The Design of Aluminium Bolted Flange Connections *

(Part 1 Relationship between Leak Properties and
Gasket Loads in Integral Hub Flanges)

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Water leak tests were carried out on aluminium and steel bolted integral hub flange connections of the Japanese Industrial Standards (J.I.S.) 10 kgf/cm² class and a nominal pipe diameter of 150 mm.

Bolt loads were measured by strain gauges. Rates of water leakage are detected by a leak detector made of copper wires covered with cotton cloth. The leak detector is very sensitive to leakage and can detect leakage early before a droplet is observed.

It is shown that the leak detector output correlates with the calculated gasket load.

It is proposed that the highest operating pressure for aluminium flanges be made equal to that of steel flanges of identical dimensions, after confirming that the stress conditions in the flange satisfy the J.I.S. rules.

Key Words : Structural Analysis, Aluminium Flanges, Water Leak Tests, Bolted Flange Connections, Bolt Loads, Gasket Loads, Flexibility Analysis

1. Introduction

For the last fifty years, there has been great progress in designing of bolted flange connections (5)-(7), evaluation of local stresses (8), and prediction of gasket sealing properties (9)-(12).

The designing of bolted flange connections with ring gasket is usually based on the stress analysis specified by appropriate industrial code definitions such as J.I.S. (1), A.S.M.E. (3), D.I.N. (4) and so forth. In the code definitions, a required gasket load for preventing leakage is given as a gasket design load based on the gasket factor m for operating conditions and the gasket unit seating stress y for gasket seatings (1),(3).

The code definitions give a relatively precise and theoretical basis to the stress analysis of flange ring-hub connections; however, the definitions on gasket loads under pressurized conditions are not sufficient to investigate leakage in more detail. It is necessary to make clear the relationship between leakage and gasket loads under pressurized conditions, for further analysis of the bolted flange connections. For further analysis, changes in the gasket load from gasket seating conditions to operating conditions should be investigated by taking into consideration flange flexibility because the gasket load is to be defined as a function of the bending flexibility of the flanges.

Some investigations for evaluating the gasket factor have been carried out, but in them the effects of flange flexibility on the gasket load are not considered. The gasket factors were evaluated for very rigid bolted flange connections (9)-(10).

Currently, steel flanges specified by J.I.S. (2) have been widely used in various areas of industry without any critical leakage problems. The reason is that steels for flange materials possess an appropriate ratio of allowable stress to Young's modulus for a flange design based only on stress analysis, that is, steel flanges usually possess high rigidity. In the case of aluminium alloys in which the ratio is higher than that in steels, leakage is possible even though the stress condition of the flange satisfies the code definitions.

In this paper, the gasket load for bolted flange connections with raised face and ring gasket under pressurized conditions is investigated. Gasket properties are assumed taking the gasket as an elastic body in terms of the tangent modulus. Comparisons in leak tests between aluminium alloy and steel flanges with identical dimensions are made in order to establish the design rules for aluminium flanges particularly regarding leakage.

2. Test Unit

2.1 Flanges

Flange dimensions are determined in accordance with J.I.S. B2212 (Basic Dimensions of 10 kgf/cm² Ferrous Materials Pipe Flanges). Two sets of flanges are made of aluminium alloy A5083 and structural steel SS 41. Nominal size is 150A. Dimensions

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are shown in Figure 1 and Figure 2.

Stress conditions of both flanges satisfy the design rule at the design pressure of 1 MPa and the design bolt load W_{m2} (1) for gasket seating conditions derived from the unit gasket seating stress y shown below.

Figure 1. Dimensions of aluminium flange

Figure 2. Dimensions of steel flange

2.2 Gaskets

Gasket materials are PTFE with inorganic fillers. Dimensions and specifications are as follows:

- thickness, 2 mm
- inner diameter, 167 mm
- outer diameter, 220 mm
- gasket factor m , $m = 1.6$
- unit gasket seating stress y , $y = 18.6$ MPa

* These values are recommended by the manufacturer of the gaskets.

