Gasket Fundamentals & Installation Training

Benefits

- Correctly Bolted Flange Connections
- Increased Gasket Life
- Reduced Maintenance Costs
- Fugitive Emissions & The Environment
- Increased Plant Pipeline Safety & Reliability

DURLON®
SEALING SOLUTIONS

Contents

Causes of Gasket Failure

- Uneven loading of flanges
- Gasket load too low
- Bolt strength too low
- Torque loss
- Bolt relaxation/strength (approximately 10% torque lost in first 24 hours)
- Gasket creep
- Vibration in the system
- Thermal cycling
- Water hammer
- Elastic interaction during bolt tightening
- Improper gasket installation practice

Reducing Gasket Failures

- Use proper gasket installation practices
- Lubricate bolts, washers and nut facings
- Bring the flanges together slowly and parallel (multiple passes with increasing torque, each pass following proper tightening sequence).
- Use a 1⁄16” thick gasket up to 8” flanges and 1⁄8” for 10” and above. (1⁄16” has less gasket creep)
- Be sure there is adequate gasket stress
- Periodic re-torquing (first 4-24 hrs is critical)
- Use the correct tightening pattern/method for the job:

  Order of efficiency (least to greatest)

    1) Torque wrench
    2) Hydraulic torque wrench
    3) Hydraulic stud tension
- Use the installation procedures that follow on the bolt tightening worksheet (pg. 17)
- Refer to torque information from your gasket manufacturer torque data tables (pg. 20-23)

Joint Disassembly Warning

Prior to any joint disassembly, it is essential that plant procedures (lock-out and tag-out procedures) have been followed to depressurize and de-energize the system, including the removal of liquid head from the system, to ensure that the BFJA (Bolt Flange Joint Assembly) may be safely opened.

After reaffirming that all pressure on the joint has been released and the joint has been separated, proceed with bolt loosening and nut removal. Good general practice is to loosen the side of the joint away from yourself first to ensure in case of an accidental release that it is directed away from yourself.

Disassembly of a BFJA should be conducted in a similar fashion as the initial assembly. Bolts should be loosened in increments and also in a crisscrossed pattern to ensure an even unload. The first loosening should be done at approximately 50% of the original recommended torque. Once joint separation is achieved, proceed with the balance of the bolt loosening and nut removal. An aid such as a hydraulic or manual flange spreader may be used if necessary to separate the joint.

Torque loss is inherent in any bolted joint. The combined effects of bolt relaxation, gasket creep, vibration in the system, thermal expansion, and elastic interaction during bolt tightening, contribute to torque loss. When torque loss reaches an extreme, the internal pressure exceeds the compressive force holding the gasket in place and a leak or blowout occurs.

A key to reducing these effects is proper gasket installation. Reduced maintenance costs and increased safety can be obtained by bringing the flanges together slowly and parallel when installing a new gasket, taking a minimum of 3 bolt tightening passes, and following the correct bolt tightening sequence/pattern.

Even when installation is ideal, the bolt stress is uniformly applied to each bolt, and the gasket is properly compressed, problems can still arise.

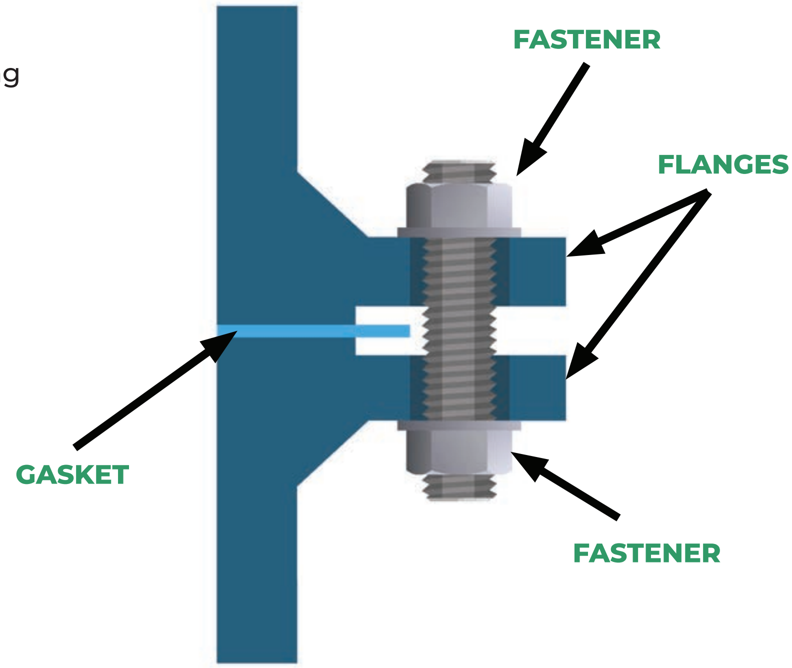
Inherently with time, loosening will occur due to gasket factors already mentioned. If other factors such as cycling, thermal upsets or vibration are present, periodic re-torquing might be necessary. Never re-tighten soft sheet gasket materials at operating pressure or temperature.

THE BOLTED JOINT

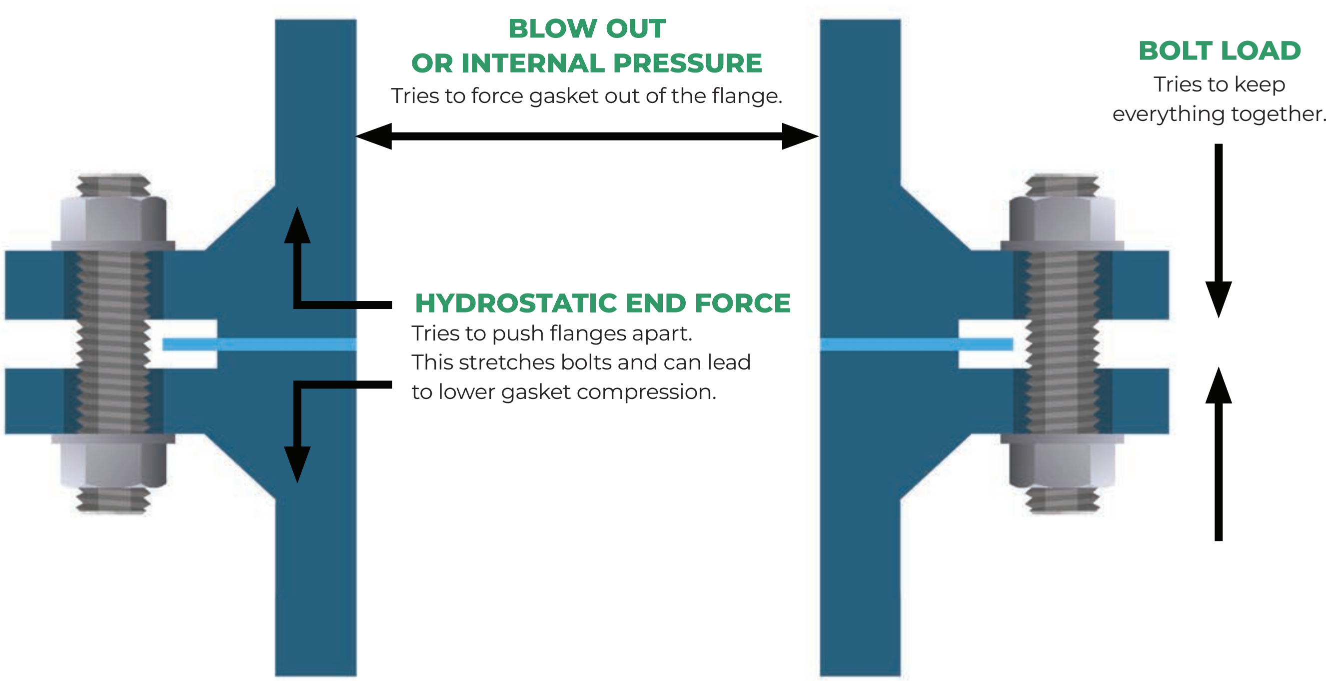
Causes of Bolted joint failures:

- Low bolt loads and/or gasket loading
- Weak fastening materials
- No torque control
- Inadequate lubrication of fasteners
- Poor installation
- Uneven gasket compression
- Poor flange design

Note: *3" 150#, 8" 150#, and 12" 150# can be problematic.*



FORCES INVOLVED IN THE BOLTED JOINT



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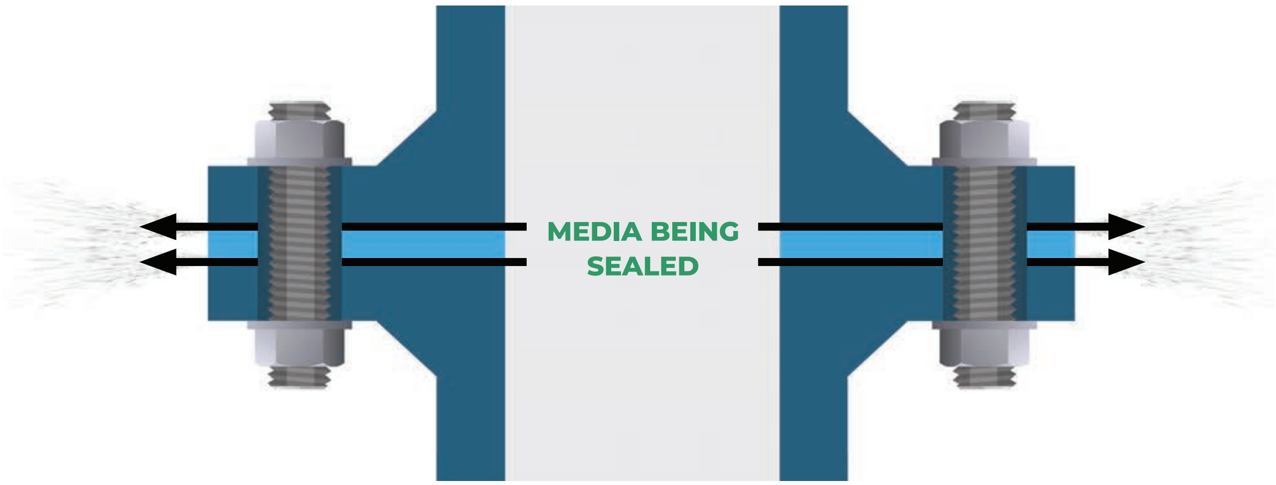
HOW/WHERE GASKETS CAN LEAK

In both cases, tangential or permeation, the leakage is a result of either insufficient compression or a loss of compression on the gasket while in service.



