Fastener Requirements

1. Applied Force

Most applications do not require more than 100 psi (0.69 MPa) to achieve an effective EMI seal. Waveguide flanges often provide ten times this amount. Hollow strips require less than 10 pounds per in. Compression deflection data for many shapes, sizes and materials is included in the Performance Data section of this handbook.

The force required at the point of least pressure, generally midway between fasteners, can be obtained by using a large number of small fasteners spaced closely together. Alternatively, fasteners can be spaced further apart by using stiffer flanges and larger diameter bolts. Sheet metal parts require more fasteners per unit length than castings because they lack stiffness.

To calculate average applied force required, refer to load-deflection curves for specific gasket materials and cross sections (see Performance Data, page 80).

2. Fastener Sizes and Spacing

Fastener spacing should be determined first. As a general rule, fasteners should not be spaced more than 2.0 inches (50 mm) apart for stiff flanges, and 0.75 inch (19 mm) apart for sheet metal if high levels of shielding are required. An exception to the rule is the spacing between fasteners found in large cabinet doors, which may vary from 3 inches (76.02 mm) between centers to single fasteners (i.e., door latches). The larger spacings are compensated for by stiffer flange sections, very large gaskets, and/or some reduction in electrical performance requirements.

The force per bolt is determined by dividing the total closure force by the number of bolts. Select a fastener with a stress value safely below the allowable stress of the fastener.

3. Flange Deflection

The flange deflection between fasteners is a complex problem involving the geometry of the flange and the asymmetrical application of forces in two directions. The one-dimensional solution, which treats the flange as a simple beam on an elastic foundation, is much easier to analyze[1] and gives a good first order approximation of the spacings required between fasteners, because most EMI gaskets are sandwiched between compliant flanges.

Variation in applied forces between fasteners can be limited to ±10 percent by adjusting the constants of the flange such that

$\beta d = 2$,

where

$$\beta = \sqrt[4]{\frac{k}{4 E_f I_f}}$$

where

k = foundation modulus of the seal

E_f = the modulus of elasticity of the flange

I_f = the moment of inertia of the flange and seal

d = spacing between fasteners

The modulus of elasticity (E_f) for steel is typically 3×10^7. The modulus for aluminum is typically 1×10^7, and for brass it is about 1.4×10^7.

The foundation modulus (k) of seals is typically 10,000 to 15,000 psi.

The moment of inertia (I_f) of rectangular sections, for example, may be obtained from the following expression[2]:

$$I_f = \frac{bh^3}{12}$$

where

b is the width of the flange in contact with the gasket (inches) and

h is the thickness of the flange (inches).

Example

Calculate the bolt spacings for flanges with a rectangular cross-section, such as shown in Figure 22,

where

h is the thickness of the flange.

b is the width of the flange.

d is the spacing between fasteners.

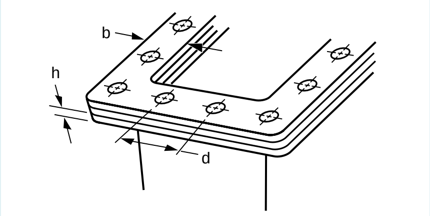

Figure 22 Bolt Spacings for Flanges

Assume the flange is to be made of aluminum.

To maintain a pressure distribution between bolts of less than ±10 percent, ßd must be equal to 2 (see Figure 23 and discussion).

Assume an average foundation modulus (k) of 12,500 psi for the seal. If the actual modulus is known (stress divided by strain), substitute that value instead.

The bolt spacings for aluminum flanges for various thicknesses and widths have been calculated for the previous example and are shown in Figure 24.

The previous example does not take into account the additional stiffness contributed by the box to which the flange is attached, so the results are somewhat conservative.

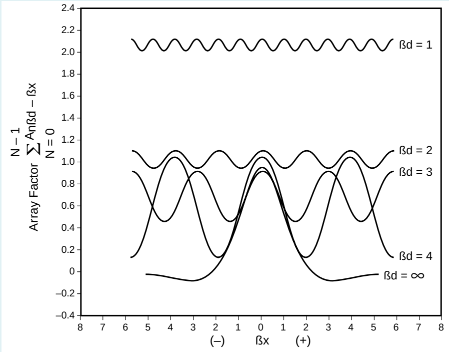

Figure 23 Array Factor vs. Spacing

References

1. Galagan, Steven, *Designing Flanges and Seals for Low EMI*, **MICROWAVES**, December 1966.

2. Roark, R.J., *Formulas for Stress and Strain,* McGraw-Hill, 4th Ed., p. 74.

Actual deflection vs. distance between fasteners may be computed from the following expression:

$$y = \frac{\beta p}{2k} \sum_{n=0}^{N-1} A_{n\beta d-\beta x}$$

where p is the force applied by the fastener, and β and k are the constants of the flange as determined previously. N represents the number of bolts in the array.