3. Experiments

3.1 Pressure tests

A pair of identical flanges were provided for the pressure tests after bolting with eight bolts to the gasket seating loads W_{m2} defined by J.I.S. B8243 Appendix II.

Water was used as pressurized. In the pressure tests, increments in pressure were 0.1 MPa and the pressure was held for 5 min. at each stage. Measurements on bolt loads and leak rates were made at each pressure stage. Bolt loads were measured at the start and the end of the holding time. Leakage was measured every minute during the holding time.

3.2 Measurements of bolt loads

Strain gauges were attached on each bolt. Bolt loads are determined as a sum

of the strains on each bolt. Bolts used for tests are designated as M20 (Metric Coarse Screw Threads) with pitch of 2.5. Each bolt strain was calibrated to loads prior to the tests.

3.3 Leak detector

The leak detector is a copper wire covered with cotton cloth. This wire is wound on outside of the gasket prior to the tests. Leakage is detected by changes in electrical resistance between the flange and the wire. A strain gauge bridge circuit is made of this wire. Changes in resistance are observed as strains.

Leak detector is positioned as shown in Figure 3, to detect the leakage from each surface of the gasket.

Figure 3. Arrangement of leak detector

3.4 Bolt seating load

The bolt seating load P_0^* was determined to be 100.9 kN according to the definitions given in J.I.S. B8243 Appendix II (12.6 kN per each bolt). The bolt seating load was not necessarily equal to the code definitions in the tests, because the purpose of the tests was to clarify the relationship between leakage and gasket loads.

3.5 Experimental conditions

Several pressure tests were carried out on aluminium alloy and steel flanges. Flanges were used repeatedly and new gaskets were provided for each test.

The gaskets came from two manufacturer's lot numbers which are denoted in the results.

4. Experimental results

4.1 Bolt loads

Changes in bolt loads are shown in Figure 4. In Figure 4, Al denotes aluminium flanges and Fe denotes steel flanges. The bolt loads of aluminium alloy flanges decrease more than those of steel flanges as the pressure increases. This comes from the difference in flange flexibility. Variance in bolt strains is within 3 per cent of the average value.

4.2 Leakage

Leak detector outputs are shown in Figure 5. At the outset of the study, the leak detector outputs were tentatively correlated to actual leak rates. A quantitative relationship was, however, not established. Leakage from the flange connections was evaluated by the leak detector outputs, assuming the outputs are proportional to actual leak rates.

It was confirmed through a series of experiments that no changes in the leak detector outputs were observed over a long time before pressure was applied. Drop-lets were observed when the leak detector outputs reached -10000 to $-20000 \mu\text{m/m}$.

It is observed from Figure 5 that the leakage does not occur abruptly. A small amount of leakage was detected even at quite low pressure. It was found that leakage occurred at the gasket-flange interface rather than through the gasket material itself.

Figure 5 shows that the leak properties are strongly affected by flange flexibility. Differences in the gasket loads between aluminium alloy and steel flanges are presumed to be considerably wide, from the changes in the bolt loads measured, as the pressure increases. The leak properties are also affected by the gasket lot and the both sides of the gasket. Differences in the leak properties between the both sides of the gasket are considered to be due to an imperfectly square inside crosssection caused by a punching of the

gasket out of the original sheet.

Figure 4 Changes in bolt load

Figure 5 Changes in leak detector output

5. Analysis

5.1 Models of flanges for calculation

Bolted flange connections with a hub are assumed as in Figure 6, in order to analyze the effect of the flange flexibility on the gasket loads. This is an approximation but would be practically useful to determine the gasket loads under gasket seating and operating conditions. The loads acting on the flange ring are shown in Figure 7.

