10.6. RING-JOINT

Also called API Ring (Figure 2.11). Both flanges have channels with walls in a 23º angle. The gasket is made out of solid metal with an oval or octagonal profile. The octagonal profile is more efficient.

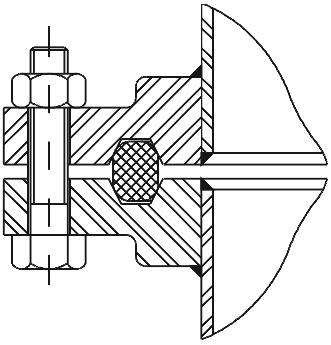

Figure 2.11

11. THE NEW GASKET CONSTANTS

Traditionally calculations for piping flanges and gaskets use values and formulas recommended by the American Society of Mechanical Engineers (ASME) Section VIII of Pressure Vessel and Boiler Code.

The ASME Code recommends values for minimum seating stress "y" and the maintenance factor "m" for various styles of gaskets. These values were determined from experimental work in 1943.

With the development of materials like Flexible Graphite, PTFE and the replacement of asbestos-based gaskets for other materials, it became necessary to determine the values of "m" and "y" for those new materials. In 1974, the Pressure Vessel Research Committee (PVRC) initiated an experimental program to better understand the behavior of a gasket in a flanged joint since there was no analytical theory that allowed determination of this behavior. This work was sponsored by thirty institutions among them ASME, American Petroleum Institute (API), Fluid Sealing Association (FSA), and American Society for Testing Materials (ASTM) among others.

The University of Montreal, Canada, was contracted to conduct the tests, and present their results and suggestions.

In the course of the research the impossibility of determining the values of "m" and "y" for the new materials was verified and it was also ascertained that the values for the traditional materials were not consistent with the experimentally obtained results.

The researchers then opted to develop, starting from an experimental basis, a methodology for gasket calculation that was coherent with the practical results. In this section this new form of calculation is demonstrated.

It is appropriate to point out that the standardization organizations (like ASME, API, ANSI, etc.) have not yet officially published a method to calculate gaskets using the

New Gasket Constants. There is a proposal put forth by the researchers now being discussed by the ASME.

11.1. GASKET TESTING

The gaskets chosen for testing were the most represented in industry.

- Metallic gaskets flat and corrugated in low carbon steel, soft copper and stainless steel.
- Metal o-rings.
- Sheet Packing: NBR and SBR binders with asbestos, aramid and glass fibers.
- Flexible Graphite and PTFE sheets.
- Spiral Wound gaskets in stainless steel asbestos, non-asbestos, flexible graphite and PTFE fillers.
- Double Jacketed carbon and stainless steel with asbestos and non-asbestos fillers.

The gaskets were tested in the device shown in Figure 2.12.