Permeation (through gasket body) leakage or Tangential (flange-surface interface) leakage

MECHANICS OF A BOLTED JOINT

Force is applied by the bolts through the flange:

- This compresses and densifies the gasket
- Reduces porosity in the gasket
- Creates a sealing barrier at the gasket ID
- Prevents fluid from penetrating and degrading the gasket
- Creates a seal

During gasket installation the bolts are tightened, this in turn transfers the force applied by the bolts, through the flange to compress the gasket. Many people are not aware that gasket materials are porous, be it compressed non-asbestos, PTFE's, flexible graphite, or spiral wound gaskets; they all have a certain amount of porosity.

This is why it's important that during the bolt tightening process a sufficient load is applied to not only compress the gasket but densify the gasket and reduce this porosity.

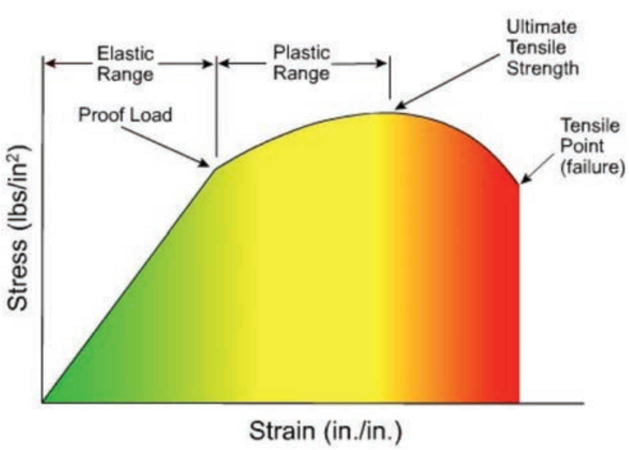
FUNCTION OF BOLTS, STUDS AND NUTS

Bolt selection is important.
Automobiles utilize SAE Grade 5 or Grade 8 fasteners, which are specifically designed for various sections of the vehicle and are chosen based on the fastener's clamping ability.

The clamping capability of a bolt, determined by its elasticity, refers to its capacity to revert to its original shape after undergoing deformation, resembling a "spring-like" behavior.

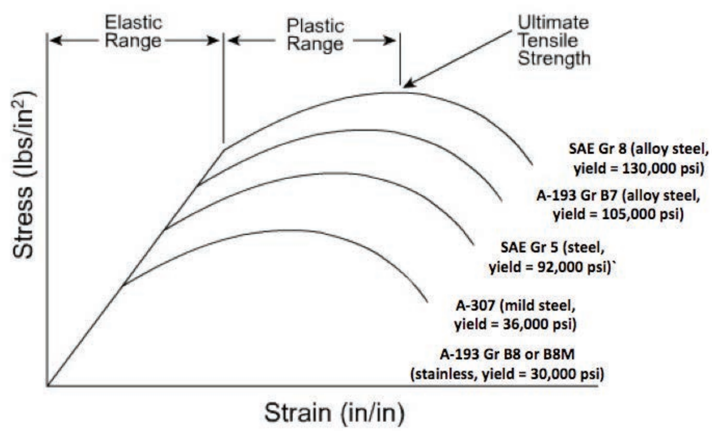
The bolts within a flange system are intended to exhibit spring-like properties, relying on their elasticity to not only withstand stretching but also to recover their initial shape after the load is released. Consequently, the appropriate selection of bolt grade holds considerable importance, often overlooked by the end user, who may not fully appreciate its significance.

It is vital to ensure that the applied stress remains within the fastener's elastic range, which encompasses the amount of stress exerted up to the fastener's proof load. Exceeding this range can lead the fastener into a plastic range, resulting in permanent deformation and rendering it unusable.



In cases where a bolt is overtightened to the point of fracture, it surpasses both the elastic and plastic ranges, reaching the tensile point.

The bolt grade affects how much load can be placed on the gasket.
The problematic aspect of a weak fastener lies in the fact that, when calculating torque, the relation between the total bolt area, the bolt strength, and the gasket area is taken into account. When a fastener is too weak, it is incapable of generating sufficient load to compress the gasket, enhance its density, and reduce porosity.
Examples:
- A-193 Gr B8 or B8M Class 1 (stainless, yield = 30,000 psi)
- A-307 (mild steel, yield = 36,000 psi)
- SAE Gr 5 (steel, yield = 92,000 psi)
- A-193 Gr B7 (alloy steel, yield = 105,000 psi)
- SAE Gr 8 (alloy steel, yield = 130,000 psi)



The Importance of Lubrication:
- The amount of friction is reduced.
- Friction works against the user.
- The lubricant should be applied to both the bolt threads, the nut threads, and the nut flange facing.
- Reducing friction is very important.
- The torsional force is greater with bolts (with heads) than studs, where both ends are free to turn.

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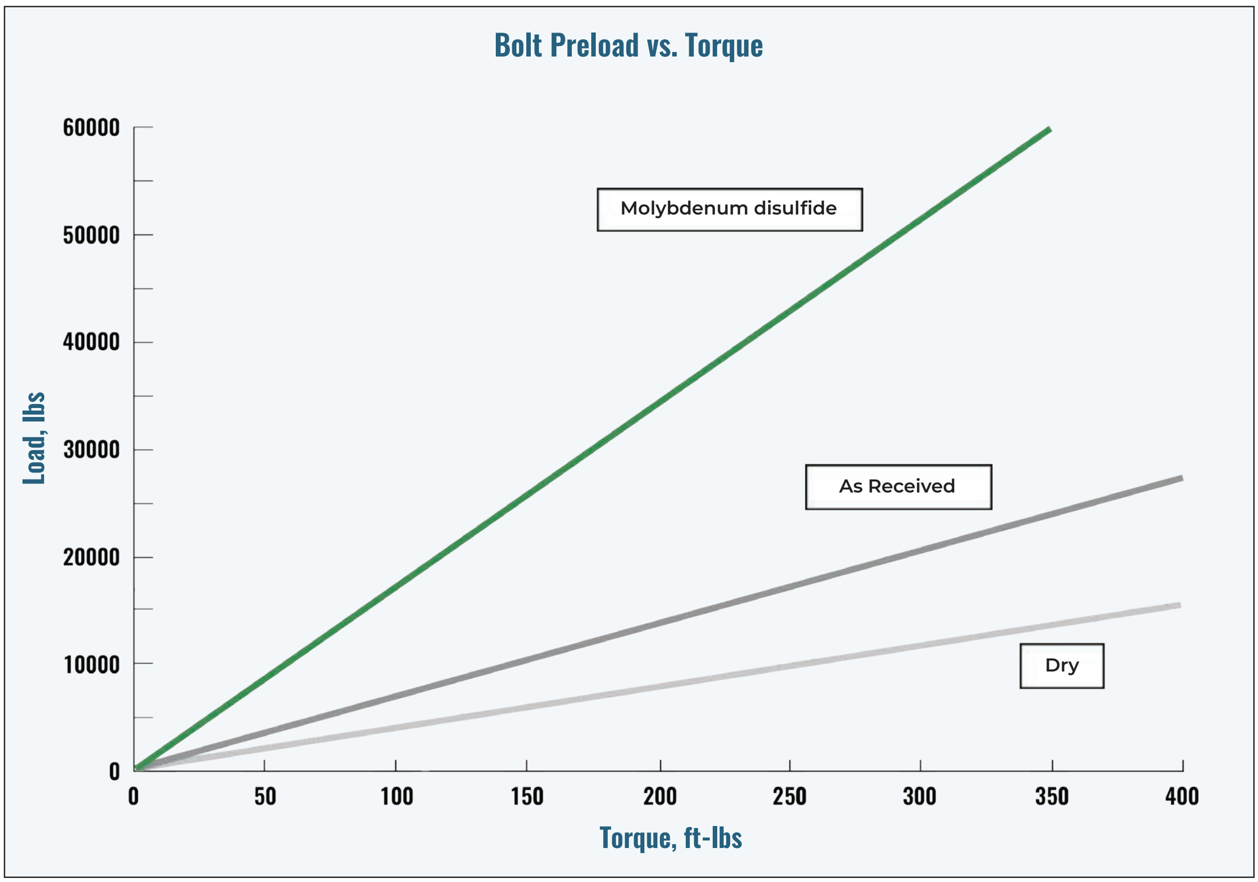
Below is an illustration of the effect of lubrication.

Showing a significant difference, this example uses a torque wrench set to 200 ft-lbs.

- Fastener is dry with no lubricant, the load on the bolt is ~8,000 lbs.
- Fastener is lightly oiled (as received), the load on the bolt is ~13,000 lbs.
- Fastener is using a very efficient lubricant such as a moly paste, the load on the bolt is ~34,000 lbs.

The effort exerted in tightening the fasteners can be significantly affected by lubrication. In each case, the nut would have appeared tight, but the user's effort was being hindered by friction.