5.2 Notations

A pair of flanges is assumed for the analysis. Lower suffix 1 is given for one side of the flanges, and lower suffix 2 for the other. Upper suffix o is given for the gasket seating condition, and upper suffix p for the operating condition.

M_t : torsional moment per unit length along the centroid of the cross section of the flange ring

P_b : bolt load (total load of eight bolts)

P_g : gasket load

P_1 : hydrostatic end force on the area between the diameter at the location of the gasket load reaction, and the inner diameter of the flange

P_o : hydrostatic end force on the area inside of the flange

θ : rotation angle of flange ring

E : modulus of elasticity of materials (suffix f for flanges, suffix b for bolt and suffix g for gasket)

l_{ob} : initial length of bolt

A_b : cross sectional area of one bolt based on groove of thread

N : number of bolts

t_g : initial thickness of gasket

A_g : contact area of gasket and flange face

p : inner pressure

$C_o = (A+B)/2$

$l_{ob} = t_{f1} + t_{f2} + t_g$

t_{f1} : thickness of flange ring

$t_1 = (t_{h1} + t_{hs})/2$

t_1 , t_{h1} and t_{hs} are shown in Figure 6.

Figure 6 Model for analysis

Figure 7 Loads acting on flange ring

5.3 Deflection of flange ring at gasket seating

Rotation of a pair of flange rings is expressed as follows:

$$\theta_i^o = -\frac{h_c^o}{2\pi} P_g^o \left/ \left(D_{fi} + \frac{B}{2} K_{fi} - \frac{B t_{fi}}{4} J_{fi} \right) \right. \quad (1)$$

where $i=1, 2$

$$h_c^o = (C - G^o)/2$$

$$D_{fi} = E_{fi} t_{fi}^3 \ln(A/B)/12$$

Values of K and J are obtained as functions of the dimensions of the hub and pipe connected, and the modulus of elasticity of the materials.

5.4 Bolt load and gasket load under pressurized conditions

Bolt loads and gasket loads under pressurized conditions are given by taking into consideration the extension of the bolts, the compression of the gasket and the relationship of $P_b = P_g + P_o + P_1$, as follows:

$$P_b = P_g^o + \frac{R_g}{R_b + R_g} \left\{ \frac{\pi (G^p)^2}{4} p - \frac{h_c^o}{R_g} (\theta_1^o + \theta_2^o) + \frac{h_c^o}{R_g} (\theta_1^p + \theta_2^p) \right\} \quad (2)$$

$$P_g = P_g^o - \frac{R_b}{R_b + R_g} \left\{ \frac{\pi (G^p)^2}{4} p + \frac{h_c^o}{R_b} (\theta_1^o + \theta_2^o) - \frac{h_c^o}{R_b} (\theta_1^p + \theta_2^p) \right\} \quad (3)$$

where $h_c^o = (C - G^o)/2$

$$R_b = l_{ob}/(N A_b E_b), \quad R_g = t_g/(A_g E_g)$$

Rotation angles under pressurized conditions θ_1^p and θ_2^p are determined by Equations (2) and (3). Diameters at the location of the gasket load reaction G^o and G^p , and contact area A_g should be introduced to determine θ_1^o and θ_2^o . G^o , G^p and A_g will be derived as functions of flange flexibility, as shown in next articles.

5.5 Determination of gasket contact area and diameter at the location of the gasket load reaction

5.5.1 Gasket seating condition

Consider the gasket compressed, as shown in Figure 8, by the rotation of flange ring. The compressed deflection of gasket at the location of the centroid of cross section of the flange ring is given as follows, by assuming a pair of flanges with identical dimensions.

$$y^0 = 2(r_i^0 - r)\theta^0 \dots\dots\dots (4)$$

where $\theta_1^0 = \theta_2^0 = \theta^0$
 $r_1^0 = r_2^0 = r_i^0$

Gasket load is calculated by the deflection of the gasket.

