DETERMINATION OF DESIGN GASKET ASSEMBLY STRESS WITH THE NEW CONSTANTS

- EXACT METHOD -

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ABSTRACT
The paper proposes and describes a direct approach for calculating the assembly stress to be applied to a gasket using the new gasket constants concept. It also proposes the introduction of the exponent “d” for calculation of the Tightness Parameter as a fourth constant in addition to “Gb”, “a” and “Gs” and the utilization of the actual gasket diameter in the calculation of the same Tightness Parameter. In order to understand the influence of these parameters on the resultant bolted flange connections design values, comparisons are made between these individual proposals and the current calculation procedure. The Exact Method including all the previous parameters is described and a comparison of the resultant bolted flange connections design values is made against the current Convenient approach.
The Exact Method allows one to obtain the actual gasket assembly load values according to the new gasket constants concept and therefore leads to the design of optimum bolts and flanges. The method is structured in a straightforward sequence and therefore is easy to understand. It also allows the design engineer to make a real analysis of his design.

INTRODUCTION

The PVRC task force, working to determine the new design procedure for bolted flange connections, has done a great job in developing a theory based on extensive experimental work, but they are still striving to determine an adequate way to establish a calculation procedure, using the concepts behind the new gasket constants.

The current draft of the new code describes the Convenient Method for doing a calculation. The results obtained by using this procedure are different from the ones obtained with the Flexible Method, an alternative calculation procedure that requires iteration to find what would be the lowest and therefore the optimum bolt load. As in some situations this iterative procedure shows no convergence, attempts are being made to modify the Convenient Method so that it would result in design bolt load figures closer to the optimum.

This is happening basically because of the approach that was taken when the calculation development began. At that time it was decided to develop a procedure, using the new constants, that would be similar to the current one in relation to the calculation sequence; in this way mixing the new concepts with the traditional procedure. Although this may be a desirable feature, it has brought together the need for several assumptions and for the introduction of correction factors that are not supported by the theory developed, in an attempt to obtain the actual figures for the assembly stress to be applied on the gasket.

The other reason, i.e. to avoid the utilization of interactions in the calculations shown with the introduction of the Convenient Method, is no longer valid since nowadays almost everybody uses a computer to do this kind of work and many calculators already have an equation solver built in them.

Also, the simplifications of using the fixed exponent of 0.5 and of considering all the gaskets with 150 mm diameter in the Tightness Parameter calculation does introduce errors in the design but does not contribute much to reduce the calculation effort.

This paper proposes and describes a direct approach for calculating the assembly stress to be applied on the gasket using the new constants concept. It proposes also the introduction of "d" the Tp exponent as a fourth constant and the utilization of the actual gasket diameter in the Tp calculation. Comparisons are made between these proposals and the current calculation procedure.

As you go through the text you will notice that the units are not consistently within the ISO System of Units. This was deliberately done in order to have a direct comparison within figures appearing in the paper and figures appearing in the references so that no conversions are necessary.

THE DIRECT APPROACH

The direct approach for calculating the design gasket assembly load and then the design bolt load, uses as the basic equation the balance of forces involved in a bolted flange connection that is:

$$Sga = Sgmin + P(Ai/Ag) \qquad \text{(eq. 1)}$$

where: Sga = Gasket assembly stress,

Sgmin = Minimum gasket stress under operating conditions,

P = Internal fluid pressure,

Ai = (3.14/4)Gi^2 = Effective internal pressure area (Gi being the internal gasket diameter in contact with the flange),

Ag = 3.14(Go-N)N = Gasket contact area.

From the new gasket constants concept (see Fig.1) it is known that:

$$Sga = Gb(Tpa)^a \quad \text{(eq. 2)}$$

and

$$Sgmin = Gs(Sga/Gs)^{(\log Tpmin/\log Tpa)} \quad \text{(eq. 3)}$$

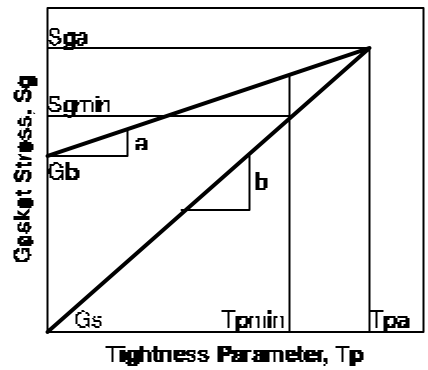

Fig.1: New gasket constants

Entering with equations 2 and 3 in equation 1 is possible to obtain an expression that is a function of Tpa. It can be written as follows:

$$f(Tpa) = Gb(Tpa)^a - Gs\{[Gb(Tpa)^a]/Gs\}^{(\log Tpmin/\log Tpa)} - P(Ai/Ag) = 0 \quad \text{(eq. 4)}$$

By knowing the operational conditions it is possible to determine Tpmin, Ai and Ag and then solve equation 4 to obtain Tpa, the tightness parameter required at the assembly stage.

