The selection of gaskets has become more critical due to a number of factors:

SPECIFICATION OF THE INDUSTRIAL PROBLEM

- Pipes and joints are now included in the pressure vessel codes
- Tighter rules for emission control
- Aggressive effort to lower costs by reducing product loss and increasing margins of safety
- The international demand for standards for evaluating asbestos-free gaskets

OBJECTIVES

Bearing in mind rising standards of emission control and restrictions on the use of gaskets containing asbestos, **ISS**'s objective is a systematic and optimized development of a better sealing system for bolted gasket flanges.

GOALS

Our goal is simply to help our customers in understanding and solving, or preventing, field problems. Our search for something better led us to an impressive new gasket called the ***GRAFTEC***. This gasket is designed to provide superior performance over other gaskets.

EXPECTED ACHIEVEMENTS

Save money and increase margins of safety by:

- Better resistance against both chemical attack and high temperatures
- Reducing product loss through leakage
- Eliminating monitoring due to excessive fugitive emission levels
- Fewer industrial accidents caused by sudden gasket failure
- Preventing costs associated with production loss through plant shutdown and environmental clean-up costs.

SUMMARY

In this handbook we feature our new ***GRAFTEC*** gasket. However, the evaluation procedure that we have developed can be applied to other types of gaskets. This is why we have a large section on Gasket Design Factors and the Appendix.

As part of our continuous improvement process we are working to improve the performance of the ***GRAFTEC*** gasket, but also to help our customers with a better sealing system. To do this, we view a gasket as merely part of a total sealing system that includes flanges, bolts and assembly procedures. Together, with our suppliers and customers, we will help our customers achieve economic, social and enviromental benefits.

As you browse through this handbook, you will get a better sense of its organization and layout. Part I is a highlight of the unique Features and Benefits of the ***GRAFTEC*** gasket. The most outstanding benefit is the ability to seal with extraordinary tightness.

Part II gives test data and performance comparison for assistance in making a gasket selection. In the testing we found that the metal core in the ***GRAFTEC*** has a high resistance to flattening. Note the page "Gasket Crush".

Part III, "Gasket Design Factors," deals with gasket design factors and the new PVRC gasket constants. It covers "m" and "y" factors, ASME Code and PVRC methods.

Part IV deals with bolt loads and torque values. The page on "Uniform Bolt Loads" is particularly important for explaining the phenomenon known as "elastic interaction."

Part V, "Appendix," has a glossary that will be helpful in understanding the sealing technology of bolted joints.

EXCLUSIVE DESIGN FEATURES

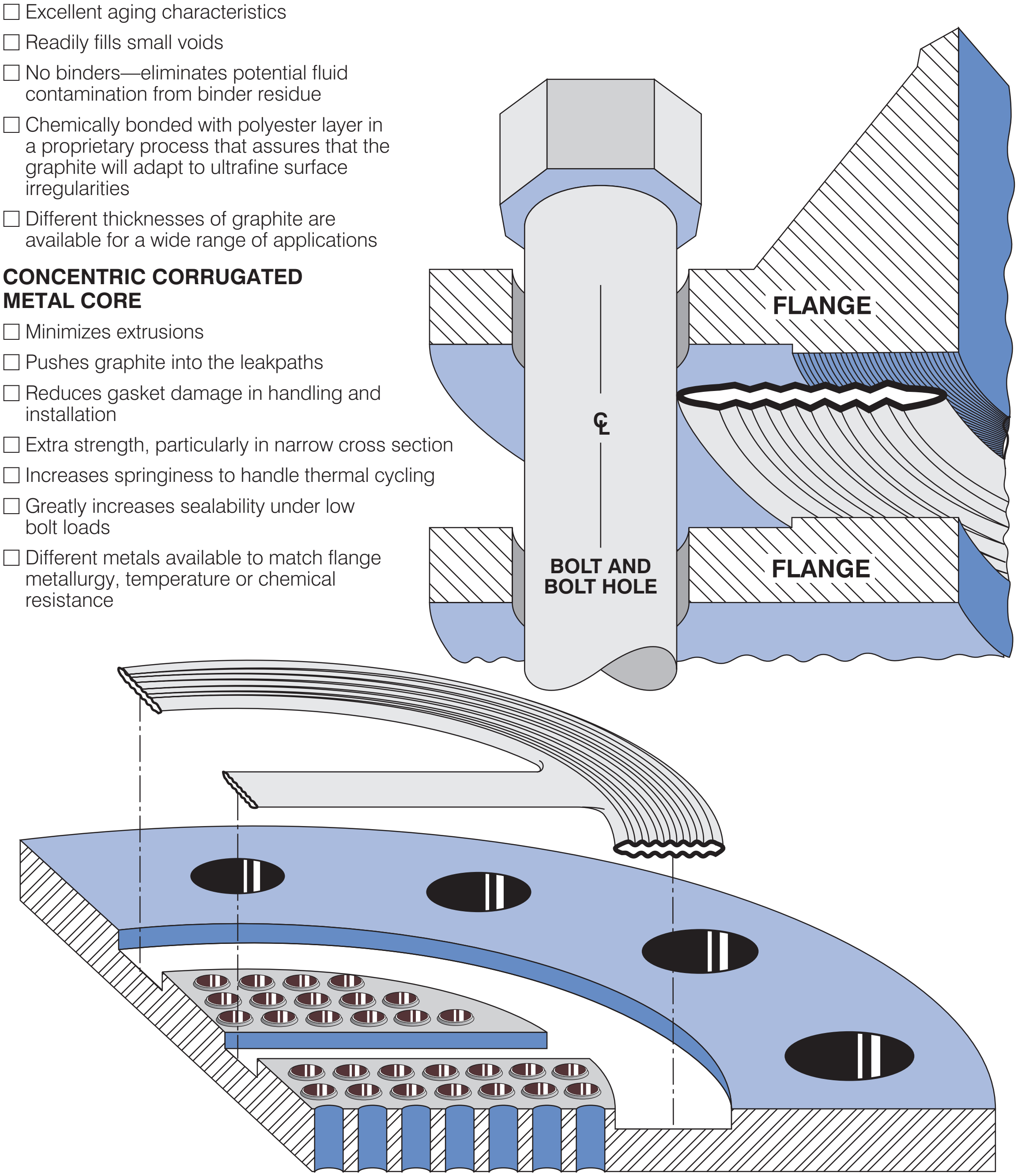
GRAFTEC Gasket Is Proving To Be The Solution For Controlling Emissions

ENCAPSULATED WITH FLEXIBLE GRAPHITE

- ☐ Graphite is resistant in high temperatures to 450°C (850°F) in hot air, 700°C (1300°F) in steam environment. In reducing environments which are almost totally oxygen exclusive, such as nitrogen or carbon dioxide, the threshold is about 900°C (1650°F). In the absence of oxygen, 3000°C (5432°C).
- ☐ Resistant to low temperature of -200°C (-328°F)
- ☐ Resistant to fluids with pH range of 0-14 (with only a few exceptions)
- ☐ Inherently resilient
- ☐ Rapid heat transfer
- ☐ Excellent aging characteristics
- ☐ Readily fills small voids
- ☐ No binders—eliminates potential fluid contamination from binder residue
- ☐ Chemically bonded with polyester layer in a proprietary process that assures that the graphite will adapt to ultrafine surface irregularities
- ☐ Different thicknesses of graphite are available for a wide range of applications

CONCENTRIC CORRUGATED METAL CORE

- ☐ Minimizes extrusions
- ☐ Pushes graphite into the leakpaths
- ☐ Reduces gasket damage in handling and installation
- ☐ Extra strength, particularly in narrow cross section
- ☐ Increases springiness to handle thermal cycling
- ☐ Greatly increases sealability under low bolt loads
- ☐ Different metals available to match flange metallurgy, temperature or chemical resistance



Piping Specifications for ***GRAFTEC*** Gaskets

1.0 General
This gasket may be substituted for compressed asbestos or spiral wound gaskets in raised face weld neck or slip-on ASME/ANSI B16.5 Class 150, 300 & 600 or other int. standards, carbon or alloy steel piping joints. Joints with these gaskets may be assembled with normal plant care.

2.0 Physical and Material Characteristics
Corrugated metal encapsulated with chemically bonded flexible graphite.
Nominal thickness: 1.6mm
Graphite layers: 2 layers, each flexible graphite nominal 0.5mm thick.
Metal inset: Nominal 0.6mm thick austinetic stainless steel with corrugations. Other types of metal are available.
Gasket dimensions: per ASME/ANSI standard B 16.5, group 1 (same as ANSI B 16.21)
other international standards or specials available

2.1 Joint Assembly Characteristics
Gasket shall achieve tightness and be suitable for joint assembly using the same manual procedures, skills, care and tools as is appropriate for compressed asbestos sheet gaskets. Bolt load control such as torque settings is optional. Torque wrench not required.

3.0 Fire Resistance: Simulated Tightness Test (FITT)
Intermech Sealing Solutions (Pty) Ltd , the manufacturer, certifies the results of two tests, conducted in accordance with the Pressure Vessel Research Council FITT procedure 1.3 (Ref. 1) on NPS 4 test gaskets exposed to a 20 minute heat-up plus 15 minute soak @ 350℃, and the following:

Required post exposure Minimum Tightness: Tpmin > 32 (Helium)
Typical post exposure Minimum Tightness: Tpmin = 2000 (Helium)

4.0 Room Temperature Tightness (Helium Media ROTT)
Intermech Sealing Solutions (Pty) Ltd, the manufacturer, certifies the results of two or more tests on 123.83mm ID by 149.23mm OD gaskets, conducted in accordance with the Pressure Vessel Research Council developed Room Temperature Tightness Test (ROTT) [Ref. 2] procedure and the following:

Requirement	Gb	a	Gs	S100	S1,000 (1.2)
Maximum:	1,300	0.273	9	3,180	5,630
Typical:	922	0.248	5	2,889	5,114

Notes:
(1) The constants Gb, a and Gs shall be determined by the ROTT test procedure of Ref [2].
(2) S100 and S1000 are stresses (psi) respectively representing the values $Gb(100)^a$ and $Gb(1000)^a$.

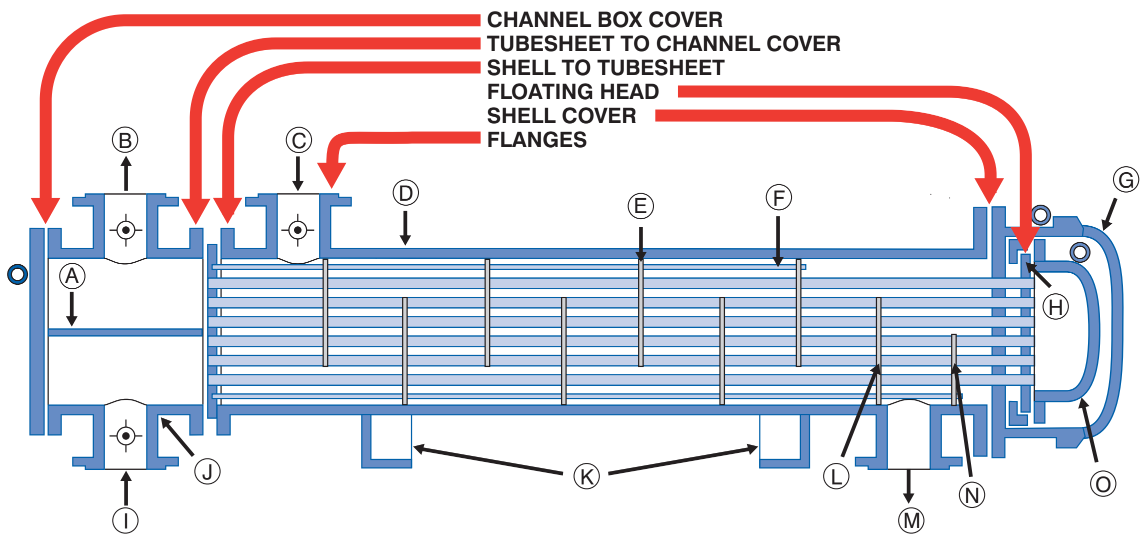
Crush Qualification Test
Intermech certifies the results of a crush test on the ***GRAFTEC*** gasket that indicates the tightness parameter, Tp: and the gasket deflection, Dg (in.) with a gasket stress, Sg (psi) up to 40,000 psi and internal pressure of 800 psi (Helium). The test demonstrates adequate resistance to crushing by excessive bolt load.

REFERENCES
(1) Derenne, M., Payne, J.R., Marchand, L., and Bazerqui, A., “On The Fire Resistance of Gasketed Joints”; WRC Bulletin No. 377, Dec. 1992
(2) Draft No. 8 Standard Test Method for Gasket Constants for Bolted Joint Design”; ASTM Committee F 3, Payne, J., April 1991; (Not approved by ASTM).

GRAFTEC Gaskets are leading the way in sealing Heat Exchangers

- ***GRAFTEC*** gaskets are primarily designed for TEMA shell and tube heat exchanger vessels.
- Maintain a tight seal in a wide range of bolt loads
- Ensure a tight seal under fluctuating temperature and pressure conditions.
- Can be supplied with partition bars in any configuration

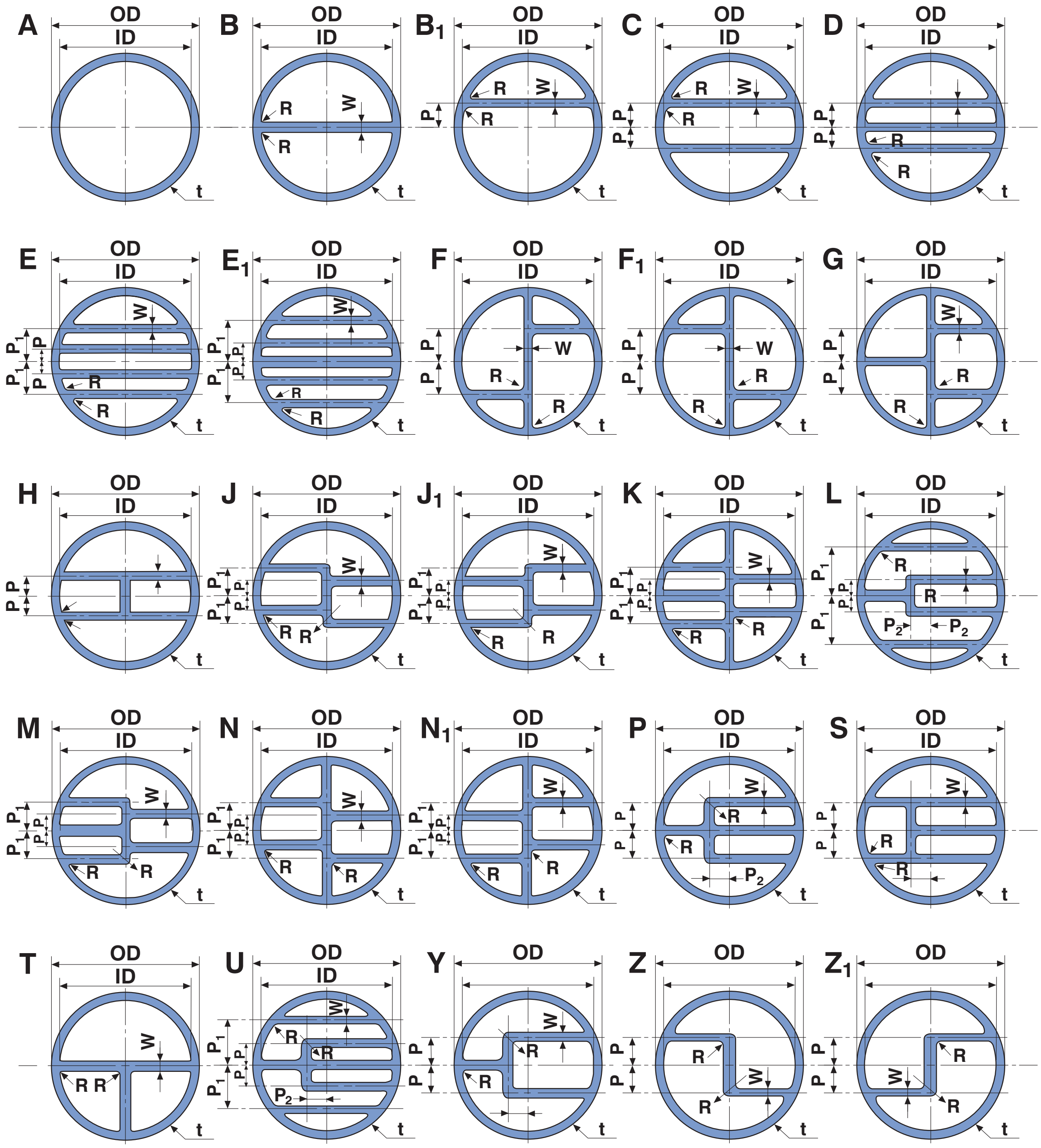
Gasket Locations



Ⓐ Pass partition	Ⓕ Tie rods and spacers	Ⓚ Support saddles
Ⓑ Tubeside fluid out	Ⓖ Shell cover	Ⓛ Last baffle
Ⓒ Shell inlet	Ⓗ Floating tubesheet	Ⓜ Shell outlet
Ⓓ Shell	Ⓘ Tubeside fluid in	Ⓝ Floating-head support plate
Ⓔ Baffles	Ⓙ Stationary-head channel	Ⓞ Floating-head cover

Shapes

Heat exchanger gaskets have complicated partition bar(s). The typical shapes of the gasket are as illustrated below.



