# Elastomer Shielding Design Guide

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This Elastomer Shielding Design Guide describes design techniques by which the gasket can be incorporated into an enclosure. These techniques cover:

a. Seam Design
b. Gasket Design
c. Groove Design
d. Fastener Spacing

## Seam Design

The primary function of an EMI seam gasket is to minimize the coupling efficiency of a seam. The reflection and absorption functions of the EMI gasket are to a large extent masked by metal cover plates and fasteners which provide the major contribution towards the restoration of the enclosure integrity. This fact does not diminish the important role of the EMI gasket in the enclosure design nor the adequate design of the enclosure to minimize enclosure discontinuities.

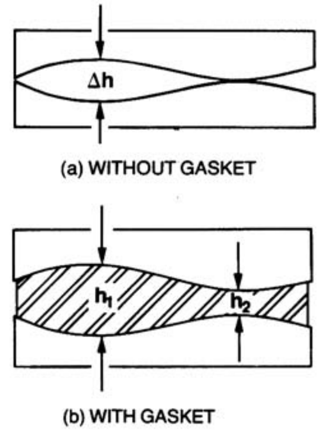
In the design of a shielding enclosure, the impedance between the mating seam surfaces should be as nearly equal to the enclosure material as possible to permit uniform current flow throughout the enclosure. Any significant difference in seam impedance, including that introduced by the gasket materials, can produce nonuniform current flow resulting in the generation of EMI voltages. These voltages can then be sources of radiated energy both into or out of the enclosure. To provide effective shielding, the seam design should incorporate the following features:

a. Mating surfaces should be as flat as economically possible.
b. Flange width should be at least five (5) times the maximum expected separation between mating surfaces.
c. Mating surfaces requiring dissimilar materials should be selected from one of the electrochemical groups shown in Table 6-3.
d. Mating surfaces should be cleaned to remove all dirt and oxide films just prior to assembly of the enclosure parts.
e. Protective coatings having conductivity much less than half that of the mating surfaces should be avoided or the coating removed in the area of mating surfaces.
f. Surfaces requiring a protective coating should be plated with tin, nickel, zinc or cadmium.
g. Fasteners should be tightened from the middle of the longest seam toward the ends to minimize buckling.
h. Bonded surfaces should be held under pressure during adhesive curing to minimize surface oxidation.
i. Edges of exposed seams should be sealed with a suitable protective compound (caulk) and preferably one which is conductive to prevent the intrusion of moisture. Even with these precautions in the manufacturing, preparation and assembly of enclosure parts, mating surfaces are seldom perfect.

## Gasket Design

In EMI shielding, many mechanical and electrical design considerations are interdependent. One of the more important ones is joint uneveness. Joint uneveness refers to the degree of mismatch between mating seam surfaces. It results when the mating surfaces make contact at irregular intervals due to surface roughness or to bowing of cover plates because of improper material selection, thinness of the cover plate, too few fasteners, excessive bolt tightening, or improper gasket selection. Ideally, gaskets should make firm, continuous and uniform contact with seam surfaces. Performance of any shielding product can be degraded by improper application. Joint uneveness is an excellent example of a mechanical restraint which can have an adverse effect on the electrical performance of a gasket.

Figure 7-1 depicts an enlarged cross sectional view of an enclosure seam. Figure 7-la shows the seam without gasketing material joining only at the irregular high spots between the surfaces. In fact, if the cover plate were weightless and zero pressure applied between parts by fasteners, the enclosure and cover plate would only make contact at the three highest points. As pressure is applied, the irregular high spots become flattened resulting in more surface area and more points coming in contact to support the plate. Basically it is the function of a resilient gasket which bridges these gaps but at a much lower closing pressure. The ideal gasket will bridge irregularities within its compression-deflection capabilities without losing its properties of resiliency, stability or conductivity.




$$\Delta h = h_1 - h_2$$

**Figure 7-1, Seam Joint Uneveness**

The maximum joint uneveness is the dimension of the maximum separation between the flanges of the seam when the two surfaces are just touching. This separation is designated as $\Delta h$ as shown in Figure 7-1a.

With a gasket in place, the maximum spacing (h1) between mating surfaces occurs at the minimum gasket compression. Conversely, the minimum spacing (h2) occurs at the maximum gasket compression. The difference between the maximum (h1) and the minimum (h2) spacing is h. The gasket under these extreme conditions undergoes its severest mechanical test at the maximum deflection and severest electrical test at the minimum deflection.

There are, therefore, four important properties of an EMI gasket which must be considered before it is incorporated into an enclosure. These properties are compression (or deflection), compression set, shielding effectiveness and environmental seal. Compression, the reduction in volume of a gasket under pressure, is usually applied to sponge materials or products that are formed with hollow cores. Deflection, the reduction of a dimension due to pressure without necessarily resulting in a change in volume, is applicable to all materials including solid elastomers. Since these terms have been used interchangeably, the term compression is used here. Compression set is the permanent loss of the original height of a gasket after being compressed. It is important therefore to understand the various types of joints in order to determine which gasket properties are most important to a particular design.

**Types of Joints:** There are traditionally three types of joints classified by usage:
**Type I** Permanently mounted cover plates or assemblies. Generally compression set is not of concern in these applications even though high pressures may be encountered. For applications requiring an environmental seal in addition to an EMI seal under high closing forces, an elastomerfilled flat gaskets such as TECKNIT Duolastic Teckfelt or Teckspan are most applicable.

**Type II** Access cover plate with high joint uneveness which is opened frequently but always closes on the same portion of the gasket. A hinged door is an example of a Type II joint. Most of the elastomeric gaskets are suitable for this type of application where the closure pressures are under 100 psi. In the lowest closure pressures, the hollow-shaped elastomers are most suitable. TECKNIT extruded conductive elastomer materials would meet these requirements for low closure force with low compression set. These gaskets need only be replaced as the result of wear and aging or whenever the gaskets are removed.

**Type III** Removable cover plate with a symmetrical mounting pattern which is replaceable but not necessarily in the original orientation. Gaskets for this type of application are removable and reusable. Gasket materials which exhibit low closure force and low compression set would be suitable in most applications.

## Environmental Seals

In many applications, it is desirable to incorporate an environmental seal (fluid or gas) such as neoprene or silicone solid or closed cell sponge elastomer. As a general rule, the degree of seal effectiveness is a function of the gasket deformation or percent compression. These seals must:

a. Be impervious to the fluid(s) or gas(es) being excluded.
b. Be compatible with the environment (including pressure, temperature and vibration) while retaining the original characteristics of resiliency, cohesion and softness (compressibility).
c. Conform uniformly to mating surface irregularities.

There are elastomeric materials besides neoprene and silicone which are suitable environmental seals. The listing below presents the most important characteristics of the more common elastomers.

**a. Neoprene** This elastomer is used commonly in EMI gaskets and will withstand temperatures ranging from —54°C to +100°C for solid and —32°C to +100°C for sponge (closed cell) elastomers. Neoprene is lightly resistant to normal environmental conditions, moisture and to some hydrocarbons. It is the least expensive of the synthetic rubber materials, and is best suited from a cost standpoint for commercial applications.