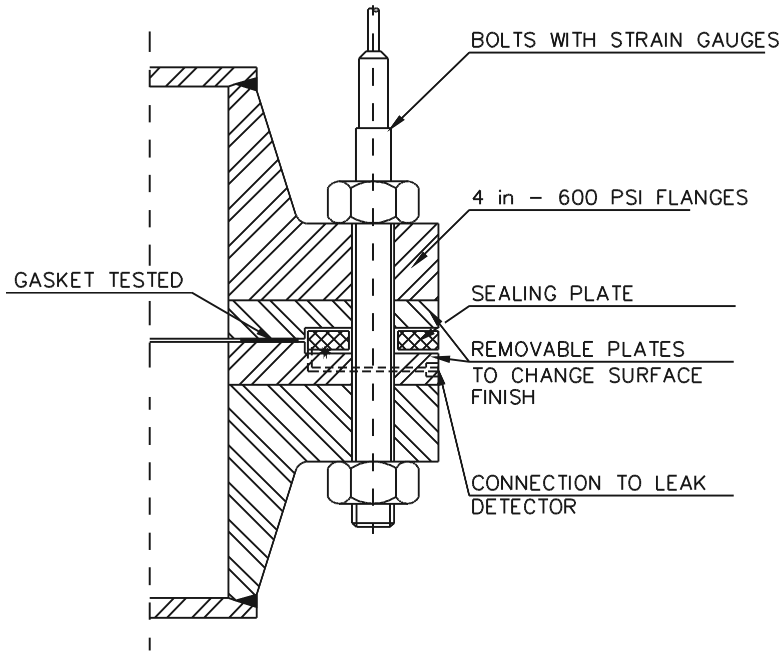

Figure 2.12

The tests were conducted under three pressures: 100, 200, and 400 psi with nitrogen, helium, kerosene and water. Sequence of steps followed during the test:

- The initial stress - part A of the chart figure 2.13 -the gasket is tightened until deflection Dg keeping Sg constant, pressure is increased to 100 psi and the leak rate Lr is measured.

- The same procedure is repeated for 200 and 400 psi.
- The operating stress - part B of the Figure 2.13 - with pressure constant (100, 200 and 400 psi) Sg is decreased at regular intervals, deflection, Dg, and leak rate, Lr, are measured.
- This procedure is repeated until Lr exceeds the leak detector measuring capacity.
- Keeping the pressure constant, Sg is increased measuring Dg and Lr at regular intervals.

The Figure 2.14 shows an example of the fluid pressure as a function of mass leak rate for each value of gasket stress.

