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

This is a corollary of the formulas exposed in ASME Boiler and Pressure Code Section VIII Division 2, 2007 edition, Chapter 4.16 “Design rules for Flanged Joints”.

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):

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

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):

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

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

$$H_G = W_0 - H = 0.785G^2P + 2b\pi GmP - 0.785G^2P = 2b\pi GmP$$

We can also note $H_G = H_G^*P$ where $H_G^* = 2\pi mbG$

The factor m is the gasket factor, $m = 0$ only for self-energized type (and accordingly, $H_G = 0$ for this case).

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

The expression of M_0 may be written as:

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

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

we are able to mathematically speculate that:

$M_{0E} = \left(H_D^* h_D + H_T^* h_T + H_G^* h_G\right)P_1$ where P_1 is dimensionally a pressure

THEN

we can write

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

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 P_1 is given by $P_1 = \dfrac{M_{0E}}{H_D^* h_D + H_T^* h_T + H_G^* h_G}$

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

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

One may note the factor:

$$F_M = \frac{I}{0.3846\, I_P + I} = \frac{1}{1 + \dfrac{1}{2(1+\nu)}\dfrac{I_P}{I}}$$

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 "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.

Finally, the equivalent pressure evaluation:

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

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 <u>*tensile*</u> *net-section axial force and to neglect the compressive net- section forces.*

Personally I interpret this requirement as specific for design case.

When using for checking the piping flange, in my interpretation, one can consider also the real effect of the compressive force (presuming a sound engineering judgment has been done in terms there isn't another unanalyzed case in which is possible to have that axial force as tensile one).

The Kellogg formula may be recovered under the following simplifications:

IF

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

AND $\dfrac{h_T}{h_D} \rightarrow 1$

AND $m = 0$

THEN

$$P_1 = P_{KELLOGG} = \frac{\dfrac{4M_E}{G} + F_A}{\dfrac{\pi}{4}B^2 + \dfrac{\pi}{4}\left[G^2 - B^2\right]} = \frac{\dfrac{4M_E}{G} + F_A}{\dfrac{\pi}{4}G^2} = \frac{16M_E}{\pi G^3} + \frac{4F_A}{\pi G^2}$$

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

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:

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

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 Kellogg limit $P + P_1 \le P_{RATING}$ is an "ad- hoc" limit, definitely not a limit based on stress. It may be considered as a practical limit in order to "feel" the leakage danger, however without providing a measure of the safety factor.

B1. STRESS LIMIT

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

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

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

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

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

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

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 stress.

Note that the Kellogg procedure has assumed that the maximum of $P + P_1$ is P_{RATING}. In "Evaluation of Flanged Connections due to Piping Load", McKeehan and Peng already evaluated the consistent stress margin of this method.

B2. 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 $\frac{\theta}{\theta_{\max}}$ where $\theta_{\max}$ is noted as K_R, subject to Code limitations.

Since $M_0 = \left[\frac{\pi}{4}B^2 h_D + \frac{\pi}{4}\left[G^2 - B^2\right]h_T + 2\pi mbGh_G\right](P + P_1)$, the following relation (valid for integral type flange) $J = \frac{52.14VM_0}{LE_{y0}g_0^2 K_R h_0} \le 1$ may be reconsidered as:

$$P + P_1 \le \frac{LE_{y0}g_0^2 h_0}{52.14V\left[\frac{\pi}{4}B^2 h_D + \frac{\pi}{4}\left[G^2 - B^2\right]h_T + 2\pi mbGh_G\right]}K_R$$

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 Kellogg 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.