**b. Silicone** This material has outstanding physical characteristics and will operate continuously at temperatures ranging from —62°C to +260°C for solid and —75°C to +205°C for closed cell sponge elastomers. Even under the severest temperature extremes these materials remain flexible and are highly resistant to water and to swelling in the presence of hydrocarbons.

**c. Buna-n** Butadiene-Acrylonitrile resists swelling in the presence of most oils, has moderate strength and heat resistance although it is not generally suited for low temperature applications.

**d. Natural Rubber** This material has good resistance to acids and alkalies (when specially treated) and can be used to 160°C, is resilient and impervious to water. Rubber will crack in a highly oxidizing (ozone) atmosphere and tends to swell in the presence of oils.

Since most seals used with EMI gaskets have elastomeric properties of stretch and compressibility, some guidelines are needed when specifying the dimensional tolerance of these materials: Figure 7-2 shows some of the common errors encountered in gasket design.

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## COMMON ERRORS IN GASKET DESIGN

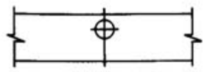
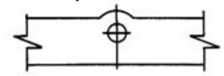
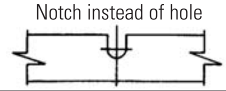
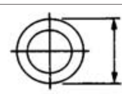
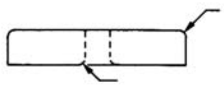
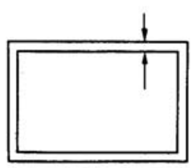
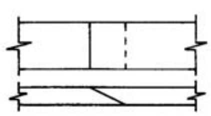
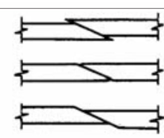
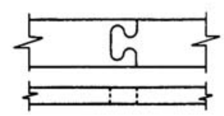
| Detail | Why faulty | Suggested remedy |
|---|---|---|
|  Bolt holes close to edge | Causes breakage in stripping and assembling | Projection or ear"  Notch instead of hole  |
|  Metalworking tolerances applied to gasket thickness, diameters, length, width, etc. | Results in perfectly usable parts being rejected at incoming inspection. Requires time and limits. Increases cost of parts and tooling. Delays deliveries. | Most gasket materials are compressible. Many are affected by humidity changes. Try standard or commercial tolerances before concluding that special accuracy is required. |
|  Transference of fillets, radii, etc., from mating metal parts to gasket | Unless part is molded, such features mean extra operations and higher cost. | Most gasket stocks will conform to mating parts without preshaping. Be sure radii, chamfers, etc., are functional, not merely copied from metal members |
|  Thin walls, delicate cross section in relation to overall size. | High scrap loss; stretching or distortion in shipment or use. Restricts choice to high tensile strength materials. | Have the gasket in mind during early design stages. |
|  Large gaskets made in sections with beveled joints |  Extra operations to skive. Extra operations to glue. Difficult to obtain smooth, even joints without steps or transverse grooves. |  Die-cut dovetail joint |

**Figure 7-2, Gasket Design Errors**

a. Minimum gasket width should not be less than one half of the thickness (height).
b. Minimum distance from bolt hole (or compression stop) to nearest edge of sealing gasket should not be less than the thickness of the gasket material. When bolt holes must be closer, use U-shaped slots.
c. Minimum hole diameter not less than gasket thickness.
d. Tolerances should be conservative whenever possible. Standard tolerances for die-rule cut gaskets (Table 7-1) should not be closer than:

**Figure 7-1, Gasket Tolerances**

| Solld elastomer: | Tolerances |
|---|---|
| Up to 150mm (6.0"): | ± 0.4mm (0.016") |
| Over 150mm (6.0"): | ± 0.8mm (0.032") |
| Holes: | ± 0.4mm (0.016) |
| **Sponge elastomer:** | **Tolerances** |
| Up to 100mm (4.0"): | ± 0.8mm (0.032") |
| Over 100mm (4.0"): | ± 1.6mm (0.063") |
| Holes: | ± 0.8mm (0.032') |

e. Cross section tolerances (Table 7-2) of elastomer strips should be:

| (1) WIdth Dimensions | Tolerance | |
|---|---|---|
| | **Solid** | **Sponge** |
| Up to 3.2mm (0.125") | ± 0.4mm (0.016") | ± 0.4mm (0.016") |
| 32 to 6.4mm (0.125"-0.250") | ± 0.4mm (0.016") | ± 0.8mm (0.032") |
| 6.4 to 19mm (0.250"-0.750") | ± 0.8mm (0.032") | ±1.2mm (0.047") |
| Over 19mm (0.7509) | ± 1.2 mm (0.047") | ± 1.6mm(0.063") |
| **(2) Height Dimensions** | **Tolerance** | |
| | **Solid** | **Sponge** |
| Up to 19mm (0.750"): | ± 0.25mm (0.010") | ± 025mm (0.010") |

Note: Check specific product data sheet specification for tolerance limitations.

## Closure Pressure

Shielding effectiveness and closure pressure have a general relationship as shown in Figure 7-3. The minimum closure force (Pmin) is the recommended applied force to establish good shielding effectiveness and to minimize the effects of minor pressure difference. The maximum recommended closure force (Pmx) is based on two criteria: (1) maximum compression set of 10% and/or (2) avoidance of possible irreversible damage to the gasket material when pressure exceeds the recommended maximum. Higher closure pressures may be applied to most knitted wire mesh gaskets when used in Type I joints, but the gaskets should be replaced when cover plates are removed, i.e., whenever the seam is opened.

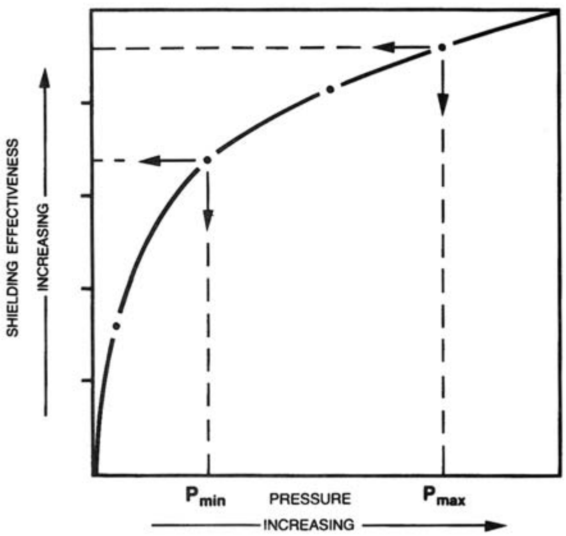



**Figure 7-3, Shielding Effectiveness Versus Closure Force (Typical characteristics at a given frequency)**

## Compression Set

Selection of a gasketing material for a seam which must be opened and closed is to a large extent determined by the compression set characteristics of the gasket material. Most resilient gasket materials will recover most of their original height after a sufficient length of time when subjected to moderate closing forces. The difference between the original height and the height after the compression force is removed is compression set. As the deflection pressure is increased, the compression set increases (See Figure 7-4).