The metal core of the ***GRAFTEC*** gasket can be selected from most types of sheet metal. The selection is generally based on chemical resistance, heat resistance and cost. The popular metals for the ***GRAFTEC*** are as follows:

Mild Steel CR210 Stainless 304
Stainless 316

Stainless 904, Monel and other approved exotic metallurgies can be used, subject to availability.

The selection of a metal to be used in a gasket that is suitably resistant to corrosive media or to high temperature can involve many considerations. No compilation of data in a table or in a few pages could possibly consider all these variables. Intermech recommends that designers contact the manufacturers of alloyed material, who conduct laboratory corrosive tests and in-plant corrosion testing.

The following comments may be helpful.

1. Concentration of Corrosive Agents: Dilute solutions are not necessarily less corrosive than those of full strength, and the reverse is often the case. Probably the most familiar example of this is the action of sulfuric acid on iron; concentrations over 90 percent acid may be handled by iron without much difficulty, but below this concentration the rate of attack will increase rapidly with an increase in dilution.

2. Purity of Corrosive Agent: By purity, in this instance, is meant not concentration, but the absence of contaminating amounts of other corrosive compounds. For example, the corrosive attack by compounds that are derivatives of an acid; in the pure state these compounds may be relatively inert, but if contaminated by any carry-over of free acid they must be handled with this in mind.

3. Temperature: Besides its effects upon the mechanical properties of the gasket, the temperature of the corrosive agent will have a marked influence upon the rate of attack.

OUTLINE FOR REVIEWING CORROSION

FORMS OF CORROSION

1. General Corrosion
2. Galvanic Corrosion
3. Concentration-Cell or Crevice Corrosion
4. Chemical Pitting
5. Intergranular Corrosion
6. Effects of Stress on Corrosion
   (a) Corrosion Fatigue
   (b) Stress-Corrosion Cracking

CORROSIVE ENVIRONMENTS

A. Atmospheric Corrosion
B. Corrosion by Water, Acids,
C. Alkalies, Salts, Fluorine,
D. Chlorines and Hydrogen
E. Chlorides

SUMMARY: Room Temperature Tightness tests (ROTT) were performed on ***GRAFTEC*** gasket specimens at the Ecole Polytechnique Gasket Test Facility.

Gb = 922 psi
a = 0.248
Gs = 5.1 psi

The tests show excellent tightness. On a range of loading and unloading stress levels, the ***GRAFTEC*** are the tightest flexible graphite gaskets tested to date. By comparison to laminated graphite sheet, the initial leak rate of the ***GRAFTEC*** gasket averaged about l00 times less at an initial gasket stress (Sg) of 8,000 psi.

Constants, Gb, a, and Gs: These are the constants used in formulas that give a design bolt load having the same meanings as the larger of Wm1 or Wm2 of the ASME Code. Gb, a and Gs are obtained by interpretation of leakage test data as plots of gasket stress (Sg) vs the tightness parameter, Tp on log-log paper. The values of Gb and Gs are determined by the intercepts of the loading and unloading lines with the Tp =1.

Gb, a: These tell us what the gasket seating load should be because Gb and a are associated with the seating load sequence (Part A data) of a gasket test. Gb represents the loading of the gasket (Intercept of the loading curve on the gasket stress Axis) at Tp = 1. The slope of the line is represented by **a**.

Gs: Gs is associated with the operating part of a gasket test, known as Part B, where the gasket is unloaded and reloaded as leakage is measured. Gs = Unloading intercept (intercept of the unloading curve on the gasket stress axis) at Tp = 1.

Tightness Parameter, Tp: The investigators discovered that test data could be summarized by use of a dimensionless tightness parameter. It is represented by Tp, expressed in terms of mass leak rate.

Tp is the pressure (in atmospheres) required to cause a helium leak of 1 mg/sec for a 150 mm (5.9 in.) OD gasket in a joint. A tightness parameter of 100 would mean that it takes an internal pressure of 1,470 psi (10.1 MPa) to create a total leak rate of about 1 mg/sec from a 6” OD gasket (152 mm) gasket. A 100 times less leak rate of 0.01 mg/sec at 1,470 psi would mean a tenfold increase in the tightness parameter to 1,000 Tp. Tp is proportional to pressure and inversely proportional to the square root of the leak rate. A higher value of Tp indicates a tighter joint.

Gb(Tp) ^ a: The value of the expression Gb(Tp) ^ a compares seating properties among gaskets when comparisons are made at representative values of Tp, such as 100 and 1,000. Such comparisons show the combined effect of Gb and a on the seating performance of a gasket. The new gasket constants will eventually replace the present ASME Code m and y factors. The new constants, Gb, a and Gs help define the behavior of the gasket under all possible stress conditions. The only design guidance emerging from this work is the concept of “tightness” levels. Once the designer has learned how to convert Gb, a Gs and selected tightness level to specific stress targets, he can design “better” flanges. The installer of gaskets will find the new gasket constants useful, since they genuinely define gasket behavior.

You can visualize the new PVRC gasket constants using an XY graph. Basically the constants work in a correlation between one set of values, Gasket Stress on a second set of values, the measured leak rate (Tp). For example ,you would plot the ***GRAFTEC,*** Gb , as 922 psi on the Gasket Stress axis and 1 on the Tp axis. As you increase the gasket stress the Tp increases. This leak rate decrease is determined by the slope which in the case of the ***GRAFTEC*** is 0.248. For example, when the Sg increases to 10,000 psi, the Tp increases to 14,950 and the Tp is now referred to as Tpa representing the tightness achieved at assembly.
The formula is:
Tpa = (Sg/Gb) ^ (1/a) Sg = Gb(Tpa) ^ a

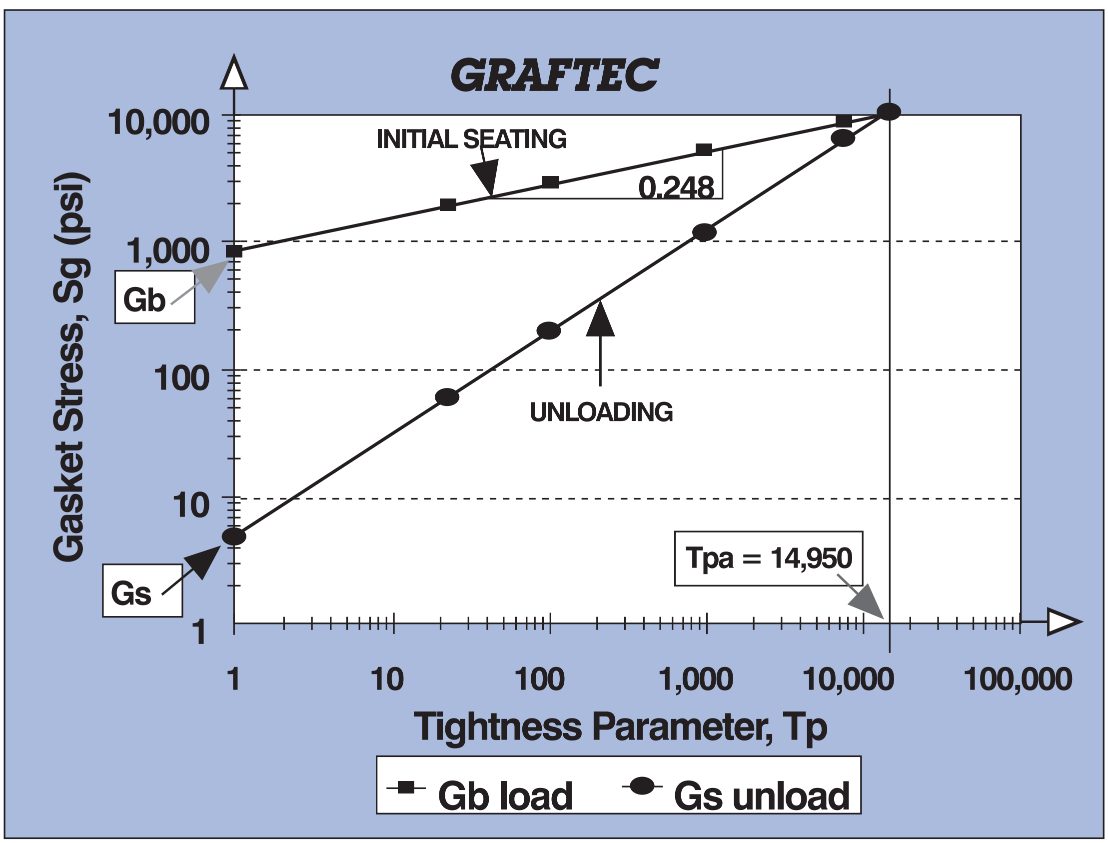
After seating, the joint is pressurized and the gasket unloads, which results in less gasket stress. This is represented by the Gs constant. It is represented on the graph as Gs = 5.1 psi on the Gasket Stress axis and 1 on the Tp axis.

We can draw a line from the high point of the load line (10,000,14,950) down to (5.1,1). This represents the unloading of the gasket as the gasket stress is decreased.

How do you determine the Tpmin when the gasket stress is reduced? You can calculate the unload line (Gs) by the following formula:
Slope,(Gs)unload line, psi (MPa) = Log(Sgmax/Sgmin)//Log(Tpa/Tpmin)
With no higher values you can use Gs for Sgmin and 1 for Tpmin.

Tpmin = (Sg/Gs) ^ (1/slope)
Sg = Gs x Tpmin ^ (slope)

Estimated Leak rate (Lr): You must convert inches to millimeters, then to a reference mass leak rate which is keyed to a normalized reference gasket of 150 mm outside diameter.
Lr = Gasket OD (in.) x 25.4mm/150mm x (Internal Pressure/Atmospheric Pressure (14.7) x 1/Tpmin) ^ 2 = mg/sec.
To convert to lb/hr, multiply mg/sec by 0.008.



GASKET CONSTANTS FOR VARIOUS GASKETS

The ***GRAFTEC*** Gasket Constants compare favorably against those of other gaskets. Low values of gasket constants (Gb), (a), (Gs) are good. The value of the expression Gb(Tp) ^ a compares compares seating properties among gaskets when comparisons are made at representative values of Tp, (measure of tightness). Such comparisons show the combined effect of (Gb) and (a) on the seating performance of a gasket. Table XX-6.2 compares the value of Gb(Tp) ^ a, which indicates the seating stress required to meet a Tp (measure of tightness) for various gaskets.

TABLE XX-6.2 (Preliminary)

Gb(Tp) ^ a = Sg, (psi) Sg = Gasket Stress						
Tp (100) Sg (psi)	**Tp (1,000) Sg (psi)**	**TYPE**	**MATERIAL**	**Gb (psi)**	**a**	**Gs (psi)**
2,889	5,114	***GRAFTEC***	SS/Graphite	922	0.248	5.1
6,851	11,823	Spiral Wound:	SS/Graphite	2,300	0.24	13
8,575	11,836	Spiral Wound:	SS/PTFE	4,500	0.14	70
7,498	12,734	Spiral Wound:	SS/Mica	2,600	0.23	15
13,536	27,007	Spiral Wound:	SS/Asbestos	3,400	0.30	7
3,615	8,875	Flexitallic “LS”	SS/Graphite	600	0.39	2
8,364	14,204	Metal Jacketed	Soft Iron	2,900	0.23	15
8,364	14,204	Metal Jacketed	Stainless Steel	2,900	0.23	15
9,021	20,196	Metal Jacketed	Soft Copper	1,800	0.35	15
		Laminated Graphite				
6,225	13,126	with Stainless:	Tanged	1,400	0.33	0.01
4,631	11,033	with Stainless:	Bonded	816	0.38	0.07
5,629	10,244	with Stainless:	Screen	1,700	0.26	15
5,686	13,765	Flexible Graphite	Unreinforced	970	0.38	0.05
		Compressed Elastomers reinforced with:				
4,988	7,046	1/16 in. thick	Asbestos fibers	2,500	0.15	117
4,978	8,105	3/32 in. thick	Aramid fibers	1,900	0.21	14
		Expanded PTFE Filled PTFE	For data on PTFE based gaskets, contact your **INTERMECH** representative.			

All data presented in this table is based on current published information from the Pressure Vessel Research Council (PVRC) project for the ASME Special Working Group for Bolted Flanged Joints. The PVRC continues to refine data techniques and values are subject to further changes.

GRAPH

After seating the gasket with an initial load, unloading occurs in the bolted joint when the system is pressurized, and with subsequent unloading due to gasket relaxation and thermal effects. Using the ROTT test procedure, the gaskets are stressed to 10,000 psi, Sya. In these calculations, the gaskets are unloaded to 1,000 psi. Comparing the various gaskets shows the ***GRAFTEC*** to be tighter in the unload cycle. This offers a higher margin of safety before a major leak. This is a tool for screening gaskets for applications that demand superior performance with lower leak rates and a margin of safety.

HOW TO MAKE THE CALCULATIONS

First step - obtain the gasket constants (Gb,a,Gs) from the manufacturer or from a table.

