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Evaluation of pipeline flanges subject to internal pressure and imposed external loads

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ABSTRACT: *Engineers involved in pipeline design are often confronted with the evaluation of external loads imposed on flange connections. This article aims to provide more clarity in the various evaluation methods and the associated conservatisms. The magnitude of the external loads has an essential influence on the leak-tightness of the flange connection. Flange joint sealing integrity is of great importance and requires special attention. The article highlights various methods that can be applied in practice depending on the criticality of the pipeline system. Because various methods as described in this article are available to the engineer concerned, the sensitivity and degree of conservatism of the applied methods can be compared and evaluated.*
The successively discussed external flange load evaluation methods are: Comprehensive flange assessment approach based on ASME VIII-1; Appendix 2, Simple "Kellogg" based equivalent pressure approach, Improved "Kellogg" approach including "Koves" factor and evaluation method in accordance with ASME Code case 2901.

KEYWORDS: *External loads, sealing integrity, degree of conservatism, external load evaluation methods.*

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I. INTRODUCTION

The traditional flange calculation as included in ASME BPVC Section VIII - Division 1; Appendix 2 [1] does not account for imposed external loads that act together with the internal pressure on the flange connection. One of the most used conservative evaluation to perform flange check on external loads is the well known "Kellogg Equivalent Pressure Method", where the axial force F_E and the bending moment M_E are converted into an equivalent pressure P_{EQ}. The equivalent pressure is then consecutively added to the internal design pressure and then compared with the rated pressure according to ASME B16.5 [2] or ASME B16.47 [3]. W.J. Koves has made a fundamental contribution to improving the conventional way of determining the equivalent pressure [4] and has been further elaborated by C.J. Dekker et al. [5]. Recently ASME Code case 2901 [6] has been published which offers an alternative for the assessment of welding neck flanges loaded by internal pressure and external loading. In some cases, however, it is desirable to perform a flange calculation that provides insight into the flange - and bolt stresses which occur in case of internal pressure in combination with external loads exerted on the flange. To accommodate this, the ASME flange calculation algorithm needs to be adjusted.

II. METHODOLOGY AND PRACTICAL APPROACH

The methodology recognizes the non-uniform load distribution due to bending moments and takes the flange rigidity into account to some extent. The flexibility of the flange, gasket and bolts should be modeled to sufficient accurately predict the gasket and bolt loads resulting from pressure and external loads. The flange can be analyzed by using the shell-and plate theory solution, which is consistent with the current ASME Code approach. The effect of axial loads on a flange joint can be handled the same way as the axial pressure thrust term in the current ASME method.
The axial force is simply added to the axial pressure - thrust force, and the ASME design procedure is followed for the computation of flange moments. Using ASME nomenclature:

$$H_D = \frac{\pi}{4} B^2 . P + F_A$$

where:

F_A = the axial applied force (only be taken into account if tensile) (N)
P = the internal design pressure (MPa)
B = the inside diameter of flange (mm)
H_D = the axial force in the flange neck (N)

External moments are more difficult to handle. The loading is not axisymmetric and cannot be addresses as easily as the axial forces. However, the ASME design approach assumes axisymmetric behaviour. Therefore, the problem is to evaluate the effect of external moment loading on the flange joint and develop a correction to be applied to the axisymmetric analysis. This is addressed by analytically solving for the forces acting on a ring flange, as a result of an external moment; then comparing it with the axisymmetric force solution.

An external applied moment is assumed to create a linear stress distribution in the flange neck. This moment can be reduced to a linear distributed load. Therefore, a moment correction factor, F_M, can be defined which adjusts for the torsional resistance of the flange to external moment loads. Using the relationship between the shear modulus (G) and the Young's modulus of elasticity (E) being: $= \frac{E}{2(1+\nu)}$, which gives the following :

$F_M = \frac{1}{[1+\frac{J}{2(1+\nu)I}]}$, where J and I are parameters related to flange configuration and rigidity.

where:

J = Polar moment of inertia (mm^4) ,
I = Second moment of inertia (mm^4)
ν = Poisson's ratio = 0.3.

F_M is the reciprocal of the "Koves" factor K_f, hence,

$$F_M = \frac{1}{K_f} \text{ with } K_f = 1 + \frac{G.J}{E.I} = 1 + \frac{I_z + I_r}{2(1+\nu)I_r}$$

where:

$I_r = \frac{1}{12}$ (effective flange width) (flange thickness)^3
$I_z = \frac{1}{12}$ (flange thickness) (effective flange width)^3
With ν = 0.3 one arrives at: $K_f = 1 + \frac{(\text{effective flange width } ,W_f)^2 + (\text{flange thickness } ,t_f)^2}{2.6 \, (\text{flange thickness } ,t_f)^2}$

The effective flange width W_f is defined as the actual flange width being: 0.5 (OD flange - ID flange) minus the reduced bolt hole diameter D_{bhd}*.

