Performance of Resilient Metal Seals

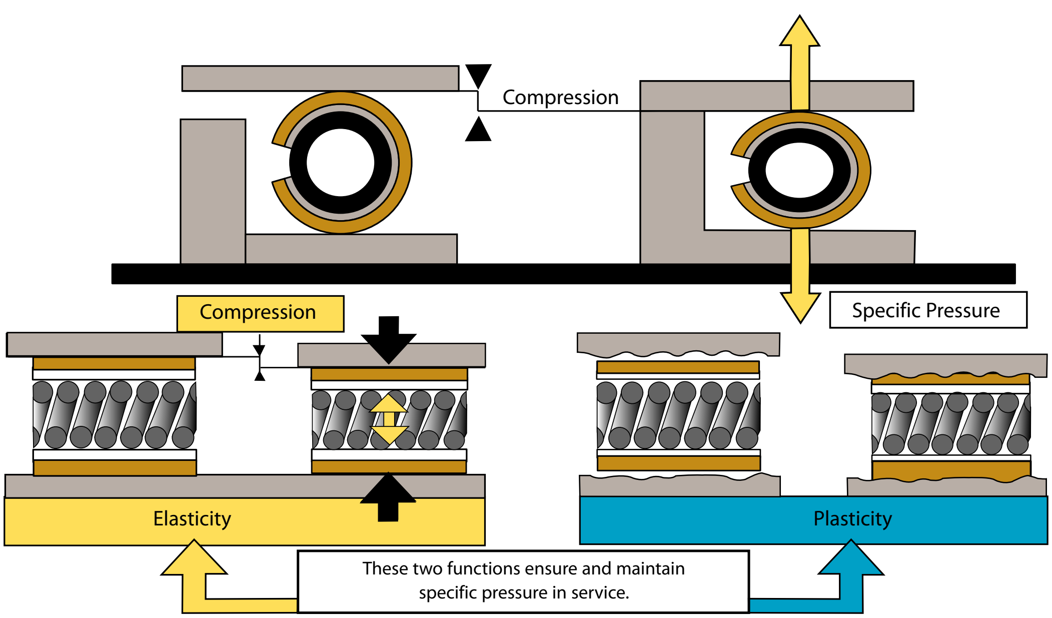
The performance of a resilient metal seal depends on two basic factors: elasticity and plasticity. The concept is similar to an elastomer seal such as Viton or Buna. The difference is that the elastomer compound serves both functions where a metal seal must use two components: a substrate and a soft outer layer.

Elasticity

Each seal has a resilient metal substrate in the form of a spring (Helicoflex®), tubing (O-Flex™), or formed strip (E-Flex™, C-Flex™). This substrate serves to provide a specific load that is used to deform a soft outer layer. The substrate also has a certain amount of spring back that helps maintain constant contact force during service. This spring back is not necessarily designed to compensate for axial or radial flange separation. Instead, it ensures that the seal maintains enough contact force to properly seal a static joint in service.

Plasticity

The soft outer layer is usually a plating/coating or a wrapped jacket. This outer layer is designed to plastically deform based on the specific load generated by the substrate. As the soft outer layer is deformed, it flows into the flange/groove imperfections and creates a seal. The tightness of the seal will depend on the amount of specific load, the ductility of the outer layer and the groove surface finish. An ideal groove/flange finish has machining marks that follow the circumference of the seal. Any radial marks or scratches may not be completely filled by the soft outer layer and could create a leak.



Bolted Joints

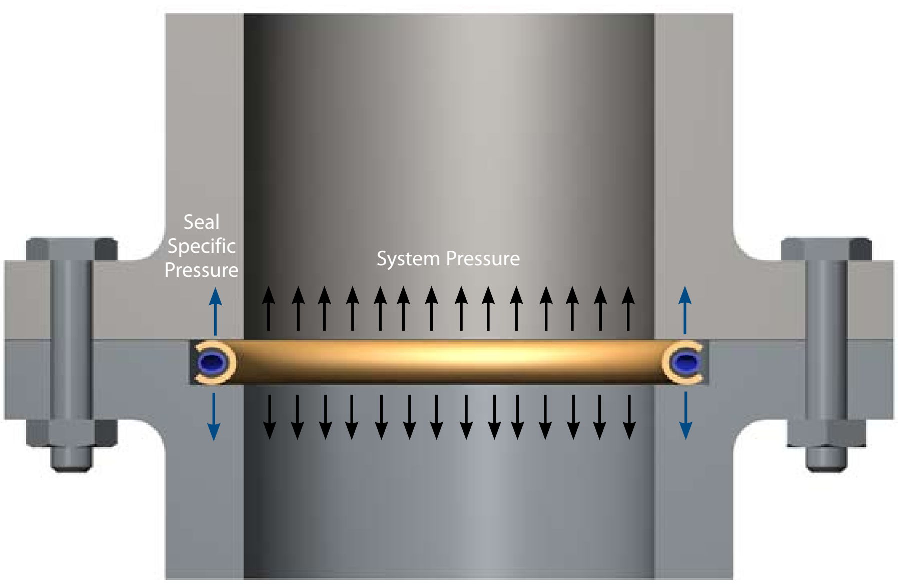
A bolted joint is an assembly that relies on each component to work properly. The performance and success of the bolted joint depends on the quality and design of each of these components. There are three major components of every bolted joint:

1. Flanges (Flange design / Groove dimensions & finish)
2. Bolts / Fasteners
3. Seal / Gasket

The above components cannot be designed mutually exclusive of each other. They must be considered together as a system during the design process. If any part of the bolted joint assembly does not perform properly, the joint as a whole will not perform to expectations and may leak.

Bolt Load and Tightening Torque

When using bolts to fasten the sealing joint the bolts must be of suitable strength and quantity to compress the seal and withstand the maximum hydrostatic load. Additionally, the bolts and flanges must be robust enough to prevent warpage, distortion or separation during service. All service factors must be considered such as thermal stresses, differential expansion, external loads and vibration.



Bolt Load Estimates

The following equations may be used to estimate required bolt loads.

NOTE: These estimates are offered as guidelines only. There are many other factors that the flange designer must consider such as: thermal cycling, vibration, cyclic fatigue, flange thickness, flange rotation, bolt stress relaxation, additional bolt preload, externally applied loads, etc. The customer is responsible for the flange design and for ensuring that the flanges, bolts and bolt loads are sufficient for the application. Please refer to Section VIII of the ASME Boiler and Pressure Vessel Code for code requirements.

Total Bolt Load ≥ Seal Seating Load + Hydrostatic Load + Safety Factor

Seal Seating Load

Total load required to compress the seal to optimal level. This information can be found for each seal type in the Performance Data sections of the catalog. It is referenced as Y_2 and is given in pounds per circumferential inch (PCI).

Seal Seating Load = Seal Diameter x π x Y_2

Hydrostatic Load

Load required to contain the system pressure.

Hydrostatic Load = Maximum system pressure x (π/4) x $(\text{Seal Diameter})^2$

Safety Factor

This is a customer determined safety factor and must consider: system temperature effects, temperature cycling/spikes, pressure cycling/spikes, vibration, etc.

NOTE: A more detailed calculation is available for the Helicoflex spring energized seals. Please see the Helicoflex Seal product section.

Example Calculation

Seal:
O-Flex metal o-ring, Material = SS321
OD = 4.000in, CS = .125in, wall thickness = .020in
Y_2 = 1142 lbs/in

Operating Conditions:
Pressure: 500 psi, Temperature: 70 F

Seating Load = 4.000in x π x 1142lbs/in = 14351 lbs
Hydrostatic Load = 500 lbs/in^2 x (π/4) x $(4.000\text{in})^2$ = 6283 lbs

Total Bolt Load Estimate ≥ 14351 lbs + 6283 lbs + customer safety factor

NOTE: each application should be reviewed to determine if additional bolt preload may be required for proper bolt stretch.

Tightening Torque and Bolt Tension

The following equation may be used to create a rough estimate of the required torque:

$$T = K \times P \times D$$

Where:
T= tightening torque (in-lbs)
K*= dynamic coefficient of friction (i.e. minimum = .15 (dry-zinc plated))
P= total bolt load / number of bolts (lbf)
D= nominal bolt diameter (in)

(* Also referred to as the "nut factor" in some texts.)

It must be understood that every bolted joint is unique and the tightening torque should be determined for each application through experimentation. A properly tightened bolt is one that is stretched, thus acting like a very rigid spring pulling the mating surfaces together. As the bolt is tightened it begins to stretch and goes into a state of tension. There are many factors that affect how much tension occurs when a given amount of tightening torque is applied. These factors include bolt diameter, bolt grade (strength), and friction. Torque calculations can have significant errors based on these factors, especially friction. Best practice indicates that bolts should be properly lubricated and hardened washers used under the head and nut.