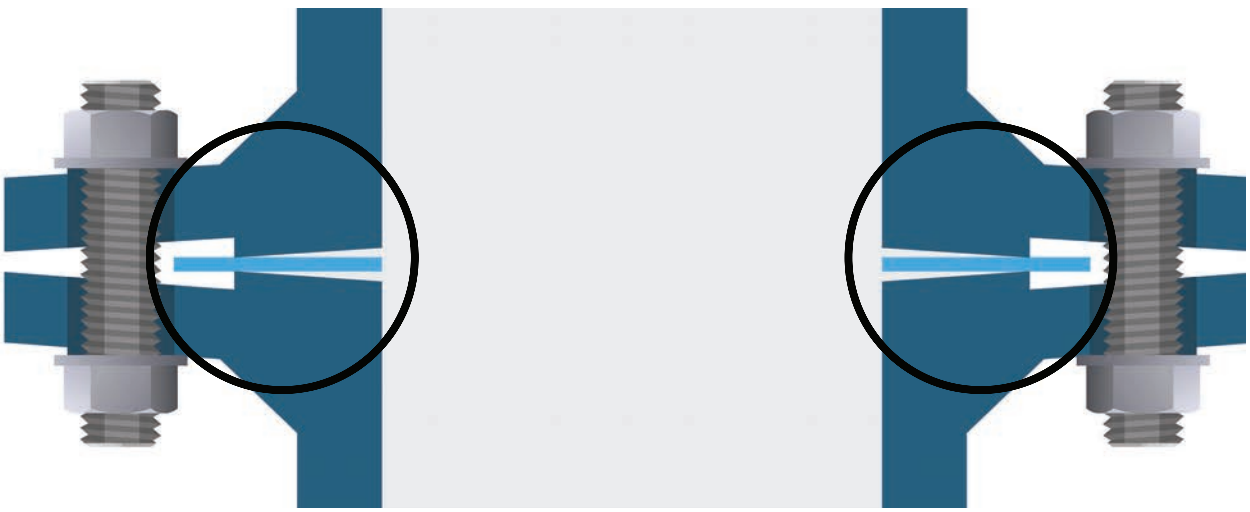
EFFECTS OF OVER-TIGHTENING

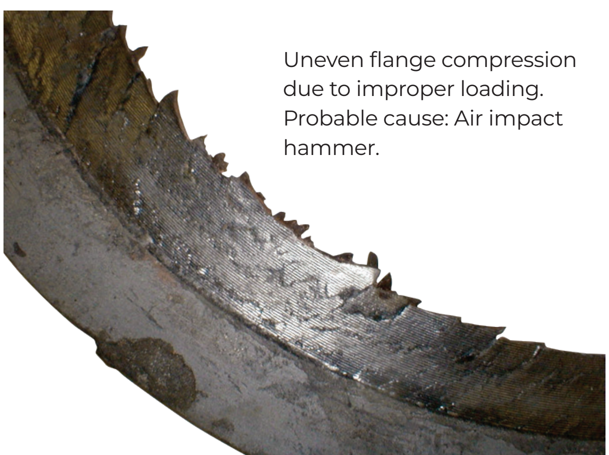
Over-tightening - Flange Rotation

The illustration below represents an over-tightening issue that can be observed on certain Class 150 flanges due to their lightweight nature. In the industry, the term "flange rotation" is used to describe this phenomenon, wherein the outer edges of the flange are "rotated" towards each other.

- Reduces the gasket contact area
- Crushes the gasket toward OD
- Allows fluid to penetrate gasket ID leading to deterioration of gasket
- Damages the flanges
- Result: leakage

In the case of Class 150 flanges, this occurrence arises due to their lightweight nature and insufficient thickness when subjected to high stress. Besides the potential damage to the flanges themselves, the over-tightening can result in gasket damage and potential leakage.





Uneven flange compression due to improper loading. Probable cause: Air impact hammer.

This gasket serves as an example of potential issues, which could be attributed to several factors. The first possibility is the absence of proper procedures. Attention is drawn to the focal point where the stress was exerted. Observing the gasket's inner diameter, you can notice that the jagged edges extend in opposite directions from the focal point.

Cut almost entirely around the gasket at the OD of the raised face due to severe over-tightening (flange rotation).

This is a 10" Class 150 ring gasket that leaked. It was cut approximately 270 degrees around by the outside edges of the mating raised faces on the flanges.

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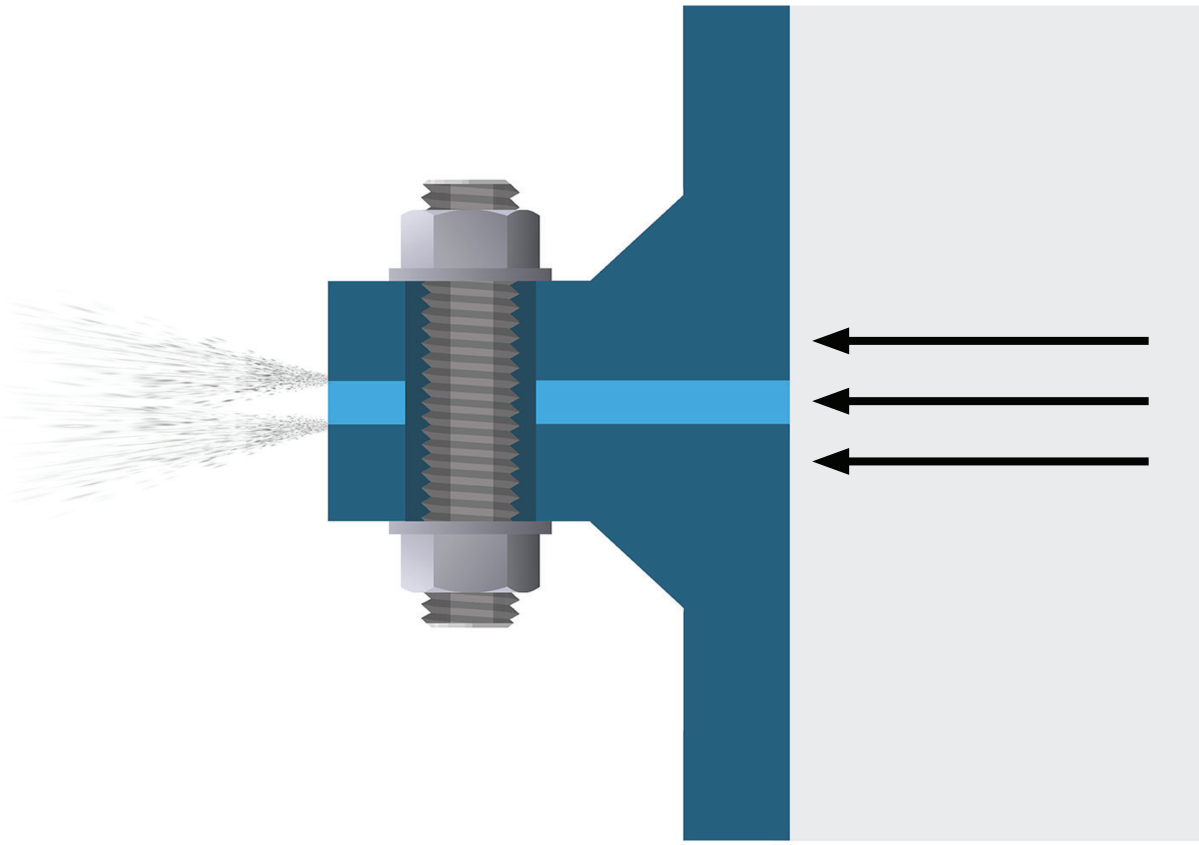
EFFECTS OF UNDER-TIGHTENING

The opposite of overloading is under-tightening, wherein an insufficient load is applied, resulting in the penetration of fluid and degradation of the gasket. It should be noted that all gaskets possess porosity and necessitate an adequate load to achieve a proper seal.

- Fluid is allowed to penetrate the gasket ID, leading to the deterioration of the gasket.
- Under-loading can lead to gasket blow-out or leakage.
- The unloading caused by temperature or pressure cycling can have an equivalent effect.



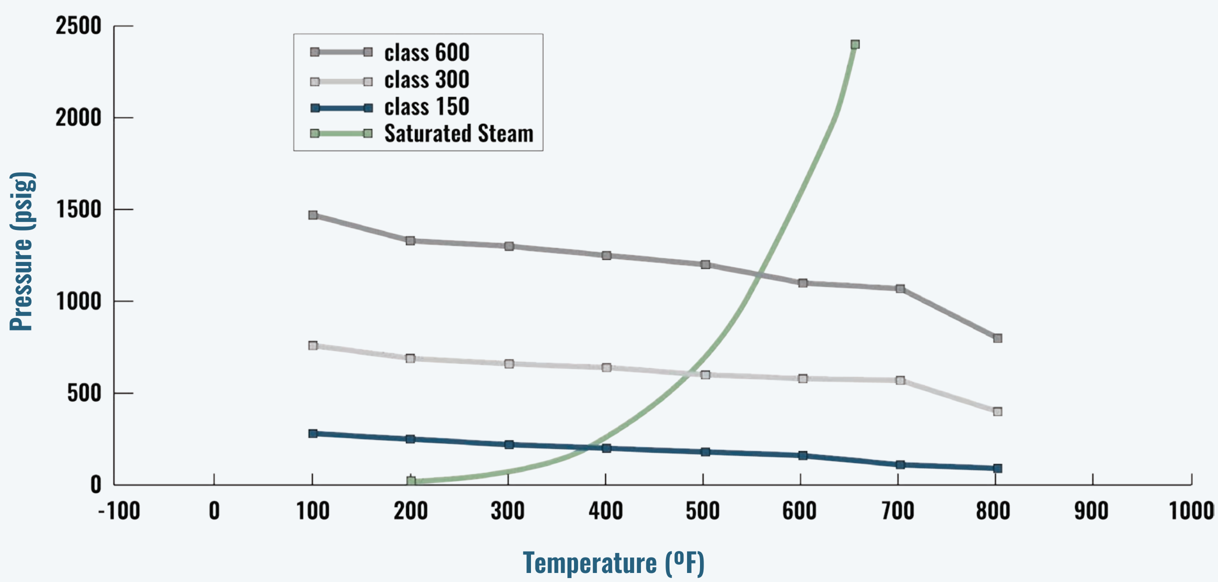
PRESSURE-TEMPERATURE (PxT FACTOR)

In all piping systems the flanges, valves, and the piping itself have a pressure – temperature relationship. This PxT factor is the result of multiplying the operating pressure times the operating temperature to arrive at a numerical value. This value is not constant and is different at each temperature and pressure combination.