The array factor denoted by the summation sign adds the contribution of each fastener in the array. The array factor for various bolt spacings (βd) is shown in Figure 23. Although any value can be selected for βd, a practical compromise between deflection, bolt spacing and electrical performance is to select a bolt spacing which yields a value βd equal to 2.

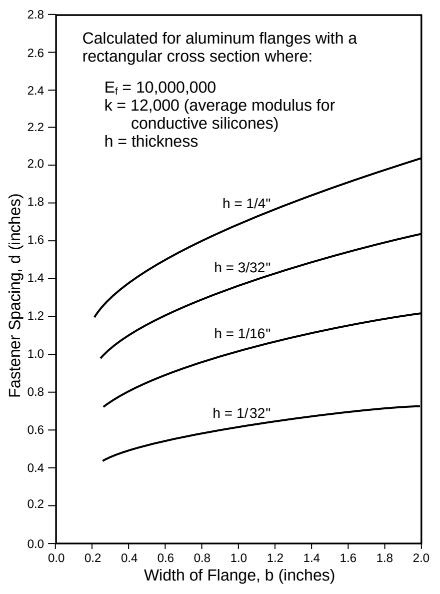

Figure 24 Fastener Spacing

For βd = 2, the flange deflection fluctuates by ±10 percent. Minimum deflection occurs midway between fasteners and is 20 percent less than the deflection directly under the fasteners. The variation in deflection is approximately sinusoidal.

Table IV lists a few recommendations for bolts and bolt spacings in various thin cross section aluminum flanges.

Bolt spacings for waveguide flanges are fixed by Military and EIA Standards. Waveguide flanges normally have bolts located in the middle of the long dimension of the flange because the flow of current is most intense at this point.

Table IV

SCREW SIZE	℄-TO-℄ (in.)	THICKNESS (in.)	MAX. TORQUE TO PREVENT STRIPPING FOR UNC-2A THREAD (in.-lbs.)
#2	⅜	0.062	4.5
#4	¾	0.125	10.0
#6	1	0.125	21.0
#8	1¼	0.156	37.5
#10	1⅜	0.156	42.5

4. Common Fasteners

Many different types of fasteners are available, but bolts are the most widely used fastening devices. The approximate torque required to apply adequate force for mild steel bolts is shown in Table V.

These values are approximate and will be affected by the type of lubricants used (if any), plating, the type of washers used, the class and finish of the threads, and numerous other factors.

The final torque applied to the fasteners during assembly should be 133 percent of the design value to overcome the effect of stress-relaxation. When torqued to this value, the gasket will relax over a period of time and then settle to the design value.

Torque may be converted to tension in the bolts by applying the formula

$$\text{Tension} = \frac{\text{Torque}}{0.2 \times \text{Bolt Dia.}}$$

Frequently the rule of thumb value of 0.2 for the coefficient of friction can result in torque and bolt estimates which may be seriously in error. Excessive bolt preload may lead to RF leakage. Therefore, if lubricants are used for any reason, refer to the literature[3] for the proper coefficient values to be applied.

In soft materials, such as aluminum, magnesium and insulating materials, inserts should be provided if the threads are "working threads." A thread is considered a "working thread" if it will be assembled and disassembled ten or more times.

Torque loss caused by elongation of stainless steel fasteners should also be considered. High tensile strength hardware is advised when this becomes a problem, but care must be taken of the finish specified to minimize galvanic corrosion.

Thermal conductivity of high tensile strength hardware is lower than most materials used in electro-mechanical packaging today, so

Table V

RECOMMENDED TORQUE VALUES FOR MILD STEEL BOLTS

Size	Threads per in.	Max. Recommended Torque (in.-lbs.)	Max. Recommended Tension* (lbs.)	Basic Pitch Dia.(inches)
#4	40	$4\frac{3}{4}$		0.0958
	48	6		0.0985
#5	40	7		0.1088
	44	$8\frac{1}{2}$		0.1102
#6	32	$8\frac{3}{4}$		0.1177
	40	11		0.1218
#8	32	18		0.1437
	36	20		0.1460
#10	24	23		0.1629
	32	32		0.1697
1/4"	20	80	1840	0.2175
	28	100	2200	0.2268
5/16"	18	140	2530	0.2764
	24	150	2630	0.2854
3/8"	16	250	3740	0.3344
	24	275	3950	0.3479
7/16"	14	400	5110	0.3911
	20	425	5250	0.4050
1/2"	13	550	6110	0.4500
	20	575	6150	0.4675
9/16"	12	725	7130	0.5084
	18	800	7600	0.5264
5/8"	11	1250	11,040	0.5660
	18	1400	11,880	0.5889

$$* \text{Tension} = \frac{\text{Torque}}{0.2 \times \text{Diameter of Bolt}^{\dagger}}$$

† Basic Pitch Diameter

3. Roehrich, R.L., *Torquing Stresses in Lubricated Bolts*, **Machine Design**, June 8, 1967, pp. 171-175.

that the enclosure expands faster than the hardware and usually helps to tighten the seal. Should the equipment be subjected to low temperatures for long periods of time, the bolts may require tightening in the field, or can be pretightened in the factory under similar conditions.