Where possible, it is recommended the fastener elongation, or stretch, be measured directly to ensure proper tension or preload, in the fastener.

NOTE: These estimates are offered as guidelines only. There are many other factors that the flange designer must consider such as: thermal cycling, vibration, cyclic fatigue, flange thickness, flange rotation, bolt stress relaxation, additional bolt preload, externally applied loads, etc. The customer is responsible for the flange design and for ensuring that the flanges, bolts and bolt loads are sufficient for the application. Please refer to Section VIII of the ASME Boiler and Pressure Vessel Code for code requirements.

Typical Bolt / Fastener Information

Size / Nominal Diameter	Nominal Diameter inches	Pitch (THD/IN)	Area at Root of Thread sq. in.	30000 PSI Stress		45000 PSI Stress		60000 PSI Stress	
				Fastener Preload lbs	Torque Req'd K= .15 lbs-in	Fastener Preload lbs	Torque Req'd K= .15 lbs-in	Fastener Preload lbs	Torque Req'd K= .15 lbs-in
#6	0.138	32	0.008	225	5	338	7	450	9
#8	0.164	32	0.012	360	9	540	13	720	18
#10	0.190	24	0.015	435	12	653	19	870	25
#12	0.226	24	0.021	618	21	927	31	1236	42
1/4"	0.250	20	0.027	807	30	1211	45	1614	61
5/16"	0.313	18	0.045	1362	64	2043	96	2724	128
3/8"	0.375	16	0.068	2034	114	3051	172	4068	229
7/16"	0.438	14	0.093	2799	184	4199	276	5598	367
1/2"	0.500	13	0.126	3771	283	5657	424	7542	566
9/16"	0.563	12	0.162	4860	410	7290	615	9720	820
5/8"	0.625	11	0.202	6060	568	9090	852	12120	1136
3/4"	0.750	10	0.302	9060	1019	13590	1529	18120	2039
7/8"	0.875	9	0.419	12570	1650	18855	2475	25140	3300
1"	1.000	8	0.551	16530	2480	24795	3719	33060	4959
1-1/8"	1.125	8	0.728	21840	3686	32760	5528	43680	7371
1-1/4"	1.250	8	0.929	27870	5226	41805	7838	55740	10451
1-3/8"	1.375	8	1.155	34650	7147	51975	10720	69300	14293
1-1/2"	1.500	8	1.405	42150	9484	63225	14226	84300	18968
1-3/4"	1.750	8	1.980	59400	15593	89100	23389	118800	31185
2"	2.000	8	2.652	79560	23868	119340	35802	159120	47736

NOTES:

1. For fasteners larger than one inch, it is often customary to use a thread pitch of 8 in place of UNC thread pitch.
2. Contact Applications Engineering for other sizes.
3. These values/estimates are offered as guidelines only. There are many other fators that the flange designer must consider such as: thermal cycling, vibration, cyclic fatigue, flange thickness, flange rotation, bolt stress relaxation, additional bolt preload, externally applied loads, etc. The customer is responsible for the flange design and for ensuring that the flanges, bolts and bolt loads are sufficient for the application. Please refer to Section VIII of the ASME Boiler and Pressure Vessel Code for code requirements.

Installation Procedures

Seal installation is as important to the performance of the bolted joint as the flange, bolt and seal design. Following these simple steps will help ensure a successful installation.

Preparation Verify the seal part number, required bolt loading and any special handling or installation instructions. Seals should remain in original protective packaging and preferably be stored in a controlled environment until time of installation. Finally, the packaging should be opened carefully to avoid scratching or damaging the seal. Be especially careful when using razor knives to open seal packaging or container.

Inspection Inspect the groove and flanges to make sure the seal track area is free of burrs, debris and any radial marks or scratches. If necessary, clean the groove carefully with acetone or alcohol using a lint free cloth. Any radial scratches must be removed by careful polishing (polishing marks must follow seal circumference). Deeper scratches may require re-cutting the groove and/or re-facing the flange. Additionally, the sealing surface of the seal should be inspected for scratches and carefully handled to avoid dings, dents and radial marks or scratches.

Seal installation Carefully, place the seal into the groove or onto the flange. Gently bring the mating flange into place taking care not to scratch or damage the seal during all steps of the process.