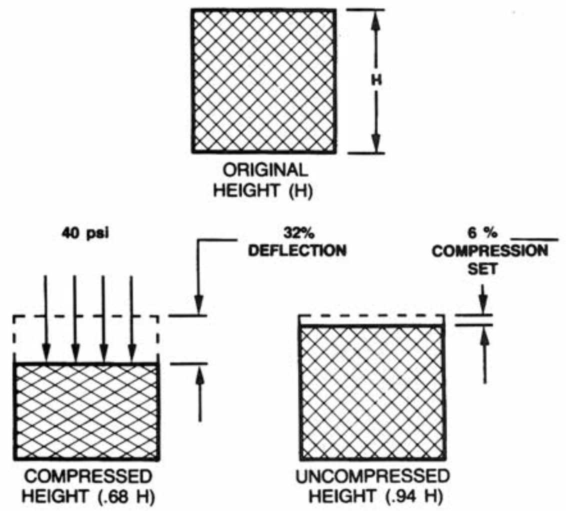



**Figure 7-4, Compression Set**

## General Compression/Deflection Curves

Compression/deflection curves can be used to determine the following gasket characteristics:

1. Gasket height needed to compensate for joint uneveness.
2. Gasket closing pressure needed to assure good shielding.
3. Gasket compression set as a function of applied pressure.

The data presented is representative of the general characteristics of the materials depicted. Variation in the values presented can be expected as a result of manufacturing tolerances, density of material, variation in hardness (durometer) and variations in cross sections. Figures 7-5 through Figure 7-8 cover knitted wire mesh, oriented wires in solid elastomer, oriented wires in a sponge elastomer and a medium durometer (45) elastomer Minimum gasket height can be calculated from the data presented for rectangular cross sections.

**Example**, Figure 7-5 knitted wire mesh gasket shows a minimum recommended closing pressure of 138 kPa (20 psi) and a maximum recommended closing pressure of 414 kPa (60 psi). Below 138 kPa (20 psi), a significant falloff in shielding effectiveness can be expected while above 414 kPa (60 psi) high compression iet may result. Using these minimum (Pmjn) and maximum (Pmax) pressure values and extending them to the compression/deflection curve, minimum and maximum compression values (percentage of original gasket height H) can be determined. In the case of the knitted wire mesh, the minimum recommended deflection is 80% of the original

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height (or 0.8H), and the maximum recommended deflection is 60% (or 0.6H). The difference in gasket height then is:

$$\Delta h = 0.8H - 0.6H = 0.2H$$

Using this value with the known or anticipated joint un- eveness, the minimum gasket height can be calculated. For purposes of this example, assume joint uneveness (h) is 0.06".

$$\Delta h = h_1 — h_2 = 0.06"$$

For minimum gasket height, the maximum compression difference (Δh) must equal the maximum joint uneveness (Δh), ΔH = Δh. Substituting for ΔH (0.2H) and for Δh (006")

$$0.2H = 0.06"$$
$$H_{min} = \frac{0.06}{0.2} = 0.30"$$

This value is the minimum gasket height which will accommodate the required pressure range, shielding effectiveness, compression set and joint uneveness when using a knitted wire mesh gasket. Any gasket with a height greater than 0.30" should be suitable for the depicted example.

