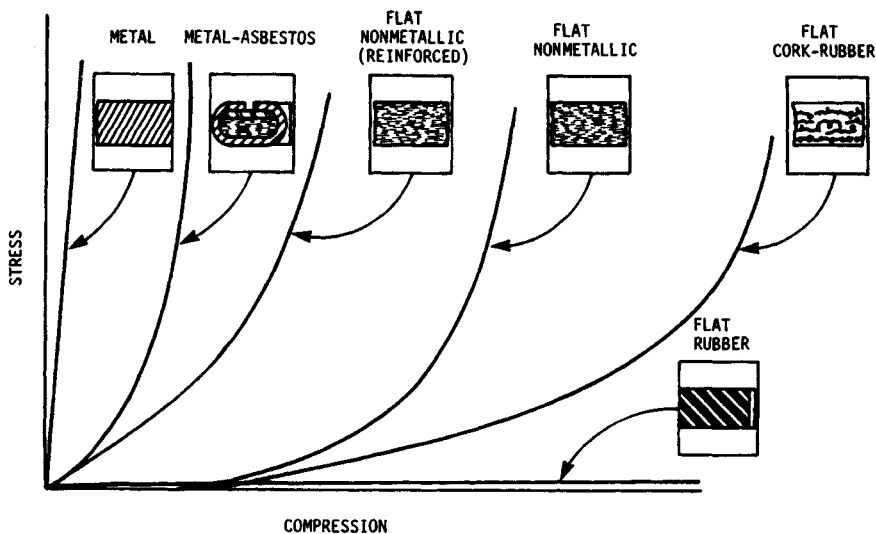


FIGURE 25.2 Stress versus compression for various gasket materials.

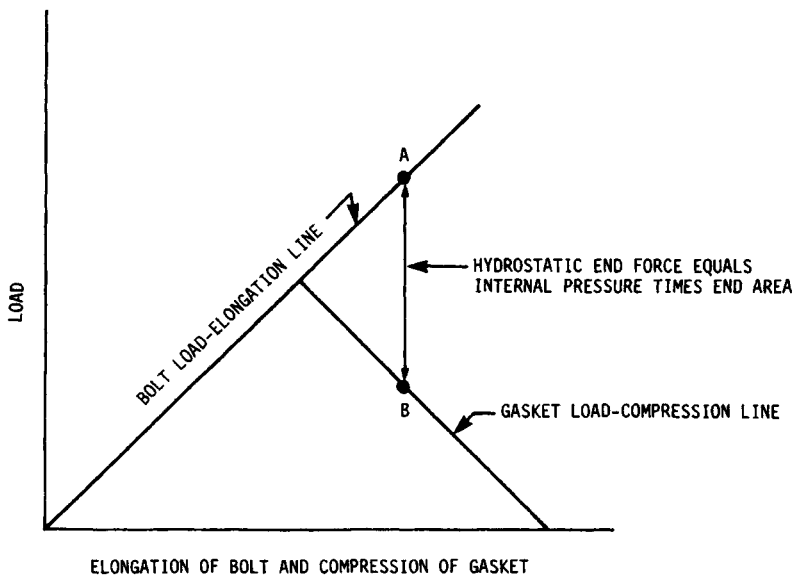
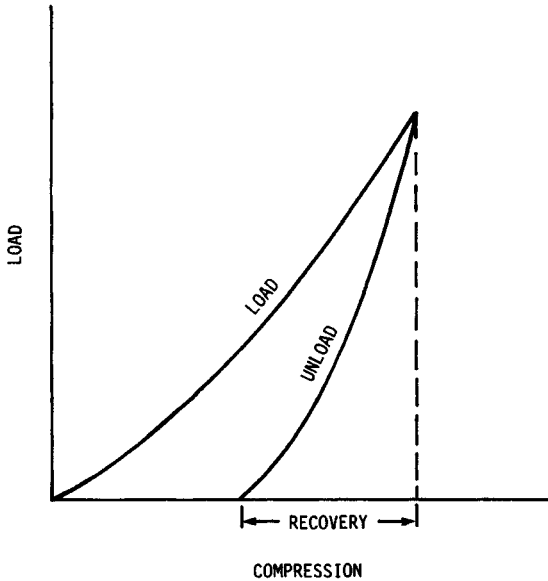


FIGURE 25.3 Graphical representation of a gasketed joint and effect of hydrostatic end force. *A*, Maximum load on gasket; *B*, minimum load on gasket.