Type	Gb (psi)	a	Gs (psi)
GRAFTEC	922	0.248	5.1
Spiral Wound Graphite	2,300	0.24	13
Flexitallic "LS" Graphite	600	0.39	2
Laminated Graphite/Tang	1,400	0.33	0.01

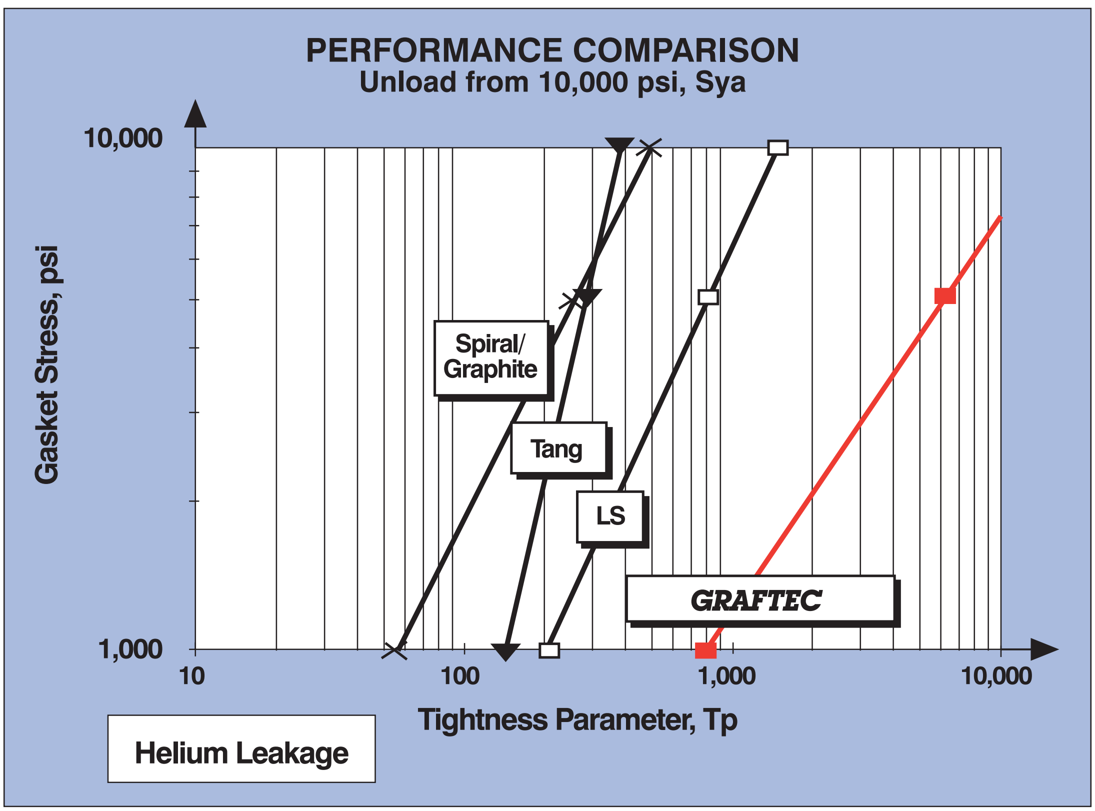
Second step - determine the Tp at the initial gasket seating of 10,000 psi = Sgmax or Sya.

Tpmax = (Sgmax/Gb) ^ (1/a)

Third step - determine the slope of the unloading from the inital seating.

Slope of the unload line = @log(Sgmax/Gs)/@log(Tpmax/1)
Note that Gs constant is at Tp = 1.

Compute, Tpmin at 1,000 psi (Sgmin).
Tpmin = (Sgmin/Gs) ^ (1/slope of the unload line)



GASKET STRESS vs MASS LEAKAGE

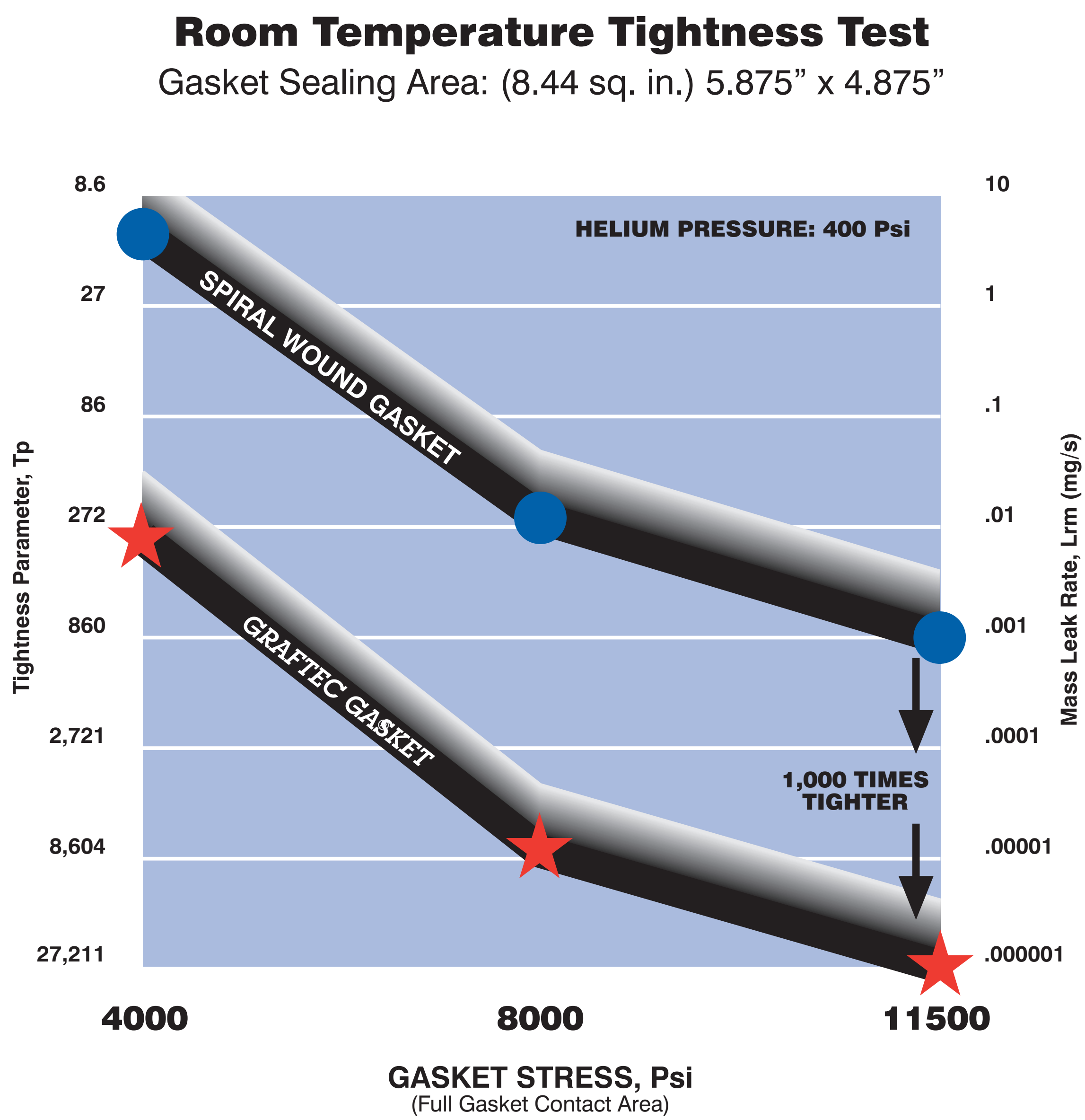


Figure 2

LOW INITIAL SEALING. At the initial assembly stage, the ***GRAFTEC*** gasket has much lower leak rate (1,000 times) under the same load conditions than the graphite filled spiral wound gasket. This advantage indicates a much more forgiving gasket in a wide range of initial bolt loading. This can translate into a higher margin safety from potential "leakers". We do not suggest that anybody use the ***GRAFTEC*** gasket solely for this advantage, rather, it gives you another reason for seriously considering the ***GRAFTEC*** gasket.

TEST METHOD: PVRC (Pressure Vessel Research Council) ROTT test method was applied in accord with the Draft ASTM Standard No. 7, April 1990.

GRAFTEC GASKETS ARE THE WINNER IN LEAK TIGHTNESS TESTS OF GRAPHITE BASED GASKETS

Tests confirm the ability of the ***GRAFTEC*** gaskets to seal at a much lower bolt load over other leading graphite type gaskets – at less than half the bolt load or gasket stress. ***GRAFTEC*** is the best gasket to get a tight seal without over-stressing the flange assembly.

MAKE *GRAFTEC* GASKETS PART OF YOUR FAILURE-AVOIDANCE STRATEGY.
GRAFTEC gaskets maintain tight leakage control with minimal force. This means you do not have to worry about exceeding allowable flange and bolt design stresses. Many metal type gaskets require bolt loads for seating more than allowable design stresses – Class 150 flanges are particularly vulnerable. Overloaded bolts can lead to (1) fractures, (2) deformation of the flange, (3) bolt stretching, or (4) decreased recovery ability of the gasket. All these things can lead to catastrophic failure. To eliminate this risk, make ***GRAFTEC*** part of your failure-avoidance strategy.

SUPERIOR SEALING CHARACTERISTICS*
On initial loading to stress levels over 11,600 psi, it was difficult to detect leak rate at 800 psi internal pressure with a detection system that is capable of resolving 1/100,000 milligrams per second of helium. This level of tightness is rarely seen in any gasket.

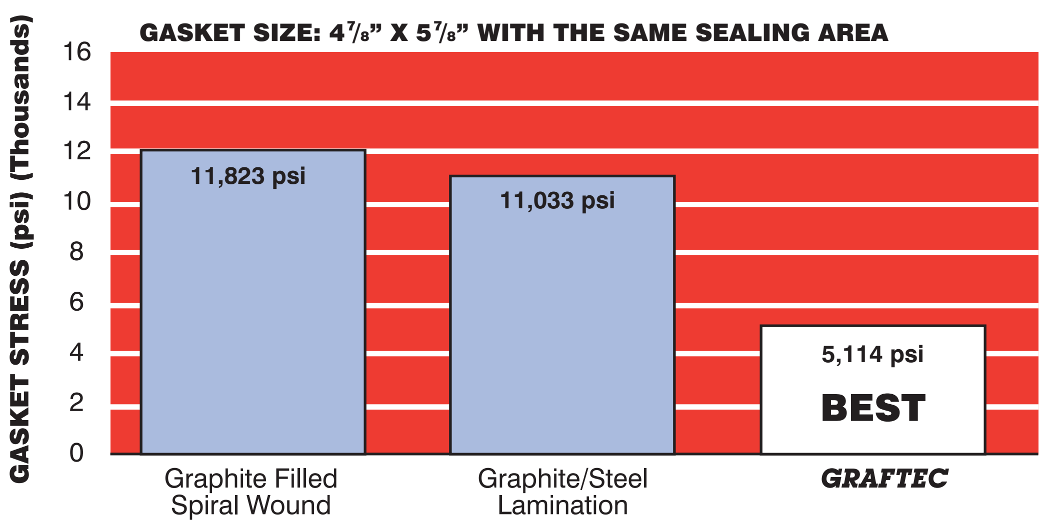
LOW (TIGHT) LEAK RATE IN THESE TESTS
Note that in volumetric terms the allowable leak rate T3 (tight) is approximately 0.45 liter/day (0.84 pint/day) of nitrogen gas at standard conditions for a 10 inch NPS joint.

ALLOWABLE LEAK RATE T3 (Tight) VS. GASKET STRESS

Lower Gasket Stress Is The Best

T3 (Tight) represents a Mass Leak Rate Per Unit Diameter (L_{RM}) of (1/50,000) 0.00002 mg/sec-mm* OR (1/248,000) 0.000004 lbm/hr-in.**

*Milligrams per second per millimeter of gasket outside diameter. **Pounds per hour per inch of gasket outside diameter.



The tests were performed at the École Polytechnique Gasket Test Facility Department of Mechanical Engineering, University of Montreal.

The test procedure is the (PVRC) Pressure Vessel Research Council Temperature Tightness Tests (ROTT) in accord with the current Draft (No. 7 of April 1990) ASTM Standard Test Method for Gasket Constants for Bolted Joint Design.

SUMMARY: A crush test was performed at room temperature on an ***GRAFTEC*** gasket specimen at the École Polytechnique Gasket Test Facility. The elastic recovery upon final unloading from the maximum stress of 40,000 psi shows that the ***GRAFTEC*** gasket sustained very well the imposed high loading. Tightness kept improving as the gasket was compressed to higher loads. Leakage resistance to unloading was good and not affected by the imposed high stresses. It appears very difficult to crush the ***GRAFTEC*** (at room temperature)

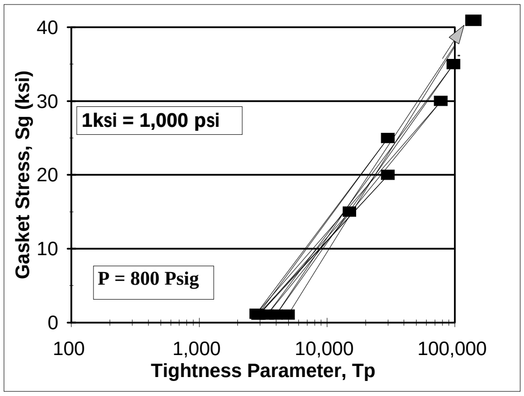


FIGURE 1: The tightness increased to a value of 124,000 (Tp) as the specimen was compressed to increasingly higher loads. The leakage resistance to unloading was good even when the gasket was crushed to the higher loads.

TEST PROCEDURE: The crush test procedure consists in cycling the gasket from a minimum load of 1,025 psi up to the required maximum load of 40,000 psi by increments of 5,000 psi. Gasket deflection and leakage (with helium at 800 psig) are measured at every step. The details of the crush test procedure are as follows:

I -The gasket specimen is initially loaded to a stress level of 15,000 psi. Gasket deflection measurements are taken at intermediate stress levels. A first leakage measurement is taken at the 15,000 psi stress level.

2 - The gasket specimen is unloaded to a stress level of 1,025 psi. The compressive stress is then increased to the next stress level incremented by 5,000 psi. The cycle is repeated up to the maximum gasket stress of 40,000 psi.

SPECIMEN *: GRAFTEC* with 316 stainless corrugated core that is encapsulated by a 0.020 in. thick layer of flexible graphite on each side. The gasket contact surface is 8.44 ^ 2 on a 5-7/8" in. OD x 4-7/8" in. ID. specimen.

TEST RESULTS AND ANALYSIS: Figure 1 is a plot of gasket stress, Sg, versus tightness, Tp, on log-log scales. The Tightness Parameter, Tp, represents the inverse of leakage and may be thought of as the number of atmospheres of pressure needed to cause a leak of 1 mg/sec of helium. Thus, a high Tp is good. Figure 2 is a plot of gasket stress, Sg, versus gasket deflection, Dg.

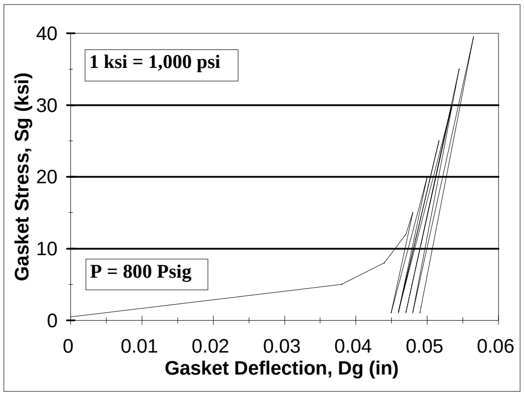


FIGURE 2: The ***GRAFTEC*** gasket had good deflection recovery in each one of the unloadings. The unload-reload lines are almost parallel, which means that the mechanical behavior of the gasket was not affected by the imposed high loads. The elastic deflection recovery upon final unloading from the maximum compressive stress of 40,000 psi to the 1,025 psi stress level is of approximately 3.7 mills.

Fire Integrity Certification of ***GRAFTEC*** Gaskets

To determine the ability of the the ***GRAFTEC*** gasket to maintain tightness in a fire, two tests were conducted at École Polytechnique, Department of Mechanical Engineering, University of Montreal, Canada, Gasket Testing Facility. The test procedure was the FITT test (Fire Tightness Test) which gives a good indication of the fire survival potential of a gasket in a real fire. It measures leakage at realistic loads while rapidly heating and soaking a gasket at 1,200° F for 15 minutes.

