Joint Integrity Calculations

This section is designed to enable a flange designer or gasket user to:

1. Calculate a bolt stress required for a particular gasket in a known flange.

2. Modify both gasket and bolting parameters in the relevant calculations to arrive at a suitable gasket type and dimension, and bolt pattern to suit a given application.

A Torque Guide is included to enable the user to obtain a torque figure once the bolt stress has been calculated.

See the installation section for a controlled bolting procedure in which to apply these torque values.

Gasket Type

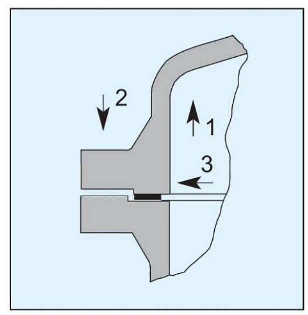
The engineer must always be aware of the abilities and limitations of the gasket types and materials. Factors such as blow out resistance, creep resistance, stress retention, recovery characteristics and cost must be considered.

Application

When determining the type of gasket to be used, design pressures and temperatures must always be considered. Media will further dictate gasket selection and what materials may or may not be utilized, ensuring chemical compatibility. Always consider special conditions such as thermal cycling, thermal shock, vibration, and erosion.

Flange Design

Attention to the flange design is critical when designing a gasket. Flange configuration, available bolt load and materials all have obvious effects on gasket selection. Flange configuration determines the style and basic dimensions of the gasket. Compatibility between flange and gasket material must be ensured, thus minimizing the possibility of galvanic corrosion.



When a joint assembly is placed in service, three basic forces become active and affect overall sealing performance.

1) END FORCE -	Which originates with the pressure of confined gases or liquids that tends to separate the flange faces.
2) GASKET LOAD -	The function of the bolting or other means which applies force upon the flange faces to compress the gasket and withstand internal pressure
3) INTERNAL PRESSURE -	Force which tends to move, permeate or bypass the gasket.

Taking the above factors into consideration, attention must be paid to the initial force applied to a joint. Firstly, the applied preload must be sufficient to seat the gasket upon the flange faces, compensating for any surface imperfections which may be present. Secondly, the force must be sufficient to compensate for the internal pressures acting against the flange assembly. i.e. the hydrostatic end force and internal pressure. Finally, the applied force must be sufficient to maintain a satisfactory residual load upon the joint assembly.

ASME Boiler and Pressure Vessel Code Calculations

Section VIII of the ASME Boiler & Pressure Vessel Code, establishes criteria for flange design and suggests values of "m" (gasket factor) and "y" (minimum gasket seating stress) as applied to gaskets. For the most part, the defined values have proven successful in actual applications. However, much confusion exists regarding these values, primarily due to a misunderstanding of the definitions of the terms and their significance in practical applications. Mandatory Appendix II, in Section VIII of the Boiler Code, requires in the design of a bolted flange connection, that complete calculations shall be made for two separate and independent sets of conditions.

Operating Conditions

Condition one (1) requires a minimum load be determined in accordance with the following equation:

$$(1) \quad Wm1 = \frac{3.14G^2P}{4} + 2b\ 3.14GmP$$

This equation states the minimum required bolt load for operating conditions and is the sum of the hydrostatic end force, plus a residual gasket load on the contact area of the gasket times a factor times internal pressure. Stated another way, this equation requires the minimum bolt load be such that it will maintain a residual unit compressive load on the gasket area that is greater than internal pressure when the total load is reduced by the hydrostatic end force.

Gasket Seating

Condition two (2) requires a minimum bolt load be determined to seat the gasket regardless of internal pressure and utilizes a formula:

$$(2) \quad Wm2 = 3.14bGy$$

The "b" in these formulae is defined as the effective gasket width and "y" is defined as the minimum seating stress in psi. For example, Section VIII of the Boiler Code suggests a minimum "y" value for a spiral wound gasket of 10,000 psi (Winter 1976 Addenda). These design values are suggested and not mandatory. The term "b" is defined as:

$b = b_o$ when $b_o \leq 1/4"$ $\quad b = 0.5\sqrt{b_o}$ when $b_o > 1/4"$

After Wm1, and Wm2 are determined, the minimum required bolt area Am is determined as follows:

$Am1 = \frac{Wm1}{Sb}$ where Sb is the allowable bolt stress at operating temperature, and

$Am2 = \frac{Wm2}{Sa}$ where Sa is the allowable bolt stress at atmospheric temperature.