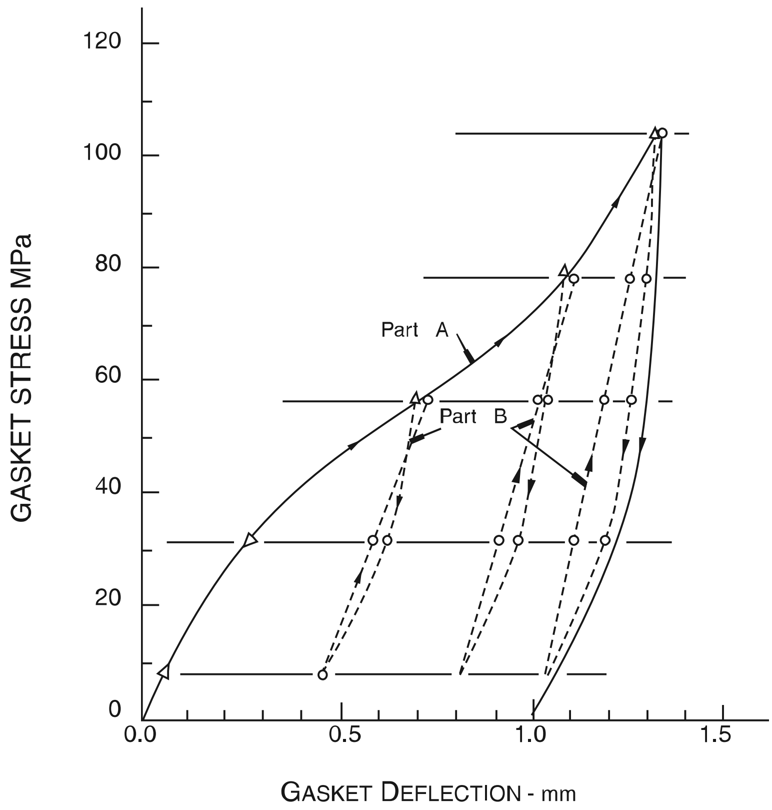

Figure 2.13

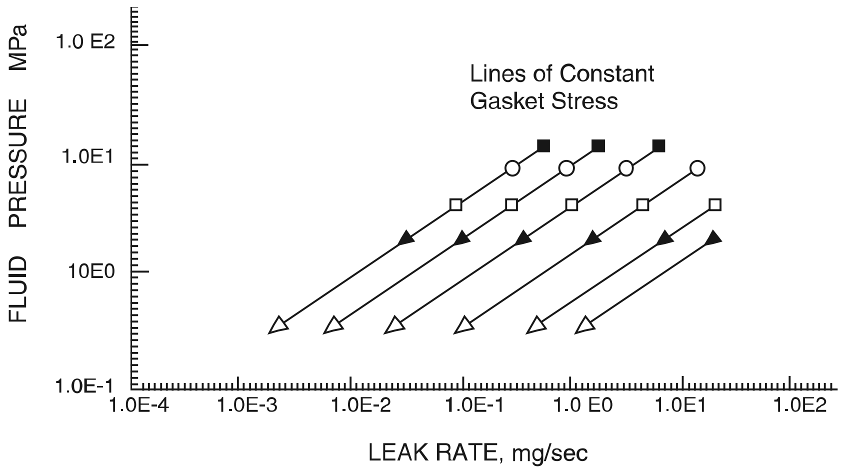

Figure 2.14

From experiments conducted at the University of Montreal various conclusions were reached:

- The gaskets demonstrate similar behavior no matter what style, or material they are made of.
- The tightness of a gasket is a direct function of the seating stress.
- The non-dimensional Tightness Parameter, Tp, was suggested as the best way to represent the behavior of the diverse styles of gaskets and materials.

$$T_p = (P/P^*) \times (L_{rm}^* / (L_{rm} \times D_t))^a$$

where:

$0.5 < a < 1.2$ being 0.5 for gases and 1.2 for liquids

P = Fluid Pressure (MPa)

P^* = Atmospheric Pressure (0.1013 MPa)

L_{rm} = mass leak rate per unit of diameter (mg/sec-mm)

L_{rm}^* = mass leak rate with 1 mg/sec-mm reference. Normally taken for a reference gasket with a 150 mm outside diameter.

D_t = gasket outside diameter (mm)

The Tightness Parameter can be defined as the pressure necessary to create a certain level of leakage. For example a Tp equal to 100 signifies that a pressure of 100 atmospheres (1470 psi or 10.1 MPa) is necessary to create a leak of 1mg/sec in a gasket with an external diameter of 150 mm (6 in).

By plotting on a scale log-log the experimental values of the Tightness Parameter in function of the Gasket Stress we have the chart in Figure 2.15.