GRAPH BELOW

At 1,500 psi Gasket Stress, the triangles represent the tightness behavior during the test at 1,200 °F. The arrow above the triangles indicate that during the monitoring, the Tightness Parameter, Tp, values increased nearly 20 fold to about Tp of 2,800. This means that the leak rate decreased over 300 fold. For comparing performance, The Tp value of 32 represents the average performance of a well aged compressed asbestos sheet material. From this test, it was concluded that the ***GRAFTEC*** has fire integrity.

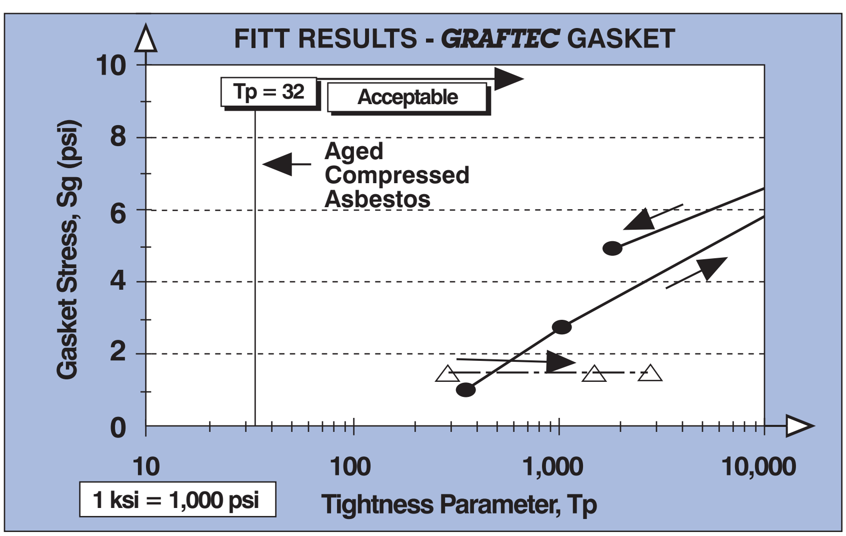
EXCEEDS THE PERFORMANCE OF FLEXIBLE GRAPHITE LAMINATE SHEET

The post-exposure tightness of the ***GRAFTEC*** specimens exceeded that of flexible graphite laminate sheet and equaled graphite filled spiral wound gaskets.

WARNING: A word of caution is needed to point out that no matter how fire-safe the gasket, the bolted joint containing that gasket may open up under certain conditions of flame or fire-water impingement during a real fire. In a fire, the bolts get sloppy and stretch, separating the flange faces. But the ***GRAFTEC*** does not burn up. It stays in place, helping control the release. The result is valuable extra time to control the fire. Tests indicate that the ***GRAFTEC*** has the ability to regain tightness when cooled.

TEST PROCEDURE

1. A 5-7/8 in. OD x 4-7/8 in. ID ***GRAFTEC*** gasket was installed in the test rig within a heavy platen assembly.
2. The gasket was compressed to 4 levels from 1,025 psi up to 8,000 psi at room temperature and the leakage was measured. The load was reduced to 5,000 psi then the pressure and load were removed.
3. The hot loading platen was heated to 800°F and the gasket assembly was introduced.
4. Gasket stress of 1,500 psi was applied.
5. Applied 400 psi internal pressure with helium.
6. Maintain pressure until temperature stabilizes at 1,200 °F.
7. Hold temperature and pressure for 15 minutes while the leakage rate was monitored.



Since mass leak rates are difficult to visualize, we have calculated Nitrogen and Helium gases, with equivalents in terms of volumetric leak rates with a 12.75" (324mm) OD sealing contact.

MASS LEAK RATES:

Tc 3 Tight	1/50,000	mg/s per mm	1/248,000 lb/hr	per in
Tc Standard	1/500	mg/s per mm	1/2,480 lb/hr	per in
NPS joint 10" with a face OD of	324	millimeter	12.75	inches

mg = milligram   mm = millimeter   s = second   in = inches

The mass leak rate is calculated on a per millimeter or inch basis of the outside diameter (OD) of the sealing contact surface.

NITROGEN weight(1)	**1.251**	**gram/liter**	**0.075261**	**lb/cu ft**
Tc 3(Tight), Allowable Leak Rate	0.00002	mg/sec.mm	0.000004	lb/hour/in
Leak Rate of the NPS joint 10" (2)	0.006477	mg/s	0.000051	lb/hour
	560	mg/day	0.001234	lb/day
	0.560	gram/day	0.016	cu ft /day
Volumetric Leak Rate (3)	**0.45**	liter/day	**0.84**	pints/day
Tc 2(Standard), Allowable Leak Rate	0.00200	mg/sec.mm	0.000403	lb/hour/in
Leak Rate of the NPS joint 10"	0.647700	mg/s	0.005141	lb/hour
	55,961	mg/day	0.123387	lb/day
	55.961	gram/day	1.639	cu ft /day
Volumetric Leak Rate	**45**	liter/day	**84**	pints/day

HELIUM, weight	**0.179**	**gram/liter**	**0.011143**	**lb/cu ft**
Tc 3(Tight), Allowable Leak Rate	0.00002	mg/sec.mm	0.000004	lb/hour/in
Leak Rate of the NPS joint 10"	0.006477	mg/s	0.000051	lb/hour
	560	mg/day	0.001234	lb/day
	0.560	gram/day	0.111	cu ft /day
Volumetric Leak Rate	**3**	liter/day	**6**	pints/day
Tc 2(Standard), Allowable Leak Rate	0.00200	mg/sec.mm	0.000403	lb/hour/in
Leak Rate of the NPS joint 10"	0.647700	mg/s	0.005141	lb/hour
	55,961	mg/day	0.123387	lb/day
	55.96	gram/day	11.07	cu ft /day
Volumetric Leak Rate	**314**	liter/day	**569**	pints/day

Note: 51.4281 pints (U.S. dry) in a cu ft.   cu ft = cubic feet

(1) Weights assume a dry gas at 0° C (32° F) and 760 mm Hg (14.70 pounds/sq inch).
(2) Leak rate of the gasket is calculated by multiplying the leak rate per mm (inch) by the smaller of the flange face or gasket OD.
(3) To calculate Volumetric Leak Rate:
Divide the leakage in gram/day by the weight of the gas (gram/liter) for liter/day.
Multiply the leakage in cu ft/day X pints in a cu ft (51.4281) for pints/day

ASSEMBLY GASKET STRESS WORKSHEET

DEFINITION OF GASKET STRESS

Gasket stress is the contact pressure between the flange and gasket bearing surface. The definition of stress is the magnitude of the applied force applied to the area of the gasket on which the force acts. In a flange it is created by the applied force from the tension in the bolts clamping the flanges.

GASKET CONTACT AREA

SYMBOLS AND UNITS

Go = The smaller of Gasket OD or flange sealing surface OD [inch]
OD = Outside diameter of sealing surface, gasket or flange face [inch]
ID = Inside diameter of sealing surface, gasket [inch]
N = Width of full gasket contact sealing used to determine the basic gasket seating width [inch]
Ag = Full gasket contact area based on the contact width [in ^ 2]

For initial seating, use the full contact area of the gasket. When the joint is pressurized, the PVRC introduces an effective (roughly half) width [n] that is the same as [b] in the ASME Code, Section VIII Table 2-5.2 to allow for flange rotation.

1. Compute the full surface sealing area of the gasket, Ag [in ^ 2]

Ag = 0.7854*(OD ^ 2-ID ^ 2) OR
3.14*(Go-N)N Ag = __________in ^ 2

HOW TO FIND GASKET STRESS AT ASSEMBLY (Sya)

SYMBOLS AND UNITS

Sg = The stress on the sealing area of the gasket [psi]
Sya = The PVRC uses this symbol for the design assembly seating stress or joint contact unit seating load. [psi]
Fp = Bolt preload in each bolt at assembly [lbs]
FGA = Total nominal clamping force on the gasket at assembly [lbs]
Sa = bolt stress at ambient temperature [psi]
K = Nut factor [dimensionless]
D = nominal diameter of bolt [in]
12 = Divide Torque by 12 to convert from ft-in to ft-lb
C = 0.0833; conversion Factor, Torque (ft-in to ft-lbs)
Ar = Root cross-section area of a bolt [in ^ 2]
Ab = Ar* number of bolts, (n)

BOLT LOAD

1. Compute, the nominal bolt preload in each bolt at assembly (lbs.). For example, if preload is specified by torque (T; ft./lbs.) then

Fp = 12*T/*K*D _________lbs

2. Compute, if preload is specified by the actual total cross-sectional area of bolts at root of thread or section of least diameter under stress, (Ar), then multiply Ar by the bolt stress, Sa.

Fp = Ar*Sa = __________lbs

You can convert the final nominal preload to nominal Torque:

T = K*D*Fp/12 = __________ Ft/lbs

3. Compute the total, nominal clamping force on the gasket at assembly (FGA; lbs.), n = number of bolts. Ab = Ar*number of bolts (n)

FGA = n*Fp = _____________lbs
OR
FGA = Ab*Sa = _____________lbs

GASKET STRESS

4. Compute the initial gasket stress(Sya)

Sya = FGA/Ag = __________ psi

After the assembly, you then calculate the pressure load on the joint, estimate how much the pressure load will partially relieve the joint and compute the net clamping force on the joint after the system has been pressurized. For reference, go to the pages titled PVRC METHOD.

THE TRADITIONAL ASME CODE METHOD

The ASME Boiler and Pressure Vessel Code, Section VIII, Div. 1 Appendix 2 provides mandatory design “Rules for Bolted Flanged Connections With Ring Type Gaskets” where the gasket is within the circle defined by the bolt holes. Appendix 2 also suggests (but does not require) the gasket factor “m” and a minimum gasket seating stress, “y”. These help establish a design bolt load for the joint which, in turn, is used to verify that the flange geometry is satisfactory for the governing design conditions. The design bolt load for the joint is calculated for operating and seating requirements from m and y and as follows:

OPERATING CONDITIONS

1) Determine the minimum bolt load required for operating conditions, Wm1:

$$Wm1 = P(Ai) + (Sg)Ag$$

This equation means that the bolts must be designed for the sum of the pressure load (also called the hydrostatic end force), as represented by P(Ai) and a gasket load sufficient to maintain a seal. P is the design pressure, psi. The Code uses:

$Ai = 0.785G^2$ as the area against which the pressure acts and

$Ag = 2b(3.14)G$ as the gasket area over which the minimum stress

$Sg = m(P)$ must be maintained. G is the gasket OD less twice the effective width of the gasket which is defined by the Code as b.

$b=b_0$ when $b_0 \leqq 1/4"$ $b = .5\sqrt{b_0}$ when $b_0 > 1/4"$

SEATING THE GASKET

2) Determine the minimum bolt load required to seat the gasket, Wm^2:

$$Wm2 = (Ag)y/2 = b(3.14)Gy$$

This equation means that the bolts must also be designed to exert a stress on the gasket that is sufficient to seat it.

3) Select the largest of Wm1 or Wm2 to determine the minimum required bolt area, Am, as:

Am = the greater of Am1 or Am2 where

$Am1 = Wm1/Sb$ and $Am2 = Wm2/Sa$

and Sb and Sa are respectively the allowable operating and seating design stresses for the bolts.

4) Define the seating design bolt load, W, from the average of the required and actual bolt areas:

$$W = 0.5(Am + Ab)Sa$$

5) Define the operating design bolt load (also W) as: W = Wm1

From this point the ASME Code requires a stress analysis of the flange in question that will verify that the flange stresses in the equation are adequate for these bolt loads.

m and y FACTOR

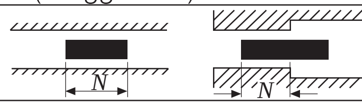
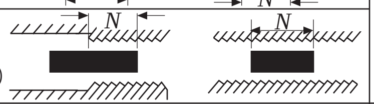
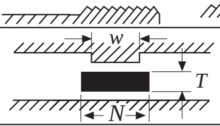
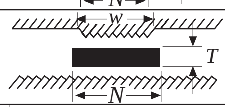
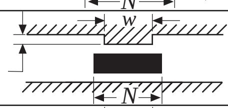
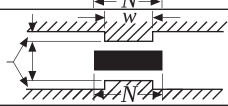
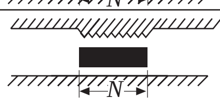
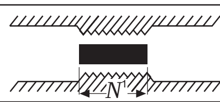
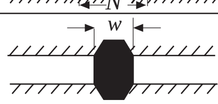
The above process requires a recommendation on the appropriate m and y for the ***GRAFTEC*** gasket. Since the ***GRAFTEC*** gasket is a laminate of flexible graphite chemically bonded to a steel core, it should perform at least as well as other similar laminated flexible graphite products that are available to the process industry.
In fact, the École Polytechnic tests on the ***GRAFTEC*** gasket samples show that it leaks less than laminated flexible graphite chemically that is bonded to a steel core.

Thus, for non-critical applications where the leak rate is not a concern, it is recommended that the m and y of 3 and 5,000 psi respectively, as recommended for similar laminated flexible graphite products, be considered for the ***GRAFTEC*** gasket. This is based on the similarity of the ***GRAFTEC*** to other graphite products,. For more critical applications, it is recommended that the alternate PVRC method be used. The more modern PVRC method should be considered for the similar laminated flexible graphite products as well graphite products as well.

Gasket factors similar to the ASME Code m and y have been in use for around 50 years. Generally they have served industry well although one shortcoming has been that there is no acceptable way to replicate or verify current factors, or to get comparable new factors for new gasket products. Also, the ASME Code m and y do not consider the leak rate of a gasketed joint.

ASME CODE CALCULATION

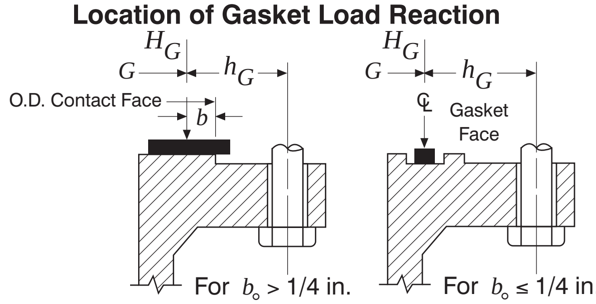
TABLE 2-5.2
EFFECTIVE GASKET SEATING WIDTH

Facing Sketch (Exaggerated)	Basic Gasket Seating Width, b_o Column I	Column II
(1a) 	$\frac{N}{2}$	$\frac{N}{2}$
(1b) See Note (1) 		
(1c)  $w \le N$	$\frac{w+T}{2};\left(\frac{w+N}{4}\ \text{max.}\right)$	$\frac{w+T}{2};\left(\frac{w+N}{4}\ \text{max.}\right)$
(1d) See Note (1)  $w \le N$		
(2) 1/64 in. Nubbin  $w \le N/2$	$\frac{w+N}{4}$	$\frac{w+3N}{8}$
(3) 1/64 in. Nubbin  $w \le N/2$	$\frac{N}{4}$	$\frac{3N}{8}$
(4) See Note (1) 	$\frac{3N}{8}$	$\frac{7N}{16}$
(5) See Note (1) 	$\frac{N}{4}$	$\frac{3N}{8}$
(6) 	$\frac{w}{8}$	...

Effective Gasket Seating Width, b_o

$b = b_o$, when $b_o \le 1/4$ in.; $b = 0.5\sqrt{b_o}$, $b_o > 1/4$ in.