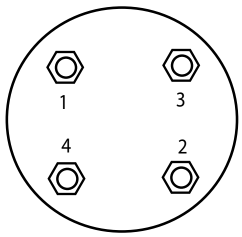
Note: Large seals (> 36") should be supported every three feet of circumference to prevent bending or crimping.

Bolts / Fasteners Bolts, bolt holes and nuts should be free of burrs, debris and galling. Bolts and nuts should be well lubricated with a process compatible lubricant. Hardened washers should be used when possible to further reduce friction. Note: for critical applications the installer may want to preload the bolts and release (without the seal) two or three times to "run in" the threads.

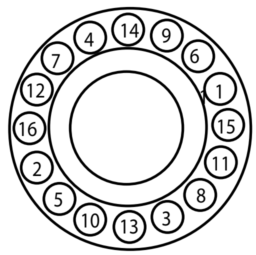
Bolt Tightening Bolts should be tightened using a star pattern (see diagram). Number the bolts with an indelible marker to make the process easier. First, tighten the nuts until "finger tight". Then, tighten bolts in one-third increments, according to the proper star bolting pattern. Make a final check pass at the final target torque value moving consecutively from bolt to bolt in a rotational order starting with bolt number one. It is recommended to re-torque 12-24 hours after initial installation, especially for high temperature applications.

Removing Used Seals Most metal seals are designed to make some light contact with the groove wall during compression and service. This helps to reinforce the seal against the system pressure. As a result, it may be difficult to remove the seal with finger force only, especially if the groove is very narrow. Ideally, a hard plastic pick can be used to remove the seal. For some seals, you may carefully drill a small hole in the top of the seal and use a small pick. In all cases, great care must be taken not to scratch the groove when using tools to remove the seal.

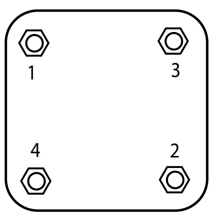
Correct Bolting Patterns



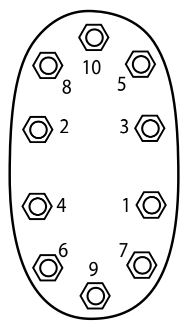
Circular Four-Bolt



Circular Multibolt



Square
Four Bolt



Noncircular
Multibolt

Jacket –vs- Plating/Coating

There are two types of soft outer layers that can be applied to metal seals to improve leakage performance. In both cases, the substrate must provide enough specific load to plastically deform the soft outer layer into the flange imperfections.

Wrapped Jacket The Helicoflex Spring Energized Seal has a soft outer jacket that consists of a metal strip that has been wrapped or formed around the spring. Typically it is much thicker than platings or coatings. For example, a Silver jacket is approximately .012" to .020" thick where Silver plating is approximately .001" to .002" thick.

There are two primary advantages of the wrapped jacket. First, there is greater flexibility in material choice since the jacket is not limited by available plating technology. The Helicoflex seal can be made with most metals available in strip or sheet form which helps match the seal material to temperature and corrosion requirements. Secondly, because the jacket is thicker, it typically performs better on rougher surface finishes. This is especially helpful for older vessels, such as aging nuclear reactor pressure vessels, where the grooves may have been polished or refinished.

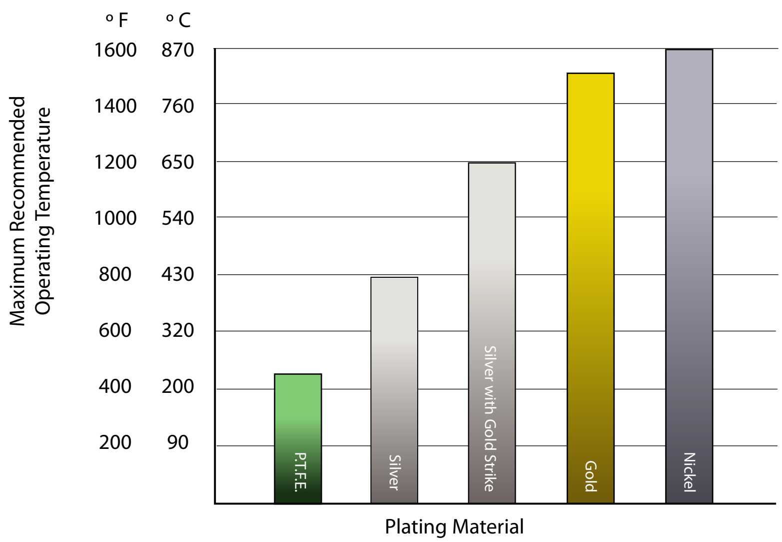
The Helicoflex seal spring is specifically designed for each jacket material to ensure plastic deformation is achieved.

