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Design Parameters Affecting Bolt Load in Ring-Type Gasketed Joints

A simplified formula for predicting the change in bolt load from preload to operating condition in a ring-type bolted flange connection is presented. The results clearly demonstrate the important dimensionless parameters that affect the magnitude and direction of the bolt load change.

Introduction

Standard flange design methods [1, 2] assume that bolt loads remain unchanged from "seating condition" to "operating condition." The seating condition corresponds to bolt preload with the gasket "seated" on the flange face. The operating condition implies a pressurized state of the flanged joint. Reference [3] contains an extensive list of publications which deal with the problem of bolt load variation due to changes in internal pressure. The significance of this effect is illustrated in a recent work by Singh [4, 5] where the bolt load variation is examined from the point of view of its effect on the thermal performance of heat exchangers. Papers by Bernhard [6], Murray and Stuart [7], among others, also explore the implications of the bolted joint flexibility on its sealworthiness. These studies are quite helpful in aiding a detailed stress analysis of flanged joint. However, they are of limited value to a designer seeking to optimize the structural characteristics of a flanged joint without the benefit of sophisticated tools of stress analysis. Our object, herein, is to study the anatomy of a simplified flanged joint, and to derive some nondimensional parameters which indicate whether bolt load will increase or decrease as the joint is pressurized. The physical significance of these parameters on sealing characteristics of the joint is also explained.

It is recognized that the long-term performance of a flanged joint depends on a host of other factors, such as bolt creep, gasket relaxation and hysteresis, temperature ramps in the joints, etc. The effect of these factors should be carefully considered in the flanged joint design process. Nevertheless, an insight into the gasket surface pressure and bolt load fluctuation with pressure changes based on an idealized linear elastic model is undeniably valuable.

Analysis and Discussion

Consider the flanged joint in Fig. 1 which shows two identical lap joint ring flanges prestressed with a total bolt preload W. This joint is commonly used in piping connections. The gasket is compressed by the bolt load W. The resultant gasket load is transmitted to the flange ring via the contact load between the flange and lap rings. The contact load is offset from the bolt load line in the radial direction by an amount h_1, and is equal to the gasket compression load in magnitude. Thus the flange ring is subject to a radially symmetric distributed moment Wh_1. The freebody of the flange ring is shown in Fig. 2. l denotes the prestressed bolt length; and θ denotes initial flange ring rotation due to the preload. Elementary ring theory relates ring rotation θ to applied moment Wh_1 [8].

$$\theta = \frac{Wh_1 D_m}{4\pi EI} \tag{1}$$

where

D_m = mean flange ring diameter
E = Young's modulus of flange ring
I = moment of inertia of rings section about its neutral axis of bending (0–0 in Fig. 2)

Force equilibrium requires

$$W = H_g \tag{2}$$

where H_g is the gasket preload. Fig. 3 shows the pressurized joint; the gasket bearing load is reduced to $H_g{}'$ and the flange rotation acquires a new value θ'.

Axial equilibrium now yields

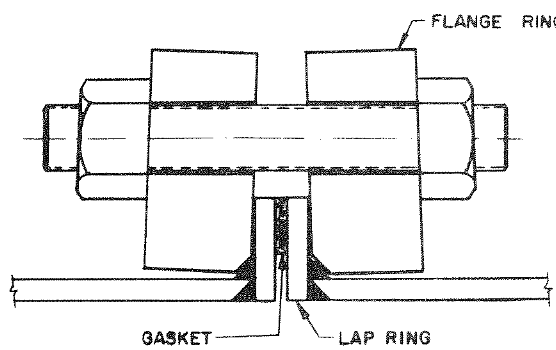

Fig. 1 Lap joint flange assembly

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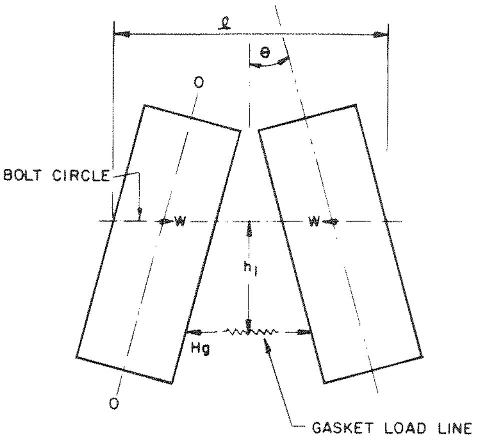