Then Am is equal to the greater of Am1 or Am2. Bolts are then selected so the actual bolt area, Ab, is equal to or greater than Am.

At this point, it is important to realize the gasket must be capable of carrying the entire compressive force applied by the bolts when prestressed unless provisions are made to utilize a compression stop in the flange design or by the use of a compression gauge ring. For this reason, FLEXITALLIC's standard practice is to assume W is equal to Ab Sa.

We are then able to determine the actual unit stress on the gasket bearing surface. This unit stress Sg is calculated as follows:

$$(3) \quad Sg\ (psi) = \frac{Ab\ Sa}{.785\ [(do - .125^*)^2 - (di)^2]}$$

*Note: Based on 4.5mm (.175") thick spiral wound gasket. The “v” or Chevron shape on the gasket O.D. is not part of the effective seating width, therefore .125” is subtracted from the actual gasket O.D.

Using the unit stress we can assign construction details which will lead to the fabrication of a gasket having sufficient density to carry the entire bolt load.

ASME Boiler and Pressure Vessel Code Calculations

Gasket Seating Stress "y"

Defined as the applied stress required to seat the gasket upon the flange faces. The actual required seating stress is a function of flange surface finish, gasket material, density, thickness, fluid to be sealed and allowable leak rate.

Gasket Factor "m"

Appendix II, Section VIII, of the Boiler Code makes the statement the "m" factor is a function of the gasket material and construction. We do not agree entirely with this interpretation of "m". Actually, the gasket does not create any forces and can only react to external forces. We believe a more realistic interpretation of "m" would be "the residual compressive force exerted against the gasket contact area must be greater than the internal pressure when the compressive force has been relieved by the hydrostatic end force". It is the ratio of residual gasket contact pressure to internal pressure and must be greater than unity otherwise leakage would occur. It follows then, the use of a higher value for "m" would result in a closure design with a greater factor of safety. Experience has indicated a value of 3 for "m" is satisfactory for flanged designs utilizing Spiral Wound gaskets regardless of the materials of construction. In order to maintain a satisfactory ratio of gasket contact pressure to internal pressure, two points must be considered. First, the flanges must be sufficiently rigid to prevent unloading the gasket due to flange rotation when internal pressure is introduced. Secondly, the bolts must be adequately prestressed. The Boiler Code recognizes the importance of pre-stressing bolts sufficiently to withstand hydrostatic test pressure. Appendix S, in the Code, discusses this problem in detail.

Notations

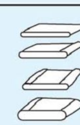
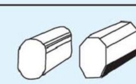
A_b	=	Actual total cross sectional root area of bolts or section of least diameter under stress; square inches
Am	=	Total required cross sectional area of bolts, taken as greater of Am1 or Am2; square inches
Am1	=	Total required cross sectional area of bolts required for operating conditions; square inches
Am2	=	Total required cross sectional area of bolts required for gasket seating; square inches
b	=	Effective sealing width; inches
b_o	=	Basic gasket seating width; inches
2b	=	Joint-contact-surface pressure width; inches
G	=	Diameter of location of gasket load reaction; inches
m	=	Gasket factor
N	=	Radial flange width of spiral wound component
P	=	Design pressure; psi
Sa	=	Allowable bolt stress at atmospheric temperature; psi
Sb	=	Allowable bolt stress at design temperature; psi
W	=	Flange design bolt load; pounds
Wm1	=	Minimum required bolt load for operating conditions; pounds force
Wm2	=	Minimum required bolt load for gasket seating; pounds force
y	=	Minimum gasket seating stress; psi
Sg	=	Actual unit stress at gasket bearing surface; psi
do	=	Outside diameter of gasket; inches
di	=	Inside diameter of gasket; inches

The ASME boiler and pressure vessel code is currently under review by the Pressure Vessel Research Council. Details of these proposed improvements, including the effects on gasket design procedures are highlighted under the PVRC Method.