Platings/Coatings Platings and coatings are applied directly to the seal substrate. Typically these treatments are very thin and are usually .001/.002" thick. Therefore, they require a smooth groove/flange finish for optimal performance. Platings such as Silver and Nickel are applied by an electroplating process while coatings such as PTFE are typically applied by a spray or dip process. It is more difficult to match materials to temperature and corrosion requirements because platings and coatings are limited in choice by available deposition technologies.

It is important to note that each plating material requires a minimum amount of specific load to plastically deform. Below are some guidelines for Silver plated non-spring energized seals.

Cross sections: 0.063 to 0.156 = minimum load of 400 lbs per inch of circumference.
Cross sections: 0.188 to 0.250 = minimum load of 800 lbs per inch of circumference.

Maximum Recommended Operating Temperatures for Platings and Coatings



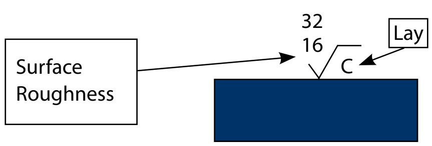
Contact Applications Engineering for additional platings and coatings.

Surface Finish

The leak rate of any joint is largely influenced by the condition of the surfaces in the joint. Leak paths are inherent in any sealing surface. Both the surface roughness of the seal and the surface roughness of the mating flange surfaces will affect sealing performance.

Surface roughness, also called surface texture or finish, is a trait of any surface. The design engineer usually specifies the required surface roughness of a flange sealing surface to ensure proper function of the flange in the joint.

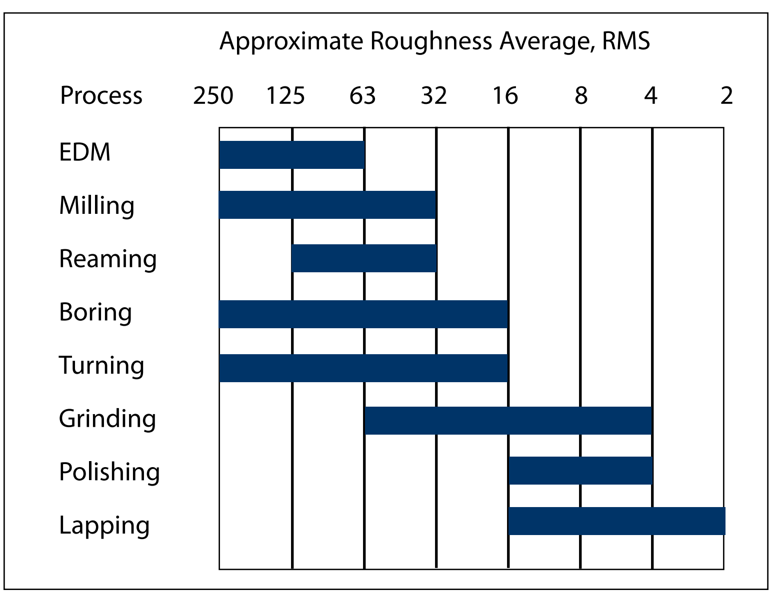
Surface roughness is usually specified with a "check mark" symbol on a drawing as shown in the figure below. Surface roughness is typically indicated in RMS or microinches (μin) and is located on the left side of the symbol above the check mark. In the example below the roughness value is 32 RMS maximum and 16 RMS minimum. If a single value is specified, this value is interpreted as a maximum value.



/ / parallel lay
⊥ perpendicular lay
C circular lay*
R radial lay
M multi-directional lay
X angular lay in both directions

* Most metal seal applications require a circular or circumferential lay

The directional lay of a finished surface refers to the direction of the machining or polishing marks. The lay of a sealing surface is specified under the surface roughness symbol as shown in the figure above.



Understanding Leakage

Leakage is the flow of a fluid through an orifice or permeation through a material and typically occurs as a result of a pressure differential. It is important to understand that all materials and mechanical joints permit some leakage over a period of time. This leakage may range from as much as several gallons or cubic feet per minute to as little as a bubble of air in several years.