**FIGURE 25.4** Load-compression and unload curves for a typical nonmetallic gasket material.

Some advantages of thin gaskets over thick gaskets are

1. Reduced creep relaxation and subsequent torque loss
2. Less distortion of mating flanges
3. Higher resistance to blowout
4. Fewer voids through which sealing media can enter, and so less permeability
5. Lower thickness tolerances
6. Better heat transfer

A common statement in the gasket industry is, "Make the gasket as thin as possible and as thick as necessary."

The following paragraphs describe some of the gasket's design specifications which need to be considered for various applications. A large array of gasket designs and sealing applications are used, and more are coming into use daily. Gaskets are constantly being improved for higher and higher performance.

In high-pressure, clamp load, and temperature applications, a high-spring-rate (stress per unit compression) material is necessary in order to achieve high loading at low compression, thereby sealing the high pressures developed. These applications generally rely on sealing resulting from localized yielding under the unit loading. In addition to the high spring rate, high heat resistance is mandatory. To economically satisfy these conditions, metal is the most commonly used material.

In applications where close tolerances in machining (surface finish and parallelism) are obtainable, a solid steel construction may be used. In those situations where close machining and assembly are not economical, it is necessary to sacrifice some gasket rigidity to allow for conformability. In such cases, conformability

exceeding that resulting from localized yielding must be inherent in the design. The metal can be corrugated, or a composite design consisting of asbestos could be used to gain the conformability required.

In very-high-pressure applications, flat gaskets may not have adequate recovery to seal as the hydrostatic end force unseats the gaskets [25.6]. In these cases, various types of self-energized metal seals are available. These seals utilize the internal pressure to achieve high-pressure sealing. They require careful machining of the flanges and have some fatigue restrictions.

In applications where increased surface conformity is necessary and lower temperatures are encountered, asbestos and/or other nonmetallic materials can be used under the limitations noted earlier.

Elastomeric inserts are used in some fluid passages where conformity with sealing surfaces and permeability are major problems and high fluid pressures are encountered. Since the inserts have low spring rates, they must be designed to have appropriate contact areas and restraint in order to effect high unit sealing stresses for withstanding the internal pressures. The inserts also have high degrees of recovery, which allow them to follow high thermal distortions normally associated in the mating flanges. Compression set and heat-aging characteristics must also be considered when elastomeric inserts are used.

### 25.5.2 Creep and Relaxation

After the initial sealing stress is applied to a gasket, it is necessary to maintain a sufficient sealing stress for the designed life of the unit or equipment. All materials exhibit, in varying degrees, a decrease in applied stress as a function of time, commonly referred to as *stress relaxation*. The reduction of stress on a gasket is actually a combination of two major factors: stress relaxation and creep (compression drift). By definition,

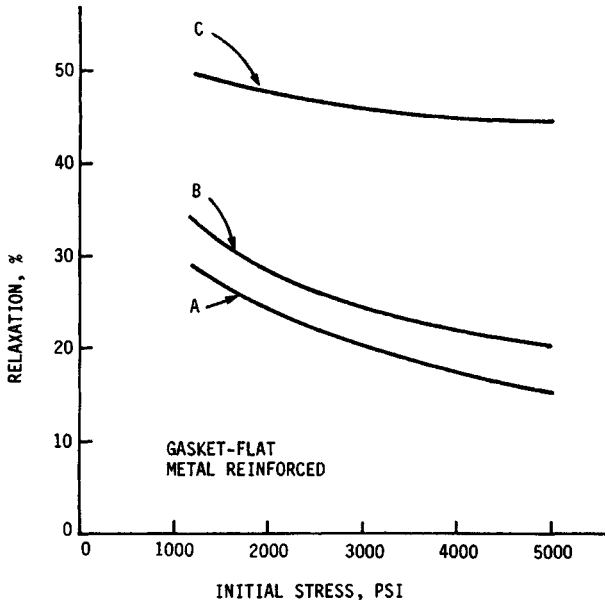
*Stress relaxation* is a reduction in stress on a specimen under constant strain ( $d\sigma/dt; e = \text{constant}$ ).

*Creep* (compression drift) is a change in strain of a specimen under constant stress ( $de/dt; \sigma = \text{constant}$ ).