ASME Boiler and Pressure Vessel Code Calculations

Gasket Materials and Contact Facings

Gasket factors (m) for Operating Conditions and Minimum Design Seating Stress (y)

Gasket Material		Gasket Factor (m)	Minimum Design Seating Stress (y) (psi)	Sketches and Notes	Seating Width (See Table)	
					Gasket Group	Column
Self-Energizing Types O-rings, metallic, elastomer, and other gasket types considered as self-sealing		0	0			
Elastomers without fabric Below 75A Shore Durometer 75A or higher Shore Durometer		 0.50 1.00	 0 200			
Elastomers with cotton fabric insertion		1.25	400		(1a), (1b) (1c), (1d), (4), (5)	
Vegetable fiber		1.75	1100			
Flexicarb products	NR SR ST	2.00 2.00 2.00	900 900 2,500		(1a) (1b)	
MRG		2.00	2,500		(1a) (1b)	
Flexpro		2.00	2,500		(1a) (1b)	
Spiral wound metal, with filler		3.00	10,000		(1a), (1b)	II
Spiral wound Style LS		3.00	5,000		(1a) (1b)	
Corrugated metal with filler or Corrugated metal jacketed with filler	Soft aluminum Soft copper or brass Iron or soft steel Monel or 4%-6% chrome Stainless steels & Nickel based alloys	2.50 2.75 3.00 3.25 3.50	2900 3700 4500 5500 6500		(1a), (1b)	
Corrugated metal	Soft aluminum Soft copper or brass Iron or soft steel Monel or 4%-6% chrome Stainless steels & Nickel based alloys	2.75 3.00 3.25 3.50 3.75	3700 4500 5500 6500 7600		(1a), (1b), (1c), (1d)	
Flat metal jacketed, with filler	Soft aluminum Soft copper or brass Iron or soft steel Monel 4%-6% chrome Stainless steels & Nickel based alloys	3.25 3.50 3.75 3.50 3.75 3.75	5500 6500 7600 8000 9000 9000		$(1a)_2$, $(1b)_2$, (1c), (1d), (2)	
Grooved metal	Soft aluminum Soft copper or brass Iron or soft steel Monel or 4%-6% chrome Stainless steels & Nickel based alloys	3.25 3.50 3.75 3.75 4.25	5500 6500 7600 9000 10100		(1a), (1b), (1c), (1d), (2), (3)	
Solid flat metal	Soft aluminum Soft copper or brass Iron or soft steel Monel or 4%-6% chrome Stainless steels & Nickel based alloys	4.00 4.75 5.50 6.00 6.50	8800 13000 18000 21800 26000		(1a), (1b), (1c), (1d), (2), (3), (4), (5)	I
Ring Joint	Iron or soft steel Monel or 4%-6% chrome Stainless steels & Nickel based alloys	5.50 6.00 6.50	18000 21800 26000		(6)	

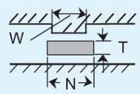
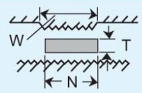
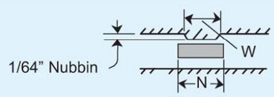
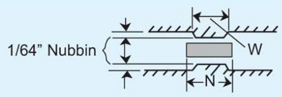
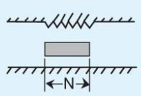
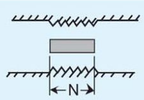
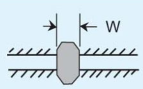
Notes:

This table gives a list of many commonly used gasket materials and contact facings with suggested design values of m and y that have generally proved satisfactory in actual service when using effective gasket seating width b given in the table on the next page. The design values and other details given in this table are suggested only and are not mandatory.

The surface of a gasket having a lap should not be against the nubbin.