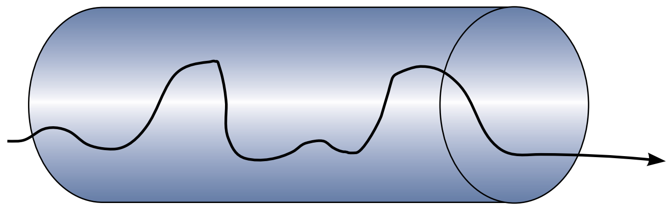
Helicoflex designs and manufactures a wide range of seals to satisfy a broad range of sealing requirements including leakage rate. Therefore, it is necessary to establish leakage rate criteria so that a suitable seal can be selected or designed. A specification that defines a "no leak" or "zero leakage" requirement is, in a technical sense, unrealistic and may lead to costly attempts at sealing. Leak tightness must be considered in relation to the medium being sealed, the normal operating conditions, and the sealing requirements regarding safety, contamination, and reliability.

Gas Flow

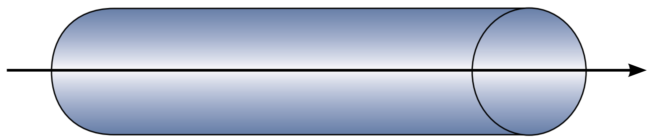
Gas flow is used in characterizing leakage and performing leakage testing. Even at very low pressures, gases behave and flow like fluids. Gas flow is categorized into different types of flow modes as follows:

Flow Mode	Flow Description	Leakage Rate (std cc/sec)
Turbulent Flow (Viscous Flow)	Flow through a passage that is typified as a large leak and at high pressure differentials. Leaks with turbulent flow are large and can be readily located and repaired.	Greater than 10^{-2}
Laminar Flow (Viscous Flow)	Flow in a passage that is typified by slow movement of fluid in a relatively straight path along the centerline of a passage.	10^{-1} to 10^{-6}
Transitional Flow	Flow that occurs between the laminar and molecular flow regimes.	10^{-4} to 10^{-7}
Molecular Flow	At molecular flow each molecule travels independently of other molecules. However, the general flow is in direction of the lower pressure.	Less than 10^{-7}

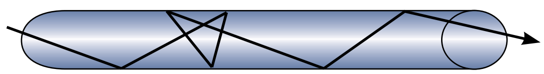
Note: Both turbulent flow and laminar flow are types of viscous flow.



Path of a molecule through a leak path in turbulent flow.



Path of a molecule through a leak path in laminar flow.



Path of a molecule through a leak path in molecular flow.

Viscosity: Why liquids and gases have different leakage rates

Viscosity is the internal friction of molecules of a liquid or gas and characterizes the resistance of a fluid to flow at a given temperature. High viscosity indicates a greater resistance to flow and low viscosity indicates a lesser resistance to flow. Therefore, fluids with a low viscosity have a higher probability of leaking or flowing at a higher rate.
Examples of typical fluid viscosities at room temperature (68°F, 20°C):

Fluid	Viscosity (in centipoises) at 68ºF, 20ºC
SAE 10 Grease	65
Water	0.95
Gasoline	0.6
Liquid Propane	0.11
Helium	0.019
Air	0.018
Hydrogen	0.009

From the above viscosity values it can be seen that at ambient temperature, water has a viscosity that is approximately 53 times greater than air. Therefore, at low pressure, the volume of water flow will be 53 times less than that of air.

Equivalent Leakage Rates

Std cc/sec*	mbar-l/sec	Torr Liters/sec	Time for one cc to Leak	Time for one bubble** to leak
10^{-1}	1.01×10^{-1}	7.6×10^{-2}	10 seconds	0.25 seconds
10^{-2}	1.01×10^{-2}	7.6×10^{-3}	100 seconds	2.5 seconds
10^{-3}	1.01×10^{-3}	7.6×10^{-4}	16.7 minutes	25 seconds
10^{-4}	1.01×10^{-4}	7.6×10^{-5}	2.8 hours	4 minutes
10^{-5}	1.01×10^{-5}	7.6×10^{-6}	28 hours	40 minutes
10^{-6}	1.01×10^{-6}	7.6×10^{-7}	11.5 days	7 hours
10^{-7}	1.01×10^{-7}	7.6×10^{-8}	3.8 months	3 days
10^{-8}	1.01×10^{-8}	7.6×10^{-9}	3.2 years	1 month
10^{-9}	1.01×10^{-9}	7.6×10^{-10}	32 years	9 months
10^{-10}	1.01×10^{-10}	7.6×10^{-11}	320 years	8 years
10^{-11}	1.01×10^{-11}	7.6×10^{-12}	3200 years	80 years