In a gasketed joint, stress is applied by tension in a bolt or stud and transmitted as a compressive force to the gasket. After loading, stress relaxation and creep occur in the gasket, causing corresponding lower strain and tension in the bolt. This process continues indefinitely as a function of time. The change in tension of a bolt is related to the often quoted "torque loss" associated with a gasket application. Since the change in stress is due to two primary factors, a more accurate description of the phenomenon would be *creep relaxation*, from now on called *relaxation*.

Bolt elongation, or stretch, is linearly proportional to bolt length. The longer the bolt, the higher the elongation. The higher the elongation, the lower the percentage loss for a given relaxation. Therefore, the bolts should be made as long as possible for best torque retention.

Relaxation in a gasket material may be measured by applying a load on a specimen by means of a strain-gauged bolt-nut-platen arrangement as standardized by ASTM F38-62T. Selection of materials with good relaxation properties will result in the highest retained torque for the application. This results in the highest remaining stress on the gasket, which is desirable for long-term sealing.



**FIGURE 25.5** Relaxation versus stress on a gasket: A, 0.030 in–0.035 in thick; B, 0.042 in–0.047 in thick; C, 0.062 in–0.065 in thick.

The amount of relaxation increases as thickness is increased for a given gasket material. This is another reason why the thinnest gasket that will work should be selected. Figure 25.5 depicts the relaxation characteristics as a function of thickness for a particular gasket design.

Note that as clamping stress is increased, relaxation is decreased. This is the result of more voids being eliminated as the stress level is increased.

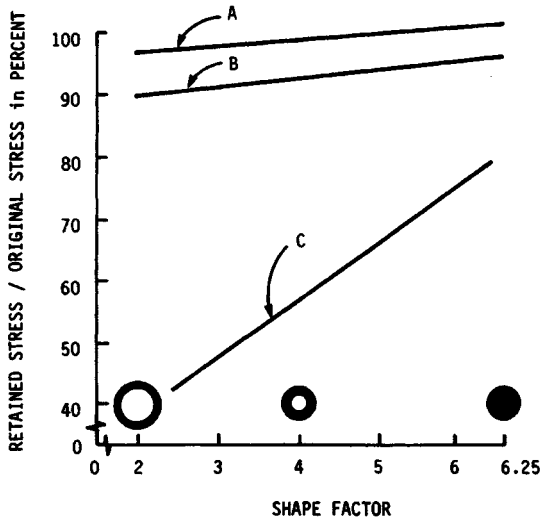
### 25.5.3 Effect of Geometry

The gasket's shape factor has an important effect on its relaxation characteristics. This is particularly true in the case of soft packing materials.

Much of the relaxation of a material may be attributed to the releasing of forces through lateral expansion. Therefore, the greater the area available for lateral expansion, the greater the relaxation. The *shape factor* of a gasket is the ratio of the area of one load face to the area free to bulge. For circular or annular samples, this may be expressed as

$$\text{Shape factor} = \frac{1}{4t} (\text{OD} - \text{ID}) \quad (25.1)$$

where  $t$  = thickness of gasket  
 OD = outside diameter  
 ID = inside diameter



**FIGURE 25.6** Retained stress for various gasket materials versus shape factor of the gasket. A, Asbestos fiber sheet; B, cellulose fiber sheet; C, cork-rubber.

As the area free to bulge increases, the shape factor decreases, and the relaxation will increase as the retained stress decreases. Figure 25.6 depicts the effect of shape factor on the gasket's ability to retain stress.

Note that the shape factor decreases with increasing thickness. Therefore, the gasket should be as thin as possible to reduce relaxation. It must be thick enough, however, to permit adequate conformity. The clamp area should be as large as possible, consistent with seating stress requirements. Often designers reduce gasket width, thereby increasing gasket clamping stress to obtain better sealing. Remember, however, that this reduction might decrease the gasket's shape factor, resulting in higher relaxation over time.

## 25.6 ENVIRONMENTAL CONDITIONS

Many environmental conditions and factors influence the sealing performance of gaskets. Flange design details, in particular, are most important. Design details such as number, size, length, and spacing of clamping bolts; flange thickness and modulus; and surface finish, waviness, and flatness are important factors. Application specifics such as the medium being sealed, as well as the temperatures and pressures involved, also affect the gasket's sealing ability. The material must withstand corrosive attack of the confined medium. In particular, flange bowing is a most common type of problem associated with the sealing of a gasketed joint. The amount of bowing can be reduced by reducing the bolt spacing. For example, if the bolt spacing were cut in half, the bowing would be reduced to one-eighth of its original value [25.7]. Doubling the flange thickness could also reduce bowing to one-eighth of its original value. A method of calculating the minimum stiffness required in a flange is available [25.8].