ASME Boiler and Pressure Vessel Code Calculations

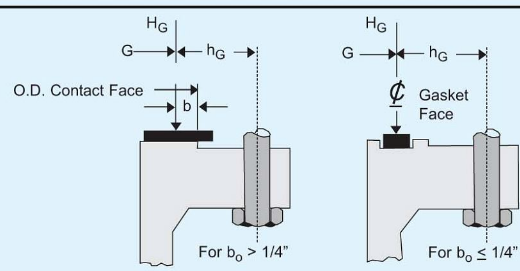
Effective Gasket Seating Width - See Note (1)

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

Effective Gasket Seating Width, b

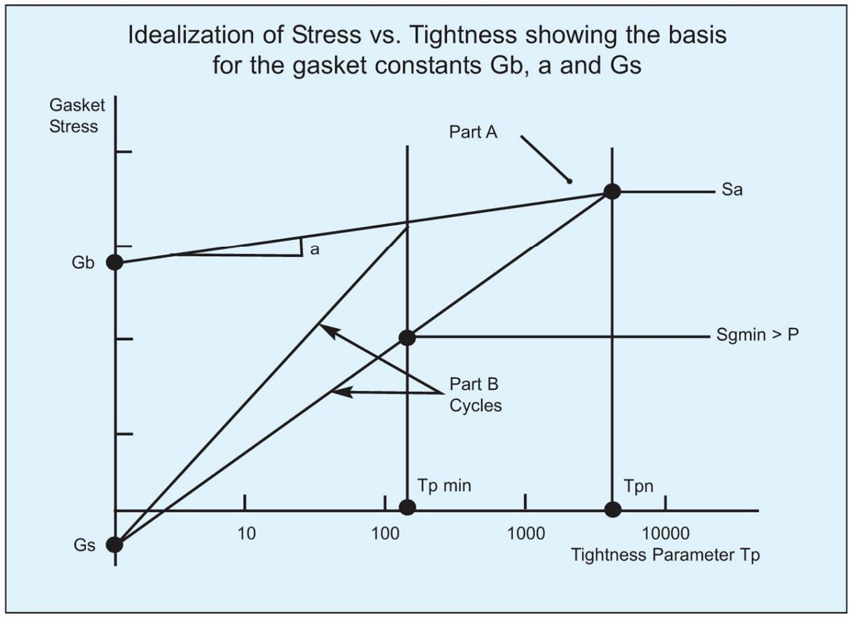
$b = b_o$, when $b_o \leq 1/4$”; $b = 0.5\sqrt{b_o}$, when $b_o > 1/4$”

Location of Gasket Load Reaction



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

PVRC METHOD



Current gasket design calculations for bolted joints such as ASME VIII, DIN 2505, etc., have many shortcomings surrounding the expected tightness and optimum operating stress levels to ensure against joint leakage. In general, current design methods only ensure that the optimum bolt load is available to seat the gasket and accommodate the hydraulic loads created by the internal pressure. Little information is given regarding the tightness of the joint in service or the optimum level of gasket stress to fulfill the legislative, environmental and company emission requirements at the source of application.

Flexitallic financially supports, and is actively involved in the research efforts of the ASME's Pressure Vessel Research Council (PVRC) to review and update current gasket design methodology. The PVRC has, through many years of research and development (involving hundreds of actual gasket tests), conceived a new philosophy that addresses the mechanisms of sealing that will benefit gasket manufacturers, vessel designers and the operators of process equipment in general. The result is a package that recommends minimum levels of gasket assembly stress to fulfill the operational requirements of the user. The new procedure is similar to the existing ASME Section VIII calculation, except it incorporates new gasket factors (to replace the traditional m & y gasket factors) that have been determined through an extensive test program.

The new gasket factors are (Gb), (a), and (Gs).

(Gb) and (a) represent the initial gasket compression characteristics and relate to the initial installation, while (Gs) represents the unloading characteristics typically associated with the operating behavior.

