A more consistent "Pressure Equivalent Method" for piping flanges, following the 2007 ASME Section VIII Division 2 relations

Second draft

This is a corollary of the formulas exposed in *ASME Boiler and Pressure Vessel Code Section VIII Division 2, 2007 edition, Chapter 4.16 "Design rules for Flanged Joints"* [1].
The Section VIII Division 2 relations are developed for flanges' design, but they may be used also as relations for flanges' checking under the internal pressure and external loads.

One may observe that the basic relations are quite independent of the Code, being based on elasticity theory and FEA simulations. Even true, for formal aspects, the developed formulas remain valid within the Code that contains them, i.e. Section VIII Division 2, 2007, referred afterward as Code.

The nomenclature used is given by paragraph 4.16.12 of the Code. Details are given in the text when supplementary notations have been used.

A. THE EQUIVALENT PRESSURE DUE TO PIPING LOAD

In the flanges section of the Code, one central notion is M_0 "the flange design moment for the operation condition". It includes both internal pressure and external loads applied on the flange

The expression is (4.16.14):

$$M_0 = \text{abs}\left[\left(H_D h_D + H_T h_T + H_G h_G + M_{0E}\right)F_S\right]$$

where M_{0E} is the component of flange design moment resulting from a net section bending moment and/or axial force.
We may note that $F_s = 1$ for the flanges normally used in piping. A different value is considered only for "Split Loose Type Flanges".

If we consider this formula as for flanges' checking, the internal pressure coincident with external axial force and bending moments must be considered.

The first three terms in the above written formula are forces due to internal pressure multiplied by arms.
The pressure forces are given by $(4.16.10) \div (4.16.13)$:

$$H_D = 0.785 B^2 P = H_D^* P \qquad \text{where } H_D^* = \frac{\pi}{4}B^2$$

$$H_T = H - H_D = 0.785 G^2 P - 0.785 B^2 P = H_T^* P \qquad \text{where } H_T^* = \frac{\pi}{4}\left[G^2 - B^2\right]$$

$$H_G = W_0 - H = 0.785 G^2 P + 2 b \pi G m P - 0.785 G^2 P = 2 b \pi G m P \qquad \text{where the factor } m \text{ is the gasket}$$
factor.

However, the last expression for the gasket load is specifically written for the design case.
A specific expression must be considered for flange checking case.

The real behavior of the flange joint is done by setting up the bolts load when the flange is not subject of the pressure. Under pressure, the flange joint must remain tight.

For a more detailed discussion we can follow **[5],** *chapter "Factors m, y, b"*.

The discussion involves the m factor which (in **[5]**) is defined as the factor needed to hold the seal under internal pressure.

Suppose the ratio of the compressive stress of the gasket to the internal pressure of the fluid trying to escape is designated as m' at the instant when leakage starts. In other words, the compressive stress of the gasket, at the instant when leakage starts, is just m' x *(fluid pressure).*

Under the pressure condition, the correct equilibrium condition (as described in **[5]**) is:

$\text{bolts load} = (\text{fluid pressure}) \times (\text{aria subject to fluid pressure}) + m' \times (\text{fluid pressure}) \times (\text{aria subject to gasket pressure})$

The m factor given by Codes is derived from m' by adding a safety margin.

In fact, the above written equation is used for setting-up the bolts load, considering for m factor a specific value, associated to the gasket type. This shall assure a considerable value for gasket load when the flange is not subject of the pressure. Under pressure, the gasket load is **decreasing**, but finally the joint must remain tight at maximum allowable pressure.

Consequently, when the flange is under pressure, the load on the gasket is:

$gasket\ load = bolts\ load - (fluid\ pressure) \times (aria\ subject\ to\ fluid\ pressure)$
   where:

$bolts\ load = (max\ fluid\ pressure) \times (aria\ subject\ to\ fluid\ pressure) + m \times (max\ fluid\ pressure) \times (aria\ subject\ to\ gasket\ pressure)$

For a piping flange, in the language of **[1]** the relation is:

$$H_G = W_0 - H = W_0 - 0.785 G^2 P = W_0 - H^* P$$
$$\text{where } H^* = \frac{\pi}{4} G^2.$$

The arms' expressions are given in table (4.16.6).