In the table below, the PxT factors for carbon steel piping per ANSI B16.34 and saturated steam are shown. The fact that PxT values exists for piping should indicate that such values also exist for gasketing, and just like piping, those values change with differences in the pressure and temperature.

PRESSURE-TEMPERATURE RELATIONSHIPS						
Temp	(Carbon Steel) Class 150		(Carbon Steel) Class 300		Saturated Steam	
°F	psi	(P x T)	psi	(P x T)	psi	(P x T)
100	285	(28,500)	740	(74,000)	1	(100)
200	260	(52,000)	675	(135,000)	12	(2,400)
300	230	(69,000)	655	(196,500)	68	(20,400)
400	200	(80,000)	635	(254,000)	250	(100,000)
500	170	(85,000)	600	(300,000)	680	(340,000)
600	140	(84,000)	550	(330,000)	1550	(930,000)
700	110	(77,000)	530	(374,500)	3100	(2,170,000)

The following chart graphically represents the information presented above.
Pressure - Temperature Ratings for ANSI Class 150, 300 and 600 WCB Piping Components (ASME/ANSI B16.34-1988)



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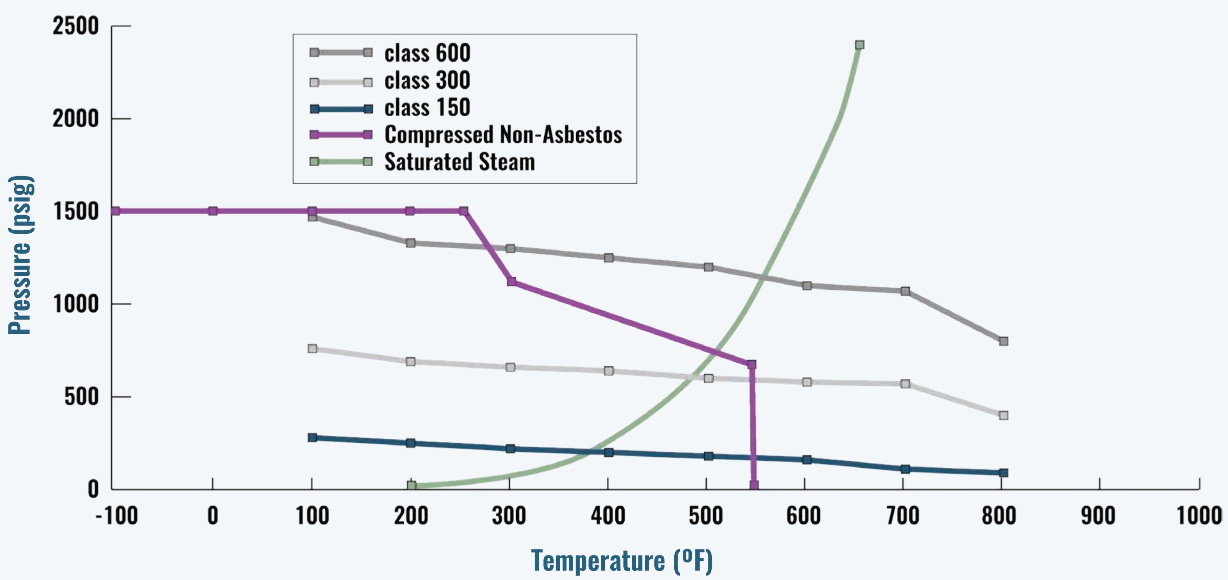
Now we can look at how sheet gaskets fit. As stated to the left, just like piping, the PxT relationship for gaskets changes with each pressure – temperature combination and therefore is not a constant.

The following chart shows compressed non-asbestos and compressed asbestos gasketing vs. 3 different pressure classes and saturated steam for reference. All sheet non-asbestos gasketing should be limited to Class 300 and below. Pressure - Temperature Ratings for ANSI Class 150, 300 and 600 WCB Piping Components (ASME/ANSI B16.34-1988) vs. Sheet Gasketing



Physical and Mechanical Properties

ASTM F104, the Standard Classification System for Non-metallic Gasket Materials includes a line call-out encompassing ASTM test methods for evaluating the physical and mechanical properties of non-metallic gasket materials. Some of these ASTM tests are:

F 36 – Compressibility and Recovery
F 2378 – Sealability
F 38 – Creep relaxation
F 146 – Fluid Resistance
F1574 – Compressive Strength

In addition to ASTM tests, we also do testing to BSI (British Standards), DIN (German Institute for Standardization) and FSA (Fluid Sealing Association) standards. These tests include:

ASTM – F2837 – Hot Compression
DIN – 3535 – Gas Permeability
FSA – NMG-204 – High Pressure Saturated Steam Test

Other Considerations

Fire safe capability. There is no standard for "fire safe" gasket materials. Durlon® 8500 passed the API 6FB, Durlon® 8900 passed the API 607 and Durlon® 9000 passed the API 6FA fire tests - all done by an independent lab. API Spec 6FB, Fire Test for End Connections, and API Bulletins, 6F1 and 6F2, do discuss fire testing but for metal gaskets and API rings, not soft gasket material.

Gasket design factors. The m and Y values established by ASME and the newer design factors being developed by the PVRC for fugitive emissions, are additional considerations. The m and Y values do not take fugitive emissions into account whereas the newer tightness parameters (Tp) do.

These gasket factors recognize that all joints leak to some extent. Therefore, an acceptable level of leakage is defined. A leak rate of 1/2,480 lb/hr per inch of OD (0.002 mg/sec. mm) has been defined as a "standard" acceptable leak rate and is known as T2.

Tp classes and their associated leak rates:

T1 – Economy
1/25 lb/hr per inch of OD (0.2 mg/sec. mm)
T2 – Standard
1/2,480 lb/hr per inch of OD (0.002 mg/sec. mm)
T3 – Tight
1/248,000 lb/hr per inch of OD (0.00002 mg/sec. mm)

Torque values for Durlon® products are calculated using a tightness parameter of T3.

PROPER GASKET INSTALLATION

Make sure system is at ambient temperature and depressurize.

Visually examine and clean flanges, bolts nuts and washers.
Replace defective components if necessary.

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Lubricate bolt threads, nut threads & facing, and washers.

Install a new gasket.
Do not reuse old gaskets or multiple gaskets.

PROPER GASKET INSTALLATION - CONT'D

Number bolts in a cross-pattern sequence according to this diagram.
Pre-tightening; Hand tighten all nuts and if possible snug nuts with a hand wrench.

Use the appropriate cross pattern tightening sequence.