Different gasket materials and types require different surface finishes for optimum sealing. Soft gaskets such as rubber sheets can seal surface finishes in the vicinity of 500 microinches ( $\mu\text{in}$ ), whereas some metallic gaskets may require finishes in the range of 32  $\mu\text{in}$  for best sealing. Most gaskets, however, will seal adequately in the surface finish range of 63 to 125  $\mu\text{in}$ , with 90 to 110  $\mu\text{in}$  being preferred. There are two main reasons for the surface finish differences: (1) The gasket must be able to conform to the roughness for surface sealing. (2) It must have adequate bite into the mating flange to create frictional forces to resist radial motion due to the internal pressure, thereby preventing blowout. In addition, elimination of the radial micro-motion will result in maintaining the initial clampup sealing condition. Micromotion can result in localized fretting, and a leakage path may be created [25.9].

Because of the complexity that results from the wide variety of environmental conditions, some gaskets for specific applications will have to be designed by trial and error. Understanding Sec. 25.7 will enable a designer to minimize the chance for leaks. Since the factors are so complex, however, adherence to the procedure will not ensure adequate performance in all cases. When inadequate gasket performance occurs, gasket manufacturers should be contacted for assistance.

## **25.7 GASKET DESIGN AND SELECTION PROCEDURE**

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### **25.7.1 Introduction**

The first step in the selection of a gasket for sealing in a specific application is to choose a material that is both chemically compatible with the medium being sealed and thermally stable at the operating temperature of the application. The remainder of the selection procedure is associated with the minimum seating stress of the gasket and the internal pressure involved. In these regards, two methods are proposed: the American Society of Mechanical Engineers (ASME) Code method and the simplified method proposed by Whalen.

### **25.7.2 ASME Code Procedure**

The ASME Code for Pressure Vessels, Sec. VIII, Div. 1, App. 2, is the most commonly used design guide for gasketed joints. An important part of this code focuses on two factors: an  $m$  factor, called the *gasket material factor*, which is associated with the hydrostatic end force, and a  $y$  factor, which is the minimum seating stress associated with particular gasket material. The  $m$  factor is essentially a safety factor to increase the clamping load to such an amount that the hydrostatic end force does not unseat the gasket to the point of leakage. The factors were originally determined in 1937, and even though there have been objections to their specific values, these factors have remained essentially unchanged to date. The values are only suggestions and are not mandatory.

This method uses two basic equations for calculating required bolt load, and the larger of the two calculations is used for design. The first equation is associated with  $W_{m2}$  and is the required bolt load to initially seat the gasket:

$$W_{m2} = \pi b G y \quad (25.2)$$

The second equation states that the required bolt operating load must be sufficient to contain the hydrostatic end force and simultaneously maintain adequate compression on the gasket to ensure sealing:

$$W_{m1} = \frac{\pi}{4} G^2 P + 2b\pi GmP \quad (25.3)$$

where  $W_{m1}$  = required bolt load for maximum operating or working conditions, lb

$W_{m2}$  = required initial bolt load at atmospheric temperature conditions without internal pressure, lb

$G$  = diameter at location of gasket load reaction, generally defined as follows: When  $b_0 \leq \frac{1}{4}$  in,  $G$  = mean diameter of gasket contact face, in; when  $b_0 > \frac{1}{4}$  in,  $G$  = outside diameter of gasket contact face less  $2b$ , in

$P$  = maximum allowable working pressure, psi

$b$  = effective gasket or joint-contact-surface seating width, in

$2b$  = effective gasket or joint-contact-surface pressure width, in

$b_0$  = basic gasket seating width per Table 25.4 (the table defines  $b_0$  in terms of flange finish and type of gasket, usually from one-half to one-fourth gasket contact width)

$m$  = gasket factor per Table 25.3 (the table shows  $m$  for different types and thicknesses of gaskets ranging from 0.5 to 6.5)

$y$  = gasket or joint-contact-surface unit seating load, psi (per Table 25.3, which shows values from 0 to 26 000 psi)

Tables 25.3 and 25.4 are reprints of Tables 2-5-1 and 2-5-2 of the 1980 ASME Code [25.10].