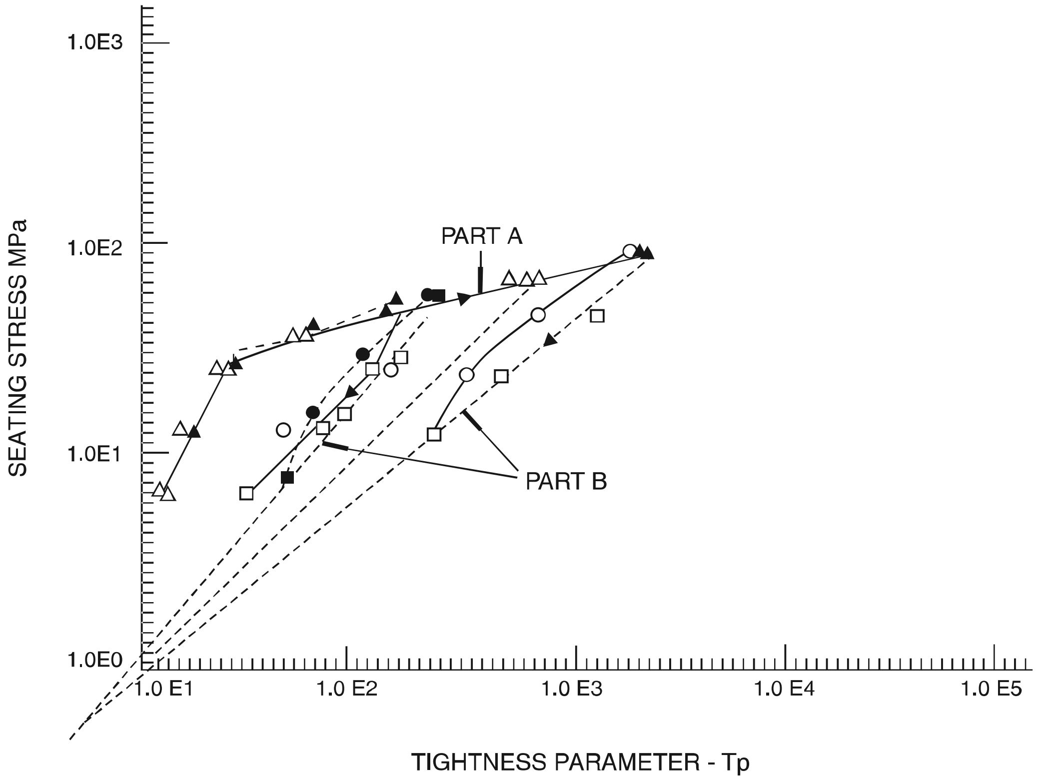

Figure 2.15

From the chart we can establish the Gasket Constants, obtained experimentally, that determine the gasket behavior. The constants are:

G_b = intersection point of the seating stress line (part A of the test)
a = inclination of the seating stress line
G_s = focal point of the gasket stress relief lines (part B of the test)

In Table 2.5 are the constants obtained in the PVRC study. The ASTM is developing a method to determine the gasket constants.

Table 2.5
Gasket Constants

Gasket Material	Gb (MPa)	a	Gs (MPa)
Compressed Asbestos Sheet			
1/16" thick	17.240	0.150	0.807
1/8" thick	2.759	0.380	0.690
Compressed Non-Asbestos Sheet 1/16" (1.6 mm) thick			
Teadit NA 1001	0.938	0.45	5 E-4
Teadit NA 1080			
Teadit NA 1081			
Teadit NA 1100	0.903	0.44	5.4 E-3
Expanded PTFE Sheet Teadit 24SH 1/16" thick	2.945	0.313	3 E-4
Expanded PTFE Cord Teadit 24B	8.786	0.193	1.8 E-14
Tealon® Restructured PTFE Sheet			
TF 1570	244	0.31	1.28×10^{-2}
TF 1580	114	0.447	1.6×10^{-3}
TF 1590	260	0.351	6.3
Flexible Graphite - Graflex®			
Monolithic - style TJB	6.690	0.384	3.448 E-4
Tanged Core - style TJE	9.655	0.324	6.897 E-5
Stainless Steel insert - style TJR	5.628	0.377	4.552 E-4
Polyester insert – style TJP	6.690	0.384	3.448 E-4
Spiral Wound Gasket Graflex® filled			
Without inner ring (style 913)	15.862	0.237	0.090
With inner ring (style 913 M)	17.448	0.241	0.028
Spiral Wound Gasket PTFE filled			
Without inner ring (style 913)	31.034	0.140	0.483
With inner ring (style 913 M)	15.724	0.190	0.462
Jacketed Gasket Graflex® filled			
Flat (style 923)	20.000	0.230	0.103
Corrugated (style 926)	58.621	0.134	1.586
Flat Metal Gasket (style 940)			
Aluminum	10.517	0.240	1.379
Copper or Brass	34.483	0.133	1.779

In Figure 2.16 the chart of a spiral wound gasket with Flexible Graphite filler.