Eq. 4 is solved by iteration using the equation solver present in all popular spreadsheets and in many advanced calculators.

Knowing Tpa, it is possible to go back to equations 2 and 3 and determine the assembly stress Sga and the minimum operational gasket stress, Sgmin.

The design gasket assembly load or the design bolt load is then determined by:

$$Wmo = Sga\ Ag \quad \text{(eq. 5)}$$

Figures 2 and 3 show comparisons of design assembly gasket loads as calculated by the Direct method for two extreme conditions, in comparison with results obtained for the same set of operational conditions using the Convenient and the Flexible methods and also comparing with a modification of the Convenient method, also being studied.(Ref.3).

The operational conditions used in the calculation are:

Design pressure, P	1000 psi	Tightness class, Tc	1
Gasket outside diameter, Go	24 in	Assembly efficiency, Ae	1
Gasket width, N	1 in		

Gasket constants:	Gb	a	Gs
Flat soft copper 1/16”	5000 psi	0.133	258 psi
SWG-Graphite-#300	2300 psi	0.237	13 psi

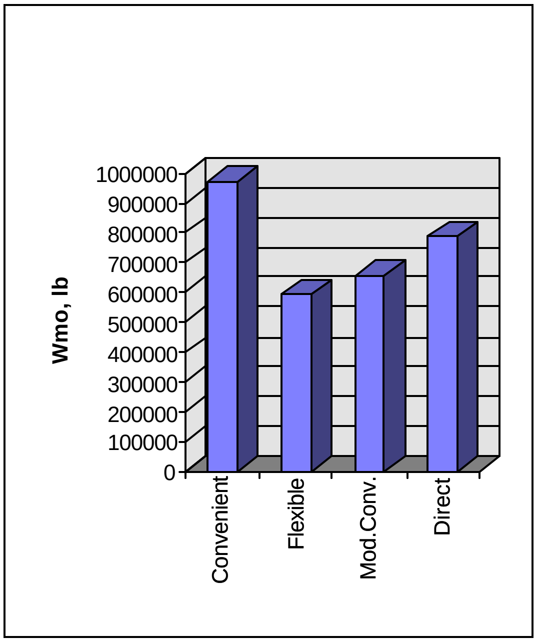

Fig. 2: Design gasket assembly load
Flat Soft Copper 1/16”

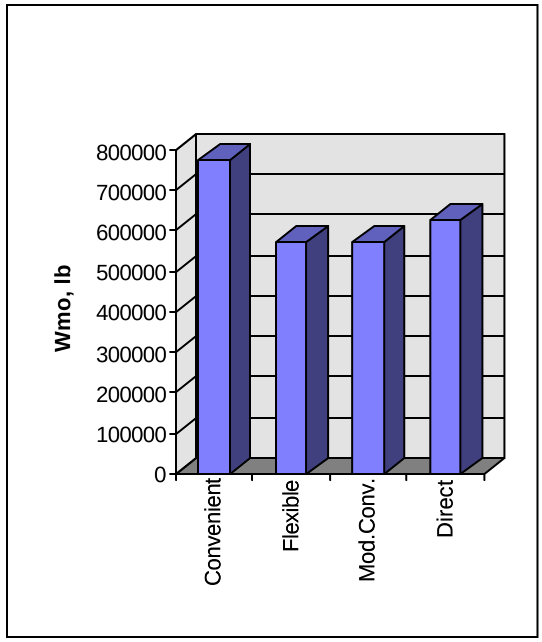

Fig. 3: Design gasket assembly load
SWG Graphite filled #300

The computed results obtained by the various calculation methods are shown in Tables 1 and 2.

Method	Tpmin	Tpa	Sga (Sya)	Sgmin (Sm1)	Wmo
Convenient	124.3	186	10,022	7,546	971,346
Flexible	124.3	10,633	17,160	2,290	591,579
Mod.Conv.	124.3	2,081	13,813	3,182	656,073
Direct	124.3	349	11,147	5,632	786,668

Table 1: Computed values for Flat Soft Copper 1/16”

Method	Tpmin	Tpa	Sga (Sya)	Sgmin (Sm1)	Wmo
Convenient	124.3	186	7,941	4,828	775,000
Flexible	124.3	2,804	15,095	946	570,638
Mod.Conv.	124.3	2,025	13,975	1,082	570,638
Direct	124.3	2,276	9,006	3,456	629,547

Table 2: Computed values for SWG Graphite filled, #300

Looking at the figures shown in these tables, it is possible to observe the discrepancies of values between the different methods and also their difference from the actual figures computed by the direct approach.