To determine bolt diameter based on required load and a specified torque for the grade of bolt, the following is used:

$$W_b = 0.17DT \quad (\text{for lubricated bolts}) \quad (25.4)$$

or 
$$W_b = 0.2DT \quad (\text{for unlubricated bolts}) \quad (25.5)$$

where  $W_b$  = load per bolt, lb

$D$  = bolt diameter, in

$T$  = torque for grade of bolt selected, lb · in

Note that  $W_b$  is the load per bolt and must be multiplied by the number of bolts to obtain total bolt load.

To determine the bolt diameter based on the required load and the allowable bolt stress for a given grade of bolt, use

$$W_b = \sigma_b A_b \quad (25.6)$$

where  $W_b$  = load per bolt, lb

$\sigma_b$  = allowable bolt stress for grade of bolt selected, psi

$A_b$  = minimum cross-sectional area of bolt, in<sup>2</sup>

### 25.7.3 Simplified Procedure

A simpler method of calculation has been suggested by Whalen [25.11]. This method is also based on the seating stress  $\sigma_g$  on the gasket, as shown in Table 25.5, and on the

hydrostatic end force involved in the application. Basically, Whalen's equations accomplish the same thing as the Code, but they are simplified since they use the full gasket contact width, regardless of the flange width and the surface finish of the sealing faces.

This method is based on the total bolt load  $F_b$  being sufficient to

1. Seat the gasket material into the flange surface
2. Prevent the hydrostatic end force from unseating the gasket to the point of leakage

In the first case, Table 25.5 lists a range of seating-stress values. The ranges shown were found in a search of the literature on gasket seating stresses. Gasket suppliers can be contacted to confirm these values.

Table 25.6 depicts various gasket types and comments on them. As a starting point in the design procedure, the mean value of  $\sigma_g$  could be used. Then, depending on the severity of the application and/or the safety factor desired, the upper and lower figures could be utilized.

Two equations are associated with this procedure. The first is

$$F_b = \sigma_g A_g \quad (25.7)$$

where  $F_b$  = total bolt load, lb  
 $\sigma_g$  = gasket seating stress, psi (from Table 25.5)  
 $A_g$  = gasket contact area, in<sup>2</sup>

This equation states that the total bolt load must be sufficient to seat the gasket when the hydrostatic end force is not a major factor. The second equation associated with the hydrostatic end force is

$$F_b = KP_t A_m \quad (25.8)$$

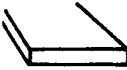
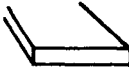





where  $P_t$  = test pressure or internal pressure if no test pressure is used  
 $A_m$  = hydrostatic area on which internal pressure acts (normally based on gasket's middiameter)  
 $K$  = safety factor (from Table 25.7)

The safety factors  $K$  from Table 25.7 are based on the joint conditions and operating conditions but not on the gasket type or flange surface finish. They are similar to the  $m$  factors in the ASME Code. Equation (25.8) states that the total bolt load must be more than enough to overcome the hydrostatic end force. The middiameter is used in  $A_m$  since testing has shown that just prior to leakage, the internal pressure acts up to the middiameter of the gasket.






After the desired gasket has been selected, the minimum seating stress, as given in Table 25.5, is used to calculate the total bolt load required by Eq. (25.7). Then the bolt load required to ensure that the hydrostatic end force does not unseat the gasket is calculated from Eq. (25.8). The total bolt load  $F_b$  calculated by Eq. (25.7) must be greater than the bolt load calculated in Eq. (25.8). If it is not, then the gasket design must be changed, the gasket's area must be reduced, or the total bolt load must be increased.

Both the ASME procedure and the simplified procedure are associated with gasketed joints which have rigid, usually cast-iron flanges, have high clamp loads, and generally contain high pressures. A great many gasketed joints have stamped-metal covers and splash or very low fluid pressure. In these cases, the procedures do not

**TABLE 25.3** Gasket Materials and Contact Facings<sup>1</sup>  
*Gasket Factors m for Operating Conditions and Minimum Design Seating Stress  $\gamma$*