The expression of M_0 may be written as:

$$M_0 = \left(H_D^* h_D + H_T^* h_T - H^* h_G\right) P + W_0 h_G + M_{0E}$$

and note that all H_D^*, H_T^*, H^*, h_D, h_T, h_G are independent relatively to pressure and external loads.

Now,

IF

we are able to mathematically speculate that:

$$M_{0E} = (H_D^* h_D + H_T^* h_T - H^* h_G) \, P_1 \quad \text{where } P_1 \text{ is dimensionally a pressure}$$

THEN

we can write

$$M_0 = \left(H_D^* h_D + H_T^* h_T - H^* h_G\right) P + \left(H_D^* h_D + H_T^* h_T - H^* h_G\right) P_1 + W_0 h_G =$$
$$= \left(H_D^* h_D + H_T^* h_T - H^* h_G\right)(P + P_1) + W_0 h_G$$

We can compare the moment expression for the case when the flange is loaded only with internal pressure P, i.e.:

$$M_0 = \left(H_D^* h_D + H_T^* h_T - H^* h_G\right) P + W_0 h_G$$

with the moment expression, after the mathematical substitution $M_{0E} \rightarrow P_1$, for the case when the flange is loaded with internal pressure P and M_{0E}

$$M_0 = \left(H_D^* h_D + H_T^* h_T - H^* h_G\right)(P + P_1) + W_0 h_G$$

It appears that the flange is loaded only with a total pressure $P + P_1$, hence the P_1 pressure is the equivalent pressure due to piping load.

The equivalent pressure P_1 is given by $P_1 = \dfrac{M_{0E}}{H_D^* h_D + H_T^* h_T - H^* h_G}$

The M_{0E} evaluation is given by the relation (4.16.16) of the Code.

$$M_{0E} = 4 M_E \left[\frac{I}{0.3846 \, I_P + I}\right]\left[\frac{h_D}{C - 2 h_D}\right] + F_A h_D$$

One may note the factor:

$$F_M = \frac{I}{0.3846 \, I_P + I} = \left.\frac{1}{1 + \dfrac{1}{2(1+v)} \dfrac{I_P}{I}}\right|_{v=0.3}$$

with I_P and I are the flange moments of inertia as given in table 4.16.7.

As it is explained in the ***Companion Guide to the ASME Boiler and Pressure Vessel Code*** **[2]** "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".

This is an important real fact for understanding the stress in flanges, as well the consequences for flanges rigidity.

Actually, this is Dr. Koves's treatment of external moment **[3],** now assimilated by the Code.

Finally, the equivalent pressure evaluation:

$$P_1 = \frac{\left[\dfrac{4M_E F_M}{C-2h_D}+F_A\right]h_D}{H_D^* h_D + H_T^* h_T - H^* h_G} = \frac{\dfrac{4M_E F_M}{C-2h_D}+F_A}{H_D^* + H_T^* \dfrac{h_T}{h_D} - H^* \dfrac{h_G}{h_D}} = \frac{\dfrac{4M_E F_M}{C-2h_D}+F_A}{\dfrac{\pi}{4}B^2 + \dfrac{\pi}{4}\left[G^2-B^2\right]\dfrac{h_T}{h_D} - \dfrac{\pi}{4}G^2 \dfrac{h_G}{h_D}}$$

This expression is the equivalent pressure as possible to be calculated by following the Sect VIII Div2 relations.

Note that the Code specifically asks to consider only the value of the external *tensile* net-section axial force and to neglect the compressive net- section forces.

This requirement may be interpreted as specific for design case. When using the above written relations for checking the piping flange, one may consider also the real effect of the compressive force.