* Std cc/sec = One cubic centimeter of gas flow per second at 14.7 psi of pressure and a temperature of 77ºF

** Bubble diameter is 3mm

Leak Legend	Approximate Leak Rates per meter of circumference	Actual leak rate in service will depend on the following:
Ultra-Helium	≤ 1×10^{-11} std.cc/sec He	**Seal Load:** Wall Thickness or Spring Load
Helium	≤ 1×10^{-9} std.cc/sec He	**Surface Finish:** Seal and Cavity
Bubble	≤ 1×10^{-4} std.cc/sec He	**Surface Treatment:** Coating/Plating/Jacket Material
Low Bubble	≤ 25 cc/sec @ 50 psig Nitrogen per inch of diameter	

Conversion of helium leakage rate to leakage rates of other gases

To Convert to Leakage Rate of:	Multiply Helium Leakage Rate by:	
	Laminar Flow	Molecular Flow
Argon	0.88	0.316
Air	1.08	0.374
Nitrogen	1.12	0.374
Water vapor	2.09	0.469
Hydrogen	2.23	1.410

Sources:

1. Leakage Testing Handbook, Prepared for Liquid Propulsion Section, Jet Propulsion Laboratory, National Aeronautics and Space Administration, Pasadena, California
2. Nondestructive Testing Handbook, Volume One, Leaktesting, American Society for Nondestructive Testing.
3. Leakage Testing Handbook, Revised Edition, July 1969, General Electric.
4. Fluid Flow in Small Passages, Mars Hablanian, J.W.Marr, Varian

Common Conversion Tables

Length

Multiply \ To Obtain		Inch	micron	mm	cm	meter
inch	by	1	2.5400E+04	25.4000	2.5400	2.5400E-02
micron	by	3.9370E-05	1	1.0000E-03	1.0000E-04	1.000E-06
mm	by	3.9370E-02	1.0000E+03	1	1.0000E-01	1.000E-03
cm	by	3.9370E-01	1.0000E+04	10.0000	1	1.0000E-02
meter	by	39.3700	1.0000E+06	1.0000E+03	1.0000E+02	1

Pressure

Multiply \ To Obtain		bar	pascal	Mpascal	torr	psi	inches mercury 0ºC	inches water 4ºC
bar	by	1	1.0000E+05	1.0000E-01	7.5006E+02	14.5040	29.5300	4.0146E+02
pascal	by	1.0000E-05	1	1.0000E-06	7.5006E-03	1.4504E-04	2.9530E-04	4.0146E-03
Mpascal	by	10.0000	1.0000E+06	1	7.5006E+03	1.4504E+02	2.9530E+02	4.0146E+03
torr	by	1.3332E-03	1.3332E+02	1.3332E-04	1	1.9337E-02	3.9370E-02	5.3524E-01
psi	by	6.8948E-02	6.8948E+03	6.8948E-03	51.7150	1	2.0360	27.6800
inches mercury 0ºC	by	3.3863E-02	3.3863E+03	3.3863E-03	25.4000	4.9115E-01	1	13.5950
inches water 4ºC	by	2.4909E-03	2.4909E+02	2.4909E-04	1.8683	3.6127E-02	7.3556E-02	1

Vacuum Leak Rate

Multiply \ To Obtain		$torr.l.s^{-1}$	$atm.cm^3.s^{-1}$	$mbar.l.s^{-1}$	$Pa.m^3.s^{-1}$
$torr.l.s^{-1}$	by	1	1.316	1.333	1.333E-01
$atm.cm^3.s^{-1}$	by	7.600E-01	1	1.013	1.013E-01
$mbar.l.s^{-1}$	by	7.501E-01	9.862E-01	1	1.000E-01
$Pa.m^3.s^{-1}$	by	7.501	9.869	10.000	1

Mass

Multiply \ To Obtain		Kgf	N	lbf
Kgf	by	1	9.8067	2.2046
N	by	1.0197E-01	1	2.2481E-01
lbf	by	4.5359E-01	4.4482	1

Torque

Multiply \ To Obtain		lb.in	Kg.m	N.m
lb.in	by	1	1.1521E-02	1.1298E-01
Kg.m	by	86.7962	1	9.8067
N.m	by	8.8507	1.0197E-01	1