Gasket material	Gasket factor m	Minimum design seating stress $\gamma$ , psi	Sketches	Facing sketch and column to be used from Table 25-4
Self-energizing types (O-rings, metallic, elastomer, other gasket types considered as self-sealing)	0	0		
Elastomers without fabric or high percentage of asbestos fiber: Below 75A Shore Durometer 75A or higher Shore Durometer	0.50 1.00	0 200		(1a), (1b), (1c), (1d), (4), (5); column II
Asbestos with suitable binder for operating conditions: $\frac{1}{8}$ in thick $\frac{1}{16}$ in thick $\frac{1}{32}$ in thick	2.00 2.75 3.50	1 600 3 700 6 500		(1a), (1b), (1c), (1d), (4), (5); column II
Elastomers with cotton fabric insertion	1.25	400		(1a), (1b), (1c), (1d), (4), (5); column II
Elastomers with asbestos fabric insertion (with or without wire reinforcement): 3-ply 2-ply 1-ply	2.25 2.50 2.75	2 200 2 900 3 700		(1a), (1b), (1c), (1d), (4), (5); column II
Vegetable fiber	1.75	1 100		(1a), (1b), (1c), (1d), (4), (5); column II
Spiral wound metal, asbestos-filled: Carbon Stainless or Monel	2.50 3.00	10 000 10 000		(1a), (1b); column II
Corrugated metal, asbestos inserted or corrugated metal, jacketed asbestos-filled: Soft aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	2.50 2.75 3.00 3.25 3.50	2 900 3 700 4 500 5 500 6 500		(1a), (1b); column II

**TABLE 25.3** Gasket Materials and Contact Facings<sup>†</sup>Gasket Factors  $m$  for Operating Conditions and Minimum Design Seating Stress  $y$  (Continued)

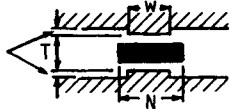
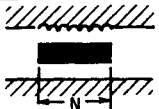
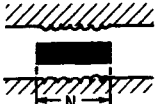

Gasket material	Gasket factor $m$	Minimum design seating stress $y$ , psi	Sketches	Facing sketch and column to be used from Table 25-4
<b>Corrugated Metal:</b> Soft aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	2.75 3.00 3.25 3.50 3.75	3 700 4 500 5 500 6 500 7 600		(1a), (1b), (1c), (1d); column II
<b>Flat metal, jacketed asbestos-filled:</b> Soft aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	3.25 3.50 3.75 3.50 3.75 3.75	5 500 6 500 7 600 8 000 9 000 9 000		(1a), (1b), (1c), † (1d), ‡ (2) †; column II
<b>Grooved metal:</b> Soft aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	3.25 3.50 3.75 3.75 4.25	5 500 6 500 7 600 9 000 10 100		(1a), (1b), (1c), (1d), (2), (3); column II
<b>Solid flat metal:</b> Soft aluminum Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	4.00 4.75 5.50 6.00 6.50	8 800 13 000 18 000 21 800 26 000		(1a), (1b), (1c), (1d), (2), (3), (4), (5); column I
<b>Ring joint:</b> Iron or soft steel Monel or 4-6% chrome Stainless steels	5.50 6.00 6.50	18 000 21 800 26 000		(6); column I

† This table gives a list of many commonly used gasket materials and contact facings with suggested design values of  $m$  and  $y$  that have generally proved satisfactory in actual service when using effective gasket seating width  $b$  given in Table 25.4. The design values and other details given in this table are only suggested and are not mandatory.

‡ The surface of a gasket having a lap should not be against the nubbin.

**TABLE 25.4** Effective Gasket Width<sup>†</sup>

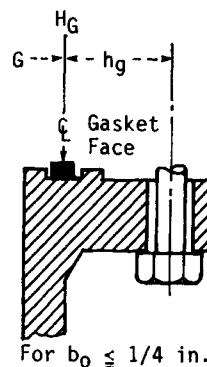
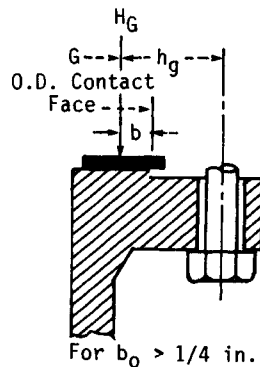
Facing sketch (exaggerated)		Basic gasket seating width $b_0$	
		Column I	Column II
(1a)		$\frac{N}{2}$	$\frac{N}{2}$
(1b)‡			
(1c)		$\frac{w + T}{2} \left( \frac{w + N}{4} \text{ max.} \right)$	$\frac{w + T}{2} \left( \frac{w + N}{4} \text{ max.} \right)$
(1d)‡			
(2) ¼" Nubbin		$\frac{w + N}{4}$	$\frac{w + 3N}{8}$