There is a big effort being spent trying to optimize results and to define the bolt load calculation procedure. The need for all this effort was generated by the initial approach of trying to mix the concepts of the new gasket constants with the calculation sequence used with the traditional “m” and “y” factors. This can be avoided if the Direct calculation procedure, as previously shown, is adopted.

Although some years ago it could be difficult, the iterative calculation is currently easy to do and should be preferred instead of the utilization of difficult-to-demonstrate factors and mainly due to the exact and therefore optimized results that are obtained with it.

There is no need to be concerned about oversized flanges due to overestimation of the load or about undesirable leakage due to underestimation of the gasket load.

THE ACTUAL EXPONENT FOR Tp CALCULATION

The procedure for obtaining the new gasket constants is rather time consuming and when a new material is to be tested, it is advisable to determine in advance the relationship between leakage and internal pressure for that specific material. In this way, it is possible to anticipate the magnitude of the leakage to be measured during the determination of the constants, and to better plan the execution of the test.

In the global context the determination of this relationship is not time consuming and it allows us to determine the actual exponent “d” for the calculation of the Tightness Parameter Tp.

During the determination of the constants for a non-asbestos compressed sheet gasket material, two sets of constants were obtained. One set of constants using the exponent 0.5 as recommended by the current procedure and another using the actual exponent as determined experimentally. The results are shown in Table 3.

Constant	d = 0.5	d = 0.6
Gb	318	381
a	0.57	0.49
Gs	0.025	0.090

Table 3: Gasket constants (psi) with standard and actual Tp exponents.

Using these two sets of constants and the same operational conditions as in the previous example a design gasket assembly load calculation was made for various internal pressures. The results are shown in Tables 4 and 5.

Pressure psi	Tpmin 0.1	Tpmin 1	Tpmin 10	Tpa 0.1	Tpa 1	Tpa 10	Sgmin 0.1	Sgmin 1	Sgmin 10	Sga 0.1	Sga 1	Sga 10
125	1.8	17.5	175	4	21	189	3.7	954	4239	741	1691	4977
250	3.5	35.0	350	16	49	396	7.3	1081	5673	1482	2555	7147
500	7.0	70.0	700	65	125	850	11.4	1119	7442	2960	4068	10391
1000	14.0	140.1	1401	269	376	1896	16.9	1070	9500	5914	6967	15397
2000	28.0	280.2	2802	1105	1301	4485	24.5	1009	11685	11818	12803	23479

Table 4: Design values computed with actual Tp exponent
and corresponding set of constants.

Pressure psi	Tpmin 0.1	Tpmin 1	Tpmin 10	Tpa 0.1	Tpa 1	Tpa 10	Sgmin 0.1	Sgmin 1	Sgmin 10	Sga 0.1	Sga 1	Sga 10
125	1.6	15.5	155	4	18	164	0.5	918	5080	738	1656	5818
250	3.1	31.1	311	15	41	338	2.5	1144	7301	1477	2619	8775
500	6.2	62.2	622	50	96	704	5.8	1346	10382	2954	4295	13330
1000	12.4	124.3	1243	169	250	1492	10.9	1498	14560	5908	7395	20457
2000	24.9	248.6	2486	569	712	3243	18.7	1621	20054	11813	13415	31848

Table 5: Design values computed with standard Tp exponent of 0.5
and corresponding set of constants.

In order to better visualize the difference in the design results between the two approaches, Figures 4, 5 and 6 show the ratio of calculated values with the actual Tp exponent to those computed with the standard exponent of 0.5 plotted against operational pressure for the three standard tightness classes.

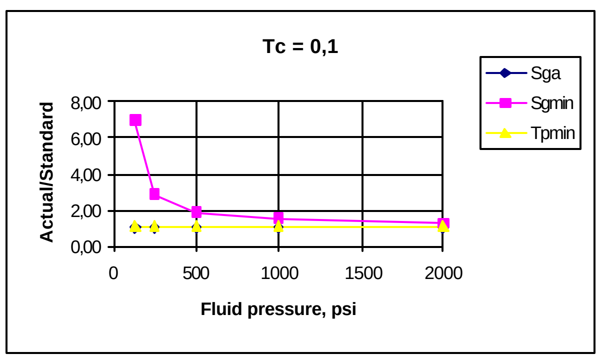

Fig. 4: Influence of Tp exponent on bolted flange design parameters for Tc = 0.1.

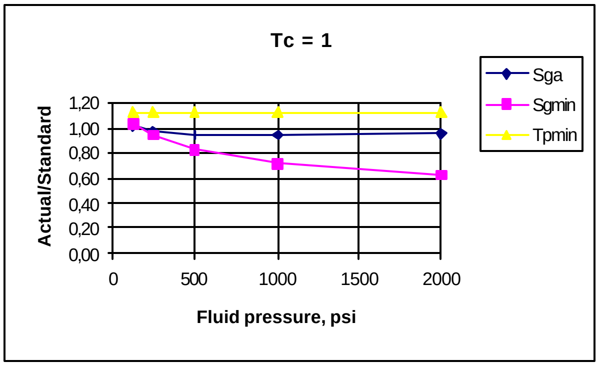

Fig. 5: Influence of Tp exponent on bolted flange design parameters for Tc = 1.