The reduced bolt hole diameter follows from:

D_{bhd}* = max [Bolt hole diameter "D_{bhd}" (Inside diameter of flange "B" /1000 ; 0.5 x bolt hole diameter "D_{bhd}")]

Therefore:

$$K_f = 1 + \left[\frac{(t_f^2 + W_f - D_{bhd} *)}{(2.6 \, t_f^2)} \right] \text{ and } F_M = \frac{1}{K_f}$$

The greater the torsional resistance, relative to the bending resistance, the less the induced circumferential bending stress and corresponding flange rotation as a result of the external moment.

Using the terminology of ASME Section VIII, Division 1 - Appendix 2 [1], external moments and forces can be included in the design by defining the operating design moment as follows:

$$H_D = \frac{\pi}{4} B^2 . P + F_A + \frac{4 F_M . M_E}{G} \quad (N)$$

$$H_G = (2b)(\pi.G.m.P) \quad (N)$$

$$H_T = \frac{\pi}{4}(G^2 - B^2)\,P \quad (N)$$

$$M_O = H_D.h_D + H_T.h_T + H_G.h_G \quad (Nmm)$$

The rest of the flange calculation is performed in accordance with the procedure described in ASME VIII-1, Appendix 2[1].

A fully elaborated example is given in Section III of this article.

III. WORKED EXAMPLE

A welding neck flange connection in accordance with ASME B16.47 Series A Class 300 is part of an NPS 36 " (NB 900) gas transmission pipeline.

The design conditions are:

Design pressure 30 bar, design temperature 50° C, corrosion allowance nil.

The flange connection is provided with a spiral wound gasket according to ASME B16.20 [7]. The flange material is A105, bolt material is A193 B7 and the pipeline material is A515 Grade 70.

The external loads acting on the flange connection are respectively: $F_A = F_E$ = 350 kN and the resulting bending moment M_E = 330 kNm.

Moment M_E is initially based on a bending stress in the adjoining pipe of 6000 psi = 41.37 MPa whilst the axial tensile force F_A is based on approximate 0.2 times the hydrostatic end force. This rule of thumb is often used by practicing engineers as a starting point.

Calculation

The execution of the calculation is limited to the operating condition and is shown in table 1 below.

Table 1.

Flange Calculation (Type Integral Flange) Welding Neck Flange according ASME B16.47 ; NPS 36" ; Series A ; Class 300				
System Design Parameters	Parameter	Symbol	Value	Equation
	Design Pressure	P	3 MPa	
	Design Temperature	-	50°C	
	Axial Force	$F_A = F_E$	350 kN	
	Bending Moment	M_E	330 kNm	
Flange Data	Flange outside diameter	A	1270 mm	
	Flange inside diameter	B	889 mm	
	Bolt circle diameter	C	1168.4 mm	
	Flange thickness	t_f	103.2 mm	
	Hub thickness small end	g_o	12.7 mm	
	Hub thickness large end	g_1	51 mm	
	Hub length	h	136.8 mm	
	Bolt hole diameter	D_{bhd}	54 mm	
Bolting data	Bolt size	D_b	2" UN	
	Number of bolts	n_b	32	
Gasket data	Gasket material	-	SPW	
	Gasket factor	m	3	
	Gasket seating stress	y	69 MPa	
	Outside diameter gasket (Effective)	G_{OD}	1003.6 mm (include bead of 1.5 mm)	
	Inside diameter gasket	G_{ID}	955.8 mm	
	Gasket width	N	23.9 mm	$(G_{OD} - G_{ID})/2$
	Basic gasket seating width	b_o	11.95 mm	N/2
	Effective gasket seating width	b	8.71 mm	$2.52\ \sqrt{b_o}$
	Gasket load reaction diameter	G	986.18 mm	G_{OD} - 2.b