(3) ¼" Nubbin	 $w \leq \frac{N}{2}$	$\frac{N}{4}$	$\frac{3N}{8}$
(4)‡		$\frac{3N}{8}$	$\frac{7N}{16}$
(5)‡		$\frac{N}{4}$	$\frac{3N}{4}$
(6)		$\frac{w}{8}$	

S81 Effective gasket seating width  $b$ :

$$b = b_o \text{ when } b_o \leq \frac{1}{4} \text{ in} \quad b = 0.5 \sqrt{b_o} \text{ when } b_o > \frac{1}{4} \text{ in}$$

Location of gasket load reaction:








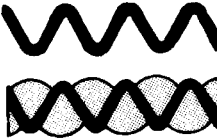


†The gasket factors listed apply only to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.  
‡Where separations do not exceed 1/8-in-depth and 1/2-in-width spacing, sketches (1b) and (1d) shall be used.

**TABLE 25.5** Minimum Recommended Seating Stresses for Various Gasket Materials

	Material	Gasket type	Minimum seating stress range ( $S_p$ ), psi†	
Nonmetallic	Asbestos fiber sheet $\frac{1}{8}$ in thick $\frac{1}{16}$ in thick $\frac{1}{2}$ in thick	Flat	1400 to 1600 3500 to 3700 6000 to 6500	
	Asbestos fiber sheet $\frac{1}{2}$ in thick	Flat with rubber beads	1000 to 1500 lb/in on beads	
	Asbestos fiber sheet $\frac{1}{2}$ in thick	Flat with metal grommet	3000 to 4000 lb/in on grommet	
	Asbestos fiber sheet $\frac{1}{2}$ in thick	Flat with metal grommet and metal wire	2000 to 3000 lb/in on wire	
	Cellulose fiber sheet	Flat	750 to 1100	
	Cork composition	Flat	400 to 500	
	Cork-rubber	Flat	200 to 300	
	Fluorocarbon (TFE) $\frac{1}{8}$ in thick $\frac{1}{16}$ in thick $\frac{1}{2}$ in thick	Flat	1500 to 1700 3500 to 3800 6200 to 6500	
	Nonasbestos fiber sheets (glass, carbon, aramid, and ceramics)	Flat	1500 to 3000 depending on composition	
	Rubber	Flat	100 to 200	
	Rubber with fabric or metal reinforcement	Flat with reinforcement	300 to 500	
	Metallic	Aluminum	Flat	10 000 to 20 000
		Copper	Flat	15 000 to 45 000 depending on hardness
Carbon steel		Flat	30 000 to 70 000 depending on alloy and hardness	
Stainless steel		Flat	35 000 to 95 000 depending on alloy and hardness	
Aluminum (soft)		Corrugated	1000 to 3700	
Copper (soft)		Corrugated	2500 to 4500	
Carbon steel (soft)		Corrugated	3500 to 5500	
Stainless steel		Corrugated	6000 to 8000	
Aluminum		Profile	25 000	
Copper		Profile	35 000	
Carbon steel		Profile	55 000	
Stainless steel	Profile	75 000		
Jacketed metal-asbestos	Aluminum	Plain	2 500	
	Copper	Plain	4 000	
	Carbon steel	Plain	6 000	
	Stainless steel	Plain	10 000	
	Aluminum	Corrugated	2000	
	Copper	Corrugated	2500	
	Carbon steel	Corrugated	3000	
	Stainless steel	Corrugated	4000	
	Stainless steel	Spiral-wound	3000 to 30 000	

†Stresses in pounds per square inch except where otherwise noted.

TABLE 25.6 Typical Gasket Designs and Descriptions

Type	Cross section	Comments
Flat		Basic form. Available in wide variety of materials. Easily fabricated into different shapes.
Reinforced		Fabric- or metal-reinforced. Improves torque retention and blowout resistance of flat types. Reinforced type can be corrugated.
Flat with rubber beads		Rubber beads located on flat or reinforced material afford high unit sealing pressure and high degree of conformability.
Flat with metal grommet		Metal grommet affords protection to base material from medium and provides high unit sealing stress. Soft metal wires can be put under grommet for higher unit sealing stress.
Plain metal jacket		Basic sandwich type. Filler is compressible. Metal affords protection to filler on one edge and across surfaces.
Corrugated or embossed		Corrugations provide for increased sealing pressure and higher conformability. Primarily circular. Corrugations can be filled with soft filler.
Profile		Multiple sealing surfaces. Seating stress decreases with increase in pitch. Wide varieties of designs are available.
Spiral-wound		Interleaving pattern of metal and filler. Ratio of metal to filler can be varied to meet demands of different applications.