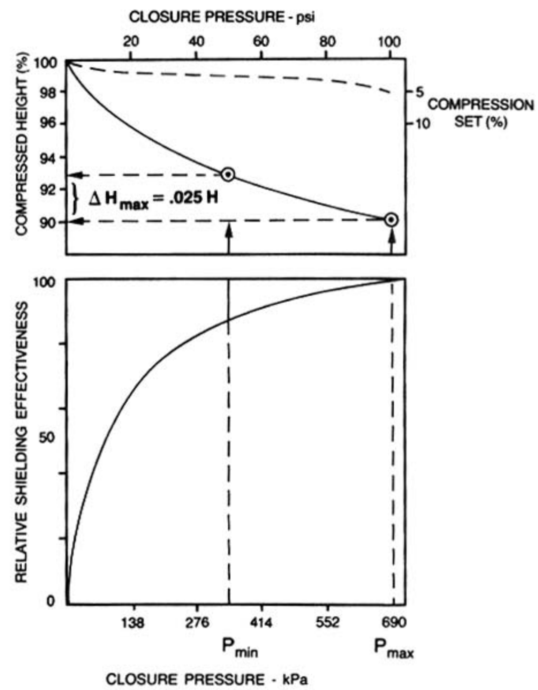


**Figure 7-5, Knltted Wire Mesh Gasket**

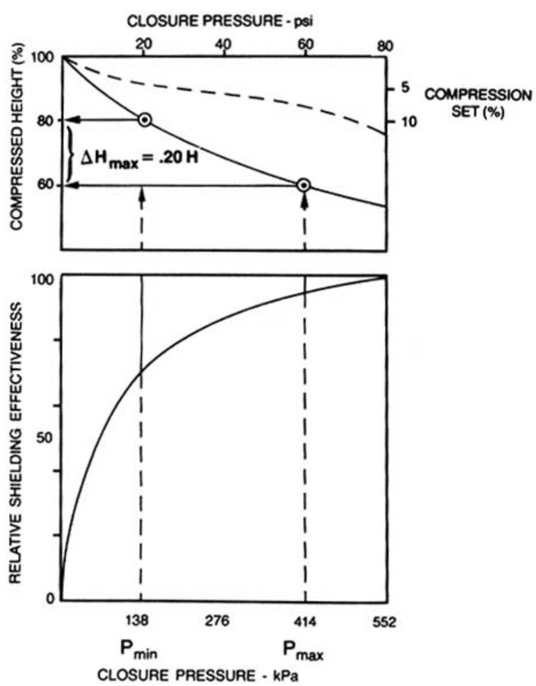


**Figure 7-6, Orlented Wires in Solid Elastomer**

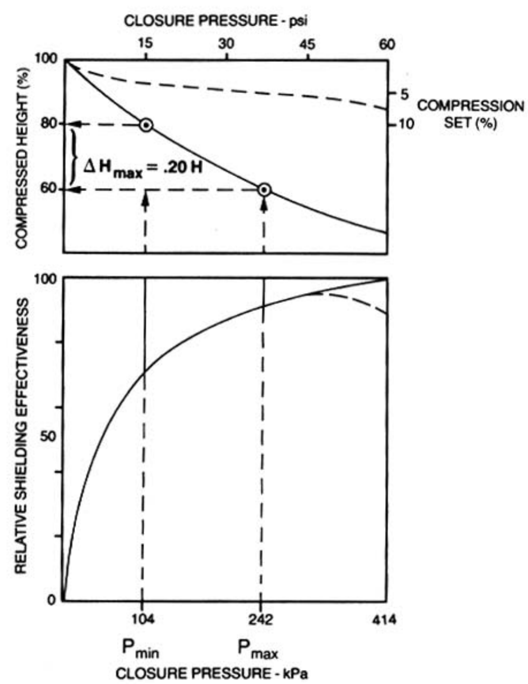


**Figure 7-7, Oriented Wires in Sponge Elastomer**

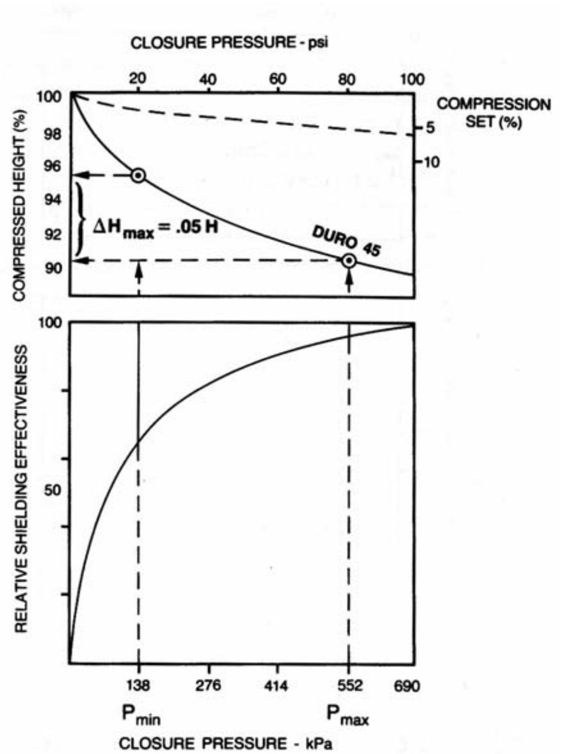


Figure 7-8, Conductive Solid Elastomer (CONSIL)

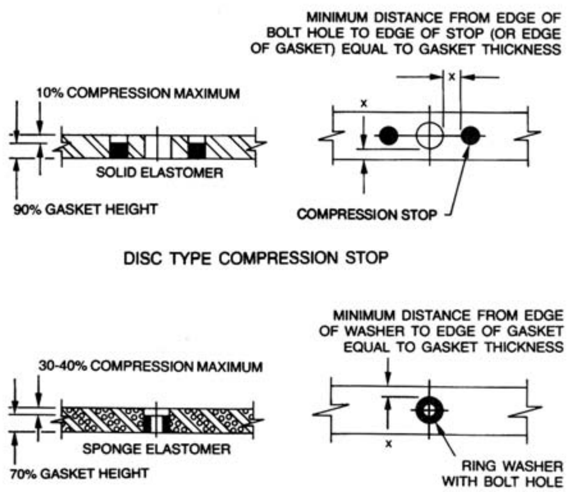


Figure 7-9, Compression Stops

## Compression Stops

In order to avoid damage to the gasket or excessive bowing of the cover plate from gasket overcompression, discs or washer type compression stops can be provided as an integral part of the gasket assembly. Compression stops are stamped out from standard gauge sheet or Cut to thickness from rod or tubing. Materials commonly used are aluminum and stainless steel. For sponge elastomers, such as DUOSTRIPS/DUOGASKETS or ELASTOFOAM, compression stops should be cut to a maximum of 80% of the elastomer thickness and a minimum of 65%. For solid elastomers, such as ELASTOMET or CONSIL materials the compression stops should be 90% to 95% of the gasket height.

Some typical compression stop assemblies are shown in Figure 7-9. Another form of compression stop is to confine the gasket by means of a groove such that the cover plate flange mates with enclosure flange, thereby effecting a compression stop.

## Groove Design

A groove for retaining a gasket assembly provides several advantages:

1. Can act as a compression stop.
2. Prevents overcompression.
3. Provides a fairly constant closure force under repeated opening and closing of the seam.
4. Provides a moisture and pressure seal when properly designed.
5. Cost effective in lowering assembly time and cost of gasketing material.
6. Best overall EMI sealing performance.

Solid elastomers are not compressible. They are easily deformed but do not change in volume as do sponge elastomers. Therefore, allowance for material flow must be considered in the groove design. If the groove cross section (volume), when the cover flange is fully closed, is insufficient to contain the fully deflected material, proper closure of the flange may be difficult. In addition, over- stressing of the material may degrade electrical and physical properties of the shielding material. Figure 7-10 depicts the various conditions of groove design.

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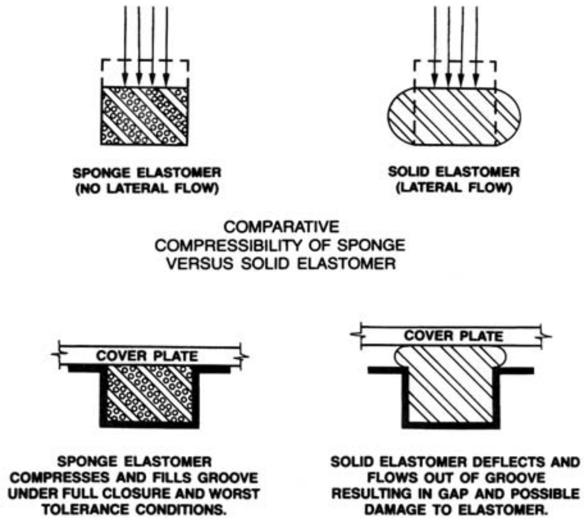
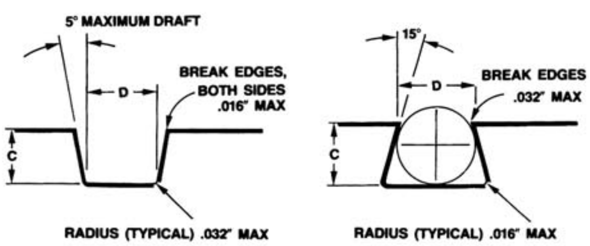



GROOVE DESIGNS

**Figure 7-10, Groove Design Considerations**

Figure 7-11 shows the design for two different grooves. Figure 7-11a depicts a typical rectangular groove, while Figure 7-11b shows a design which can mechanically retain circular cross section (cords) gaskets by side friction.




UNRETAINED-GASKET GROOVE (MOST CROSS SECTIONS)

RETAINED-GASKET GROOVE

**Figure 7-11, Groove Designs**

The design of the rectangular groove is relatively simple. The critical dimension is dimension "C", the depth of the groove as shown in Figure 7-11.

Groove design must also take into account the dimensional tolerances of the groove and the elastomer gasket. For small gasket cross sections up to 2.5 mm (0.10"), the best tolerances are obtained from extruded materials. Table 7-2 lists typical standard tolerances for strip and molded products and Table 7-3 (below) lists typical tolerances for extruded products such as CONSIL-E and SC-CONSIL.

**Table 7-3, Extruded Product Tolerances**

| DIMENSIONS | TOLERANCES |
|---|---|
| Under 2.5 mm (0.10"): | ± 0.13 mm (0.005") |
| 2.5 to 5.1 mm (0.10" to 0.20"): | ± 0.25 mm (0.010") |
| 5.1 to 7.6 mm (0.20" to 0.30"): | ± 0.38 mm (0.015") |
| Over 7.6 mm (0.30"): | ± 0.51 mm (0.020") |

Use the following steps to calculate the "C" and "D" groove dimensions:

1. Determine the maximum useful compression as a percentage of the original gasket height. This value should be the maximum compression which will not result in permanent damage to the gasket shielding or sealing properties (refer to Figures 7-5 through 7-8 for typical properties and to specific data sheets where applicable).
2. Determine the minimum useful compression value from Figures 7-5 through 7-8.
3. Calculate the maximum cross section of the gasket by adding the plus tolerance to the nominal value. Table 7-4 provides form-factors for three common cross sections.