Location of Gasket Load Reaction



NOTES:
(1) Where serrations do not exceed 1/64in. depth and 1/32in. width spacing sketches (1b) and (1d) shall be used.
(2) The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes

Section VII, of the ASME Boiler & Pressure Vessel Code, establishes criteria for flange design and suggests values of “m” for load maintenance and “y” for gasket seating. The big shortcoming is that the “m” and “y” **cannot be verified by test nor can they be rationally extended to new products because there is no workable standard test procedure.** The ASME code “m” and “y” do not consider the leak rate of a gasketed joint. It has been suggested that the only reason that traditional values of “m” and “y” have worked is because, in actual practice, applied bolt stresses are significantly higher than ASME Code design values.

“m” is the gasket factor for load maintenance. It is the ratio of contact pressure to contained pressure. The Code equation defines this term as the ratio of residual gasket load to fluid pressure at leak. Stated another way, it is the factor which defines how many times the residual load (original load less internal fluid pressure) must exceed the internal fluid pressure. Experiments show that the liquid or gaseous pressure a joint will contain is proportional to the amount of residual contact pressure exerted by the joint surfaces on the gasket—and that the contact pressure on the gasket must usually be larger than the pressure being contained.

“y” factor is the minimum gasket seating stress in either psi or megapascals that is required to preload or seat the gasket to prevent leaks in the joint as the system is pressurized. It is the flange pressure to compress the gasket enough to eliminate pores. It can be compared to an actual unit stress at gasket bearing surface (gasket assembly stress).

The “y” gasket factors in the ASME Code are experimentally determined. Many attempts have been made to confirm or “improve” them with often conflicting results. The following from John H. Bickford, “An Introduction To The Design And Behavior Of Bolted Joints,” Marcel Dekker, Inc. 1990, page 510, is an excellent summary of the “y” gasket factor.

> **The Code was never intended to define leak-free behavior of a gasketed joint.** It was intended, instead, to force designers to design pressure systems which would not “blow up” in service, to eliminate the frequent boiler explosions which occured—with often fatal results—before the Code was written. Leaks were of no real concern to the original authors in the 1930s.
>
> To support the statement that the Code is not written to define leak-free behavior, **note that the gasket factors listed in the Code are suggestions only. They are not mandatory and—this is less obvious—they are not intended to define assembly limits on gasket stress.**
>
> The implications of all this are significant for people who design or assemble bolted flanged joints: The current code “m” and “y” gasket factors do not really define the characteristics or behavior of gaskets. As a result, they do not provide a firm foundation for either design or assembly decisions.

A quiet revolution in the technology of gaskets for process plants has occurred. Early efforts were driven by a quest for meaningful gasket design factors (“m” & “y”) needed for improved ASME Boiler & Pressure Vessel Code designed joints. Directed by the PVRC (Pressure Vessel Research Council) committee on Bolted Flanged Connections - Welding Research Council, this effort was further expanded in 1985 as the number of asbestos substitutes grew. More recently fugitive emissions from flanged joints are stimulating further gasket tightness testing development.

GRAFTEC **gasket test data is based on the new PVRC gasket test programs that consider leakage and make the tightness of the joint a design criterion.**

To help improve designs, an ASME Special Working Group (SWG/BFJ) is working to implement gasket constants derived from hundreds of PVRC sponsored gasket tests. Also, an ASTM task group (3.40.21) is evaluating the PVRC gasket tightness performance test as a draft ASTM Standard Test. The new ASTM standard gasket test will be designed to elicit the gasket constants Gb, a and Gs for gasket materials.

(A) GENERAL REQUIREMENTS
(1) In the design of a bolted flange connection, calculations shall be made for design conditions including pressure,external loads and tightness.

(B) DESIGN REQUIREMENTS
(1) ***Tightness.*** Tightness is the internal pressure needed to cause a certain small leak rate in a joint. Tightness is expressed through a Tightness Parameter, Tp, (See NOTE 1). Joints shall be designed to satisfy a tightness requirement that is established by the selection of a Tightness Class that is appropriate for the service conditions. The minimum required tightness Tpmin recognizes a maximum permitted leak rate for the selected class. (a) The Tightness Class is a value of Tc for use in Formula (1). Tc shall be selected for the desired tightness class.

TABLE XX- 6.1

TIGHTNESS CLASSES AND CATEGORIES		
Category	Tightness Class	Tc
Air,Water (Note 2)	1	0.1
Standard	2	1.0
Tight	3	10.0

NOTE 1: The Tightness Parameter, Tp is a measure of tightness that has been defined as proportional to pressure and inversely proportional to the square root of leak rate. More precisely, Tp is the pressure (in atmospheres) required to cause a helium leak of 1 mg/sec for a 5-7/8 in. (1 50mm) OD gasket in a joint. The pressure to cause a leak of 1 mg/sec of helium for that joint is its tightness. A higher value of Tp means a tighter joint. Because of the square root, a joint that is 10 times tighter leaks 100 times less. More specifically:

$$Tp = (P/p^*)(1/Lr)^{0.5}$$

P is the design pressure, p* is atmospheric pressure and Lr is the leak rate (mg/sec/mm) for a 150 mm OD gasket. For design purposes, Lr must be defined in order to make use of Tp.

NOTE 2: Tightness Class 1 may be considered for non-critical services such as water and air.

(b) A minimum tightness requirement, Tpmin shall be satisfied in operation after the application of pressure P and any external loads. (Other loads that further diminish the stress at the gasket during operation may be taken into account in Formula (7) if appropriate.) Tpmin shall be determined by Formula (7):

$$Tpmin = 0.1243(Tc)P \quad \textbf{(1)}$$

(c) The assembly tightness, Tpa, is a value of the tightness parameter greater than Tpmin. Tpa must be achieved by sufficient compression of the gasket at the time of joint assembly to assure that Tpmin is achieved in operation. Accordingly by Formula (2) Tpa determines the value of the design assembly seating gasket stress Sya in Formula (6).

$$Tpa = X\ TPmin \quad \textbf{(2)}$$

$$Tr = Log(Tpa) / Log(TPmin) \quad \textbf{(3)}$$

X takes the value 1.5 (Sa/Sb) minimum. X > 1.5 may be used to adjust Tr in Formula (3) so that Sml = Sm2. This requires interaction and finds the lowest permitted value of Wmo in most cases. (d) For joints with compression stops where Ss < Sya, X in Formula (2) must be adjusted to increase Wmo in most cases. In this case X is a constant:

$$X = (1/TPmin)\ (Ae(Ss)/Gb)^{(1/a)} \quad \textbf{(4)}$$

(2) *Operating Conditions.* The conditions required to resist the hydrostatic end force of the design pressure tending to part the joint, and to maintain on the gasket or joint-contact surface sufficient compression to assure a joint that meets the required tightness at the design conditions. The minimum load is a function of the design temperature. The minimum operating gasket stress, Sml, is a function of the design pressure, the tightness class (see Table X)(-6.1), the seating gasket stress Sm2, the assembly stress, the gasket material, and the gasket contact area, Ag, to be kept tight under pressure, per Formula (5) in below, and determines one of the requirements for the amount of the bolting Am.

Sm1 = Gs[(Gb/Gs)(Tpa) ^ a] ^ (1/Tr) **(5)**

(3) *Gasket Assembly.* The conditions existing when the gasket or joint-contact surface is seated by applying an initial load with the bolts when assembling the joint, at atmospheric temperature and pressure. The minimum initial gasket assembly stress; Sya, considered to be adequate for proper seating, is a function of the gasket material, the design pressure, the tightness class, the assembly method, and the effective gasket contact area, Ag, per Formula (6) in below. Changing Sya effects both Sm1 and Sm2 and as a result the amount of bolting required will usually be minimized by finding that value of Sya which makes Sm1 = Sm2. That value is found through the application of Formulas (2) and (3) to obtain the optimum value of Tpa.

Sya = (Gb/Ae)(Tpa) ^ a **(6)**

NOTES: Ae = Assembly Method Efficiency. The use of "Ae" provides the means to credit the uniformity gained from use of the more sophisticated assembly methods when these are cost effective. Ae = 1.0 assumes "ideal" bolt-up such as for hydraulic tensioners and ultrasonics. Ae = 0.75 assumes manual bolt-up.

(C) Design Gasket Stresses

(1) *Seating.* The gasket seating stress, Sya is related to the design operating gasket stress Sm1 of Formula (1) by means of the seating design gasket stress Sm2 of Formula (7).

Sm2 = (Sb/Sa)(Sya/1.5) - P Ai/Ag **(7)**

(D) Required Bolt Load

The bolt loads used in calculating the required cross-sectional area of bolts shall be determined as follows: (1) The required bolt load operating Wmo shall be sufficient to resist the hydrostatic end force H exerted by the maximum allowable working pressure on the area bounded by the diameter gasket reaction, and, in addition, to maintain on the gasket or joint-contact surface a compression load Hp. (This compression load Hp is based on the design gasket stress, Smo which is the larger of Sm1 or Sm2 or 2P and therefore a function of the assembly load as well as the gasket material, service and loads and construction. (See NOTE 3.)

Note 3: Tables XX-6.2 give a list of commonly used gasket material, with suggested values of Gb,a, and Gs that have been established by test. These values are suggested only and are not mandatory. Values that are too low may result in excessive emissions and even gross leakage at the joint without affecting the safety of the design.

The required bolt load Wmo is determined by Formula (8) and (9).

Smo > Sm1 or Sm2 or 2P **(8)**

Wmo = H + Hp = P(Ai) + Smo(Ag) **(9)**

The equation for Wmo means that the bolts are designed for the sum of the pressure load (also called the hydrostatic end force), as represented by P(Ai), plus a gasket load that is sufficient to maintain a seal and adequate seating.

(E) Flange Design Bolt Load, W

The bolt load used in the design of the flange shall be the value obtained from formula (10).

W = Ab (Sa) **(10)**

ADD:

a = slope of gasket loading line, a gasket constant associated with seating a gasket such that the required minimum thghtness may be achieved

Ae = Assembly efficiency, a factor related to bolt load and gasket stress variation which depends on the method of joint assembly. The “ideal” is 1.0. A good estimate is 0.75 for manual bolting, 0.85 for calibrated Torque, 0.95 for multipoint hydraulic tensioning and 1.0 if bolt stretch is measured

Ag = Gasket contact area, in^2 (mm^2) based on the contact width,N. Ag = 0.7854 x (Od^2 - ID^2) or Ag = 3.14 x (Go-N)xN

Ai = Pressurized (hydraulic) area, square inches (mm^2) en-circled by the effective diameter, G. Ai = 0.7854 x G^2

Gb = Gasket constants associated with seating a gasket such that the required minimum tightness may be achieved. Gb represents the pressure on the gaskets, psi (MPa) and a represents the slope of the Gb line in relation to the Tp

Go = The smaller of the Gasket OD or Flange face sealing surface OD

Gs = Gasket constant associated with maintaining the required minimum tightness after the application of fluid pressure, psi (MPa)

n = Effective gasket sealing width. n = b’

N (Ae) = Assembly efficiency, a factor related to bolt load and gasket stress variation which depends on the method of joint assembly

Sm1 = Design gasket stress for the required operating tightness, psi (MPa) Sm1 = Gs[Gb/Gs x (Tpa)^a]^(1/Tr) or Gs[AeSya/Gs]^(1/Tr)

Sm2 = Design Gasket Stress related to the initial seating condition, psi MPa). Sm2 = [Sya/(1.5Sa/Sb)] - P(Ai/Ag)

Smax = The maximum permitted value of Sya equal to 110% the maximum gasket stress in the test that determined the gasket constants

Smin = The minimum permitted value of Sm1 equal to 90% the minimum gasket stress in the test that determined the gasket constants. Smin is 900 psi for the standard test procedure

Smo = Design gasket stress, psi (MPa)

Sya = Design assembly seating stress or joint contact unit seating load, psi (MPa)

Ss = gasket stress associated with the load needed to initiate contact by a compression limiting device

Tp = Tightness Parameter expressing the ratio of pressure to the square root of leak rate in dimensionless form. It is based on mass leak rate. Leakage is proportional to gasket diameter (leak rate per unit diamaeter) Tp is the pressure (in atmospheres) required to cause a helium leak rate of 1 mg/sec for a 5.9” (150 mm) OD gasket in a joint.

Tpmin = Minimum required tightness,value of Tp required to assure satisfactory leakage performance in operation for the specified tightness class. Tpmin. must = 0.1243 x (Tc) x P

Tpa = The tightness achieved at assembly, value of Tp required to assure that Tpmin is achieved in operation.

Tc = Tightness class factor. Tc1 represents a mass leak rate per unit diameter of 0.2 mg/sec-mm, Tc2 represents 0.002 mg/sec-mm, Tc3 represents 0.0002 mg/sec-mm.

Tr = Tightness ratio, a value greater than 1.0

Tr = Log(Tpa)/Log(Tpmin)

Wmo = Minimum required bolt load, lb(kg)

X = The ratio Tpa/Tpmin used in equation X = ?? X takes the value 1.5 minimum, X > 1,5 may be used to adjust Tr so that Sm1 = Sm2. This requires iteration and finds the lowest permitted value of Wmo in most cases. For joints with compression stops X must be adjusted to increase Wmo if Ss < Sya, X must be adjusted to increase Wmo

MODIFY:

Am = Total required cross-sectional area of bolts at root of thread or section of least diameter under stress, Square inches. Am = Wmo/Sb

W = Flange design bolt load, lb(kg)

H = Total hydrostatic end force, lb(kg) H = P x Ai

PROPOSED ASME CODE METHOD (PVRC CONVENIENT) FOR THE *GRAFTEC*

ASME CODE DATA:	psi	MPa	As per table 2-5.2 in the CODE		
Design Pressure, P:	500	3.45	**GASKET**	inches	mm
Allowable Bolt Stress:			Outside Dia, Go:	30.00	762.0
Ambient,Sa:	25,000	137.9	Width,N:	0.625	15.9
Design Temp., Sb:	25,000	137.9	Effective Width, n:	0.280	7.1
Sa/Sb:	1.0		Eff.Diameter,G:	29.44	747.8
			G = Go-2n		
***GRAFTEC* GASKET CONSTANTS**			Full contact area, Ag:	57.7 ^ in	372.1 ^ cm
Load Gb:	922 psi	6.4 MPa	Ag = 3.14 (go-N)N		
Slope of load a:	0.248		Pressure Area, Ai:	680.76 ^ in	4,392 ^ cm
Unload Gs:	5.1 psi	0.04 MPa	Ai = (3.14/4)G ^ 2		

DESIGN REQUIREMENTS, TIGHTNESS PARAMETERS, Tp
Assume Tightness Class (Tight) of 3 represented by Tc = 10 and Assembly Efficiency of Ae = 0.75 for manual bolt up. Note that for an “ideal” bolt-up the Ae = 1.0.
Pr: 34.04 = Design Pressure/Atmospheric Pressure = (500 psi/14.7 psi)
Minimum Required Tightness,Tpmin
1.8257 x Tc x Pr or .1243 x Tc x P = Tpmin: 622

SOLUTION METHOD: Either of solution method “convenient” or “flexible” is acceptable. In this example we are using an explicit “Convenient” method which sets X = 1.5, or 1.5 Sa/Sb if Sa > Sb
The Assembly Tightness, Tpa when X = 1.5 X x Tpmin = Tpa: 932
Tightness Parameters Ratio of the logarithms, Tr = Log (Tpa) / Log (Tmin) = Tr: 1.063

GASKET DESIGN STRESSES, psi or MPa
Seating, Sya Gb/Ae(Tpa) ^ a Sya: 6,701 psi 46.2 MPa
Note: Keep Sya < 1.2 Max test stress

Operating stress, Sm1 Gs[(Gb/Gs)(Tpa) ^ a] ^ (1/Tr) Sm1: 3,339 psi 23.0 MPa

Seating stress, Sm2 (Sb/Sa)(Sya/1.5) -P(Ai/Ag) Sm2: (1,439) psi (9.9) MPa
Keep Sm1 and Sm2 > 800 psi which is within 20% of the range of test data