**TABLE 25.7** Safety Factors for Gasketed Joints

<i>K</i> factor	When to apply
1.2 to 1.4	For minimum-weight applications where all installation factors (bolt lubrication, tension, parallel seating, etc.) are carefully controlled; ambient to 250°F (121°C) temperature applications; where adequate proof pressure is applied.
1.5 to 2.5	For most normal designs where weight is not a major factor, vibration is moderate and temperatures do not exceed 750°F (399°C). Use high end of range where bolts are not lubricated.
2.6 to 4.0	For cases of extreme fluctuations in pressure, temperature, or vibration; where no test pressure is applied; or where uniform bolt tension is difficult to ensure.

apply, and the compression and stress distribution discussed next should be considered by the designer.

### ***25.8 GASKET COMPRESSION AND STRESS-DISTRIBUTION TESTING***

After a gasket has been selected and designed for a particular application, two simple tests can be performed to determine the gasket's compressed thickness and stress distribution. Inadequate compression or nonuniform stress distribution could result in a leaking joint. The tests can be performed to check for these possibilities and permit correction to ensure leaktight joints.

**1. Lead pellet test** In this test, lead pellets are used to accurately indicate the compressed thicknesses of a gasketed joint. The pellets, commonly called *lead shot*, are available from local gun supply stores. A size approximately twice the thickness of the gasket should be used. Lead solid-core solder can also be used if desired; the size requirements are the same. Pellets or solder are particularly well suited for doing this test, as they exhibit no recovery after compression, whereas the actual gasket material will almost always exhibit some recovery. The degree of nonuniform loading, flange bowing, or distortion will be indicated by the variations in the gasket's compressed thickness.

To begin, the original thickness of the gasket is measured and recorded at uniformly selected points across the gasket. At or near these points, holes are punched or drilled through the gasket. Care should be taken to remove any burrs. The punched holes should be approximately 1½ times the pellet diameter.

Then the gasket is mounted on the flange. A small amount of grease can be put in the punched holes to hold the lead pellets, if required. The pellets are mounted in the grease, and the mating flange is located and torqued to specifications.

Upon careful disassembly of the flange and removal of the pellets, their thicknesses are measured, recorded, and analyzed. Comparison of the pellets' compressed thicknesses to the gasket's stress-compression characteristics permits the desired stress-distribution analysis.

2. *NCR paper test* This test utilizes no-carbon-required (NCR) paper for visual determination of the stress distribution on a gasket. NCR paper is a pressure-sensitive, color-reactive paper. The intensity of color is proportional to the stress imposed on the paper, which is the same as the stress on the gasket.

NCR paper is available from the NCR Corp., the 3M Company, and other paper companies. Various grades are available, but the medium grade is usually chosen. Some papers are only one sheet, whereas others are composed of two sheets. Either type can be used.

To begin, the bolt holes are pierced in a piece of the impression paper. The pierced holes in the paper are made slightly larger than the bolts. The paper is placed on the flange, and the mating flange is assembled per torque specifications.

When you are using the two-piece carbonless paper, make sure to keep the two papers oriented to each other as they were purchased; otherwise, no impression may result. Upon torquing, the impression is made on the paper. The flange is removed, and the impression is inspected for stress distribution. A judgment of the gasket's sealing ability can now be made. Further analysis can be done by calibrating the load versus the color intensity of the paper. Various known stresses can be applied to the paper and the resulting color impressions identified. The impressions can be compared to the test sample, and then the stress on the sample can be determined.

In both the lead pellet and NCR paper tests, gasket manufacturers can be contacted for further interpretation of the results and more detailed analysis.

## 25.9 INSTALLATION SPECIFICATIONS

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An installation is only as good as its gasket; likewise, a gasket is only as good as its installation [25.1]. The following are some recommendations associated with gasket installation:

1. Be sure that mating surfaces are clean and in specification with regard to finish, flatness, and waviness.
2. Check gasket for damage before installing it.
3. Make certain the gasket fits the application.
4. Specify lubricated bolts. Bolt threads and the underside of the bolt head should be lubricated.
5. Specify the torque level.
6. Specify the torquing sequence. In addition to the sequence, two or three stages of torque before reaching the specified level are recommended.

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