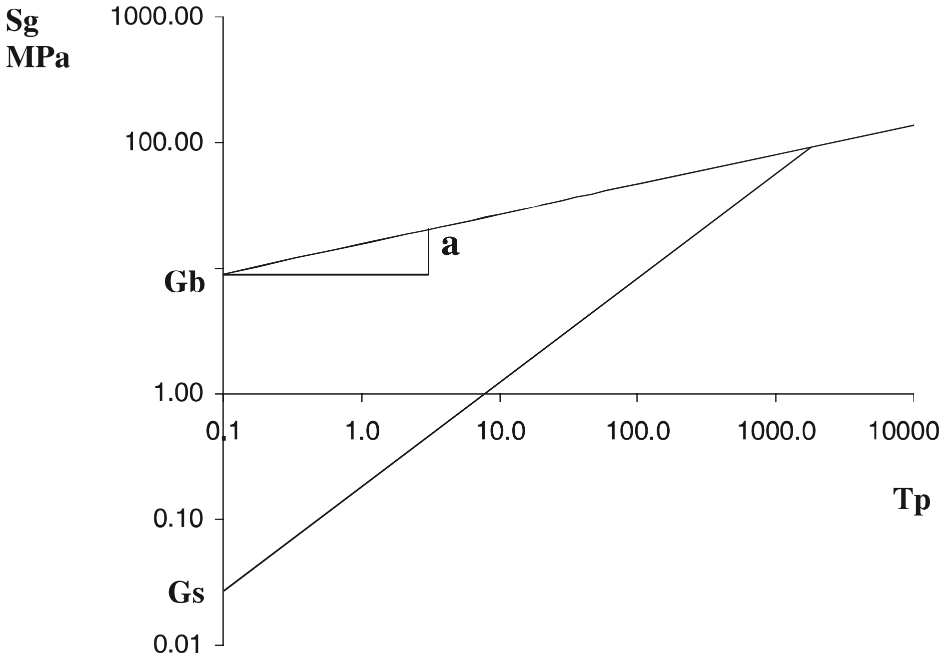

Figure 2.16

11.2. TIGHTNESS CLASS

One of the most important concepts introduced by the PVRC studies is of the Tightness Class. As it is not possible to have a perfect seal as suggested by the factors "m" and "y", the PVRC has proposed the introduction of the Tightness Class corresponding to three levels of maximum leak rates.

Table 2.6
Tightness Class

Tightness Class	Leak Rate (mg / sec-mm)	Tightness Constant - C
Air, Water	0.2 (1/5)	0.1
Standard	0.002 (1/500)	1.0
Tight	0.000 02 (1/ 50 000)	10.0

It is possible to have a classification of the different fluids by tightness class, taking in consideration the danger to the environment, fire hazards, explosions, etc.

Environmental authorities have not published such classification at the time of the publication of this book.

We can visualize the proposed values with a practical example. A spiral wound gasket with dimensions per ASME B16.20 for an ASME B16.5 4 in class 150-psi flange, with a standard tightness class, leaks 0.002 mg/sec-mm. The overall leak of this gasket is:

Leak Rate (L_{rm}) = 0.002 x outside diameter

L_{rm} = 0.002 x 149.4 = 0.2988 mg/sec = 1.076 g/hour

As mass leak rate is difficult to visualize, below are some practical tables for a better understanding.

Table 2.7
Volumetric Equivalent

Fluid	Volumetric Equivalent	
	Mass - mg / seg	**Volume - l / h**
Water	1	0.036
Nitrogen	1	3.200
Helium	1	22.140

Table 2.8
Bubble Equivalent

Mass Leak Rate	Volumetric Equivalent	Bubble Equivalent
10^{-1} mg / sec	1 ml each 10 seconds	Constant flow
10^{-2} mg / sec	1 ml a each 100 seconds	10 bubbles per second
10^{-3} mg / sec	3 ml per hour	1 bubble per second
10^{-4} mg / sec	1 ml each 3 hours	1 bubble each 10 seconds

11.3. JOINT ASSEMBLY EFFICIENCY

Studies show a great variation in the bolt load of each bolt even in situations where the torque is applied in a controlled form. The PVRC has suggested the introduction of an Assembly Efficiency Factor as shown in Table 2.9.