DESIGN STRESS, Smo, Select largest of 2P, Sm1 or Sm2 for Smo Smo: 3,339 psi 23.0 MPa
Mo, Select largest of 2P, Sm1/P or Sm2/P Mo: 6.68

DESIGN BOLT LOAD, Wmo Smo(Ag)+P(Ai/Ag) or P(Ai+AgSmo/P) Wmo: 532,984 lbs 242,265 kg
or you can use Mo: Wmo = P(Ag x Mo + Ai) Wmo: 532,984 lbs 242,265 kg

ASME CODE LIKE FACTORS: m = (Smo/P)(N/2b’): m: 7.467
Note: n = b/ ya = N Sya/b’/1.5 ya: 9,989 psi 68.9 MPa
Wm1 = (2b’)3.14(m)GP+AiP Wm1: 533,416 lbs 242,462 kg
Wm2 = 3.14 b’(ya)G Wm2: 258,229 lbs 117,377 kg

END FORCE RATIO: Wmo/PAi=Wr Wr: 1.565

GASKET STRESS RATIOS: Sm1 / P: 6.679
Sm2 / P:-2.869

PROPOSED ASME CODE METHOD (PVRC FLEXIBLE) FOR THE *GRAFTEC*

ASME CODE DATA:	psi	MPa	As per table 2-5.2 in the CODE		
Design Pressure, P:	500	3.45	**GASKET**	inches	mm
Allowable Bolt Stress:			Outside Dia, Go:	30.00	762.0
Ambient,Sa:	25,000	137.9	Width,N:	0.625	15.9
Design Temp., Sb:	25,000	137.9	Effective Width, n:	0.280	7.1
Sa/Sb:	1.0		Eff.Diameter,G:	29.44	747.8
			G = Go-2n		
***GRAFTEC* GASKET CONSTANTS**			Full contact area, Ag:	57.7 ^ in	372.1 ^ cm
Load Gb:	922 psi	6.4 MPa	Ag = 3.14 (go-N)N		
Slope of load a:	0.248		Pressure Area, Ai:	680.76 ^ in	4,392 ^ cm
Unload Gs:	5.1 psi	0.04 MPa	Ai = (3.14/4)G ^ 2		

DESIGN REQUIREMENTS, TIGHTNESS PARAMETERS, Tp
Assume Tightness Class (Tight) of 3 represented by Tc = 10 and Assembly Efficiency of Ae = 0.75 for manual bolt up. Note that for an “ideal” bolt-up the Ae = 1.0.
Pr: 34.04 = Design Pressure/Atmospheric Pressure = (500 psi/14.7 psi)
Minimum Required Tightness,Tpmin
1.8257 x Tc x Pr or .1243 x Tc x P = Tpmin: 622

SOLUTION METHOD: Either of solution method “convenient” or “flexible” is acceptable. In this example we are using a “Flexible” method which increases the ratio X = Tpa/Tpmin form 1.5 minimum so that Sm1 = Sm2. This method requires iteration and finds the lowest (optimum) bolt load.
The Assembly Tightness, Tpa when X = 9.53 X x Tpmin = Tpa: 5,962
Tightness Parameters Ratio of the logarithms, Tr = Log (Tpa) / Log (Tmin) = Tr: 1.3515

GASKET DESIGN STRESSES, psi or MPa
Seating, Sya Gb/Ae(Tpa) ^ a Sya: 10,616 psi 72.3 MPa
Note: Keep Sya < 1.2 Max test stress

Operating stress, Sm1 Gs[(Gb/Gs)(Tpa) ^ a] ^ (1/Tr) Sm1: 1,176 psi 8.1 MPa

Seating stress, Sm2 (Sb/Sa)(Sya/1.5) -P(Ai/Ag) Sm2: 1,176 psi 8.1 MPa
Keep Sm1 and Sm2 > 800 psi which is within 20% of the range of test data

DESIGN STRESS, Smo, Select largest of 2P, Sm1 or Sm2 for Smo Smo: 1,176 psi 8.1 MPa
Mo, Select largest of 2P, Sm1/P or Sm2/P Mo: 2.532

DESIGN BOLT LOAD, Wmo Smo(Ag)+P(Ai/Ag) or P(Ai+AgSmo/P) Wmo: 408,213 lbs 185,551 kg
or you can use Mo: Wmo = P(Ag x Mo + Ai) Wmo: 408,213 lbs 185,551 kg

ASME CODE LIKE FACTORS: m = (Smo/P)(N/2b’): m: 2.63
Note: n = b/ ya = N Sya/b’/1.5 ya: 15,826 psi 109.14 MPa
Wm1 = (2b’)3.14(m)GP+AiP Wm1: 408,365 lbs 185,621 kg
Wm2 = 3.14 b’(ya)G Wm2: 409,130 lbs 185,968 kg

END FORCE RATIO: Wmo/PAi=Wr Wr: 1.199

GASKET STRESS RATIOS: Sm1 / P: 2.352
Sm2 / P: 2.352

The examples are for DESIGN BOLT LOADS using the PVRC convenient and flexible methods. It is important to remember that these calculations are for designing bolted flange joints. When you assemble the joint there are additional factors to consider.

PVRC METHODS

CONVENIENT		FLEXIBLE
622	Tpmin, psi	622
1.5	X, ratio	9.53
1.063	Tr, logs	1.3515
932	Tpa, psi	5,962
922	Gb, psi	922
0.75	Ae, manual bolt-up	0.75
1,229	Gb/Ae, psi	1,229
0.248	a (slope of Gb)	0.248
5.1	Gs, psi	5.1
6,701	Sya, psi	10,616
3,339	Sm1, psi	1,176
(1,439)	Sm2, psi	1,176
3,339	Smo, psi	1,176

All torque tables should have the following warning: For general use only: make experiments of your own to determine the true values for your application. Why is this? The torque-tension relationship is influenced by many factors such as tool accuracy, elastic interaction, operator skill, bolting procedure, etc., as well as more obvious factors such as lubricity and condition of the threads. In equations, the value, K is a convenient catch-all that sums up everything which affects the torque-tension relationship. How do you determine K? Obviously, it is a soft number and to achieve an accurate K would require experiments on that application itself. This would be very expensive and time-consuming. Therefore, some kind of table can be useful as a guide for selecting a tool of the appropriate size and estimating torque.

The following table is based on the assumption that:
1. Alloy steel stud bolts
2. The nut factor (K) is 0.2; bolts and nuts are used in as-received condition and are neither cleaned nor lubricated.

CALCULATING TORQUE VALUE

Based on the appendix below, [nut factor = 0.2] the minimum assembly bolt load for a 1" bolt [Ar = .551] with a calculated bolt stress of 45,000 psi is **368** ft-lb.

T(ft-lb) = Fp x K x D/12

Fp = bolt load (lb) calculated by bolt stress[Sa] of 45,000 psi x 0.55[Ar] = 24,795 lbs

K = 0.2(dimensionless) D = 1(in.) T(ft-lb) = [24,795 lbs] x [0.2] x [1"] /12 = 368 T(ft-lb)

NUT FACTOR (K) = O . 2 (DIMENSIONLESS)				
NOMINAL DIAMETER OF BOLT (Inches) D	NUMBER OF THREADS (Per Inch) n	AREA AT ROOT OF THREAD (Sq. Inch) A_r	STRESS 45,000 PSI	
			TORQUE (ft-lbs) F_p	COMPRESSION (lbs) T
1/4	20	0.027	6	1,215
5/16	18	0.045	12	2,025
3/8	16	0.068	18	3,060
7/16	14	0.093	30	4,185
1/2	13	0.126	45	5,670
9/16	12	0.162	68	7,290
5/8	11	0.202	90	9,090
3/4	10	0.302	150	13,590
7/8	9	0.419	240	18,855
1	8	0.551	368	24,795
1-1/8	8	0.728	533	32,760
1-1/4	8	0.929	750	41,805
1-3/8	8	1.155	1,020	51,975
1-1/2	8	1.405	1,200	63,225
1-5/8	8	1.680	1,650	75,600
1-3/4	8	1.979	2,250	89,100
1-7/8	8	2.303	3,000	103,680
2	8	2.652	3,300	119,340
2-1/4	8	3.422	4,770	154,035
2-1/2	8	4.291	6,600	193,140
2-3/4	8	5.258	8,880	236,655
3	8	6.324	11,580	284,580

Research has shown that a non-lubricated bolt has about 50% of the efficiency of a well-lubricated bolt. It has further been found that different lubricants produce results varying between the limit of 50% and 100% of the tabulated stress figures. Please turn to page 36 for further data.

How much torque is needed to tighten this bolt? There is no precise "over-the-phone" answer. When bolts are tightened they become stressed in tension. As far as torque balance is concerned, engineers frequently say that approximately 90% of the tightening torque is needed to overcome the friction between the nut and the joint and the mating threads. The remaining 10% develops useful tension in the bolt. Complicated formulas are structured to describe this relationship between torque and preload. The short-form equation is defined as follows:

$T(\text{ft-in}) = KDF_p$ and $T(\text{ft-lb}) = KDF_p/12$

WHERE:

T = Applied torque, (lb.-in, N-mm) To convert to ft/lbs divide by 12.

K = Nut factor (dimensionless)

F_p = Initial applied force, bolt tensile load (lb., N)

D = Nominal diameter of the bolt (in., mm)

OTHER EQUATIONS:

$F_p = S_a \times A_r$

$S_a = F_p/A_r$

$A_r = .7854 \times (D - 1.3 / n)^2$

$A_s = .7854 \times (D - 0.9743 / n)^2$

S_a = Bolt stress, (psi, MPa)

A_r = Root diameter area, mandated by the ASME Code. (in^2, mm^2)

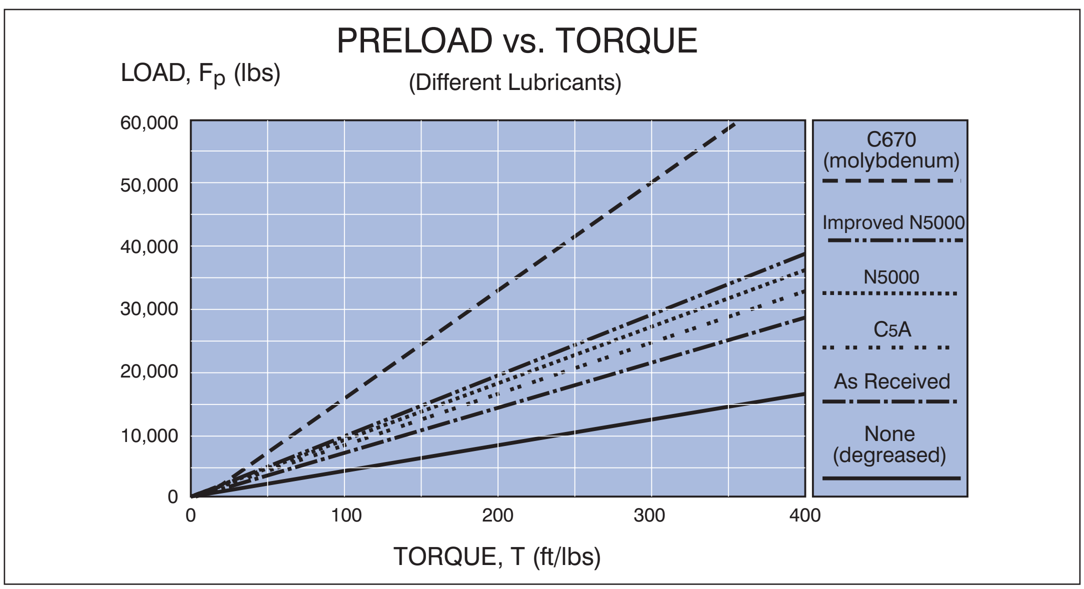
n = Number of threads per inch

A_s = Tensile stress area (in^2, mm^2)

NUT FACTOR(K): The value, K, is not a coefficient of friction. Rather, it is a convenient catch-all constant encompassing the many factors and features influencing the torque – tension relationship. However, the most dominate influence in most cases is the coefficient of friction.

EFFECT OF LUBRICANT ON TORQUE: The importance of correctly evaulating K when establishing a tightening torque is best illustrated with a real example. As part of an experimental investigation, A193 B7 7/8" bolts and nuts were evenly lubricated with a high-performance lubricant (Fel-Pro C-670) that resulted in an average K of 0.086. To reach a measured bolt stress of 50,000 psi only took 307 lb-ft of torque. When bolts and nuts were torqued in as-received condition, the measured K was 0.170, and required over twice as much torque 624 lb-ft to reach the same bolt stress of 50,000 psi. Tensile stress area (A_s) is used, rather than A_r.The stress in the bolts was measured with strain gages.*

*Source: Chaaban A., Derenne M., Bouzid A., Schafer W., "Evaluation of torque coefficients and gasket stress distributions in a bolted flanged joint using different types of lubricants." 3rd International Symposium on Fluid Sealing. Biarritz, France. September 16, 1993.



Alloy: A homogeneous combination of two or more metals.

ANSI: American National Standards Institute.

API: American Petroleum Institute.

Asbestos: A fibrous mineral characterized by its ability to resist high temperatures and the actions of acids.

Ash: An impurity found in natural and other types of graphite and are ordinarily expressed in parts per million (ppm), or percent ash.

ASME: American Society of Mechanical Engineers, founded 1880, is an educational, technical and professional society of mechanical engineers and other qualifying individuals. ASME is an internationally recognized voluntary standards setting organization.

ASTM: American Society of Testing and Materials.

Atmospheric Pressure: The weight of a column of air per area unit as measured from the top of the atmosphere to the reference point being measured. Atmospheric pressure decreases as altitude increases. ICAO sea level standard values = 14.696 pounds per square inch (0.1014 MPa).

Boiler and Pressure Vessel Code: A large document, maintained and published by the American Society of Mechanical Engineers (ASME). The code describes rules, material properties, inspection techniques, fabrication techniques, etc., for boilers and pressure vessels. It is sometimes referred to as the “Code”.

Bolt Load (pounds): It is a means of applying compressive load that flows the gasket material into surface imperfections to form a seal.

BSS: British Standards Specifications.

Calender: A machine containing rollers used in the flexible graphite industry, rubber industry and others, for compressing materials into continuous rolls, or sheets.

Cold flow: Continued deformation under stress.

Compression: Stress from forces acting toward each other.

Compressibility: The extent to which a gasket is compressed by a specified load. Permanent set is the unit amount, in percentage of the compressibility, that the material fails to return to the original thickness when the load is removed. Recovery is the amount of return to the original thickness in a given time, and is usually less under a prolonged load.

Corrosion: In broad terms, it is the destructive alteration of metal by chemical or electrochemical reaction within its environment, which encompasses not only atmospheric exposure but all the interacting conditions associated with a service application.

Corrugated, functionally: To create memory, springiness and resilience, to wrinkle up with a parallel series of ridges, grooves and hollows, or a parallel series of peaks and valleys or troughs.

Creep: The slow, plastic deformation of a body under heavy loads. Independent variables which affect creep are time under load, temperature and load or stress level. It is the loss of tightness in a gasket measurable by torque loss.

Deflection: The deviation from zero shown by the indicator of a measuring device. The movement of a part as a result of stress.

Density: The ratio of mass of a body to its volume or mass per unit volume of the substance. For ordinary practical purposes, density and specific gravity may be regarded as equivalent.

Din: Deutsch Industrie Norman. English translation is Germany Industry Standard – one of the European equivalents to ASTM.

Double-jacketed: A metal-jacketed gasket design that is entirely enclosed by the metal outer cover over a filler.