**Table 7-4 Gasket Configuration**

| Shape | Maximum Height (H max) | Minimum Height (H mm) | Form Factor | Maximum Cross Section Area (S max) |
|---|---|---|---|---|
| Rectangular (H x W) | (H + tol)* | (H - tol) | 1 | (H + tol)(W + tol) |
| Round (dia) | (dia + tol) | (dia - tol) | 0.785 | $0.785\,(dia + tol)^2$ |
| "D" Shape (A) | (A + tol) | (A - tol) | 0.893 | 0.893 (A + tol)2 |

tol = tolerance = one half of the total allowable tolerance around the nominal value.

After determining the maximum and minimum gasket height and the maximum cross section area of the gasket (see Table 7-4), the C-dimension can be calculated from the following relationships:

$C_{min}$ = minimun groove depth
$C_{max}$ = maximum groove depth
$C_{nom}$ = nominal groove depth (average)
$C_{01}$ = maximum compression as a fraction of original height

$C_{02}$ = minimum compression as a fraction of original height and

$$C_{min} = (C_{01})\ (H_{max}),$$

where $H_{max}$ = nominal height ($H_o$) of gasket before compression plus the upper tolerance ($H_o$ + tol).

$$C_{max} = (C_{02})\ (H_{min}),$$

where $H_{min}$ = nominal height ($H_o$) of gasket before compression minus the lower tolerance ($H_o$ - tol).

$$C_{nom} = \frac{C_{min} + C_{max}}{2}$$

The D-dimension (groove width) can be calculated from:

$$D_{min} = \frac{S_{max}}{C'_{min}},$$

where $S_{max}$, maximum cross sectional area of gasket (reference Table 7-4, and:

$C'_{min} = C_{nom}$ — lower tolerance
$D_{nom} = D_{min}$ + lower tol + allowance
$D_{max} = D_{nom}$ + upper tol,

where the upper tolerance is the value of the positive tolerance, and:

$D_{nom}$ = nominal value of the groove width
$D_{max}$ = maximum value of the groove width
Allowance = an added value to account for the use of adhesives and for groove design features such as inside radii.

**EXAMPLE**, calculate the groove dimensions for a 0.125" diameter round cross section solid elastomer gasket with a diameter tolerance of plus and minus 0.010". Determine first $C_{min}$ and $C_{max}$ from a 70% maximum compression ($C_{01}$) and a 90% minimum compression ($C_{02}$):

$C_{min} = (C_{01})\ (H_{max})=(0.7)\ (0.125+0.010)=0.0945"$
$C_{max} = (C_{02})\ (H_{min})=(0.9)\ (0.125-0.010)=0.1035"$
$C_{nom} = \frac{0.0945+0.1035}{2}=0.099\pm0.0045$

The tolerance on the C-dimension is critical in maintaining the compression range within the limits specified, especially for the smaller cross sections. A maximum tolerance for the C-dimension for this size gasket should be limited to ± 0.0045.

It is sometimes desirable to specify a unilateral (one directional) tolerance which is permitted to vary in only one direction from the nominal or design size. Unilateral tolerances should be used in the design of the groove depth where it is important to ensure that the design favors either the high compression or low compression forces. A negative (minus) unilateral tolerance tends to favor slightly higher compression forces while a positive (plus) unilateral tolerance tends to favor slightly lower compression forces.

In the groove example, since the tolerance is tight, it is desirable to use a unilateral tolerance for the depth dimension to ensure that the gasket is not overcompressed. Using a unilaterial tolerance of + 0.006", which should favor the lower compression forces, the C-dimension would be expressed as 0.096," + 0.006/—0.000 and the $C_{min}$ would equal 0.096", the $C_{max}$ would equal 0.102, well within the mm/max dimensions calculated.

The groove width (D) can now be calculated using the groove width equations above and Table 7-4. For the above example:

$$D_{min}=\frac{S_{max}}{C'_{min}}=\frac{(.785)\ (.125 + .010)^2}{(0.096 - 0.000)} = 0.149"$$

$D_{min}=D_{min}$+ lower tolerance+ allowance
=0.149 .006+.010=0.165"

where tolerance for the width dimension is ±0.006", see Table 7-6.

Tables 7-5 (rectangular strips), Table 7-6 (round strips) and Table 7-7 ("0" shape strips) provide suggested values for "C" and "D" groove dimensions with suggested tolerances which will maintain the gasket within the suggested compression range of 70% to 90% of original height.

**Table 7-5, Groove Dimensions**
**Rectangular Gasket**
$C_{01}$ = .7 (max compresson)
$C_{02}$ = .9 (min compression)

| Strip H (inches) | Strip W (inchs) | Groove Dimension (Inch) C ± tol | Groove Dimension (Inch) D ± 006" |
|---|---|---|---|
| .030 ± .005 | .125 ± .010 | .022 ± .002 - .000 | .231 |
| .060 ± .005 | .125 ± .010 | .045 + .004 - .000 | .211 |
| .093 ± .005 | .188 ± .010 | .071 + .006 - .000 | .289 |
| .125 ± 010 | .250 ± 015 | .096 + .006 - .000 | .389 |
| .188 ± .010 | .375 ± .020 | .150 ± .006 | .559 |
| .250 ± 015 | .500 ± .020 | .199 ± .006 | .730 |

Relerence TECKNIT Data Sheet D-810

# Elastomer Shielding Design Guide

*U.S. Customary*
*[SI Metric]*

**Table 7-6, Groove Dimensions**
**Round Gasket**
$C_{01}$ = .7 (max compresson)
$C_{02}$ = .9 (min compression)

| Diameter (inch) | Groove Dimensions (inch) Depth C ± tol | Width D ± .006 |
|---|---|---|
| .062 ± .005 | .046 + .006 - .000 | .093 |
| .093 ± .005 | .071 + .006 - .000 | .122 |
| .103 ± .005 | .087 | .135 |
| .125 ± .010 | .096 + .006 - .000 | .165 |
| .188 ± .010 | .150 ± .006 | .230 |
| .250 ± .015 | .199 ± .006 | .302 |
| .375 ± .020 | .298 ± .006 | .436 |

Relerence TECKNIT Data Sheet D-810

**Table 7-7, Groove Dimensions**
**"D" SHAPE Gasket**
$C_{01}$ = .7 (max compresson)
$C_{02}$ = .9 (min compression)

| A (inch) | Groove Dimensions (inch) Depth C ± tol | Width D ± .006 |
|---|---|---|
| .062 ± .005 | .046 + .006 - .000 | .103 |
| .093 ± .005 | .071 + .006 - .000 | .137 |
| .125 ± .010 | .096 + .006 - .000 | .186 |
| .188 ± .010 | .146 + .006 - .000 | .256 |
| .250 ± .015 | .199 + .006 - .000 | .336 |
| .375 ± .020 | .295 + .006 - .000 | .488 |

Relerence TECKNIT Data Sheet D-810

## Fastener Spacing

Fasteners are normally required between cover plate and enclosure to provide enough closing force along the seam length to insure adequate contact pressure and to compensate for joint uneveness. Fastener spacing, cover plate thickness, minimum-maximum pressures, gasket compressibility and material characteristics are important parameters in the cover plate design.