Table 2.9
Assembly Efficiency

Tightening Method	Assembly Efficiency "Ae"
Power impact, lever or striker (manual or power) wrench	0.75
Accurately applied torque (± 3 %)	0.85
Simultaneous multiple application of direct stud tension	0.95
Direct measurement of stud stress or strain	1.00

11.4. BOLT LOAD USING THE PVRC PROPOSED PROCEDURE

The proposed PVRC Method of bolt loading design calculations, to make the calculation easier, has several simplifications, which can generate values with variations when compared with exact values. These variations are shown in the Paper " The Exact Method", presented by Mr. Antonio Guizzo, Teadit´s Technical Director, at the 6th Annual Fluid Sealing Association Technical Symposium, Houston, TX, October 1996. The same author has presented at the Sealing Technical Symposium, Nashville, TN, April 1998, a paper showing the actual experimental results compared with the expected values from the PVRC proposed procedure. Copies of both papers can be obtained from Teadit at the address shown at the end of this book.

Important note: The ASME has not approved this method, proposed by the PVRC. Its use must be carefully applied in order to avoid personal and material injuries deriving from uncertainties that still exist in its application.

- Determine from Table 2.5, the constants G_b, a, e G_s for the gasket to be used.
- Determine from Table 2.6, the Tightness Class and the Tightness Constant, C.
- Determine from Table 2.9, the Assembly Efficiency, Ae, in accordance with the Tightening Method to be used.
- Calculate the Gasket Stress Area, A_g
- Determine from the ASME material tables the bolts allowable stress at the room temperature: Sa
- Determine from the ASME material tables the bolts allowable stress at the operating temperature: Sb
- Calculate the area affected by the action of the fluid pressure (Hydrostatic Area), A_i, according the ASME Code:

  $A_i = (\pi / 4) G^2$
  $G = OD - 2b$
  $b = .5 (b)^{0.5}$ or $b = b_o$ if b_o less than ¼" (6.4 mm)
  $b_o = N / 2$

  where G is the Effective Diameter per the ASME Code. See Tables 2.1 and 2.2.

- Calculate the Minimum Tightness Parameter, T_{pmin};

  $T_{pmin} = 18.0231 \, C \, P_d$

  where C is the Tightness Constant and P_d is the Design Pressure.

- Calculate the Assembly Tightness Parameter, T_{pa}. This value of T_{pa} must be reached during the seating of the gasket, to assure that the value of T_p, in operation be equal or higher than T_{pmin}.

$$T_{pa} = X\ T_{pmin}$$

were $X >= 1.5\ (S_a / S_b)$

where S_a is the Bolt Allowable Stress at the room temperature and S_b is the Bolt Allowable Stress at the operating temperature.

- Calculate the Tightness Parameter ratio:

$$T_r = \text{Log}\ (T_{pa}) / \text{Log}\ (T_{pmin})$$

- Calculate the minimum Operating Gasket Stress, S_{ml}. This pressure is required to resist the Hydrostatic End Force and to maintain on the gasket sufficient compression to assure the required minimum tightness, T_{pmin}.

$$S_{ml} = G_s\ [(G_b / G_s)\ (\ Tpa\ )^a\ ]^{\ (1/Tr)}$$

- Calculate the minimum Gasket Assembly Stress, S_{ya}:

$$S_{ya} = (G_b / Ae)\ (\ T_{pa}\ )^a$$

where Ae is the Assembly Efficiency, from Table 2.9

- Calculate the Seating Design Gasket Stress, S_{m2}:

$$S_{m2} = [(\ S_b / S_a\ )(\ S_{ya} / 1.5\ )] - P_d\ (A_i / A_g)$$

where A_g is the gasket contact area with the flange sealing surface.

- Calculate the minimum Bolt Load, W_{mo}:

$$W_{mo} = (\ P_d\ A_i\ ) + (\ S_{mo}\ A_g\ )$$

where S_{mo} is the larger of S_{m1}, S_{m2} or 2 P_d.