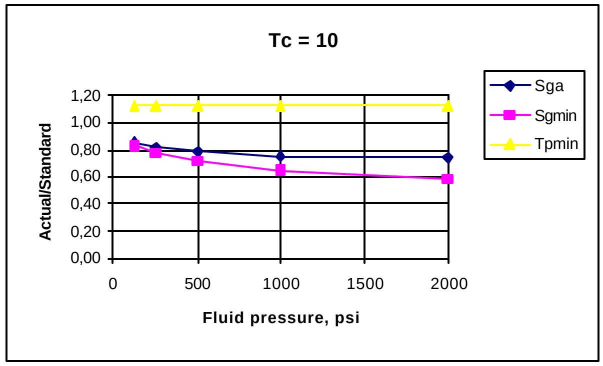

Fig. 6: Influence of Tp exponent on bolted flange design parameters for Tc = 10.

As these curves show, the design gasket assembly load is always overestimated when using the standard exponent of 0.5 in relation to the load computed with the actual exponent of 0.6. This difference grows with increasing pressure and with increasing tightness class.

In this example, the utilization of the standard Tp exponent of 0.5 and the corresponding set of constants required an assembly stress approximately 40% higher than computed with the actual exponent of 0.6 and the correspondent set of constants at a tightness class Tc = 1 and a fluid pressure of 1000 psi.

USING THE ACTUAL GASKET DIAMETER

Here again, in an attempt to obtain a simpler equation, the simplification of using a standard gasket diameter of 150 mm instead of using the actual diameter is introducing differences in the design results that may be very significant as the actual gasket diameter moves apart from the standard.

Worse than that is the fact that the differences go in one direction when the actual gaskets are smaller and go in the opposite direction when the actual gaskets are larger than the standard.

Using the same gasket constants as in the previous example and two different pressures, a simulation was made in order to compare the design results as a function of the gasket diameter.

Two sets of calculation were made: one using the standardized diameter of 150 mm for all gasket sizes and another using the actual gasket diameter in the tightness parameter determination. The results are shown in Tables 6, 7, 8 and 9.

Diameter	Tpmin			Tpa			Sgmin			Sga		
mm	0.1	1	10	0.1	1	10	0.1	1	10	0.1	1	10
35	25.7	257	2571	171	399	2913	61.9	3751	24061	5959	9648	29958
75	17.6	176	1756	170	307	2045	25.0	2407	18590	5922	8304	24487
150	12.4	124	1242	169	250	1491	10.9	1496	14551	5908	7393	20448
304.8	8.7	87	871	168	213	1086	4.6	841	11175	5902	6737	17072
609.6	6.2	62	616	168	191	804	2.0	435	8484	5899	6332	14381

Table 6: Design values computed with actual gasket diameter
for 1000 psi pressure.

Diameter	Tpmin			Tpa			Sgmin			Sga		
mm	0.1	1	10	0.1	1	10	0.1	1	10	0.1	1	10
All	12.4	124	1243	169	250	1492	10.9	1498	14560	5908	7395	20457

Table 7: Design values computed with standard gasket diameter
of 150 mm for 1000 psi pressure.

Diameter	Tpmin			Tpa			Sgmin			Sga		
mm	0.1	1	10	0.1	1	10	0.1	1	10	0.1	1	10
35	3.2	32.1	321	4.9	36	335	49.7	1709	7988	787	2446	8725
75	2.2	22.0	220	4.4	25	230	5.7	1254	6315	743	1991	7052
150	1.6	15.5	155	4.4	18	164	0.5	918	5078	738	1655	5815
304.8	1.1	10.9	109	4.4	13	116	0.0	636	4040	737	1373	4777
609.6	0.8	7.7	77	4.4	10	83	0.0	412	3208	737	1149	3945

Table 8: Design values computed with actual gasket diameter
for 125 psi pressure.

Diameter	Tpmin			Tpa			Sgmin			Sga		
mm	0.1	1	10	0.1	1	10	0.1	1	10	0.1	1	10
All	1.6	15.5	155	4.4	18.1	164	1	918	5080	738	1656	5818

Table 9: Design values computed with standard gasket diameter
of 150 mm for 125 psi pressure.

In order to better visualize the difference in the design results between the two approaches, Figures 7, 8 and 9 show the ratio of calculated values with actual to standard gasket size against the gasket diameter for an internal fluid pressure of 1000 psi. One figure for each Tightness Class.