Maximum gasket deflection occurs at the fastener locations where the maximum compressive force is applied. Frequently the closure forces required to compress a resilient gasket is sufficient to cause bowing of the cover plate. The amount of bowing depends on several interrelated factors. Figure 7-12 shows the result of high fastener pressure on cover plate bowing. The bowing can be severe enough that insufficient pressure is applied at the mid section of the gasket resulting in little or no shielding or even the development of a slit gap. These effects can be minimized by proper spacing, proper cover plate thickness and proper selection of gasket materials. The basic equation for bolt spacing (reference Figure 7-13) is given as:

$$C = \left[ \frac{480\,(a/b)\,E\,t^3\,\Delta H}{13P_{min} + 2P_{max}} \right]^{1/4}$$

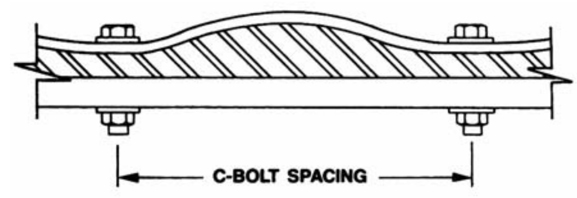



**Figure 7-12, Bowed Cover Plate**

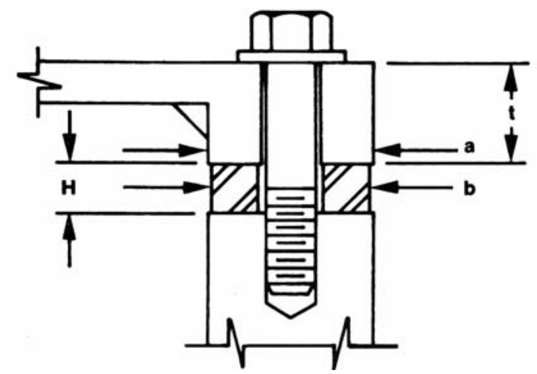



**Figure 7-13, Cover Plate arid Gasket Dimension**

where a=width of cover plate flange at seam
b=width of gasket
C=bolt spacing
E=modulus of elasticity of cover plate
$\Delta H = H_1 - H_2$
$H_1$=minimum gasket deflection
$H_2$= maximum gasket deflection
H=gasket height
$P_{min}$ /$P_{max}$ = minimum/maximum gasket pressure
t=thickness of cover plate

The equation can be tremendously simplified by making two assumptions which can be shown to have only slight affect on the result or which can be used to provide a close approximation for bolt spacing. These assumptions are:

1. **Width of gasket equals width of cover plate flange (a=b).** This condition is the limiting condition since the cover plate flange dimension (a) is always equal to or greater than the gasket width (b). For a gasket width equal to one half of the flange width, the bolt spacing correction is less than 1.19 times the value obtained for a=b or (a/b=1). The actual correction factor is the fourth root of the a/b ratio or (a/b)1/4. Using (a=b) actually provides a safety factor over any other relationship between (a) and (b).

2. **Maximum pressure ($P_{max}$) equals three times the minImum pressure ($P_{min}$)** For almost all resilient

gaskets, $P_{max}$ is usually greater than twice $P_{min}$ Using the ratio $P_{max}/P_{min} = 3$, bolt spacing is reduced by less than 7% for $P_{max}$ to $P_{min}$ ratio of 6. Actual correction factors for other values of $P_{max}$ to $P_{min}$ ratios and $P_{min}$ are given in Table 7-8.

**Table 7-8**
**Correction Factors For Bolt Spacing**
**(Reference FIgures 7-14 and 7-15)**

**$P_{max}/P_{min}$ Correction**

| $P_{max}/P_{min}$ | Correction Factor |
|---|---|
| 2 | 1.02 |
| 3 | 1.00 |
| 4 | .98 |
| 5 | .95 |
| 6 | .94 |

**$P_{min}$ Correction**

| $P_{min}$ | Correction Factor |
|---|---|
| 10 | 1.19 |
| 20 | 1.00 |
| 30 | .90 |
| 40 | .84 |
| 50 | .80 |

Incorporating these two assumptions into the basic equation, the bolt spacing is:

$$C = 2.242 \left[ \frac{Et^3 \Delta H}{P_{min}} \right]^{1/4} \quad \text{where } a/b = 1, \quad P_{max}/P_{min} = 3$$

and for $P_{min} = 20$ psi, typical of elastomeric gaskets,

$$C = 59.62 \left[ t^3 \Delta H \right]^{1/4}, \text{ aluminum plate } (E = 10^7 \text{ psi})$$

$$C = 78.46 \left[ t^3 \Delta H \right]^{1/4}, \text{ steel plate } (E = 3 \times 10^7 \text{ psi})$$

Figures 7-14 and 7-15 show sets of curves representing deflection as a percentage of gasket height (ΔH) for aluminum and steel plates respectively. The ΔH value is the difference between the maximum and the minimum gasket height under compression (reference Figures 7-5 through 7-8). Knowing the cover plate thickness and the gasket differential (ΔH), the bolt spacing can be easily determined. Since the $P_{min}$ for both figures has been selected as 20 psi, a correction factor is provided for $P_{min}$ values from 10 through 50 psi (see Table 7-8).

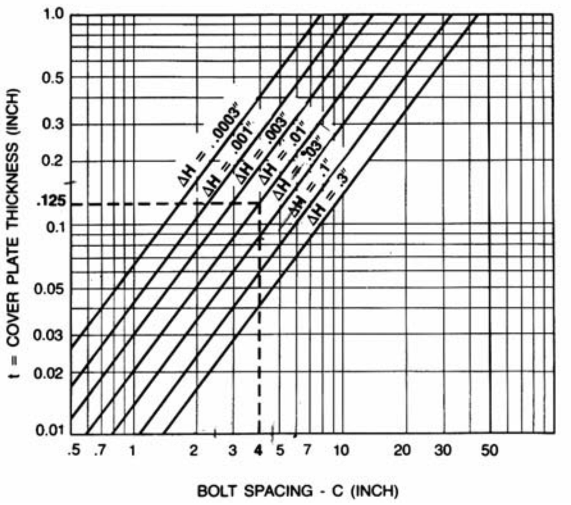



**Figure 7-14, Bolt Spacing - Aluminum Cover Plate**

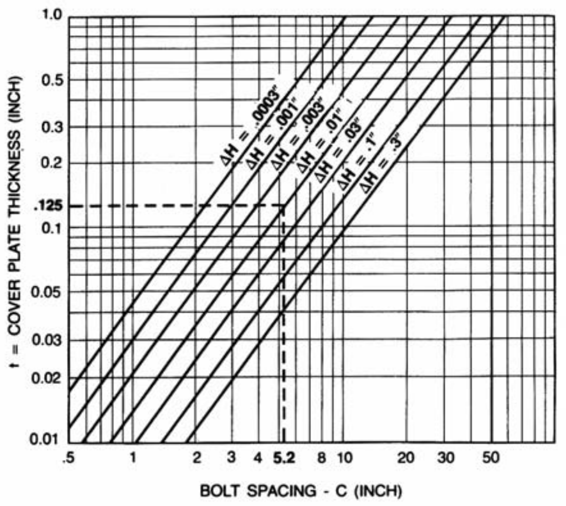



**FIgure 7-15, Bolt Spacing - Steel Cover Plate**

**EXAMPLE**, assume a design which uses a steel cover plate thickness of .125" with an anticipated gasket variation of .010" (ΔH) under a minimum pressure ($P_{min}$) of 20 psi and a maximum pressure ($P_{max}$) of 60 psi. Figure 7-15 shows a bolt spacing for steel at the stated pressure range of 20 to 60 psi to be 5.2 inches. For the same conditions using an aluminum plate, the bolt spacing is 4.0 inches (see Figure 7-14). The charts can also be used in reverse. For example, when it is desired to limit the number of fasteners for easier disassembly or removal of a cover plate. Select the desired bolt spacing and gasket differential (ΔH) to determine the required cover plate flange thickness.