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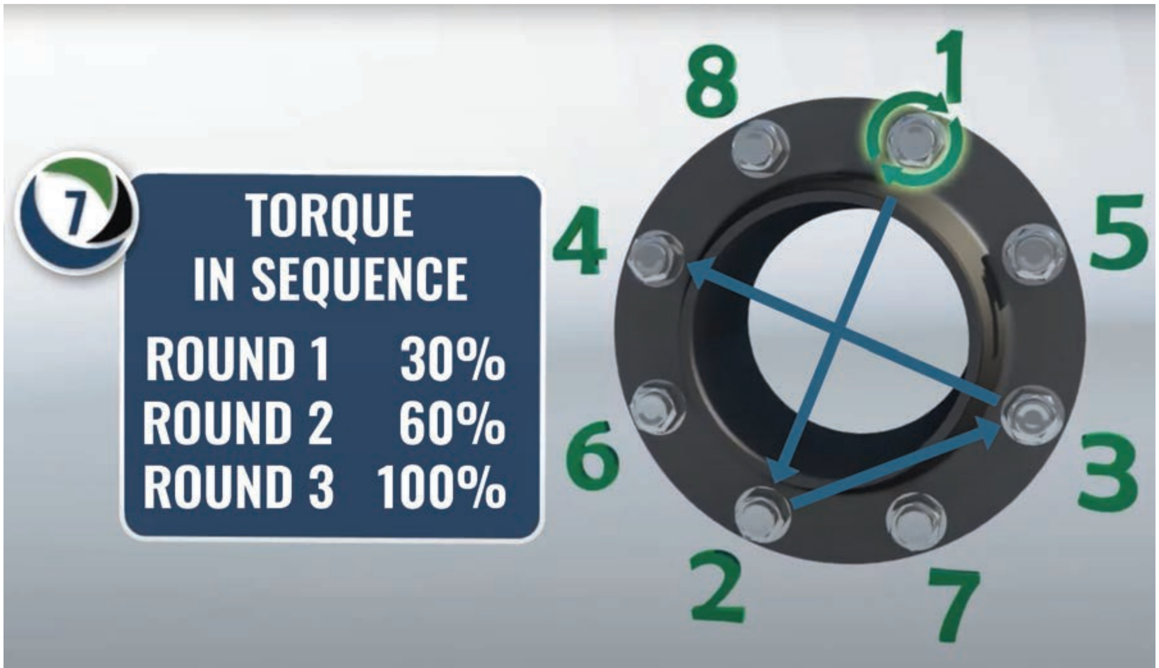
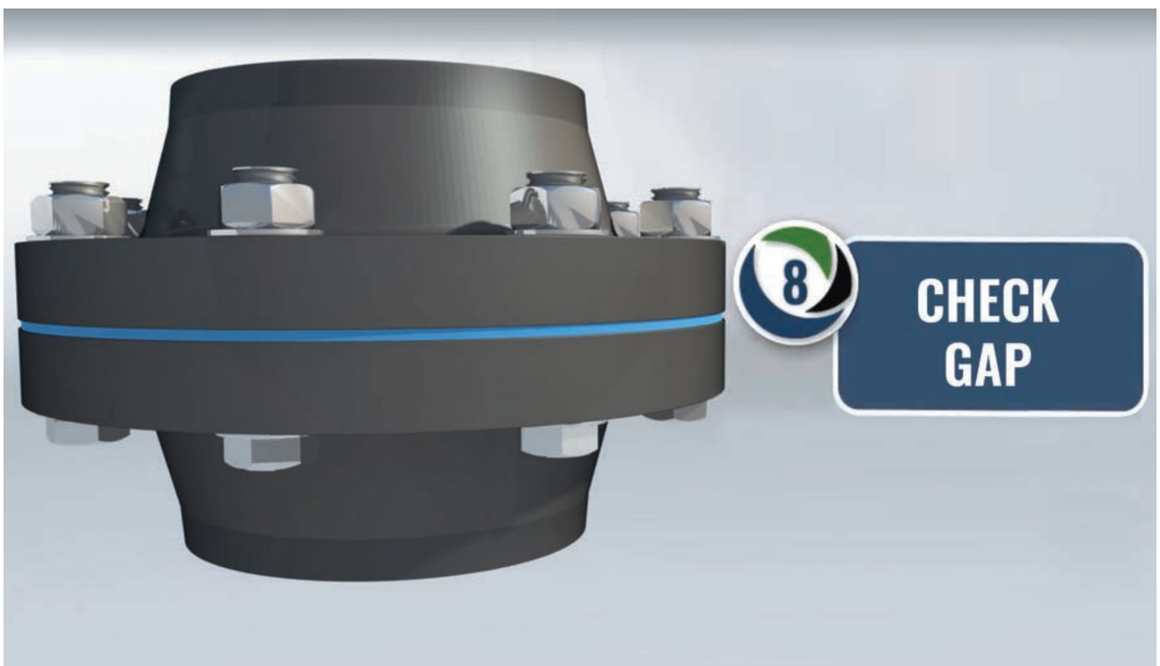


Round 1: 30% of target torque. Round 2: 60% of target torque.
Round 3: 100% of target torque.



Check gap between each of these rounds, measured at every other bolt. If the gap is not reasonably uniform, make the appropriate adjustments by selective bolt tightening before proceeding.

PROPER GASKET INSTALLATION - CONT'D

Use a rotational clockwise tightening sequence starting with bolt 1 for one complete round and continue until no further nut rotation occurs at 100% of the target torque value for any nut.

Final-round is to re-torque after 4-24 hours at ambient temperature if possible. A large percentage of short-term preload loss occurs within 24 hours after initial tightening. Repeat rotational round to recover this loss. This re-torquing round covers this loss and is especially important for PTFE gaskets.

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BOLT TIGHTENING WORKSHEET

We recommend the completion of any installation assembly worksheet with the details, including the installer signature and date for verification. You can use the Durlon worksheet for easy adoption into your QC program. Note: If unsure about flange finish, defects, alignment or alternative tightening procedure, please refer to ASME PCC-1 Guideline for pressure boundary bolted flange joint assembly. For other specific applications or general procedures please contact the Durlon Technical Department at: tech@durlon.com

Location/Identification: ______________________________ Nominal Bolt Size: __________

Gasket Contact Surface Finish on Flange: __________; Lubricant Used: ____________________

(Initial each step.)

____ 1. Be sure system is at ambient temperature and depressurized. Follow local safety rules.

____ 2. Visually examine and clean flanges, bolts, nuts and washers. Replace components if necessary.

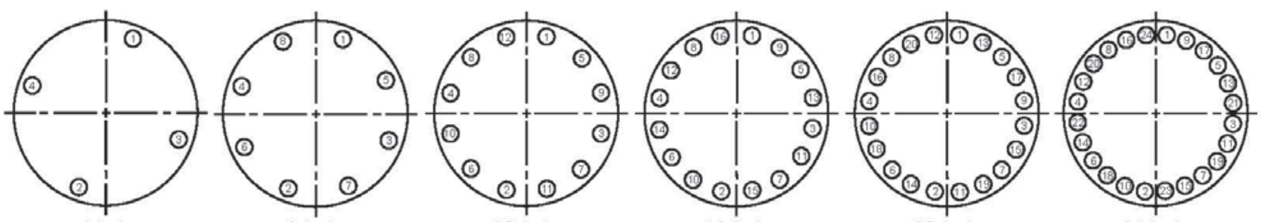
____ 3. **Lubricate bolts, nuts, and nut bearing surfaces**. Use of hardened steel washers are recommended.

____ 4. Install new gasket. DO NOT REUSE OLD GASKET, OR USE MULTIPLE GASKETS.

____ 5. Number bolts in cross-pattern sequence according to the appropriate sketch below.

____ 6. **IMPORTANT! HAND TIGHTEN NUTS, then using a hand wrench SNUG BOLTS 1/8 to 1/4 turn, following the appropriate cross pattern tightening sequence for the number of bolts below.**

____ 7. Starting at the #1 bolt, use the appropriate cross-pattern tightening sequence in the sketch below for Rounds 1, 2, and 3 (each sequence constitutes a "Round").



Final Torque: _______ ft-lbs

LUBRICATE, HAND TIGHTEN, PRE-TIGHTEN BOLTS

Round 1 - Tighten to ________ ft-lbs - **1st torque** value in torque chart* (30% of final torque)

Round 2 - Tighten to ________ ft-lbs – **2nd torque** value in torque chart (60% of final torque)

Round 3 - Tighten to ________ ft-lbs - **Final torque** value in torque chart (100% of final torque)

*Refer to torque chart on next page

Check gap at 90° intervals around the flange between each of these rounds. Larger flanges may require checking the gap in smaller intervals. If the gap is not reasonably uniform, make the appropriate adjustments by selective bolt tightening before proceeding.

___ **Rotational Round** - 100% of Final Torque (same as Round 3). Use ROTATIONAL, clockwise tightening sequence, starting with Bolt No. 1, for at least two complete rounds and continue until no further nut rotation occurs at 100% of the Final Torque value for any nut.

___ **Retorque** - Short-term bolt preload loss can occur between four to twenty-four hours after initial tightening due to bolt relaxation and/or gasket creep. Repeating the Rotational Round recovers this loss. This is especially important for PTFE gaskets.

Joint Assembler: ______________________________ Date: ______________

For torque questions, or tightening patterns for large diameter flanges, contact info@durlon.com

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GASKET FACTORS

Gasket factors are very important to understand but unfortunately can be difficult to understand or are easily misinterpreted. This section contains some of the more popular versions of Gasket Factors used in determining the recommended torque for gasket installation.

EN 13555 is a working standard, much like ASME PVRC gasket factors, in the EU. It provides the testing procedures to allow persons to derive the gasket parameters: Qsmax, Q(minL), Qsmin(L), PQR, and EG so they can be used in design equations found in EN 1591-1 (Flanges and Their Joints - Design Rules for Gasketed Circular Flange Connections - Part 1: Calculation).

For a further definition of the gasket parameters see chart to right.

Gasket Parameters	
Q_{smax}	Maximum seating stress required on the gasket at a given temperature without crushing the material.
$Q_{(minL)}$	Minimum seating stress that is required in assembly at ambient temperature to seat gasket into the flange serrations and seal internal leakage, based on tightness class, L, and specified test pressure.
$Q_{smin(L)}$	Minimum gasket seating stress required in service conditions after unloading gasket (at service temperature), so that the specified tightness class L, is maintained based on internal test pressure.
P_{QR}	This factor allows for the gasket's effect on the load applied and the relaxation of the gasket from start (final bolt-up) and after the extended life term of the material`s intended service temperature.
E_G	This is the unloading moduli, which is derived from the recovery of the gasket thickness between the initial compression seating stress and unloading the gasket to ⅓ of its initial seating stress.

When the final torque values are calculated using the previous gasket parameters, leakage can be classified into three tightness classes.

See chart to right.

Tightness Class	Specific Leak Rates (mg/s-m)
$L_{1.0}$	1.0
$L_{0.1}$	0.1
$L_{0.01}$	0.01

M & Y

M and Y values are for flange design only and are not meant to be used as gasket seating stress values in actual service. "M" is known as the maintenance factor or the multiplier. The "Y" factor is the minimum stress required (psi) over the sealing area of the gasket to provide a seal at an internal pressure of 2 psig. "Y" is not considered to be the minimum seating stress for the gasket in service. These values are used in formulas in the ASME Boiler and Pressure Vessel Code, Division 1, Section VIII, Appendix 2, to give a WM1 (minimum required bolt load for operating conditions, psi) or WM2 (minimum required bolt load for gasket seating, psi) value, based on either gasket seating or the internal pressure. The flange is designed based on the greater of these two values (WM1 or WM2). This will ensure that the flange is robust enough to maintain adequate gasket seating stress, which can decline due to flange rotation with weaker flanges when bolted up and internal pressure is introduced. These values do not take fugitive emissions into account and therefore, based on leakage, newer gasket constants Gb, a, and Gs, are being developed, based on leakage to take this into account.