Calculation parameters	Radial distance	R	88.7 mm	R = (C-B)/2 - g_1
	Lever arm	h_D	114.2 mm	R + 0.5 g_1
	Lever arm	h_T	115.41 mm	(R + g_1 + h_G)/2
	Flange width	W_f	190.5 mm	(A - B)/2
	Reduced bolt hole diameter	D_{bhd*}	48 mm	D_{bhd*} = max [D_{bhd} (B/1000 ; 0.5 D_{bhd})
	Moment correction factor	F_M	0.7195	$F_M = \frac{1}{K_f} ; K_f = 1 + \left[\frac{(t_f^2 + W_f - Dbhd*)}{(2.6\, t_f^2)}\right]$
	Lever arm	h_G	91.11 mm	(C- G)/2
	Hydrostatic end force inside	H_D	3175200 N	$H_D = \frac{\pi}{4} B^2 . P + F_A + \frac{4 . F_M . M_E}{G}$
	Difference H - H_D is pressure force at flange face	H_T	429369 N	$H_T = \frac{\pi}{4}(G^2 - B^2)\, P$
	Gasket load for sealing	H_G	485732 N	$H_G = (2b)(\pi . G . m . P)$
	Flange moment	M_O	456416359 Nmm	$M_O = H_D . h_D + H_T . h_T + H_G . h_G$

Flange factors
Bolt spacing factor B_{SC} = 1.0 (as per code)
K = A/B = 1.4286 ; $h_O = \sqrt{B g_0}$ = 106.2558 ; h / h_O = 1.2875 ; T = 1.7418 ; Z = 2.9216 ; Y = 5.6113 ;
U = 6.1662 ; F = 0.6124 ; V = 0.0415 ; f = 1.0 ; e = 0.00576 (1/mm) ; g_1 / g_O = 4.0157 ;
d = 2545760.9284 (mm^3) ; L = 1.3474 ; E = 200666 MPa ; K_I = 0.3

Stresses under operating condition		
Equation	Value	Allowable stress
$S_H = \frac{f\, M_O}{L\, g_1^2\, B}$	146.49 MPa	1.5 x 138 = 207 MPa
$S_R = \frac{(1.33\, t_f\, e + 1) M_O}{L\, t_f^2\, B}$	64.06 MPa	138 MPa
$S_T = \frac{Y\, M_O}{t_f^2\, B} - Z\, S_R$	83.34 MPa	138 MPa
S_C = Max [($S_H + S_R$)/2 ; ($S_H + S_T$)/2]	114.91 MPa	138 MPa
Flange rigidity check		
$J = \frac{52.14\, V\, M_O}{L\, E\, g_0^2\, K_I h_O}$	0.7104	≤ 1.0

Verification of bolt stress during operating
Bolt force required for operating: $W_{m1(tot)} = H + H_P + F_A + \frac{4\, M_E}{C} = \frac{\pi}{4} G^2 P + (2b)(\pi . G . m . P) + F_A + \frac{4\, M_E}{C}$
$W_{m1(tot)}$ = 2777251 + 350000 + 4 x 330000000 / 1168.4 = 4257001 N
Bolt stress S_{bolt} = $W_{m1(tot)}$ / (n_b x A_{bolt}) = 4257001 / 32 x 1710.9 = 77.76 MPa < 172 MPa (A193 B7 bolting)

IV. DISCUSSION

For comparison with the case detailed in Section III, an evaluation will be performed using:

A. Kellogg's equivalent pressure method
B. Kellogg's method including "Koves" factor
C. ASME Code Case 2901

The results of the tabulated comparison are shown below in Table 2.

Table 2.

Method "A"	**Method "B"**	**Method "C"**
By the "Kellogg" Equivalent Pressure Method, a P_{eq} is determined, added to the design pressure P and then compared to the flange rated pressure at the operating / design temperature.	As method "A" however the moment term is divided by the "Koves" factor "K_f "	Method developed by Warren Brown and released in ASME code as case 2901.
Condition to be satisfied	Condition to be satisfied	Condition to be satisfied
P + P_{eq} ≤ P_{rating} = 5.01 MPa $P + \frac{4\, F_A}{\pi\, G^2} + \frac{16\, M_E}{\pi\, G^3}$ = 5.21 MPa > 5.01MPa NOT OK!	P + P_{eq} ≤ P_{rating} = 5.01 MPa K_f = 1 / F_M = 1.39 $P + \frac{4\, F_A}{\pi\, G^2} + \frac{16\, M_E}{\pi\, G^3\, K_f}$ = 4.72 MPa < 5.01 MPa OK!	16 M_E + 4 F_E G ≤ π G^3 [(P_R - P_D) + F_M P_R] 6660652000 < 7565979451 OK! Where: M_E = External moment F_E = External tensile axial force G = Gasket reaction diameter