# Elastomer Shielding Design Guide

*U.S. Customary*
*[SI Metric]*

**EXAMPLE**, assume a 10 inch bolt spacing is desirable and a maximum gasket differential (ΔH) of 0.03” is anticipated. The questions which need to be answered are (1) what is the necessary thickness of the cover plate flange and (2) what gasket materials are most suited for that application. Referring to Figure 7-14 (aluminum), draw an imaginery line from the point at the bottom of the chart which represents a 10 inch bolt spacing to the point representing H0.03” at the intersect of the 10 inch bolt spacing line and ΔH=.03”, draw an imaginery horizontal line to the left scal (coordinate) to find the minimum thickness of the cover plate flange (t). In this case, t=.3 inch. The larger the ΔH values for the specific compression conditions established by the pressure range of 20 to 60 psi, the softer and more resilient are the gaskets needed to satisfy the large variations in joint uneveness caused by flange bowing.

# CONSIL SILICONE ELASTOMER PRODUCT CHART

TECKNIT

## Tecknit Consil Silicone Elastomer Product Chart

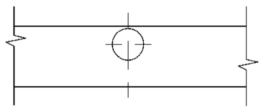
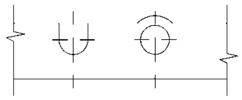
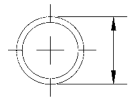
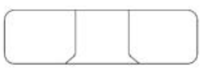
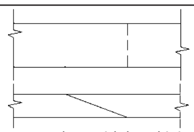
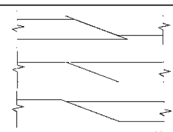
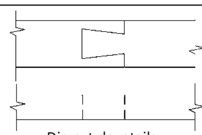
| | SC Consil FR861/ FR862 | SC Consil 860/861 862/864 | Consil A 895 | Consil A 897 | Consil NC 750/751 770 | Consil NC FR750/ FR751 | Consil E 811/815 | Consil II 841/842 | Consil R 855 | Consil R 856/857 | Consil N 891 | Consil C 871/874 | Consil C 873 | Consil C 875 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Material Grade | Commercial | | MIL-G-83528 Type B / Comm. | MIL-G-83528 Type D / Comm. | Commercial | | | | | | | MIL-G-83528 Type A | Commercial | MIL-G-83528 Type C |
| Elastomer | Silicone | | | Fluoro-silicone | Silicone | | | | | | | | | Fluoro-silicone |
| Filler | Carbon | | Silver plated Aluminum particles | | Nickel coated Graphite | | Silver plated Glass particles | | Pure Silver | | Silver plated Nickel particles | Silver plated Copper particles | | |
| Temperature | -60°F to 351°F [-51°C to 177°C] | -60°F to 351°F [-51°C to 177°C] | -67°F to 350°F [-55°C to 177°C] | -67°F to 350°F [-55°C to 177°C] | -67°F to 350°F [-55°C 160°C] | -67°F to 350°F [-55°C 160°C] | -60°F to 351°F [-51°C to 177°C] | -60°F to 351°F [-51°C to 177°C] | -67°F to 392°F [-55°C to 200°C] | -60°F to 351°F [-51°C to 177°C] | -67°F to 257°F [-55°C to 125°C] | -67°F to 257°F [-55°C to 125°C] / -55°F to 125°F [-48°C to 52°C] | -49°F to 257°F [-45°C to 125°C] | -67°F to 257°F [-55°C to 125°C] |
| Specific Gravity ASTM-D-297 | 1.16 ±0.03 | 1.28 ±0.03 / 1.2 ±0.03 | 2.0 ±13% | 2.0 ±13% | 2.0 ±13% | 2.1 / 2.0 ±13% | 1.86 ±0.25 | 1.80 ±0.25 / 1.86 ±0.25 | 3.5 ±13% | 1.7 / 2.5 ±0.25 | 4.0 ±13% | 3.5 / 3.7 ±13% | 3.5 ±13% | 4.0 ±13% |
| Hardness Shore A ASTM-D-2240 | +10 / -5 | 70 ±5 | 65 ±7 | 70 ±7 | 55 ±7/70 ±7 | 60 ±10 | 60 ±5 | 47 ±7/70 ±7 | 65 ±5 | 40 ±5/50 ±5 | 75 ±7 | 65 ±7 | 85 ±7 | 75 ±7 |
| Tensile Strength, Min. ASTM-D-412 | 650 psi | 500 psi / 650 psi | 200 psi | 180 psi | 150 psi | 150 psi | 50 psi | 100 psi / 120 psi | 300 psi | 100 psi | 200 psi | 200 psi | 400 psi | 180 psi |
| Elongation, Min.-ASTM-D-412 | 100 % | 100 % | 100% | 60% | 100% | 100% | 50% | 120% | 200% | 100% | 100% | 100% | 100% | 100% |
| Tear Strength, Min. ASTM-D-624 | 50 ppi | 50 ppi / 60 ppi | 30 ppi | 35 ppi | 50 ppi / 40 ppi | 50 ppi | 35 ppi / 20 ppi | 45 ppi | 40 ppi | 25 ppi / 44 ppi | 30 ppi | 25 ppi | 40 ppi | 35 ppi |
| Forms 1- extruded, 2- molded, 3- injection molded | 2/1 | 1/2/1/3 | 1 & 2 | | 2/1/1 | 2/1 | 1 | 2 & 3 | 2 | | | 1 & 2 | 2 | 1 & 2 |
| Volume Resistivity- ohm-cm max. | 15/24 | 3-24 | 0.008 | 0.012 | 0.1 | 0.1 | 0.03 | 0.01 | 0.002 | 0.015/0.006 | 0.005 | 0.004 | 0.005 | 0.01 |
| Shielding Effectiveness 1GHz (E-field) dB | 55 | 65 | 110 | 100 | 100 | 90 | 100 | 100 | 120 | 100 | 110 | 115 | | 115 |
| Flammability Rating | UL94 V0 | NONE | | | | UL94 V0 | NONE | | | | | | | |
| Recommended Adhesive | Teckbond NC | | Teckbond A | | Teckbond NC | | CON/RTV-II | | Cond. Adhesive 72-00002 or CON/RTV-II | | CON/RTV-Ni | Teckbond C | | |