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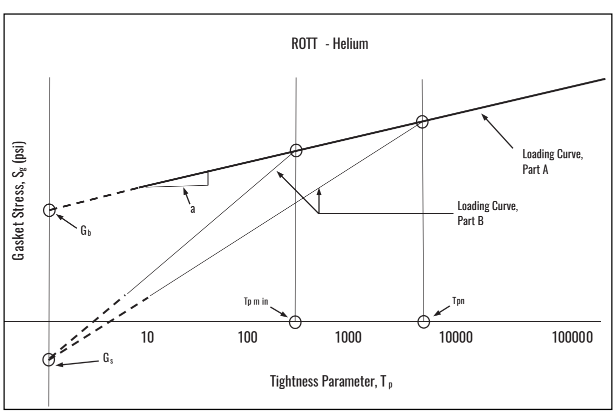
Alternative ASME PVRC Gasket Factors: Gb, a and Gs

New gasket factors to replace the ASME Code m and Y are currently being developed by the Pressure Vessel Research Council (PVRC) and ASME. The current m and Y are difficult to replicate for non-asbestos gaskets and do not take joint leakage into account. The new approach to bolted joint design makes the tightness of the joint a design parameter. In a manner similar to that of the traditional ASME Code method, the design bolt load for a joint is calculated for operating and seating requirements from the new constants Gb, a, and Gs, and the required tightness class associated with the minimum tightness.

"Gb" and "a" provide the gasket seating load; similar to Y in the present Code. "Gs" is associated with the operating stress and is similar to the m value in the present Code. The proposed ASME constants Gb, a, and Gs give a design bolt load, obtained by interpretation of leakage test data as plots of gasket stress, Sg, versus a tightness parameter, Tp. Tp is the pressure (in atmospheres) normalized to the atmospheric pressure required to cause a helium leak rate of 1 mg/sec for a 150 mm OD gasket in a joint. Since this is about the same as the OD of an NPS 4 joint, the pressure to cause a leak of 1 mg/sec of helium for that joint is its tightness. A standard test procedure, the PVRC Room Temperature Tightness Test (ROTT), has been designed to produce the constants Gb, a, and Gs. Low values for Gb, a, and Gs are desirable while a higher value of Tp means a tighter joint.

<table>
<tr><th colspan="9">Table for m ; Y, MPa (psi)</th></tr>
<tr><td>Tightness Type</td><td colspan="2">Economy</td><td colspan="2">Standard</td><td colspan="2">High</td><td colspan="2">Exceptional</td></tr>
<tr><td>Tightness Class</td><td>T1</td><td>T1.5</td><td>T2</td><td>T2.5</td><td>T3</td><td>T3.5</td><td>T4</td><td>T4.5</td></tr>
<tr><td>Leak Rate (mg/s/mm)</td><td>2.0E-01</td><td>2.0E-02</td><td>2.0E-03</td><td>2.0E-04</td><td>2.0E-05</td><td>2.0E-06</td><td>2.0E-07</td><td>2.0E-08</td></tr>
<tr><td rowspan="4">Compressed Non-Asbestos & PTFE Materials</td><td colspan="3" rowspan="4">2.5;20 (2,900)</td><td>14.2;20 (2,900)</td><td>32.3;60 (8,700)</td><td rowspan="3">14.3;100 (14,500)</td><td rowspan="4">11.5;160 (23,200)</td><td rowspan="4">98.9;160 (23,200)</td></tr>
<tr><td>7.9;40 (5,800)</td><td>7.9;80 (11,600)</td></tr>
<tr><td rowspan="2">2.7;60 (8,700)</td><td>2.4;100 (14,500)</td></tr>
<tr><td>1.7;160 (23,200)</td><td>3.4;160 (23,200)</td></tr>
<tr><td rowspan="3">Graphite Filled SWG</td><td colspan="5">8;20 (2,900)</td><td>7;80 (11,600)</td><td rowspan="3">8;120 (17,400)</td><td>-</td></tr>
<tr><td colspan="5">4;40 (5,800)</td><td rowspan="2">4;160 (23,200)</td><td>-</td></tr>
<tr><td colspan="5">2.5;60 (8,700)</td><td>-</td></tr>
</table>

Recommended Value	Possible Value	Not Recommended Value



Tightness Class	Mass Leak Rate/Unit Diameter (L_{rm}) mg/sec-min (lb/hr per " of OD)
T1	2×10^{-1} (0.04)
T2	2×10^{-3} (0.0004)
T3	2×10^{-5} (0.000004)
T4	2×10^{-7} (0.00000004)
T5	2×10^{-9} (0.0000000004)

TORQUE VALUES - SHEET GASKET MATERIAL

Pipe Size	ASME B16.21 Ring Gasket, Ft-Lbs (N-M)					
	Fasteners: A193-B7 or B16 lubricated with a never seize type lubricant, k=0.17					
	Class 150			Class 300		
	Min. [1, 2]	Optimal [1, 3]	Max. [4, 5]	Min. [1, 2]	Optimal [1, 3]	Max. [4, 5]
½”	10 (14)	25 (33)	25 (33)	10 (14)	25 (34)	25 (34)
¾”	15 (20)	30 (41)	35 (48)	15 (20)	45 (61)	45 (61)
1”	15 (20)	35 (48)	50 (68)	20 (27)	55 (75)	60 (81)
1 ¼”	25 (34)	45 (61)	75 (102)	35 (47)	80 (108)	90 (122)
1 ½”	30 (41)	60 (82)	80 (109)	50 (68)	140 (190)	145 (197)
2”	65 (88)	120 (163)	160 (217)	35 (47)	80 (108)	100 (136)
2 ½”	80 (108)	120 (163)	160 (217)	50 (68)	125 (169)	135 (183)
3”	115 (156)	150 (203)	160 (217)	75 (102)	180 (244)	200 (271)
3 ½”	65 (88)	120 (163)	160 (217)	85 (115)	180 (244)	225 (305)
4”	80 (109)	120 (163)	160 (217)	105 (142)	215 (292)	285 (386)
5”	120 (163)	215 (292)	280 (380)	140 (190)	215 (292)	285 (386)
6”	155 (211)	230 (312)	285 (386)	120 (163)	195 (264)	285 (386)
8”	215 (291)	285 (386)	285 (386)	195 (264)	315 (427)	460 (624)
10”	210 (284)	345 (468)	460 (624)	215 (292)	385 (522)	490 (664)
12”	280 (380)	400 (542)	460 (624)	330 (447)	570 (773)	735 (997)
14”	355 (481)	515 (698)	685 (929)	295 (400)	570 (773)	640 (868)
16”	340 (461)	515 (698)	675 (915)	420 (569)	795 (1078)	900 (1220)
18”	500 (678)	755 (1024)	1010 (1369)	465 (630)	885 (1200)	1020 (1383)
20”	460 (624)	755 (1024)	1010 (1369)	530 (719)	885 (1200)	1120 (1519)
22”	610 (827)	1060 (1437)	1415 (1918)	760 (1030)	1425 (1932)	1600 (2169)
24”	670 (909)	1060 (1437)	1415 (1918)	850 (1152)	1425 (1932)	1740 (2359)

Disclaimer: This is a general guide only and TFC/GRI, does not accept responsibility for negligence or misuse of this information.

General Notes:

a) Torque Values are in ft.-lbs. and assume new A193 Gr. B7 or B16 fasteners with 2H heavy hex nuts; with studs, nuts and the nut bearing surfaces lubricated with a never-seize type lubricant (k = 0.17)

b) A193 Gr. B7 & B16 fasteners have the same yield strength up to 4” diameter. There are “no” fasteners above 4” diameter in this chart.

c) All torque values are based on using a “calibrated” torque wrench.

d) All torque values in the chart above are based on using the tensile area of the fastener.

e) All torque values in chart are rounded to nearest 5 ft.-lbs.

Footnotes:

[1] Torque values are based using ASME B16.5-2017 MAWP (Maximum Allowable Working Pressure) at ambient in the gasket stress calculation.

Min. Torque Values:

[2] Min. torque values are based achieving 4,800 psi gasket stress without exceeding 80,000 psi bolt stress or PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding.

Optimal Torque Values:

[3] Optimal torque values are based on a target of 7,000 to 12,000 psi optimum gasket stress without exceeding 80,000 psi bolt stress or PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding.

Cases where torque equals ≤ 60,000 bolts stress

Cases where torque equals > 60,000 ≤ 75,000 bolts stress

Cases where torque equals > 75,000 ≤ 80,000 bolts stress

Max. Torque Values:

[4] Max. torque values are based on; max allowable 15,000 psi gasket stress; PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding, or 80,000 psi bolt stress, whichever occurs 1st.

Note: In some cases the max. torque values may be equal to the optimal torque values in order to optimize gasket stress levels.

[5] ½” through 1 ½” NPS & 3 ½” NPS due to “No Data”on flange yielding; Max. torque values are set to achieve max gasket stress of 15,000 or 80,000 psi bolt stress, whichever occurs 1st.

Cases where torque is based on 80,000 bolts stress.

Cases where torque is based on PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding.

Cases where torque is based on max. allowable 15,000 psi gasket stress.