EPA: Environmental Protection Agency. A regulatory agency of the United States of America.

Elastic interaction: involves bolt stress relaxation when adjacent bolts are tightened. Essentially, the bolts “talk” to each other making it difficult to get uniform bolt loading.

Elasticity: The ability of a material to return to its original form after the removal of the deforming force (stresses). A substance is highly elastic if it is easily deformed and quickly recovers. Metals, if deformed only a few percent, can be considered purely elastic.

Elongation: The increase in length of a stressed material.

Envelope-gasket: The filler material is enclosed in an outer cover, typically of PTFE material to enhance corrosion resistance.

Extrusion: Pressure forces a metal or plastic into a gap or opening.

Eyelet: Metallic inner eyelet are used to protect the gasket material from the sealed media. Blowout resistance and gas sealability can be improved depending on the correct choice of eyelet geometry and metal.

Fastener: A mechanical device for holding two or more bodies in definite positions with respect to each other.

Flange: The rigid members of a gasket joint that contact the sides or edges of the gasket.

Flat ring: A flange gasket lying wholly within the ring of bolts.

Flow, or creep: The gradual continuous distortion of a material under continued load.

Foot-pound: A unit of work equal to the energy required to raise one pound one foot.

Fluid: A fluid has the ability to flow and possesses mass. Examples of fluids are liquids such as water and blood. An example of a gas is air.

Fulcrum: The point on which a lever turns.

Full-face gasket: Gasket covering the entire flange surface extending beyond the bolt holes.

Gases: Unlike molecules of a solid or liquid, gas molecules are not easily attracted to one another. They tend to remain separated. Gas molecules must be housed in a container or they will disperse and lose their integrity.

Gasket constants(Gb,a Gs): Gb represents the initial loading curve relationship with tightness while Gs obeys another with the unloading curve. The slope of the loading curve is represented by a.

Graftec **:** Registered trademark of Intermech Sealing Services (Pty) Ltd for a corrugated metal gasket with flexible graphite overlay.

Heat Exchanger, Shell and Tube: Metal shell with tubes inside. Most frequently the process stream fluid flows through the tubes and the heating or cooling fluid around the outside of the tubes in the shell.

Hooke's Law: Applying Hooke's Law, steel elongates 0.001 in. per 30,000 psi of applied stress.

Hydrostatic end force: It comes to the flange from the closed end of the pipe system to which it is welded. The end force reaches the flange through the hub and pulls on the ring portion of the flange mid-hub at its large end if it is a tapered hub.

Hydrostatic test pressure: A pressure used to test the integrity of a system, the value of which is one and a half times the anticipated system working pressure.

Hydrostatics: A branch of physics which deals with the pressure of fluids at rest.

ID: Symbol for inside diameter.

IFI: Industrial Fasteners Institute.

Initial preload: The tension created in a single bolt as it is tightened. It is usually modified by subsequent assembly operations and in-service loads and conditions.

Inorganic: Chemicals which do not contain carbon.

Initial preload: The tension created in a single bolt when the nut is first tightened. It is usually modified by subsequent assembly operations and/or by in-service loads and conditions.

Iteration: To do it repeatedly

JIS: Japanese Industrial Standards.

Jointing: Common term in Europe for Gasketing.

Gasket stress: The contact stress exerted on the gasket by the flange members.

Leakage Rate: The quantity, either mass or volume, of fluid passing through and/or over the faces of gaskets in a given length of time.

Litre: A metric unit of volume equal to a cubic decimeter (1,000 cm^3). or approximately 1.056 U.S. liquid quart. 1 liter contains 1,000 cubic centimeters of approximately 1 kilogram of water at 4°C (40°F)

"M" Maintenance value: An empirical design constant of a flange gasket used in the ASME Boiler and Pressure Vessels Code. The Code equation defines this term as the ratio of residual gasket load to fluid pressure at leak, dimensionless. The definition of "M" has varied in successive editions of the Code, according to the method employed for computing residual gasket load.

Mass: The measure of the quantity of matter.

Milligram: One thousandth of a gram.

Millimetre: One milliliter is 1/1,000th of a liter, it is equivalent to one cubic centimeter (1 cm^3)

Modulus of elasticity: The ratio of the unit stress to unit strain within the elastic limit without fracture.

MTI: Materials Technology Institute of the Chemical Process Industries.

MSS: Manufacturer's Standardization Society of the Valve & Fittings Industry.

Nut Factor: (K) An experimental constant used to evaluate or describe the ratio between the torque applied to a fastener and the Preload achieved as a result. For example, torque vs. Preload, (short-form equation)
(T) torque = **(Fp)** achieved preload (lb, N) **X (K)** nut factor **X (D)** nominal diameter (in., mm).

OD: Symbol for outside diameter.

Oxidation: The act of uniting, or causing a substance to unite with oxygen chemically.

Pascal: A SI unit of pressure equivalent to one Newton per square meter.

Pascal's law: The ability of gas or liquid to transmit pressure equally in all directions throughout itself is known as Pacal's law.

Permeability: The quality or condition of allowing passage of fluid through a material.

Pi: The symbol which denotes ratio of the circumference of a circle to its diameter.

Pitch: The nominal distance between two adjacent thread roots or crests.

Preload: A clamping force expressed in pounds, which denotes the amount of tension force created that holds two or more pieces together when a fastener is tight.

Pressure: A measure of a force's intensity. To determine pressure, the total force is divided by the area (usually square inches) on which it is acting. The result is the pressure (amount of force per square inch).

Pressure, atmospheric: Pressure exerted by the atmosphere at any specified location. (Seal level pressure is approximately 14.7 pounds per square inch absolute).

Pressure, gauge: Pressure differential above or below atmospheric pressure. Expressed as pounds per square inch gauge (P.S.I.G.).

Proof load: The maximum, safe, static, tensile load which can be placed on a fastener without yielding it. Proof load is an absolute value, not a maximum or minimum. Sometimes given as a force (lb, N) sometimes as a stress (psi MPa).

PTFE: Polytetrafluoroethylene plastic.

PVRC: Pressure Vessel Research Council sponsored by the Welding Research Council.

Raised-face flange: A flange which contacts its mating joint member only in the region in which the gasket is located. The flanges do not contact each other at the bolt circle.

Recovery: The ability of the gasket to spring back after the compressive load is reduced.

Relaxation: The loss of tension, and therefore clamping force in a bolt and joint as a result of creep, thermal expansion, embedment, etc.

Residual load: The tension is less than the amount in preload by the amount of combined fluid relied and creep-relaxation.

Resilience: The property of a material (stiffness/ recovery) that enables it to resume its original shape or position after becoming bent, stretched, or compressed; elasticity.

Resilience: The energy of elasticity-the energy stored in a material under the strain within its elastic limit which will cause it to resume its original shape when the stress is removed.

Ring gasket: A flange gasket lying wholly within the ring of bolts. Also flat ring or raised face gasket.

Ring joint gasket: A shaped seal used in conjunction with flanges that are grooved to accept the ring joint gasket.

Root diameter area: ASME Code mandates the use of root diameter area rather than tensile stress area. The expression for this area is:
Ar = 0.7854(D-1.3/n)^2
D = is the nominal diameter
n = is the number of threads per inch

SI (International System of Units): is a modernized and internationally standardized version of the metric system based on the meter, second, kilogram, ampere, degree Kelvin, and candela.

Sealability: is a measure of fluid leakage through and across both faces of a gasket.

Spiral-wound gasket: A gasket which is formed by winding spring-like metal, usually "V" shaped, and a suitable filler layer into a spiral.

Springback: It is a measure (percent) of the distance a gasket recovers from an initial compressive load.

Strain: A measure of the deformation that stress causes.

Stress: It is the applied force divided by the area. An applied force or system of forces that tends to strain or deform a body.

Stress corrosion cracking (SCC): A common form of stress cracking in which an electrolyte encourages the growth of a crack in a highly stressed bolt.

Stress relaxation: In application the gasket creates a resulting stress (stored energy) back against the sealing (flange) face. The resulting stress is known as the sealing contact force which is some value in pounds per square inch. This decrease in sealing contact force (stress) over time at a constant strain is known as stress relaxation. Example: You press a 100-pound barbell over your head at full arm's extension as long as possible. Stress relaxation determines when the barbell would no longer be higher than your head.

TEMA: Tubular Exchanger Manufacturers Association.

Tensile: Pertaining to extension or tension. Tensile strength is that strength necessary to enable a bar or structure to resist a tensile strain.

Tensile strengths: They are normally expressed in terms of stress-pounds per square inch (psi).

Tensile stress area: The effective cross-sectional area of the threaded section of a fastener. Used to compute average stress levels in that section. Based on the mean of pitch and minor diameters. $(A_s = 0.7854(D - 0.9743/n)^2$

Tension: Stress from forces that are acting away from each other.

Tension, bolt: Tension (tensile stress) created in the bolt by assembly preloads and/or such things as thermal expansion, service loads, etc.

Tensioner: A hydraulic tool used to tighten a fastener by stretching it rather than by applying a substantial torque to the nut.

TEFTEC **:** Registered trademark of Intermech Sealing Solutions (Pty) Ltd for a family of PTFE gaskets.

Tex-O-lon: Register trademark of Tex-O-lon Mfg., Inc. for a gasket composed of perforated steel encapsulated with Teflon.

Thermal: relating to heat; caused by heat.

Tightness: A measure of the mass leak rate from a gasketed joint.

Tightness parameter: A dimensionless parameter which defines the mass leakage of a gasket as a function of contained pressure and a contained fluid constant.

Tongue-and-groove joint: A flange joint in which one flange is provided with a tongue (male) and the other with a groove (female).

Torque: The twisting moment, product of force and wrench length, applied to a nut or bolt.

Ultimate strength: The maximum tensile strength a bolt or material can support prior to rupture. Always found in the plastic region of the stress strain of force-elongation curve, and so is not a design strength. Also called tensile strength and ultimate tensile strength.

UNS: United Numbering System is an alphanumeric designation to identify any metal or alloy. It is not a specification. UNS consists of a single uppercase letter, followed by five digits.

Vaaler awards: The competition is sponsored by Chemical Processing magazine. It is named after John C. Vaaler (1899-1963) who served as Editor from 1946 to 1963. The competition was developed to recognize significant technical advances in the Chemical Process Industries.

Viscous: Liquids such as water or mineral oil can be considered viscous. Viscous materials dissipate the energy used to deform them as heat, such as water which has no-memory.

Viscoelasticity: Materials such as elastomers which exhibit both viscous and elastic characteristics.

Viscosity: A measure of the resistance of a liquid's molecules to flow or slide past each other.

Volume: The size or extent of a three-dimensional object or region of space.

Yield strength: The tension-applied load required to produce a specified amount of permanent deformation in a solid material.

Y factor: (psi, MPa) The y factor is the initial gasket stress of surface pressure required to preload or seat the gasket to prevent leaks in the joint as the system is pressurized.

CELSIUS AND FAHRENHEIT

The Celsius scale was formerly called the Centigrade scale. It uses the melting point of ice as 0 °C (32 °F) and the boiling point of water as 100 °C (212 °F). The Celsius scale is commonly used for scientific work throughout the world.

Locate temperature in the central column. If in degrees Celsius, read Fahrenheit equivalent in right-hand column; if in degrees Fahrenheit, read Celsius equivalent in left-hand column.

°C		°F	°C		°F	°C		°F	°C		°F
-56.7	-70	-94	71.1	160	320	182.2	360	680	293.3	560	1040
-51.1	-60	-76	73.9	165	329	185.0	365	689	298.9	570	1058
-45.6	-50	-58	76.7	170	338	187.8	370	698	304.4	580	1076
-40.0	-40	-40	79.4	175	347	190.6	375	707	310.0	590	1094
-34.4	-30	-22	82.2	180	356	193.3	380	716	315.6	600	1112
-28.9	-20	-4	85.0	185	365	196.1	385	725	321.1	610	1130
-23.3	-10	14	87.8	190	374	198.9	390	734	326.7	620	1148
-20.6	-5	23	90.6	195	383	201.7	395	743	332.2	630	1166
-17.8	0	32	93.3	200	392	204.4	400	752	337.8	640	1184
-15.0	5	41	96.1	205	401	207.2	405	761	343.3	650	1202
-12.2	10	50	98.9	210	410	210.0	410	770	348.9	660	1220
-9.4	15	59	101.7	215	419	212.8	415	779	354.4	670	1238
-6.7	20	68	104.4	220	428	215.6	420	788	360.0	680	1256
-3.9	25	77	107.2	225	437	218.3	425	797	365.6	690	1274
-1.1	30	86	110.0	230	446	221.1	430	806	371.1	700	1292
1.7	35	95	112.8	235	455	223.9	435	815	376.7	710	1310
4.4	40	104	115.6	240	464	226.7	440	824	382.2	720	1328
7.2	45	113	118.3	245	473	229.4	445	833	387.8	730	1346
10.0	50	122	121.1	250	482	232.2	450	842	393.3	740	1364
12.8	55	131	123.9	255	491	235.0	455	851	398.9	750	1382
15.6	60	140	126.7	260	500	237.8	460	860	404.4	760	1400
18.3	65	149	129.4	265	509	240.6	465	869	410.0	770	1418
21.1	70	158	132.2	270	518	243.3	470	878	415.6	780	1436
23.9	75	167	135.0	275	527	246.1	475	887	421.1	790	1454
26.7	80	176	137.8	280	536	248.9	480	896	426.7	800	1472
29.4	85	185	140.6	285	545	251.7	485	905	432.2	810	1490
32.2	90	194	143.3	290	554	254.4	490	914	437.8	820	1508
35.0	95	203	146.1	295	563	257.2	495	923	443.3	830	1526
37.8	100	212	148.9	300	572	260.0	500	932	448.9	840	1544
40.6	105	221	151.7	305	581	262.8	505	941	454.4	850	1562
43.3	110	230	154.4	310	590	265.6	510	950	460.0	860	1580
46.1	115	239	157.2	315	599	268.3	515	959	465.6	870	1598
48.9	120	248	160.0	320	608	271.1	520	968	471.1	880	1616
51.7	125	257	162.8	325	617	273.9	525	977	476.7	890	1634
54.4	130	266	165.6	330	626	276.7	530	986	482.2	900	1652
57.2	135	275	168.3	335	635	279.4	535	995	487.8	910	1670
60.0	140	284	171.1	340	644	282.2	540	1004	493.3	920	1688
62.8	145	293	173.9	345	653	285.0	545	1013	498.9	930	1706
65.6	150	302	176.7	350	662	287.8	550	1022	504.4	940	1724
68.3	155	311	179.4	355	671	290.6	555	1031	510.0	950	1742

To convert Celsius to Fahrenheit: °F = 9/5 °C + 32
To convert Fahrenheit to Celsius: °C = 5/9(°F - 32)

Mass density: With leak rates expressed in terms of mass leak rates (milligrams or pounds), the following data will help us visualize the equivalents in volumetric leak rates. Mass density is defined as the mass per unit volume of a substance. Thus, if a substance occupies a space of 1 cubic foot and has a mass of 1/2 pound, its mass density is 1/2 pound per cubic foot. The mathematical equation is :

$$\text{mass density (D)} = \frac{\text{mass (m)}}{\text{volume (V)}}$$

With this equation you can find any one of the three physical quantities (D, m,V) if the measurements for the other two are given.