Figures 10, 11 and 12 show similar curves for an internal fluid pressure of 125 psi.

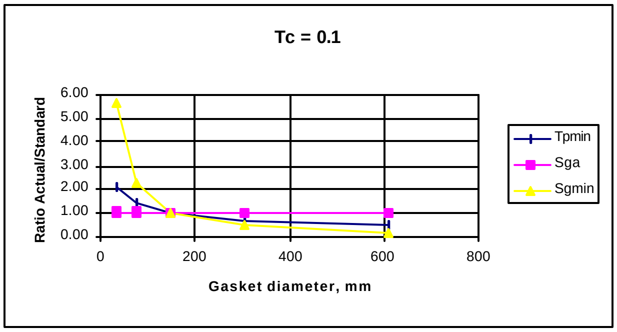

Fig. 7: Influence of gasket size on bolted flange design parameters - 1000 psi.

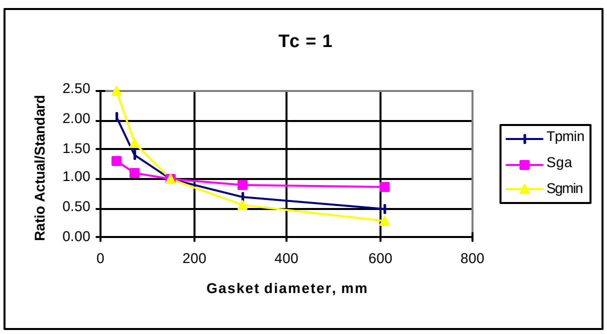

Fig. 8: Influence of gasket size on bolted flange design parameters - 1000 psi

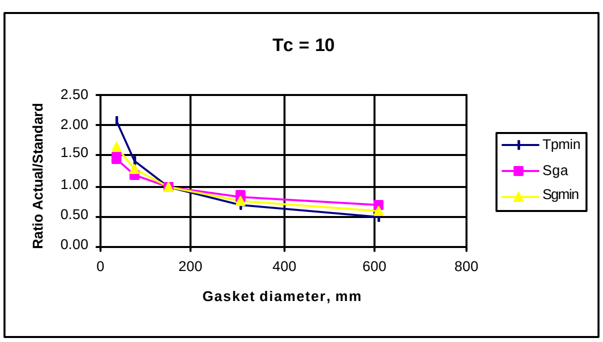

Fig. 9: Influence of gasket size on bolted flange design parameters - 1000 psi

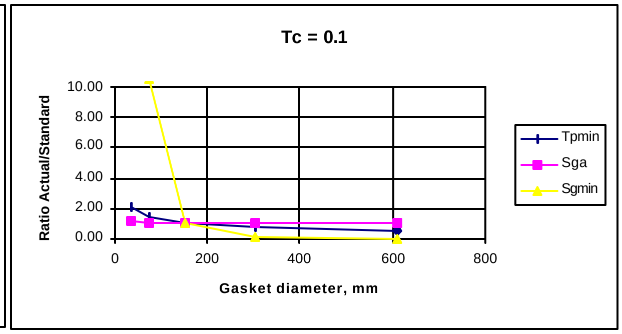

Fig.10: Influence of gasket size on bolted flange design parameters - 125 psi

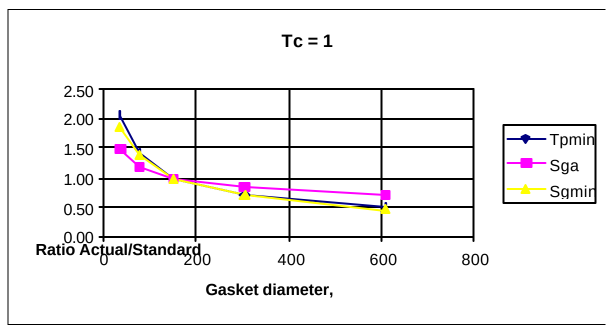

Fig.11: Influence of gasket size on bolted flange design parameters - 125 psi

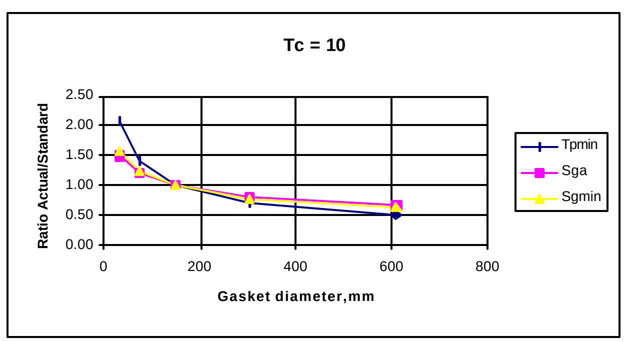

Fig.12: Influence of gasket size on bolted flange design parameters - 125 psi

As it can be seen in the above tables and curves, the design gasket assembly stress as computed using the actual gasket diameter are up to 50% higher than those computed with the standard gasket diameter of 150 mm for smaller size gaskets.