- Calculate the minimum Bolt Stress Area, A_m:

$$A_m = W_{mo} / \ S_b$$

- Number of bolts:

The actual Bolt Stress Area, A_b, must be equal or larger than A_m

11.5. EXAMPLE OF CALCULATION BY THE PVRC METHOD

A Spiral Wound Gasket with a Nominal Diameter 6 inches, Pressure Class of 300 psi, dimensions per Norma ASME B16.20, with stainless steel and Flexible Graphite and Carbon Steel guide ring. Flange with 12 bolts of 1 inch diameter in ASTM AS193-B7.

- Design Pressure: P_d = 2 MPa (290 psi)
- Test Pressure: P_t = 3 MPa (435 psi)
- Design Temperature: 450º C, (842 F)
- Bolts ASTM AS 193-B7, Allowable Stresses:
- Room temperature: Sa = 172 MPa
- Operating temperature: Sb = 122 MPa
- Quantity: 12 bolts
- From Table 2.5:

  G_b = 15.862 MPa
  a = 0.237
  G_s = 0.090 MPa
- Tightness Class: standard, L_{rm} = .002 mg/sec-mm
- Tightness Constant: C = 1
- Bolting with a Torque Wrench: Ae = 0.75
- Gasket Contact Area, A_g:

  $A_g = (\pi/4)\,[(od - 3.2)^2 - id^2] = 7271.390$ sqmm
  OD = 209.6 mm
  ID = 182.6 mm

- Hydrostatic Area. A_i :

  $A_i = (\pi/4)\ G^2 = 29711.878$ sqmm
  $G = (OD - 3.2) - 2b = 194.50$ mm
  $b = b_0 = 5.95$ mm
  $b_o = N/2 = ((OD - 3.2) - ID)/4 = 5.95$ mm

- Minimum Tightness Parameter:

  $T_{pmin} = 18.0231\ C\ P_d = 36.0462$

- Assembly Tightness Parameter:

  $T_{pa} = X\ T_{pmin} = 1.5\ (172 / 122)\ 36.0462 = 76.229$

- Tightness Parameter ratio:

  $T_r = \text{Log}\ (T_{pa}) / \text{Log}\ (T_{pmin}) = 1.209$

- Minimum Gasket Assembly Stress:

$$S_{ml} = G_s \, [(G_b / G_s) (T_{pa})^a]^{1/Tr} = 15.171 \text{ MPa}$$

- Gasket Assembly Stress:

$$S_{ya} = [Gb/Ae] (T_{pa})^a = 59.069 \text{ MPa}$$

- Seating Design Gasket Stress:

$$S_{m2} = [(S_b / S_a)(S_{ya} / 1.5)] - P_d (A_i / A_g) = 19.759 \text{ MPa}$$

- Minimum Bolt Load:

$$W_{mo} = (P_d A_i) + (S_{mo} A_g)$$

where S_{mo} is the larger value of

$S_{m1} = 15.171$
$S_{m2} = 19.759$
$2 P_d = 4$

$$W_{mo} = (P_d A_i) + (S_{mo} A_g) = 203\,089 \text{ N}$$

12. GASKET MAXIMUM SEATING STRESS

Sections 4 and 11 of this Chapter show how to calculate the minimum bolt load to assure a good sealing. However, as the PVRC studies have shown the more tightened is the gasket the better is the sealing. If the gasket is installed with the maximum possible stress, the sealability will also be the best possible for the operating conditions.

Gaskets damaged by excess torque are also a very frequent problem. For all gasket styles it is possible to calculate the maximum seating stress, which is the maximum allowable by the gasket without damaging it.

12.1. GASKET MAXIMUM STRESS CALCULATION PROCEDURE

This procedure can be used to calculate the gasket maximum seating stress.

- Calculate the Gasket Contact Area, A_g.