TORQUE VALUES - SWGs – CLASS 150 & 300

Pipe Size	ASME B16.20 SWG Gasket, Ft-Lbs (N-M)					
	Fasteners: A193-B7 or B16 lubricated with a never seize type lubricant, k = 0.17					
	Class 150			Class 300		
	Min. [1, 2]	Optimal [1, 3]	Max. [4, 5]	Min. [1, 2]	Optimal [1, 3]	Max. [4, 5]
½”	20 (27)	35 (47)	55 (75)	20 (27)	35 (47)	55 (75)
¾”	25 (34)	50 (68)	80 (109)	30 (41)	65 (88)	100 (136)
1”	35 (47)	70 (95)	80 (109)	45 (61)	90 (122)	135 (183)
1 ¼”	40 (54)	*75 (102)	80 (109)	50 (68)	100 (136)	150 (203)
1 ½”	55 (75)	*75 (102)	80 (109)	85 (115)	165 (224)	250 (339)
2”	90 (122)	*150 (203)	160 (217)	45 (61)	90 (122)	115 (156)
2 ½”	105 (142)	*150 (203)	160 (217)	65 (88)	125 (169)	170 (230)
3”	150 (203)	**150 (203)	160 (217)	100 (136)	185 (251)	225 (305)
3 ½”	85 (116)	*150 (203)	160 (217)	110 (149)	210 (285)	285 (386)
4”	110 (150)	*150 (203)	160 (217)	145 (197)	*265 (359)	285 (386)
5”	160 (218)	*265 (359)	280 (380)	180 (244)	*265 (359)	285 (386)
6”	230 (313)	**265 (359)	285 (386)	170 (230)	*265 (359)	285 (386)
8”	285 (386)	***285 (386)	285 (386)	285 (386)	*430 (583)	460 (624)
10”	315 (427)	*430 (583)	460 (624)	310 (420)	560 (759)	675 (915)
12”	430 (583)	**435 (590)	460 (624)	480 (651)	865 (1173)	990 (1342)
14”	545 (739)	*645 (875)	685 (929)	425 (576)	760 (1030)	795 (1078)
16”	545 (739)	**645 (875)	675 (915)	635 (861)	*1105 (1498)	1115 (1512)
18”	870 (1180)	**945 (1281)	1005 (1363)	740 (1003)	*1200 (1627)	1210 (1641)
20”	775 (1051)	*945 (1281)	1005 (1363)	830 (1125)	*1290 (1749)	1300 (1763)
22”	635 (861)	*930 (1261)	1415 (1918)	1050 (1424)	*1830 (2481)	2330 (3159)
24”	1135 (1539)	**1325 (1796)	1415 (1918)	1325 (1796)	*2150 (2915)	2165 (2935)

Disclaimer: This is a general guide only and TFC/GRI, does not accept responsibility for negligence or misuse of this information.

General Notes:

a) Torque Values are in ft.-lbs. and assume new A193 Gr. B7 or B16 fasteners with 2H heavy hex nuts; with studs, nuts and the nut bearing surfaces lubricated with a never-seize type lubricant (k = 0.17).

b) A193 Gr. B7 & B16 fasteners have the same yield strength up to 4” diameter. There are “no” fasteners above 4” diameter in this chart.

c) All torque values are based on using a “calibrated” torque wrench.

d) All torque values in the chart above are based on the use of a inner/outer ring style (DRI) spiral wound gasket.

e) All torque values in the chart above are based on using the tensile area of the fastener.

f) All torque values in chart are rounded to nearest 5 ft.-lbs.

Footnotes:

[1] Torque values are based using ASME B16.5-2017 MAWP (Maximum Allowable Working Pressure) at ambient in the gasket stress calculation.

Min. Torque Values:

[2] Min. torque values are based achieving 12,500 psi gasket stress or at minimum above 10,000 psi gasket stress without exceeding 80,000 psi bolt stress or PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding.

Cases where gasket stress is >10,000 < 12,500 psi

Optimal Torque Values:

[3] Optimal torque values are based on 25,000 psi optimum gasket stress without exceeding 80,000 psi bolt stress or 500 psi below PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding.

Cases where torque equals ≤ 60,000 bolts stress

Cases where torque equals > 60,000 ≤ 75,000 bolts stress

Cases where torque equals > 75,000 ≤ 80,000 bolts stress

*Cases where gasket stress is ≥ 15,000 < 25,000 psi, **Cases where gasket stress is ≥ 12,500 < 15,000 psi, ***Cases where gasket stress is ≥ 10,000 < 12,500 psi

Max. Torque Values:

[4] Max. torque values are based on; max allowable 40,000 psi gasket stress; PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding, or 80,000 psi bolt stress, whichever occurs 1st.

Note: In some cases the max. torque values may be equal to the optimal torque values in order to optimize gasket stress levels.

[5] ½” through 1 ½” NPS & 3 ½” NPS due to “No Data”on flange yielding; Max. torque values are set to achieve max gasket stress of 40,000 or 80,000 psi bolt stress, whichever occurs 1st.

Cases where torque is based on 80,000 bolts stress.

Cases where torque is based on PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding.

Cases where torque is based on max. allowable 40,000 psi gasket stress.

TORQUE VALUES - SWGs – CLASS 400 & 600

Pipe Size	ASME B16.20 SWG Gasket, Ft-Lbs (N-M)					
	Fasteners: A193-B7 or B16 lubricated with a never seize type lubricant, k = 0.17					
	Class 400			Class 600		
	Min. [1, 2]	Optimal [1, 3]	Max. [4, 5]	Min. [1, 2]	Optimal [1, 3]	Max. [4, 5]
½”	20 (27)	35 (47)	55 (75)	20 (27)	35 (47)	55 (75)
¾”	35 (47)	65 (88)	100 (136)	35 (47)	65 (88)	100 (136)
1”	45 (61)	90 (122)	135 (183)	45 (61)	90 (122)	135 (183)
1 ¼”	50 (68)	100 (136)	150 (203)	55 (75)	100 (136)	150 (203)
1 ½”	90 (122)	170 (230)	250 (339)	95 (129)	170 (230)	250 (339)
2”	50 (68)	90 (122)	125 (169)	50 (68)	95 (129)	135 (183)
2 ½”	70 (95)	130 (176)	185 (251)	75 (102)	135 (183)	185 (251)
3”	100 (136)	190 (258)	260 (353)	110 (149)	195 (264)	275 (373)
3 ½”	180 (244)	345 (468)	455 (617)	190 (258)	355 (481)	455 (617)
4”	210 (285)	395 (536)	420 (569)	225 (305)	410 (556)	455 (617)
5”	270 (366)	*430 (583)	455 (617)	335 (454)	600 (813)	685 (929)
6”	250 (339)	*430 (583)	455 (617)	305 (414)	550 (746)	685 (929)
8”	405 (549)	*645 (875)	685 (929)	500 (678)	880 (1193)	1005 (1363)
10”	475 (644)	*785 (1064)	795 (1078)	580 (786)	1010 (1369)	1370 (1857)
12”	705 (956)	**735 (997)	745 (1010)	620 (841)	1070 (1451)	1300 (1763)
14”	600 (813)	*885 (1200)	930 (1261)	735 (997)	1250 (1695)	1620 (2196)
16”	860 (1166)	*1250 (1695)	1260 (1708)	1045 (1417)	1775 (2407)	2165 (2935)
18”	920 (1247)	*1440 (1952)	1515 (2054)	1455 (1973)	2470 (3349)	3195 (4332)
20”	1135 (1539)	*1650 (2237)	1665 (2257)	1385 (1878)	2305 (3125)	2930 (3973)
22”	1230 (1668)	*2075 (2813)	2145 (2908)	1535 (2081)	*2450 (3322)	2905 (3939)
24”	1670 (2264)	*2450 (3322)	2710 (3674)	2055 (2786)	3335 (4522)	4050 (5491)

Disclaimer: This is a general guide only and TFC/GRI, does not accept responsibility for negligence or misuse of this information.

General Notes:

a) Torque Values are in ft.-lbs. and assume new A193 Gr. B7 or B16 fasteners with 2H heavy hex nuts; with studs, nuts and the nut bearing surfaces lubricated with a never-seize type lubricant (k = 0.17).

b) A193 Gr. B7 & B16 fasteners have the same yield strength up to 4” diameter. There are “no” fasteners above 4” diameter in this chart.

c) All torque values are based on using a “calibrated” torque wrench.

d) All torque values in the chart above are based on the use of a inner/outer ring style (DRI) spiral wound gasket.

e) All torque values in the chart above are based on using the tensile area of the fastener.

f) All torque values in chart are rounded to nearest 5 ft.-lbs.

Footnotes:

[1] Torque values are based using ASME B16.5-2017 MAWP (Maximum Allowable Working Pressure) at ambient in the gasket stress calculation.

Min. Torque Values:

[2] Min. torque values are based achieving 12,500 psi gasket stress or at minimum above 10,000 psi gasket stress without exceeding 80,000 psi bolt stress or PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding.

Optimal Torque Values:

[3] Optimal torque values are based on 25,000 psi optimum gasket stress without exceeding 80,000 psi bolt stress or 500 psi below PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding.

- Cases where torque equals ≤ 60,000 bolts stress
- Cases where torque equals > 60,000 ≤ 75,000 bolts stress
- Cases where torque equals > 75,000 ≤ 80,000 bolts stress

*Cases where gasket stress is ≥ 15,000 < 25,000 psi, **Cases where gasket stress is ≥ 12,500 < 15,000 psi, ***Cases where gasket stress is ≥ 10,000 < 12,500 psi

Max. Torque Values:

[4] Max. torque values are based on; max allowable 40,000 psi gasket stress; PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding, or 80,000 psi bolt stress, whichever occurs 1st.

Note: In some cases the max. torque values may be equal to the optimal torque values in order to optimize gasket stress levels.

[5] ½” through 1 ½” NPS & 3 ½” NPS due to “No Data”on flange yielding; Max. torque values are set to achieve max gasket stress of 40,000 or 80,000 psi bolt stress, whichever occurs 1st.