For larger gaskets, the actual assembly stress went as low as 32% lower than those with the standard calculation.

The difference tends to increase with the tightness class and with the decrease of the internal fluid pressure.

THE EXACT CALCULATION METHOD

The Exact Method considers that the bolted flanged connection design engineer will be able to develop a better design if he clearly understands what is behind what he is doing. Therefore, sound judgments can be made.

This calculation procedure includes all the points mentioned earlier, in a logical sequence, so that is possible to clearly visualize the objective of each step.

The exact calculation method sequence is as follows.

1. Determine the minimum tightness parameter for operational conditions, Tpmin:

Tpmin = Tc (P/14.7) [1/(0.002*D)]^d

where: Tc = Tightness Parameter

P = Internal fluid pressure, psi

D = Outside gasket diameter, mm

d = Tightness exponent

The Tightness exponent “d” is the fourth gasket constant and shall be supplied together with “Gb”, “a” and “Gs”.

2. Determine the tightness parameter for assembly, Tpa > Tpmin:

Use the Tpa function (eq. 4) and an equation solver to find the tightness parameter for assembly through an iterative process. Equation solver is a popular tool in spread sheets and is present in many scientific calculators. The equation is:

f(Tpa) = Gb(Tpa)^a - Gs{[Gb(Tpa)^a]/Gs}^(log Tpmin/log Tpa) - P(Ai/Ag) = 0

3. Determine the design gasket assembly stress, Sga:

Sga = Gb (Tpa)^a

4. Determine the minimum gasket stress at operational conditions, Sgmin:

Sgmin = Sga - P (Ai/Ag)

5. Determine the design bolt load, Wmo

Wmo = (Ag Sga)/Ae where: Ae = Assembly efficiency

From this point, follow the design of bolts and the corresponding bolt load to be used in the flange design the same way as described in the current draft of the new ASME Code.

In opposition to the ones currently being studied, this method is very straightforward and clearly shows the exact gasket stresses that must be achieved with the assembly and operational conditions, in order to obtain the desired sealing efficiency.

A COMPARISON BETWEEN THE CONVENIENT AND EXACT METHODS

In order to visualize the difference between the two procedures, a simulation was made using the example below, with the following application conditions:

Fluid pressures: 125 and 1000 psi   Flange class 600

Assembly efficiency: 1   Flange sizes: 3/4, 1.1/2, 4, 8, 18 in

Allowable bolt stress: Sa = 25000 psi

Sb = 17000 psi

Gasket material: compressed non-asbestos sheet

Gasket constants:

	d	Gb	a	Gs
Convenient	-	318	0.57	0.025
Exact	0.6	381	0.49	0.090

As mentioned earlier these two sets of constants were obtained for the same gasket material using the actual exponent in the tightness parameter calculation for the Exact method and the standard exponent of 0.5 for the Convenient set of constants.

The results are shown in the tables below.

Diameter	Tpmin			Tpa			Sgmin			Wmo		
mm	0,1	1	10	0,1	1	10	0,1	1	10	0,1	1	10
43	30	297	2970	34	316	3044	1438	5682	18703	2805	8334	25298
73	22	216	2158	26	234	2231	1040	4676	15823	6627	19367	58422
157	14	136	1362	20	157	1442	461	3345	12266	22695	62088	183934
270	10	98	985	33	134	1104	64	2139	9750	61541	121580	341740
533	7	65	654	67	121	801	9	1011	7103	260266	347154	875530

Table 10: Design values computed with the Exact method for P = 1000 psi.

Diameter	Tpmin			Tpa			Sgmin			Wmo		
mm	0,1	1	10	0,1	1	10	0,1	1	10	0,1	1	10
43	12	124	1243	19	186	1865	360	2384	11083	4119	4620	15952
73	12	124	1243	19	186	1865	360	2384	11083	11576	12921	43398
157	12	124	1243	19	186	1865	360	2384	11083	50029	55274	174092
270	12	124	1243	19	186	1865	360	2384	11083	131300	142409	394055
533	12	124	1243	19	186	1865	360	2384	11083	474723	508029	1262494

Table 11: Design values computed with the Convenient method for P = 1000 psi.