Fig. 2 Pair of ring flanges under seating condition

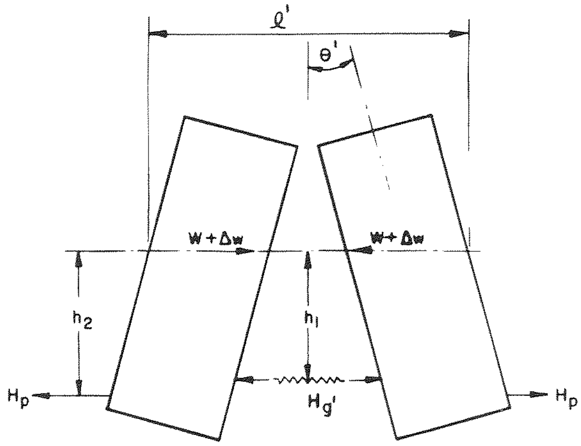

Fig. 3 Pair of ring flanges under pressurized condition

$$H_g' = W + \Delta W - H_p \tag{3}$$

where ΔW is the increase in bolt load. As before, the relationship between ring rotation and applied moment yields

$$\theta' = \frac{(H_g' h_1 + H_p h_2) D_m}{4\pi EI}$$

where

$$\theta' = \frac{[(W+\Delta W)h_1 + H_p(h_2 - h_1)]D_m}{4\pi EI} \tag{4}$$

Let K_b and K_g define the spring stiffness of the bolt and gasket (unloading modulus), respectively. Hence

$$\Delta W = K_b(l' - l) \tag{5}$$

where l' is the bolt length after pressurization. Referring to Figs. 2 and 3, the total relaxation of the gasket δ_g is given by

$$\delta_g = (l' - l) + 2(\theta' - \theta)h_1 \tag{6}$$

Furthermore

$$\delta_g = \frac{H_g - H_g'}{K_g} = \frac{H_p - \Delta W}{K_g} \tag{7}$$

Substituting for δ_g from equation (6), θ' and θ from equations (4) and (1), respectively; we have, after some manipulation,

$$\frac{\Delta W}{H_p} = \frac{1 - \dfrac{h_1(h_2 - h_1)D_m K_g}{2\pi EI}}{1 + \dfrac{K_g}{K_b} + \dfrac{h_1^2 D_m K_g}{2\pi EI}} \tag{8}$$

Let

$$\frac{K_g}{K_b} = \alpha_1 \tag{9a}$$

$$\frac{K_g D_m h_1^2}{2\pi EI} = \alpha_2 \tag{9b}$$

$$\frac{h_2 - h_1}{h_1} = \alpha_3 \tag{9c}$$

then

$$\frac{\Delta W}{H_p} = \frac{1 - \alpha_2\alpha_3}{1 + \alpha_1 + \alpha_2} \tag{10}$$

Equation (10) gives the fractional increase in bolt load with respect to the pressure load. We note that if

$$\alpha_2\alpha_3 > 1$$

then ΔW will be negative. This is clearly undesirable since, from equations (2) and (3), the drop in the gasket load is given by

$$\Delta H_g = H_g - H_g' = H_p - \Delta W$$

which may be written using equations (9) and (10) as

$$\Delta H_g = H_p\left[\frac{\alpha_1 + \alpha_2 + \alpha_2\alpha_3}{\alpha_1 + \alpha_2 + 1}\right]$$

From the foregoing equation, it can be deduced that the design goal should be to minimize α_1, α_2, and α_3. Ideally, α_3 should be made zero, although in practical designs, this is usually not achievable. The following additional deductions from the foregoing equations give useful guidelines in flange design:

(*a*) Small α_1 is a desirable design goal. Thus a gasket with small unloading modulus is preferable to one with a larger modulus. However, there are manufacturing limitations on how small K_g can be made for a particular application. However, the bolt stiffness K_b usually can be increased substantially by increasing total bolt tensile stress area. Thus,

Nomenclature

D_m = ring flange mean diameter
E = Young's modulus of flange ring material
h_1 = radial distance of bolt centerline from mean gasket line
h_2 = radial distance of bolt centerline from hydrostatic load line
H_g = gasket preload (seating condition)
H_g' = gasket load under pressurized condition
H_p = header load due to pressure
I = moment of inertia of ring
K_b = spring stiffness of body bolts
K_g = spring stiffness of gasket in unloading
l' = bolt length after pressurization
l = prestressed bolt length
W = total bolt preload
ΔW = increase in bolt load from seating to pressurized condition
θ = initial flange ring rotation due to preload
θ' = flange ring rotation under operating (pressurized condition)
δ_g = change in gasket thickness from seating to pressurized condition

a flanged joint with a greater bolt cross-sectional area and lower prestress to produce a given preload is better than one with a smaller sectional area prestressed to a proportionately larger prestress. Unfortunately, the ASME Code design rules appear to deprive the designer of this important latitude by mandating a certain bolt prestress for a bolting material. Stiffness can also be increased by reducing bolt effective length l. This can be accomplished by using tapped holes in one of the flanges instead of an end nut.

(*b*) Reduction in α_2 can be accomplished by reducing h_1. Designers, however, try to minimize h_1 to reduce flange moment. A further reduction is usually not feasible. Also, increasing the flange ring moment of inertia, I, decreases α_2 and hence helps increase ΔW.

(*c*) Our results show clearly that large diameter flanges are harder to seal than small diameter units, since α_2 increases linearly with D_m.

The foregoing results are deduced from a simplified ring flange model. The behavior of more complex flange geometries, such as a tapered hub design, is more involved. Nevertheless, the qualitative remarks stated in the foregoing still hold, and may be utilized to gain additional insight into all raised face external flange designs.

Example: To illustrate some of the concepts derived in the foregoing, let us consider a ring flange 27 in. o.d. (686 mm) by 22 in. i.d. (559 mm), 3 in. (76 mm) thick bolted to an identical flange with 24 bolts, 5/8 in. diameter (16 mm) and 7 in. (178 mm) long (effective tension length). Assuming 25000 psi (172.4 MPa) prestress in bolts, we have W = 121200 lb (539 KN). The flange ring moment of inertia, I, is 5.63 in^4 (234.3 cm^4); and the material Young's modulus is 30×10^6 psi (0.207×10^6 MPa). The gasket unloading modulus is 8.08×10^6 lb/in. (1.415 GN/m). Finally b_1 = 1 in. (2.54 cm); h_2 = 1.25 in. (3.18 cm). Results computed from equations (8)-(10) for $K_B = 20.8 \times 10^6$ lb/in. (3.642 GN/m) (7 in. or 177.8 mm long bolt); and $K_B = 32.4\ 10^6$ lb/in. (5.673 GN/m) are given in the foregoing.

$K_B \times 10^{-6}$ lb/in.	α_1	α_2	α_3	$\Delta W/H_p$	$\Delta H_g/H_g$
20.8	0.39	0.19	0.25	0.60	0.40
32.4	0.25	0.19	0.25	0.66	0.34

We note that, in both cases, the increase in bolt load is a substantial fraction of the imposed pressure load. The fractional increase is greater for the stiffer bolt geometry. The fractional drop in the gasket load is also smaller for the stiffer bolt geometry.

Conclusions

A simple formula has been evolved to show parameters which affect the change in bolt load and gasket compression load between the preload condition and the operating condition in ring-type bolted flange connections. The results can be used to quickly check the probable ability of the design to effect a leak tight joint.

While the method presented here is not intended to be a rigorous substitute for more involved design procedures, it provides a tool to perform trend studies to select the most suitable design variables for a particular problem.

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