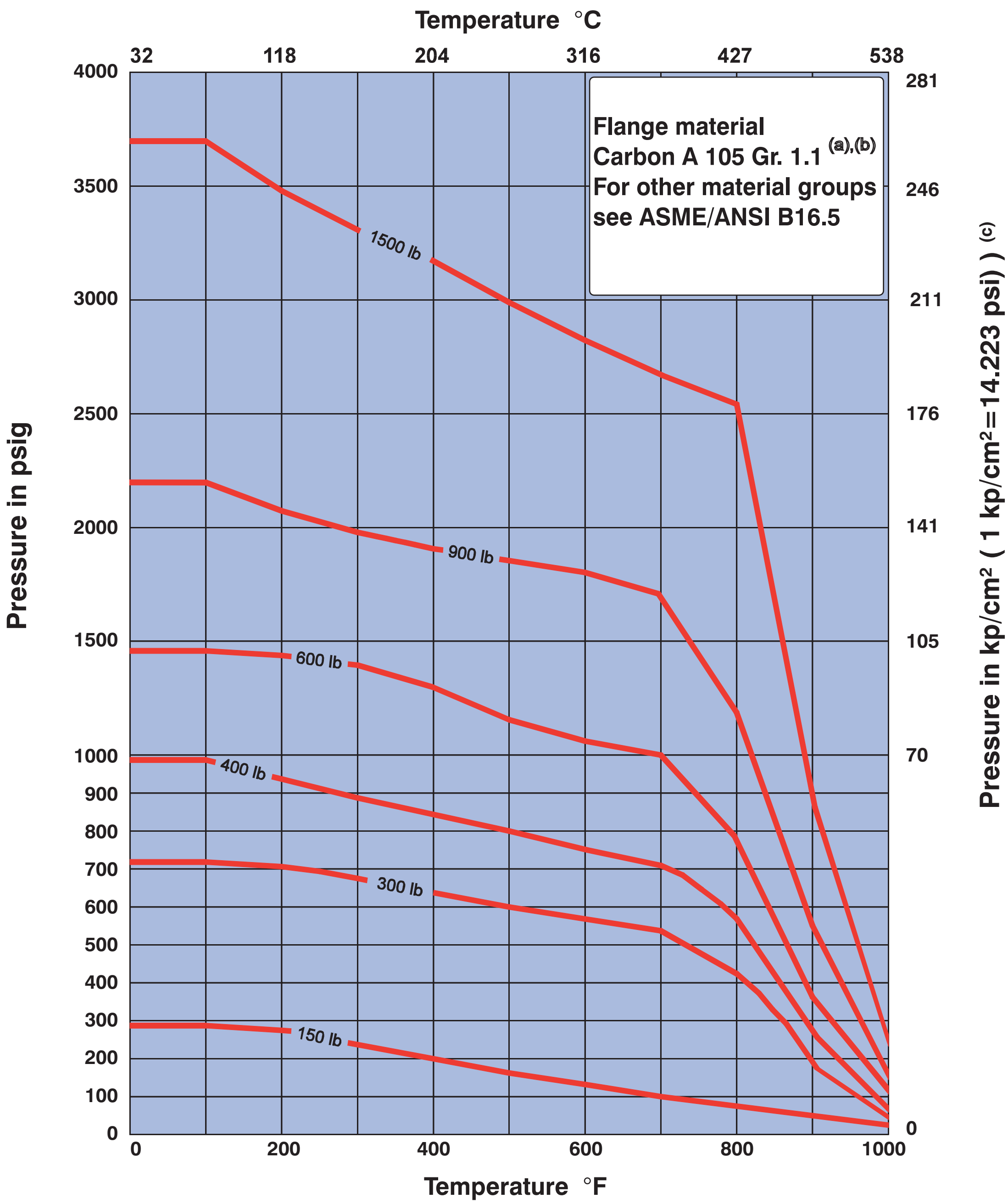
Most properties of matter are not constant. They vary with the environment. Thus, water freezes when it gets cold enough and boils when it gets hot enough. In each case, the physical properties of the water has changed. Similarily, the mass density of a gas increases when it is placed under pressure and decreases when the pressure is reduced.

MASS DENSITY

GASES	LB/CU FT	CU FT/LB	MG/CC	CC/MG
ACETYLENE (ETHYLENE)	0.0682	14.67	1.0925	0.915
AIR	0.0752	13.30	1.2046	0.830
AMMONIA	0.0448	22.32	0.7176	1.393
ARGON	0.1037	9.64	1.6611	0.602
BUTANE	0.1554	6.44	2.4893	0.402
CARBON DIOXIDE	0.1150	8.70	1.8421	0.543
CARBON MONOXIDE	0.0727	13.76	1.1645	0.859
CHLORINE	0.1869	5.35	2.9939	0.334
ETHANE	0.0789	12.67	1.2639	0.791
ETHYLENE	0.0733	13.64	1.1742	0.852
HELIUM	0.0104	96.25	0.1664	6.008
HYDROGEN CHLORIDE	0.0954	10.48	1.5282	0.654
HYDROGEN	0.0052	191.20	0.0838	11.93
HYDROGEN SULPHITE	0.0895	11.17	1.4337	0.698
METHANE	0.0417	23.98	0.6680	1.497
METHYL CHLORIDE	0.1342	7.45	2.1497	0.465
NATURAL GAS	0.0502	19.92	0.8041	1.244
NITRICOXIDE	0.0780	12.82	1.2494	0.800
NITROGEN	0.0727	13.75	1.1645	0.859
NITROUS OXIDE	0.1151	8.69	1.8437	0.542
OXYGEN	0.0831	12.03	1.3311	0.751
PROPANE	0.1175	8.51	1.8822	0.531
PROPENE (PROPYLENE)	0.1091	9.17	1.7476	0.572
SULPHUR DIOXIDE	0.1703	5.87	2.7280	0.367

Assumes a dry gas at 20° C (68° F) and a pressure of 760 mm Hg (14.70 pressure per sq inch).

Pipe Flanges and Flanged Fittings, ASME/ANSI B16.5-1988 in sizes NPS 1/2" through 24"

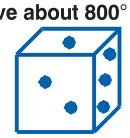


NOTES:

(a) permissible but not recommended for prolonged use above about 800° F;

(b) not to be used over 1000° F;

(c) one square centimeter is about the area of a side of a die.



SELECTED "SI" UNITS FOR GENERAL PURPOSE
SI is a Internationally standardized version of the metric system.

QUANTITY	SI UNIT	"CUSTOMARY U.S." UNIT	CONVERSION
Length (l)	Millimeter (mm)	Inch (in)	1 in = 25.4 mm 1 mm = 0.0394 in
Area (A)	Square meter (m^2)	Square inches (in^2)	1 m^2 = 1,500 in^2
Volume (Liquid)	Liter (1)	Gallon (US gal)	1 US gal = 3.79 L
Volume (Area)	Cubic meter (m^3)	Cubic inches (in^3)	1 m^3 = 61.077 in^3
Pressure (p)	Bar (assumed to be "gauge" unless otherwise stated)	Pounds per square inch (psig or psia)	1 bar = 14.72 psi
Pressure	One thousand pascals = MPa	Pounds per square inch (P/in^2 or psi)	1 MPa= 14.22 psi 1,000 psi = 70.3 Mpa 1 kp/cm^2 = 11.22 psi
Flow (mass rate)	Milligram/second (Mg/s)	Pound/hour (lb/hr)	1 mg/s = 0.00794 lb/hr 1 lb/hr = 125.98 mg/s
Flow (Liquid)	Liters per minute (1/min)	Gallons per minute (USGPM)	1 USGPM = 3.79 l/min
Force (F)	Newton (N)	Pound(f) lb(f) 1 N = 0.225 lb-ft(f)	1 lb(f) = 4.448 N
Mass (m)	Kilogram (kg)	Pound (m) lb(m) 1 lb(m) = 0.454 kg	1 kg = 2.20 lb(m)
Torque (T)	Newton-meters (N-m) lb(f)-in	Pounds (f)-inches 1 N-m = 0.74 lb-ft 1 lb-ft = 1.36 N-m	1 N-m = 8.88 lb(f)-in
Temperature (t)	Degree Celsius (°C)	Degree Fahrenheit (°F)	°C = 5/9(°F - 32) °F = 9/5 x °C + 37
Face pressure (p)	Kilogram per square centimeter (Kg/cm^2)	Pound per square inch (psi)	I kg/cm^2 = 14.22 psi I kg/mm^2 = 1,442 psi

FLANGE TIGHTNESS

The evaluation and selection of gaskets should be compared and assessed by reference to the entire flange, bolt, assembly, gasket and operating conditions system. Perfect functioning of flange joints is achieved by the combined action of these elements. (See Fig. 1) To improve the performance and integrity of bolted flange joints, emphasis should be on gasket selection, leakage behaviors, joint behaviors under pressure and temperature loads, preload specification, flange surface and bolting tools and techniques.

The tightness of a flanged joint depends on:

(1.) The gas of liquid to be contained

(2.) The pressure and temperature

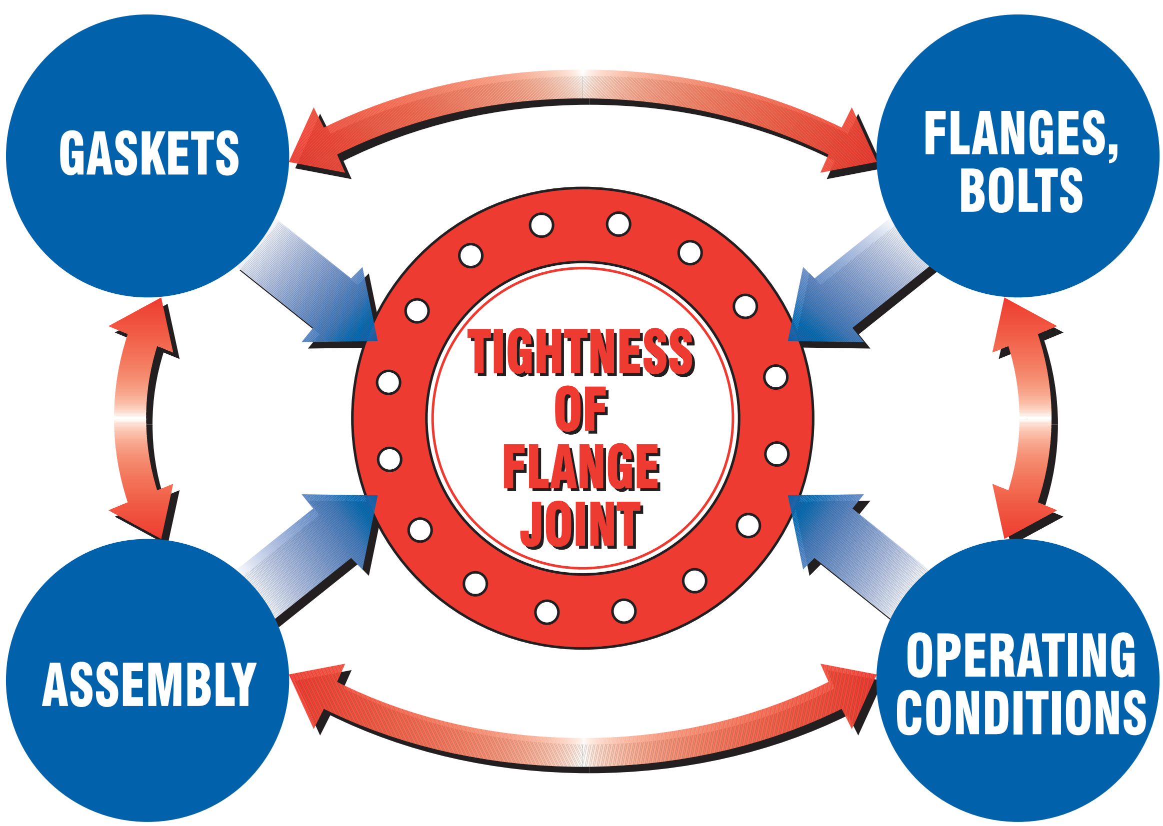
(3.) The allowable maximum leak rates

(4.) Design (stress analysis and tightness calculations

(5.) The materials of the gasket, flanges, and bolts.

(6.) Additional forces from a piping system or the apparatus

(5.) Last but not least, the assembly procedures of the parts.



First, the definition of torque when used in reference to threaded bolts is the resistance to turning or twisting force. By applying a turning motion (twisting moment) to a nut, a pull (tension) is created on the bolt; thus producing a torque/tension relationship. By turing the nut with the wrench the nut will clamp the two flanges together, as well as provide bolt tension. All wrenches produce torque. We can describe torque as the twisting moment, product of a force and wrench length, applied to a nut or bolt. The International Systems of Units [MKS] for force is newton-meter (N-m).

The symbol for torque is the Greek letter tau (τ). Torque is measured in units used. If force is in pounds and distance (wrench length) in feet, the answer is called pound-feet (lb-ft). In the metric (SI) the force factor newton is given first. A newton (abbreviated as N) is a fairly small unit of force.

4.45 newton = 1 pound
1 newton = 9.81 kilogram
1 pound = 0.225 newton

The metric (SI) uses Newton-meter [N-m].

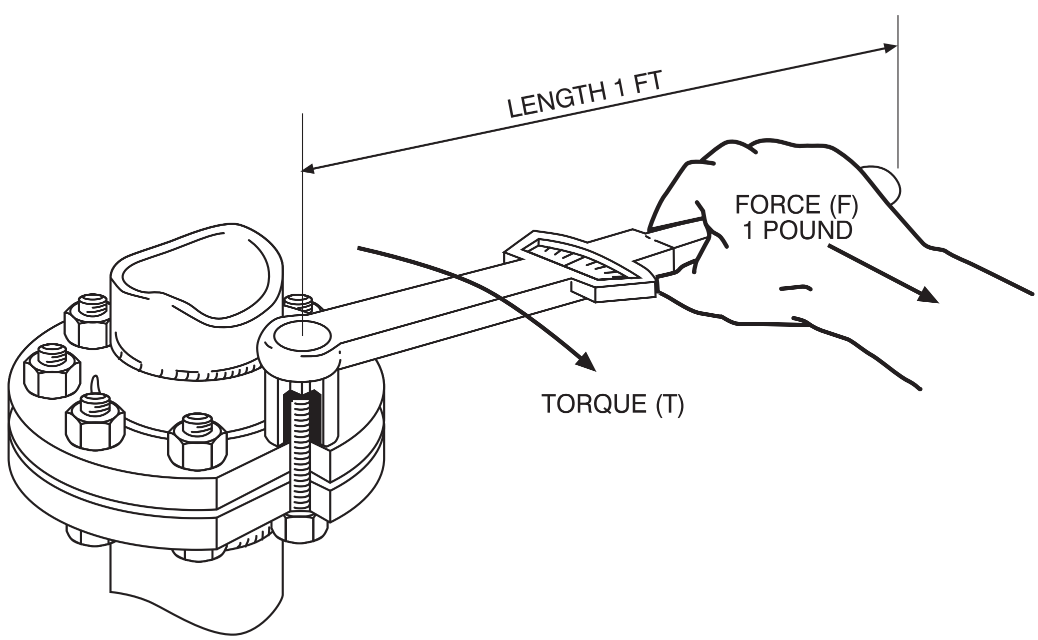
For bolting we use pounds per foot (lb-ft). However, technical and shop manuals and bolting manufacturers commonly use the foot-pound (ft-lb) term.

For example, to measure fastener torque, if 100 foot-pounds of torque is required, as wrench one foot (0.3 m) long with a pull of 100 pounds (444 N) on the end would provide it. One could also use a wrench two feet long (0.6 m) and apply 50 pounds (222 N). Torque is the product of two measurements-force and distance.

Torque = Force X Distance [lever arm]

The torque of the force (twist) is simply the force applied at a given distance from the center of the part being twisted. Force is expressed in units of pounds, ounces, kilograms, Newton.

Distance is the length of the lever arm from the axis to the line of action of the force. Distance is expressed in units of length, such as inches, feet, centimeters or meters.



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