The Kellogg formula [4] may be recovered under the following simplifications:

IF

$$\frac{F_M}{C-2h_D} = \frac{\dfrac{I}{0.3846\,I_P + I}}{C-2h_D} \rightarrow \frac{1}{G}$$

$$\text{AND } \frac{\pi}{4}B^2 + \frac{\pi}{4}\left[G^2-B^2\right]\frac{h_T}{h_D} - \frac{\pi}{4}G^2\frac{h_G}{h_D} \rightarrow \frac{\pi}{4}G^2$$

THEN

$$P_1 = P_{KELLOGG} = \frac{\dfrac{4M_E}{G}+F_A}{\dfrac{\pi}{4}G^2} = \frac{16M_E}{\pi G^3} + \frac{4F_A}{\pi G^2}$$

For a specific flange –gasket case, one can evaluate what does it means the Kellogg assumptions.

A more realistic formula is:

$$P_1 = \frac{\dfrac{4F_M M_E}{G}+F_A}{\dfrac{\pi}{4}G^2} = \frac{16F_M M_E}{\pi G^3} + \frac{4F_A}{\pi G^2} \quad \text{where, as explained above, } F_M \text{ is Dr. Koves's factor.}$$

B. LIMITS OF THE TOTAL PRESSURE

With the equivalent pressure evaluated as above, the next logical question is to find out a limit to compare with.

Since the M_0 expression can be reconsidered in terms of the total pressure:

$$M_0 = \left[\frac{\pi}{4}B^2 h_D + \frac{\pi}{4}\left[G^2-B^2\right]h_T - \frac{\pi}{4}G^2 h_G\right](P+P_1) + W_0 h_G$$

the limits of M_0 (due to Code limits for flange's stress and rigidity) can be reconsidered as limits of the total pressure.

The total pressure may be limited also to the rating value.

By imposing $P + P_1 \leq P_{RATING}$, the minimum gasket load is just limited to a value corresponding to the pressure rating value, i.e. $H_G = 2b\pi\,Gm\,P_{RATING}$.

Theoretically, increasing the pressure over P_{RATING} also means the necessity for increasing the required bolts load in order to reestablish the gasket tight ($m \cdot P_{RATING}$). Practically- and depending the m value- this wouldn't be necessary for moderate overpassing the P_{RATING} limit, because the flange joint is not immediately compromised in terms of possible leakage. The real limit of leakage is done by the ASME J factor.

It was also remarked that the bolted joint designed per Codes is normally using an allowable bolt stresses that are much lower than the desired bolt up stress (typically 50 ksi, see for a discussion the **2007 VIII Div 1 Appendix S [6]** or **2006 edn ASME VIII Div2, appendix 3, article 3.5 [7]** and **ASME PCC-1 [8]** for a description of desired flange bolting practices). The margin between allowable stress (used for flange design) and bolt up stress may be considered as a margin for the total pressure that can be over the flange rating pressure, but this real fact is quit difficult to be considered in a regular stress calculation practice.

By the other hand, it appears also that the $P + P_1 \leq P_{RATING}$ limit is not based on the flange stress.

This limit may be considered as a practical limit in order to "feel" the leakage danger, however without providing a measure of the safety factor.

B1. THE FLANGE STRESS LIMIT

The stress equations (table 4.16.8) may be reconsidered in terms of total pressure:

$$S_H = \frac{f}{Lg_1^2 B}\left[\left[\frac{\pi}{4}B^2 h_D + \frac{\pi}{4}\left[G^2 - B^2\right]h_T - \frac{\pi}{4}G^2 h_G\right](P+P_1) + W_0 h_G\right]$$

$$S_R = \frac{(1.33te+1)}{Lt^2 B}\left[\left[\frac{\pi}{4}B^2 h_D + \frac{\pi}{4}\left[G^2 - B^2\right]h_T - \frac{\pi}{4}G^2 h_G\right](P+P_1) + W_0 h_G\right]$$

$$S_T = \left[\frac{Y}{t^2 B} - Z\frac{(1.33te+1)}{Lt^2 B}\right]\left[\left[\frac{\pi}{4}B^2 h_D + \frac{\pi}{4}\left[G^2 - B^2\right]h_T - \frac{\pi}{4}G^2 h_G\right](P+P_1) + W_0 h_G\right]$$

Considering the gasket Seating Conditions are already passed, the stress acceptance criteria under operating conditions are given in table 4.16.9:

$$S_H \leq \min\left[1.5S_{f0}, 1.5S_{n0}\right]$$
$$S_R \leq S_{f0}$$
$$S_T \leq S_{f0}$$
$$S_H + S_R \leq 2S_{f0}$$
$$S_H + S_T \leq 2S_{f0}$$

By introducing the re-written stress expressions in the stress acceptance criteria, one may obtain –for a specific flange case- a limit of $P + P_1$ due to flange stress.