Diameter	Tpmin			Tpa			Sgmin			Wmo		
mm	0,1	1	10	0,1	1	10	0,1	1	10	0,1	1	10
43	3.7	37	371	3.8	38	374	643	2168	6860	954	2941	9053
73	2.7	27	270	2.8	28	272	520	1828	5842	2195	6776	20841
157	1.7	17	170	1.8	18	173	351	1402	4612	6838	21202	65050
270	1.2	12	123	1.3	13	127	170	1084	3833	12368	38818	118345
533	0.8	8	82	1.0	9	86	7	745	3008	33063	97033	293288

Table 12: Design values computed with the Exact method for P = 125 psi

Diameter	Tpmin			Tpa			Sgmin			Wmo		
mm	0.1	1	10	0.1	1	10	0.1	1	10	0.1	1	10
43	1.6	16	155	2.3	23	233	4.4	449	2790	515	1129	4193
73	1.6	16	155	2.3	23	233	4.4	449	2790	1447	3035	11277
157	1.6	16	155	2.3	23	233	4.4	449	2790	6254	11834	43966
270	1.6	16	155	2.3	23	233	4.4	449	2790	16413	25062	93116
533	1.6	16	155	2.3	23	233	4.4	449	2790	59340	76600	279642

Table 13: Design values computed with the Convenient method for P = 125 psi.

In order to better visualize the difference in the design results between the two approaches, Figures 13, 14 and 15 show the ratio Exact/Convenient of computed design figures for an internal fluid pressure of 1000 psi. One figure for each Tightness Class.

Figures 16, 17 and 18 show similar curves for an internal fluid pressure of 125 psi.

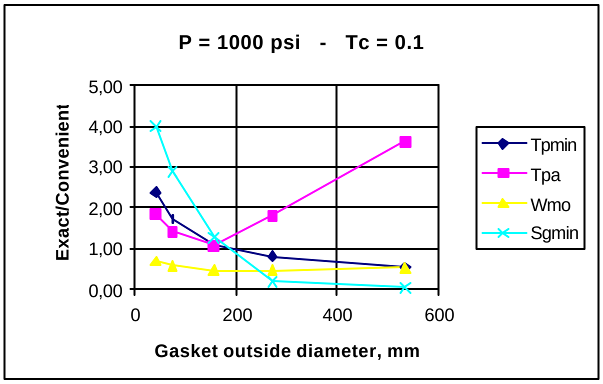

Fig.13: Exact/Convenient comparison.

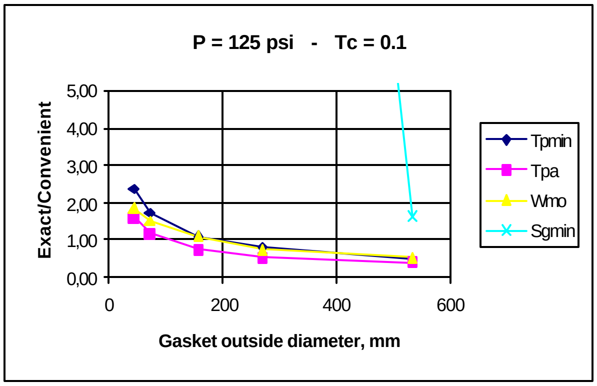

Fig.16: Exact/Convenient comparison.

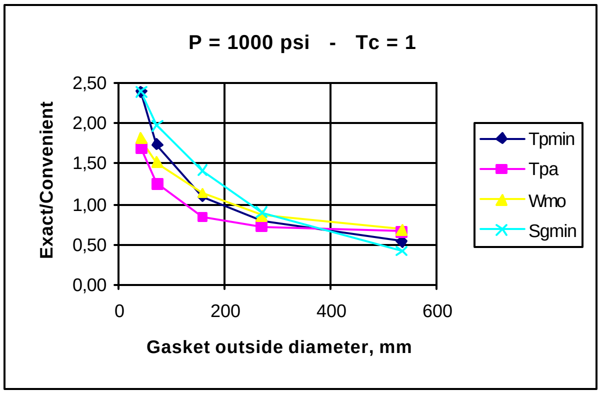

Fig.14: Exact/Convenient comparison.

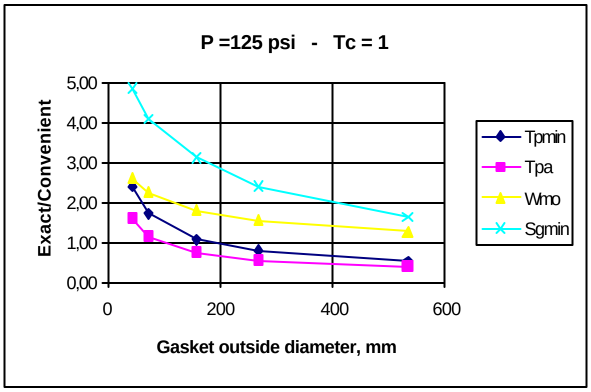

Fig.17: Exact/Convenient comparison.