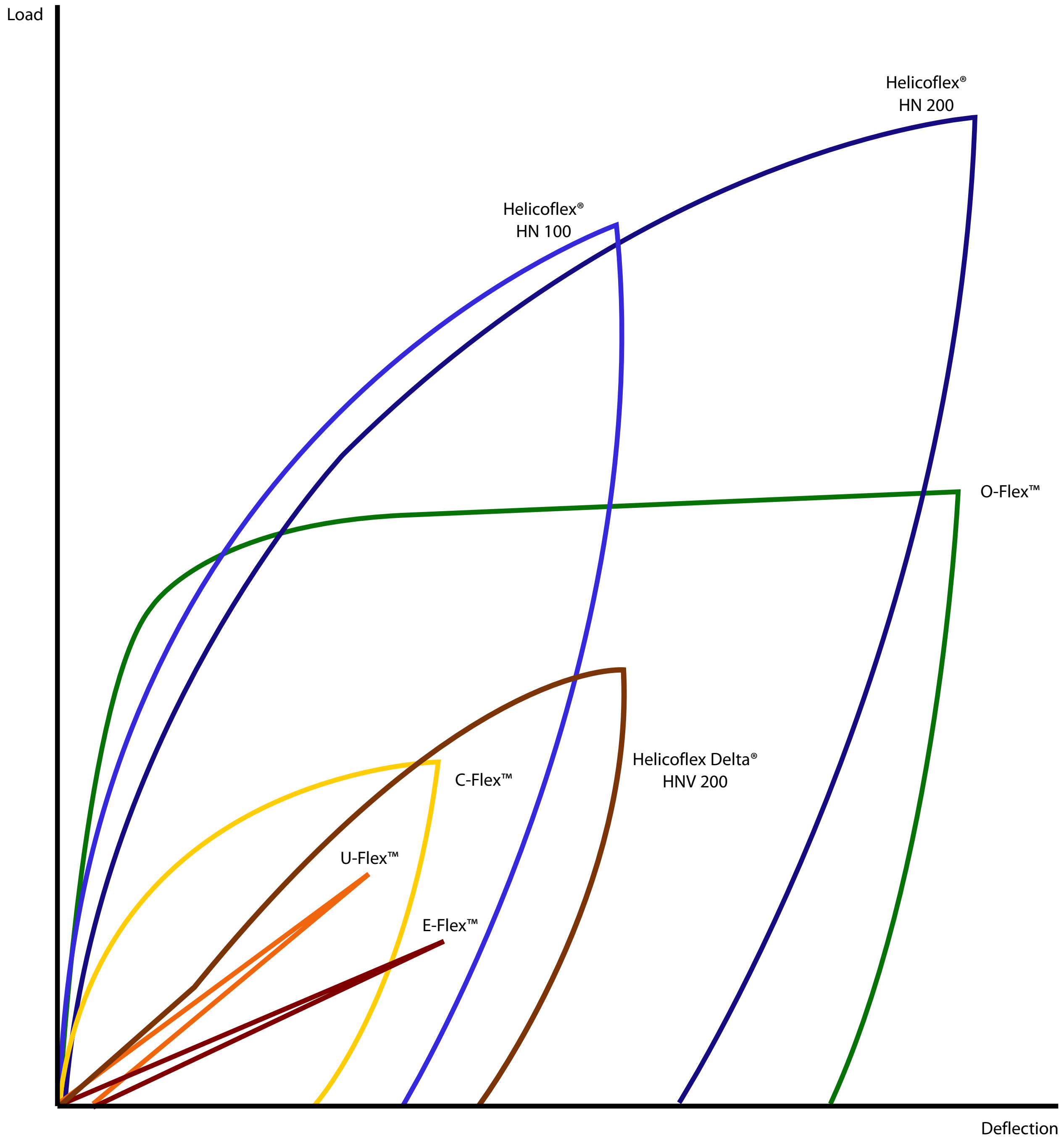
Units of Load/Unit Length

Multiply	by	To Obtain
$N.mm^{-1}$	5.71	$lb.in^{-1}$
$lb.in^{-1}$	1.75E-01	$N.mm^{-1}$

Temperature

Fahrenheit	Fº = (9/5)C+32
Celsius	Cº = 5/9 (F-32)
Kelvin	K = C+273

NOTE: The technical data contained herein is by way of example and should not be relied on for any specific application.



Load
Helicoflex®
HN 200
Helicoflex®
HN 100
O-Flex™
C-Flex™
Helicoflex Delta®
HNV 200
U-Flex™
E-Flex™
Deflection

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