- Cases where torque is based on 80,000 bolts stress.
- Cases where torque is based on PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding.
- Cases where torque is based on max. allowable 40,000 psi gasket stress.

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TORQUE VALUES - SWGs – CLASS 900, 1500 & 2500

Pipe Size	ASME B16.20 SWG Gasket, Ft-Lbs (N-M)								
	Fasteners: A193-B7 or B16 lubricated with a never seize type lubricant, k = 0.17								
	Class 900			Class 1500			Class 2500		
	Min. [1, 2]	Optimal [1, 3]	Max. [4, 5]	Min. [1, 2]	Optimal [1, 3]	Max. [4, 5]	Min. [1, 2]	Optimal [1, 3]	Max. [4, 5]
½”	30 (41)	55 (75)	85 (115)	30 (41)	55 (75)	85 (115)	35 (47)	60 (81)	85 (115)
¾”	40 (54)	80 (108)	120 (163)	45 (61)	85 (115)	120 (163)	50 (68)	90 (122)	120 (163)
1”	70 (95)	130 (176)	190 (258)	75 (102)	135 (183)	190 (258)	85 (115)	145 (197)	190 (258)
1 ¼”	110 (149)	210 (285)	315 (427)	120 (163)	220 (298)	315 (427)	155 (210)	270 (366)	360 (488)
1 ½”	160 (217)	305 (414)	450 (610)	175 (237)	315 (427)	450 (610)	225 (305)	385 (522)	505 (685)
2”	105 (142)	200 (271)	295 (400)	115 (156)	210 (285)	295 (400)	150 (203)	260 (353)	340 (461)
2 ½”	155 (210)	290 (393)	415 (563)	170 (230)	300 (407)	415 (563)	220 (298)	370 (502)	470 (637)
3”	165 (224)	295 (400)	415 (563)	260 (353)	445 (603)	585 (793)	345 (468)	555 (752)	655 (888)
4”	315 (427)	555 (752)	750 (1017)	425 (576)	715 (969)	915 (1241)	620 (841)	965 (1308)	1095 (1485)
5”	460 (624)	795 (1078)	1060 (1437)	695 (942)	1155 (1566)	1445 (1959)	1000 (1356)	1550 (2102)	1685 (2285)
6”	380 (515)	655 (888)	865 (1173)	575 (780)	935 (1268)	1145 (1552)	1565 (2122)	2355 (3193)	2505 (3396)
8”	630 (854)	1055 (1430)	1330 (1803)	975 (1322)	1550 (2102)	1830 (2481)	1530 (2074)	*2120 (2874)	2255 (3057)
10”	630 (854)	1010 (1369)	1210 (1641)	1575 (2135)	2470 (3349)	2770 (3756)	2690 (3647)	*3225 (4373)	3435 (4657)
12”	730 (900)	1190 (1613)	1460 (1979)	1665 (2257)	2510 (3403)	2655 (3600)	4180 (5667)	*5175 (7016)	5510 (7471)
14”	905 (1227)	1455 (1973)	1740 (2359)	2055 (2786)	*2665 (3613)	2815 (3817)	-	-	-
16”	1205 (1634)	1900 (2576)	2165 (2935)	3125 (4237)	*4480 (6074)	4730 (6413)	-	-	-
18”	1910 (2590)	3075 (4169)	3640 (4935)	4445 (6027)	*6230 (8447)	6670 (9043)	-	-	-
20”	2235 (3030)	3450 (4678)	3830 (5193)	5680 (7701)	*7600 (10304)	8025 (10880)	-	-	-
24”	3680 (4989)	*5425 (7354)	5730 (7769)	9180 (12446)	*11770 (15958)	12415 (16833)	-	-	-

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General Notes:

a) Torque Values are in ft.-lbs. and assume new A193 Gr. B7 or B16 fasteners with 2H heavy hex nuts; with studs, nuts and the nut bearing surfaces lubricated with a never-seize type lubricant (k = 0.17).

b) A193 Gr. B7 & B16 fasteners have the same yield strength up to 4” diameter. There are “no” fasteners above 4” diameter in this chart.

c) All torque values are based on using a “calibrated” torque wrench.

d) All torque values in the chart above are based on the use of a inner/outer ring style (DRI) spiral wound gasket.

e) All torque values in the chart above are based on using the tensile area of the fastener.

f) All torque values in chart are rounded to nearest 5 ft.-lbs.

Footnotes:

[1] Torque values are based using ASME B16.5-2017 MAWP (Maximum Allowable Working Pressure) at ambient in the gasket stress calculation.

Min. Torque Values:

[2] Min. torque values are based achieving 12,500 psi gasket stress or at minimum above 10,000 psi gasket stress without exceeding 80,000 psi bolt stress or PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding.

Optimal Torque Values:

[3] Optimal torque values are based on 25,000 psi optimum gasket stress without exceeding 80,000 psi bolt stress or 500 psi below PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding.

Cases where torque equals ≤ 60,000 bolts stress

Cases where torque equals > 60,000 ≤ 75,000 bolts stress

Cases where torque equals > 75,000 ≤ 80,000 bolts stress

*Cases where gasket stress is ≥ 15,000 < 25,000 psi, **Cases where gasket stress is ≥ 12,500 < 15,000 psi, ***Cases where gasket stress is ≥ 10,000 < 12,500 psi

Max. Torque Values:

[4] Max. torque values are based on; max allowable 40,000 psi gasket stress; PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding, or 80,000 psi bolt stress, whichever occurs 1st.

Note: In some cases the max. torque values may be equal to the optimal torque values in order to optimize gasket stress levels.

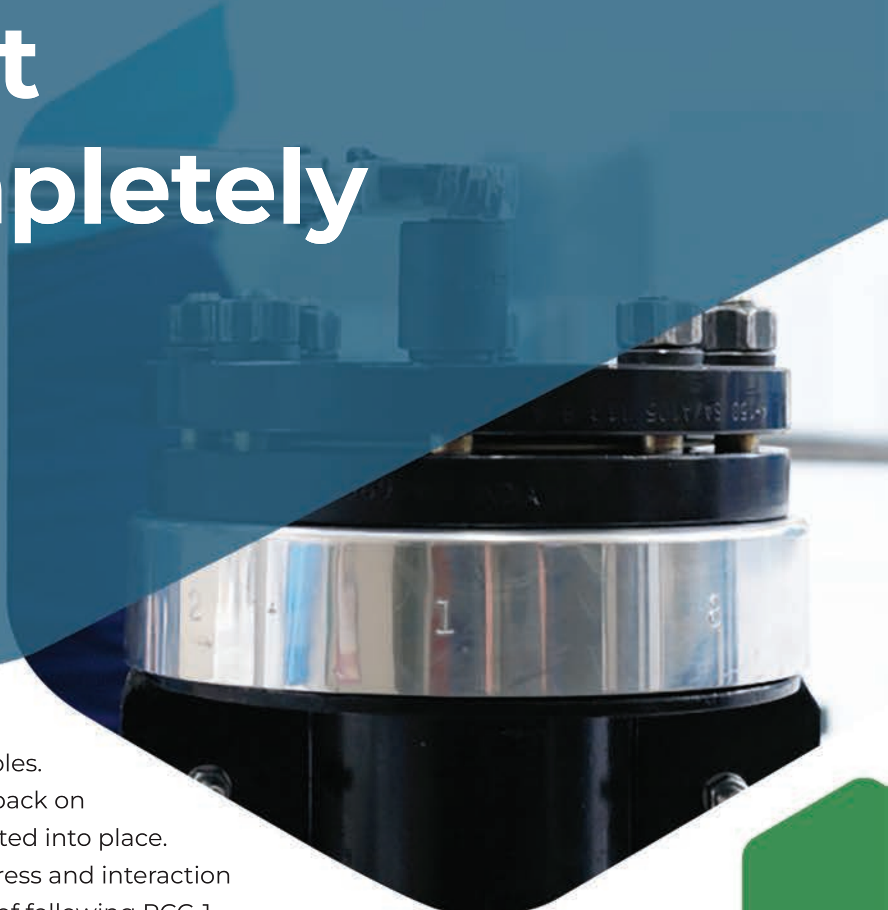
[5] ½” through 1 ½” NPS & 3 ½” NPS due to “No Data”on flange yielding; Max. torque values are set to achieve max gasket stress of 40,000 or 80,000 psi bolt stress, whichever occurs 1st.

Cases where torque is based on 80,000 bolts stress.

Cases where torque is based on PCC-1 2019 FEA SA105 Max. Stud Stress Before Flange Yielding.

Cases where torque is based on max. allowable 40,000 psi gasket stress.

Our Gasket Rig is Completely Portable



Durlon offers a bolt-up and gasket training utilizing a Flange Assembly Demonstration Unit (FADU) and based on ASME PCC-1 principles. This 90 minute training incorporates live feedback on the screen as the gasket is simultaneously bolted into place. Participants observe gasket deflection, bolt stress and interaction between bolts (crosstalk), and learn the value of following PCC-1 guidelines and get the opportunity to put their experience to the test.

WHO SHOULD ATTEND

- Pipe fitters
- Maintenance Personnel
- Gasket Fabricators
- Gasket Distributors
- Contractors
- Engineers
- Planners

AGENDA

- Scattered Bolt Load
- Bolt/Stud Tensile/yield strength
- Proper Lubrication
- Legacy / cross pattern bolt-up
- Use Of Washers
- Body Positioning
- Torque / Box Wrenches
- Proper arm movement

TRAINING BENEFITS

- Learn about bolted flange connections
- Increase gasket life
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- Become proactive to fugitive emissions and the environment
- Increase plant pipeline safety and reliability

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