Pseudo flange design pressure: **Case #1 = 5.21 MPa**	Pseudo flange design pressure: **Case #2 = 4.72 MPa**		P_R = Flange pressure rating at design temperature P_D = Flange design pressure at design temperature F_M = Moment factor = 0.1 according to table 1 of case 2901.
Stresses under operating condition **Case #1** with design pressure = 5.21 MPa and M_O = 532225703 Nmm **Case #2** with design pressure= 4.72 MPa and M_O = 482169926 Nmm			
Equation	Value Case #1	Value Case #2	Allowable stress
$S_H = \frac{f\,M_0}{L\;g_1^2\;B}$	170.83 MPa	154.77 MPa	1.5 x 138 = 207 MPa
$S_R = \frac{(1.33\,t_f\;e + 1)M_0}{L\;t_f^2\;B}$	74.73 MPa	67.70 MPa	138 MPa
$S_T = \frac{Y\,M_0}{t_f^2\;B} - Z\,S_R$	97.11 MPa	87.98 MPa	138 MPa
S_C = Max [($S_H + S_R$)/2 ; ($S_H + S_T$)/2]	133.97 MPa	121.37 MPa	138 MPa
Flange rigidity check			
$J = \frac{52.14\,V\,M_0}{L\,E\,g_0^2\;K_I h_0}$	0.83	0.75	≤ 1.0
It appears that the stresses for cases # 1 and # 2 are higher than for the case elaborated in table 1, indicating that Methods "A" and "B" are more conservative than those applied in table 1.			

In line with the previous calculations and assessments, we increase the magnitude of the external loads such that an optimum flange stress is achieved, resulting in F_A (F_E) is 562 kN and M_E is 530 kNm.

The result of this exercise is shown in table 3 below.

Table 3.

Stresses under operating condition with $F_A = F_E$ = 562 kN and M_E = 530 kNm		
Equation	Value	Allowable stress
$S_H = \frac{f\,M_0}{L\;g_1^2\;B}$	175.66 MPa	1.5 x 138 = 207 MPa
$S_R = \frac{(1.33\,t_f\;e + 1)M_0}{L\;t_f^2\;B}$	76.82 MPa	138 MPa
$S_T = \frac{Y\,M_0}{t_f^2\;B} - Z\,S_R$	99.92 MPa	138 MPa
S_C = Max [($S_H + S_R$)/2 ; ($S_H + S_T$)/2]	137.79 MPa	138 MPa
Flange rigidity check		
$J = \frac{52.14\,V\,M_0}{L\,E\,g_0^2\;K_I h_0}$	0.852	≤ 1.0
Verification of bolt stress during operating		
Bolt force required for operating: $W_{m1(tot)} = H + H_P + F_A + \frac{4\,M_E}{C} = \frac{\pi}{4}G^2P + (2b)(\pi.G.m.P) + F_A + \frac{4\,M_E}{C}$ $W_{m1(tot)}$ = 2777251 + 562000 + 4 x 530000000 / 1168.4 = 5153698 N Bolt stress $S_{bolt} = W_{m1(tot)}$ / (n_b x A_{bolt}) = 5153698 / 32 x 1710.9 = 94.13 MPa < 172 MPa (A193 B7 bolting)		

Table 4 provides insight into the permissible set of loads (F_E or F_A and M_E) for which the condition is just fulfilled.

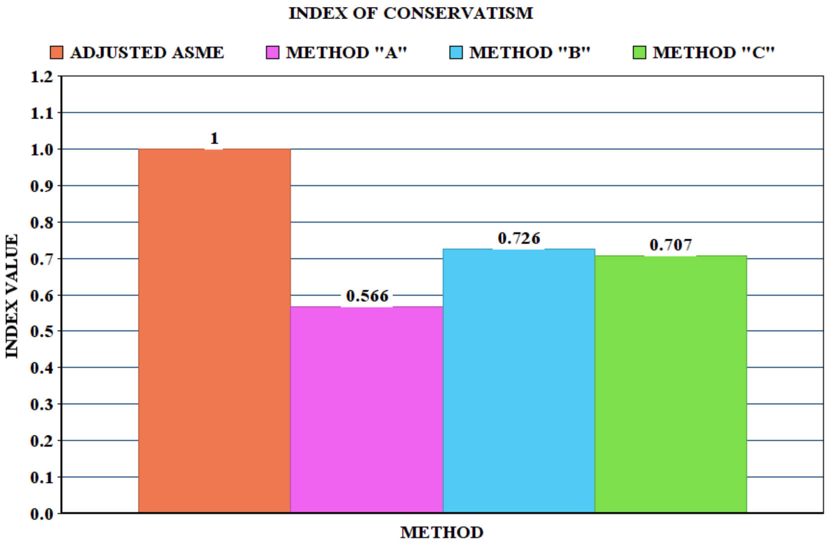
Table 4.