Note that the Kellogg procedure has assumed that the maximum of $P + P_1$ is P_{RATING}. In the

article *"Evaluation of Flanged Connections due to Piping Load"* **[9]**, McKeehan and Peng already evaluated the stress margin of this method.

B2. THE FLANGE RIGIDITY LIMIT

It was repeatedly said that "Flanges that have been designed based on allowable stress limit alone may be not sufficiently rigid to control leakage".

The ASME J factor is a measure of the flange rotation under the loads.

Basically the J factor is the report $\dfrac{\theta}{\theta_{\max}}$ where $\theta_{\max}$ is noted as K_R, subject to Code limitations.

Since $M_0 = \left[\dfrac{\pi}{4} B^2 h_D + \dfrac{\pi}{4} \left[G^2 - B^2 \right] h_T - \dfrac{\pi}{4} G^2 h_G \right] (P + P_1) + W_0 h_G$, the following relation (valid for

integral type flange) $J = \dfrac{52.14 V M_0}{L E_{y0} g_0^2 K_R h_0} \le 1$ may be reconsidered as:

$$P + P_1 \le \dfrac{\dfrac{L E_{y0} g_0^2 h_0}{52.14 V} K_R - W_0 h_G}{\left[\dfrac{\pi}{4} B^2 h_D + \dfrac{\pi}{4} \left[G^2 - B^2 \right] h_T - \dfrac{\pi}{4} G^2 h_G \right]}$$

This relation gives a "leakage limit" of the total pressure for integral type flange.
Similar relations may be written for other flange case (based on table 4.16.10 formulas).

Note that the Kellogg procedure has assumed that the maximum of $P + P_1$ is P_{RATING}.
Based on the above relations, one may evaluate the actual angle K_R corresponding to this approach. Alternatively, an inherent "$P + P_1 \le P_{RATING}$" method safety factor vs. Code requirements ($K_R = 0.3 \deg$ for integral type and $K_R = 0.2 \deg$ for loose type flange) can be evaluated.

C. A PRACTICAL APPROACH

As it was already highlighted, to further increase the $P + P_1$ over P_{RATING} means to analyze more accurately the bolts loads and the gasket load. For the piping flange case, the general opinion is overpassing P_{RATING} is not safety. However, in the same time it seems difficult to practically predict what is really happening.

In fact, this limit is the only consistent reason for which it is worth to consider the equivalent pressure concept.

The *B1* and *B2* limits are better described in their "natural" ASME VIII Div2 language. It appears to be more appropriate to consider the ASME VIII Div2 relations "as are" (for the stress evaluation and J factor evaluation or the K_R angle evaluation), or/and the Coade's flange calculator.

Probably, for piping flange checking case it is more consistent to consider the corroded thicknesses, somehow in line with the piping codes intentions.

REFERENCES

[1] 2007 ASME Boiler and Pressure Vessel Code Section VIII Division 2

[2] Companion Guide to the ASME Boiler and Pressure Vessel Code, volume 2, second edition, 2006, Edited by K. R. Rao

[3] Koves, W. J., Analysis of Flange Joints under External Loads, Journal of Pressure Vessel Technology, February 1996, volume 118, Issue 1, pp. 59-63

[4] Design of Piping Systems- the M.W. Kellogg Company, 1941, 1956, 1977, John Wiley & Sons

[5] Modern Flange Design, Taylor-Forge Bulletin 502

[6] 2007 ASME Boiler and Pressure Vessel Code Section VIII Division 1

[7] 2006 ASME Boiler and Pressure Vessel Code Section VIII Division 2

[8] 2000 ASME PCC-1, Guidelines for Pressure Boundary Bolted Flange Joint Assembly

[9] Evaluation of Flanged Connections due To Piping Load, McKeehan D.L. and Peng L-C, electronic version @ http://www.pipestress.com/Pages/PEpapers.html