Under shock and vibration, a stack up of a flat washer, split helical lockwasher and nut are the least reliable, partly because of elongation of the stainless steel fasteners, which causes the initial loosening. The process is continued under shock and vibration conditions. Elastic stop nuts and locking inserts installed in tapped holes have proven to be more reliable under shock and vibration conditions, but they cost more and are more expensive to assemble.

5. Electrical Performance as a Function of Fastener Spacing

The electrical performance (shielding effectiveness) provided by a gasket sandwiched between two flanges and fastened by bolts spaced d distance apart is equivalent to the shielding effectiveness obtained by applying a pressure which is the arithmetic mean of the maximum and minimum pressure applied to the gasket under the condition that the spacing between fasteners is considerably less than a half wavelength. For bolt spacings equal to or approaching one-half wavelength at the highest operating frequency being considered, the shielding effectiveness at the point of least pressure is the governing value.

For example, assume that a gasket is sandwiched between two flanges which, when fastened together with bolts, have a value of βd equal to 2. Figure 23 shows that a value of $\beta d = 2$ represents a deflection change of ±10 percent about the mean deflection point. Because applied pressure is directly proportional to deflection, the applied pressure also varies by ±10 percent.

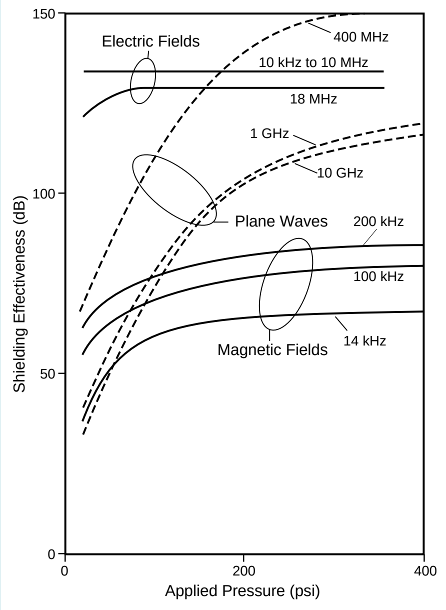

Figure 25 Shielding Effectiveness vs. Applied Pressure

Shielding effectiveness values for typical silver-plated-copper filled, die-cut gaskets as a function of applied pressure are shown in Figure 25. The curves show that the shielding effectiveness varies appreciably with applied pressure, and changes as a function of the type of field considered. Plane wave attenuation, for example, is more sensitive to applied pressure than electric or magnetic fields.

Thus, in determining the performance to be expected from a junction, find the value for an applied pressure which is 10 percent less (for $\beta d = 2$) than the value exerted by the bolts directly adjacent to the gasket. For example, examine a portion of a typical gasket performance curve as shown in Figure 26.

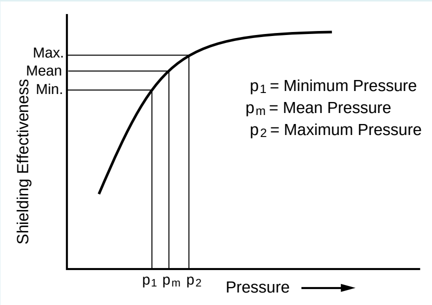

Figure 26 Typical Gasket Performance Curve

The average shielding effectiveness of the gasketed seam is a function of the mean applied pressure, p_m.

For spacings which approach or are equal to one-half wavelength, the shielding effectiveness is a function of the minimum pressure, p_1. Therefore, the applied pressure must be 20 percent higher to achieve the required performance. For this condition, the space between the fasteners can be considered to be a slot antenna loaded with a lossy dielectric. If the slot is completely filled, then the applied pressure must be 20 percent higher as cited. Conversely, if the slot is not completely filled (as shown in Figure 27), the open area will be free to radiate energy through the slot.

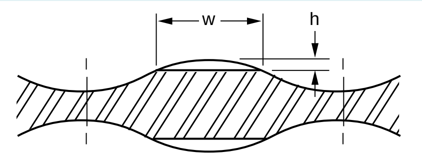

Figure 27 Unfilled Slot is Free to Radiate When Spacing is Equal to $^1/_2$ Wavelength

The cut-off frequency for polarizations parallel to the long dimension of the slot will be determined by the gap height, h. The cut-off frequency for the polarization vector perpendicular to the slot will be determined by the width of the slot, w. The attenuation through the slot is determined by the approximate formula

$$A(dB) = 54.5\ d/\lambda_c$$

where

d = the depth of the slot,

and

λ_c is equal to 2w or 2h, depending upon the polarization being considered.

This example also illustrates why leakage is apt to be more for polarizations which are perpendicular to the seam.

For large values of βd, the percentage adjustments must be even greater. For example, the