Method "A"	**Method "B"**	**Method "C"**
By the "Kellogg" Equivalent Pressure Method, a P_{eq} is determined, added to the design pressure P and then compared to the flange rated pressure at the operating / design temperature.	As method "A" however the moment term is divided by the "Koves" factor "K_f "	Method developed by Warren Brown [8] and released in ASME code as case 2901.
Condition to be satisfied	Condition to be satisfied	Condition to be satisfied
$P + P_{eq} \le P_{rating}$ = 5.01 MPa With: F_A = 318 kN and M_E = 300 kNm we end up to: $P + \frac{4\,F_A}{\pi\,G^2} + \frac{16\,M_E}{\pi\,G^3}$ = 5.01 MPa **Index** with respect to F_E is 562 kN and M_E	$P + P_{eq} \le P_{rating}$ = 5.01 MPa $K_f = 1 / F_M$ = 1.39 With: F_A = 408 kN and M_E = 385 kNm we end up to: $P + \frac{4\,F_A}{\pi\,G^2} + \frac{16\,M_E}{\pi\,G^3\,K_f}$ = 5.01 MPa	$16\,M_E + 4\,F_E\,G \le \pi\,G^3\,[(P_R - P_D) + F_M\,P_R]$ Where: M_E = External moment F_E = External tensile axial force G = Gasket reaction diameter P_R = Flange pressure rating at design temperature P_D = Flange design pressure at design temperature F_M = Moment factor = 0.1 With:

is 530 kNm according to optimum ASME flange calculation:0.566	**Index** with respect to F_E is 562 kN and M_E is 530 kNm according to optimum ASME flange calculation: 0.726	F_E = 397 kN and M_E = 375 kNm we end up to: 7566053840 ~7565979451 **Index** with respect to F_E is 562 kN and M_E is 530 kNm according to optimum ASME flange calculation: 0.707

The index of conservatism (also called the compatibility index) is a way of indicating the difference in effect between two or more methodologies. The lower the index, the higher the degree of conservatism.

The figure below shows in graphic form the conservatism indexes associated with the various approaches.



In addition the maximum allowable internal pressure has been calculated for the flange under consideration without taking into account external loads exerted on the flange. Only the operating condition has been considered and the pressure has been optimized to the allowable flange stresses and the rigidity requirement. The result led to an allowable pressure of 53.65 bar (5.365 MPa). The rated pressure according to ASME B16.47 is 50.1 bar (5.01 MPa) for this flange, which equates to a difference of about 7%.

It could be argued that the difference between the rated flange pressure or the calculated allowable pressure of the flange and the design pressure is left over for the external loads imposed on the flange connection. Often in practice the external load is converted to an equivalent pressure and added to the design pressure. This allows a resultant calculation pressure that can be used in the traditional flange calculation according to ASME. The bending moment is the dominant loading in this approach and is very popular among piping engineers and there is little doubt about the usefulness of this approach.

V. CONCLUSION

For the flange connection in question, it can be concluded that the adjusted analytical flange calculation according to ASME provides a reliable insight into the flange stresses that occur when the flange connection is loaded by internal pressure and external loads exerted on it. The flange and bolt stress is amply permissible for the initial condition considered. Adjusted in this context means that the common ASME method is augmented with terms related to the imposed external loads. In the case that this flange configuration is evaluated using the simple "Kellogg" equivalent pressure method "A", the assessment condition is found not to be met. If, on the other hand, the so-called "Koves" factor is introduced in the case of method "B", it appears that the external flange loads are permissible. This also applies when applying ASME Code case 2901 for this flange connection. Note that both Method "B" and Method "C" are fairly easy to apply and can give a quick impression of the criticality of the external loads. If we observe table 4, we discover that all three methods are more conservative than the adjusted ASME flange calculation method which include the external loads. It is also striking that for the considered flange method "A" is most conservatively followed by method "C" and "B". The mutual differences between Method "B" and "C" are relatively small.

VI . CONCLUDING REMARKS

In order to be able to state that the research conducted is sufficiently representative, it is recommended to extend the analyzes to more flange dimensions and pressure classes.

Although this investigation has been limited to one flange diameter and pressure class, the result provides sufficient starting points for further investigation and also an indication of the preferred method for evaluating external flange loads.

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