U.S.A.: 908-272-5500 • U.K.: 44-1476-590600 • Spain: 34-91-4810178

# ENGINEER'S ELASTOMER DESIGN REFERENCE CHART

**Suggested Remedies in Gasket Design**

| Fault | Why Faulty | Suggested Remedy |
|---|---|---|
|  Bolt holes close to edge. | Causes breakage in stripping and assembly. |  Use 'ear' or 'notch'. |
|  Metalworking tolerances applied to gasket. | Results in rejection of perfectly good parts. Requires time and correspondence to reach acceptable level. Costly and slow. | Most gasket materials are compressible and affected by humidity. Use commercial tolerances in preference to special tolerances. |
|  Transference of fillets and radii from metal parts to gasket. | Unless molded part, this results in unnecessary costs. | Most gasket stock will conform without shaping. Ensure features are functional, not copied from metalwork. |
| Thin walls in relation to size. | High scrap, distortion during ship or in use. High tensile materials only. | Have gasket in mind early in design process. |
|  Large gaskets with bevel joints. |  Extra ops. Smooth joint difficult. |  Die-cut dovetails. |

**Recommended Deflection of Silver Filled Elastomers**

The deflection of conductive elastomer gaskets should never exceed the maximum

- 7-10% of thickness
- 18-20% of diameter
- 12-15% of height
- Approx. 50% but not more than 100% of void width

# CONDUCTIVE ELASTOMER TOLERANCES

SHEETS, RULE DIE CUT AND MOLDED GASKETS

*U.S. Customary*
*[SI Metric]*

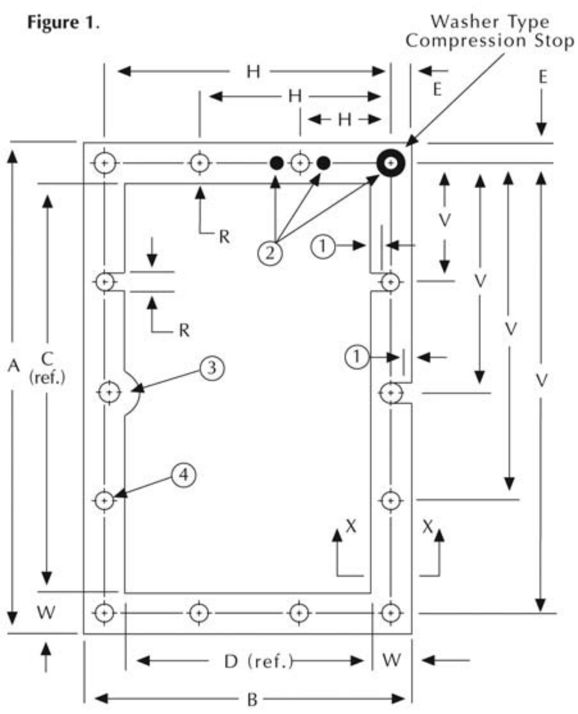
## RULE DIE CUT AND MOLDED GASKETS

The following tolerances refer to the dimensions illustrated in Figure 1.

### CUSTOM FABRICATED CONDUCTIVE ELASTOMER TOLERANCES

| SYMBOL | DIMENSION | TOLERANCE |
|---|---|---|
| A, B, V, W | up to 6 in. [152] | ± .016 [.40] |
| H | each additional 1 in. [25.4] | ± .003 [.08] |
| E, R | up to 1 in. [25.4] | ± .016 [.40] |
| | over 1 in. [25.4] | ± .031 [.79] |
| T | | See Tolerance For Sheets |

Figure 1.




NOTES:
1. Bolt holes closer to gasket edge than gasket thickness must be Ushaped slots, or see note 3.
2. Distance from compression stop to edge of sealing gasket must not be less than gasket thickness.
3. Bolt holes closer to gasket edge than gasket thickness can be with edge protrusion.
4. Holes diameter must not be less than gasket thickness, nor less than .125" in diameter.

### CONDUCTIVE ELASTOMER SHEET TOLERANCES

| THICKNESS | TOLERANCE |
|---|---|
| .020 to .032 [.51 to .81] | ± .005 [.13] |
| .033 to .045 [.84 to .14] | ± .007 [.18] |
| .046 to .062 [1.17 to 1.57] | ± .008 [.20] |
| .063 to .090 [1.60 to 2.39] | ± .010 [.25] |
| .091 to .125 [2.41 to 3.17] | ± .012[.30] |
| over .126 [3.17] | ± .015 [.38] |
| **LENGTH & WIDTH TOLERANCE** | |
| up to 12 x 18 [305 x 457] | ± .125 [3.18] |

NOTE: The above tolerances are based on gasket thickness of .125 or less. For gaskets thicker than .125, contact factory for applicable tolerances.

### TOLERANCES

### MOLDED X-SECTIONS

| SYMBOL | DIMENSION | TOLERANCE |
|---|---|---|
| T, W, D, A | under .101 [2.56] | ± .005 [0.127] |
| OD, ID, L | .101 - .250 [2.56 - 6.35] | ± .010 [2.56 ] |
| | .251 - .499 [6.37 - 12.67] | ± .015 [.381] |
| | .500 - .999 [12.7 - 25.37] | ± .020 [.508] |
| | 1.0 [25.4] and over | ± .031 [0.787] |

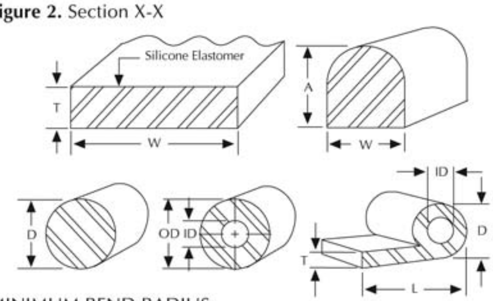
### OVERALL DIMENSION-MOLDED PARTS

| SIZE (INCHES) | | | FIXED |
|---|---|---|---|
| **Above** | | **Incl.** | |
| 0 | - | .40 (0 - 10) | ± .006 |
| .40 | - | .63 (102 - 16) | ± .008 |
| .63 | - | 1.00 (16 - 25) | ± .010 |
| 1.00 | - | 1.60 (25 - 40) | ± .013 |
| 1.60 | - | 2.50 (40 - 63) | ± .016 |
| 2.50 | - | 4.00 (63 - 100) | ± .020 |
| 4.00 | - | 6.30 (100 - 160) | ± .025 |
| 6.30 & over multiply by .004 | | | |

### EXTRUDED X-SECTIONS

| SYMBOL | DIMENSION | TOLERANCE |
|---|---|---|
| T, W, D, A | under .201 [5.10] | ± .005 [0.127] |
| OD, ID, L | .201 - .350 [5.10 - 8.89] | ± .008 [0.203] |
| | .351 - .499 [8.915 - 12.674] | ± .010 [0.254] |
| | .500 [12.7] and over | ± .015 [0.381] |

**Figure 2.** Section X-X




MINIMUM BEND RADIUS:
1/2 "W" for rectangle and "D" shape
1/2 "D" for cords 1/2 "OD" for tubes 1/2 "L" for P shape