The PVRC method has been developed over the years using the following parameters for bolted joint designs and determining gasket constants:

1. Determine the tightness class 'Tc' that corresponds to the acceptable leak rate for the application (legislative, environmental, or company emission legislation).
   T2: Standard; represents a mass leak rate per unit diameter of 0.002 mg/sec/mm-dia.
   T3: Tight; represents a mass leak rate per unit diameter of 0.00002 mg/sec/mm-dia.
2. Select the tightness constant that corresponds to the chosen tightness class
   C = 1.0 for tightness class T2 (Standard).
   C = 10.0 for tightness class T3 (Tight).
3. Select the appropriate gasket constants (Gb), a, and (Gs) for the gasket style and material, (see table, page 44).
4. Determine gasket parameters (N), (b_o), (b), and (G) as per table (page 41).
5. Gasket seating area, $Ag = 0.7854(OD^2\text{-}ID^2)$.
6. Hydraulic area, $Ai = 0.7854G^2$
7. Minimum required tightness, $Tpmin = 0.1243 \times C \times P_d$, P_d = Design Pressure
8. Assembly Tightness $T_{pa} = 0.1243 \times C \times P_t$, P_t = Test Pressure (Typically 1.5 x P_d)
9. Tightness Parameter Ratio, Tr = Log(Tpa)/Log(Tpmin)
10. Gasket Operating Stress, $Sm1 = Gs[G_b/Gs \times Tpa^a]^{1/Tr}$

PVRC Method

11. Gasket Seating Stress, $Sm2 = Gb\ (Tpa^{a})\ /\ (e \times 1.5) - Pd\ (A_i/Ag)$
    e = 0.75 for manual bolt up
    e = 1.0 for hydraulic tensioners & ultrasonic
12. Design factor, Mo = the greater of $Sm1/\ P_d$ or $Sm2\ /\ P_d$
13. Design Bolt load, $Wmo = Ag \times Smo + Ai \times P_d$
    Smo is the greater of Sm1, Sm2, 2P, S_L

S_L = A minimum permitted value of operating gasket stress equal to 90% of the minimum gasket stress in the test that determined the gasket constants. It is 6.21 MPa (900 psi) for the standard and soft ROTT test procedures, and 10.3 MPa (1500 psi) for the hard gasket procedure.

Note: Iterative method can be used for more exact results (Sm1 - Sm2).

Note: PVRC and ASME continue to refine data reduction techniques, and values are therefore subject to further review and revisions.

Gasket Factors

Type	*Material*	*Gb (psi)*	*a*	*Gs (psi)*
Spiral Wound 'LS'	SS/Flexicarb	598	0.385	0.03
(Class 150 & 300)	SS/PTFE	698	0.249	0.00128
Spiral Wound	SS/Flexicarb	2300	0.237	13
(Class 150 to 2500)	SS/Flexite Super	2600	0.230	15
	SS/Thermiculite 735	474	0.448	9.82
	SS/Thermiculite 835	2,120	0.190	49
MRG	SS/Flexicarb	813	0.338	0.2
Carrier Ring	SS/Flexicarb	1251	0.309	11
Flexpro	SS/Flexicarb	387	0.334	14
	SS/Thermiculite 845	1780	0.169	1080
Sheet Gaskets	Flexicarb	1047	0.354	0.07
(Class 150 to 300)	Flexicarb NR	818	0.347	0.07
	SF 2401	290	0.383	2.29
	SF 3300	2360	0.190	50.25
	Sigma 500	4	0.804	0.115
	Sigma 511	209	0.356	0.00498
	Sigma 522	472	0.250	0.037
	Sigma 533	115	0.382	0.000065
	Thermiculite 715	1031	0.243	9.68
	Thermiculite 815	1906	0.2	456
Corrugated Gasket	Soft Iron	3000	0.160	115
	Stainless Steel	4700	0.150	130
Soft Copper	1500	0.240	430	
Metal Jacketed	Soft Iron	2900	0.230	15
	Stainless Steel	2900	0.230	15
	Soft Copper	1800	0.350	15
Metal Jacketed Corr.	Soft Iron	8500	0.134	230