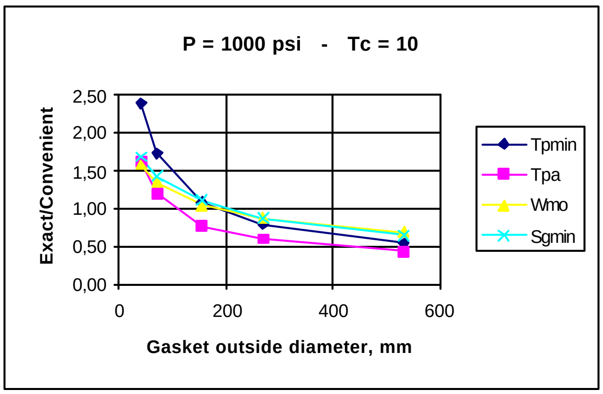

Fig.15: Exact/Convenient comparison.

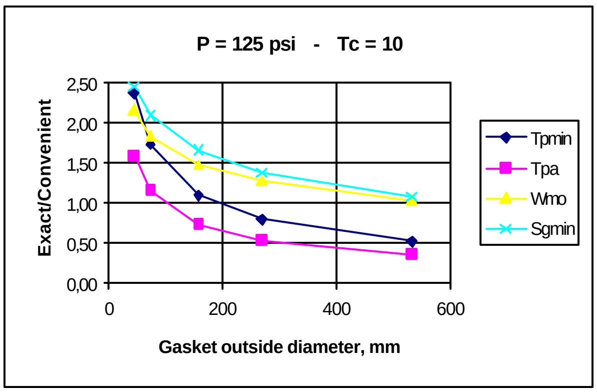

Fig.18: Exact/Convenient comparison.

As it can be seen from the above tables and curves, the difference between the two procedures is significant in all the important design parameters.

In general the Exact Method establishes higher tightness parameter and consequently higher load requirements for small flange sizes than the Convenient method. On the other hand, with large flanges, the load requirements are usually lower with the Exact method.

At low pressure, the Exact Method establishes significantly higher design bolt loads in the total range of flange sizes and especially with the small flanges.

Another interesting conclusion is that the higher the tightness class, the higher the design bolt load required by the Exact Method, in comparison with the load required by the Convenient Method.

In this example the total difference found considering only the tightness class of Tc = 1 varied between 0.68 and 2.61 for the Exact/Convenient ratio of design bolt load, Wmo. Certainly these differences are very significant and shall not be left aside.

CONCLUSIONS

The adoption of the Exact method for bolted flanged connection design brings several advantages the main being:

- **Actual gasket assembly load values** are obtained in the calculation.
- With actual gasket assembly load, the design of **bolts and flanges will be optimum** and adequate to the application; not under or over estimated.
- As the computed values for the various design parameters are exact and consistent with the theory supporting the new gasket constants concept, a **real analysis of the application** can be done.

Using the Exact method there will never be a negative value for a load estimation!

- The **Exact method is simple to use** because it is structured and its sequence is very straightforward.

There is a clear distinction and understanding of the contribution of each component of the bolted flanged connection including assembly to the overall design.

Let everybody see and feel the impact of the 33% higher bolt and flange load required due to the utilization of a manual wrench for assembling!

The disadvantage of the Exact method over the Convenient is the need of using an equation solver. But let's face it: how many design engineers are currently there that are not using a computer to perform their calculation? And within a five years time?

Is the price we are paying for having an apparently simple calculation sequence worth the less-than-optimum results, the difficulty of understanding the theory behind the procedure and the correspondent consequences that can be generated in terms of equipment design?

We still have time to decide upon and choose the way we want to go.

RECOMMENDATION FOR FURTHER STUDY

Looking at the actual values for the minimum gasket stress Sgmin, as computed in the example, it is possible to see there are situations where very low gasket stresses are allowed during operation, mainly with large flanges working at low pressures and low tightness requirements.

As insufficient stress can cause the gasket to blow out during operation, it is recommended that further studies be made in order to define and introduce in the calculation procedure, a minimum allowable stress to be applied on the gasket during operation, therefore eliminating this risk.

REFERENCES

1. Rules for Bolted Flange Connections with Ring Type Gaskets. Preliminary Committee Draft. PVRC. 5/93.
2. Nonmetallic Flat Gaskets for Pipe Flanges. ASME B16.21-1992.
3. PVRC Committee Correspondence. Bolt Load Calculation Examples & Modified Convenient Method. JRPayne. May 31, 1995.
4. As Novas Constantes de Juntas-Avaliação de uma União Flangeada. Seminário Técnico de Vedação-Segmento Petroquímico. ACGuizzo. Asberit. RJ. August 10, 1995.
5. Influência da Pressão Interna sobre o Vazamento. AMFurtado. Asberit ED-951/95. RJ. Fev/95.
6. Cálculo das Constantes para Papelões Hidráulicos. PPBatista. Asberit ED-946/95. RJ. Jan/95.