$$P_g = K_0 \int_{r_i^0}^{r_0} (r_i^0 - r) r dr \dots\dots\dots (5)$$

Putting $P_g = P_g^0$, the following third order equation about r_i^0 is obtained.

$$(r_i^0)^3 - 3r_0^2(r_i^0) + 2r_0^3 + 6P_g^0/K_0 = 0 \dots\dots\dots (6)$$

where $K_0 = 4\pi(E_{g0}/t_g)\theta^0$
 E_{g0} : modulus of elasticity of the gasket under loading conditions

In Figure 8,

- r_0 : smaller outside diameter between gasket and flange facing
- b_n : smaller dimensions of gasket width or radial contact width between gasket and flange facing before bolting

Radius at location of gasket load reaction is defined as follows:

$$\int_{r_i^0}^{r_0} (r_i^0 - r) r dr = \int_{r_g^0}^{r_0} (r_i^0 - r) r dr$$

5.5.2 Pressurized condition

The rotation angles of the flange rings increase when a pressure is applied. The displacement at the centroid between flange rings occurs by hydrostatic end loading. The compressed condition of the gasket changes from the condition shown in Figure 8 to that shown in Figure 9. Compressed deflection of the gasket at the centroid under the gasket seating condition y^0 is obtained by putting $r = C_0/2$ in Equation (4) as follows:

$$y_c^0 = 2(r_i^0 - C_0/2)\theta^0 \dots\dots\dots (10)$$

The compressed deflection of the gasket at the centroid under the pressurized condition is given based on that under the seating condition as follows:

$$y^p = 2(C_0/2 - r)\theta^p + y_c^0 - \delta_0 \dots\dots\dots (11)$$

where

$$\delta_0 = \frac{R_b R_g}{R_b + R_g} \left\{ \frac{\pi (G^p)^2}{4} p + \left(\frac{1}{R_b} \frac{C - C_0}{2} - \frac{1}{R_g} \frac{C_0 - G^0}{2} \right) (\theta_1^0 + \theta_2^0) - \left(\frac{1}{R_b} \frac{C - C_0}{2} - \frac{1}{R_g} \frac{C_0 - G^p}{2} \right) (\theta_1^p + \theta_2^p) \right\} \dots\dots\dots (12)$$

Equation (12) is given for the unloading condition of the gasket. Accordingly, R_g is determined by the unloading modulus of elasticity of the gasket E_g .

The radius at the inside contact area r_i^p is obtained by putting $y^p = 0$ in Equation (11) as follows:

$$r_i^p = \frac{C_0}{2} + \frac{y_c^0 - \delta_0}{2\theta^p} \dots\dots\dots (13)$$

The radius at the gasket load reaction under the pressurized condition r_g^p is determined by the following third order equations.

In the case of $r_i^p \geq r_0 - b_n$,

$$(r_g^p)^3 - 1.5r_i^p(r_g^p)^2 - 0.5\{(r_i^p)^3 + r_0^3\} + 0.75r_i^p\{(r_i^p)^2 + r_0^2\} = 0 \dots\dots\dots (14)$$

In the case of $r_i^p < r_0 - b_n$,

$$(r_g^p)^3 - 1.5r_i^p(r_g^p)^2 - 0.5\{(r_0 - b_n)^3 + r_0^3\} + 0.75r_i^p\{(r_0 - b_n)^2 + r_0^2\} = 0 \dots\dots\dots (15)$$

In the above equations, r_g is determined by iteration. Bolt loads and gasket loads at a given pressure are obtained as functions of the dimensions of the flange units and the modulus of elasticity of the flange materials and gasket, by using r_g determined.

Figure 8 Deflection of gasket under gasket seating

r_g^0 is obtained as a solution to the following third order equation.

$$(r_g^0)^3 - 1.5r_i^0(r_g^0)^2 - 0.5\{(r_i^0)^3 + r_0^3\} + 0.75r_i^0\{(r_i^0)^2 + r_0^2\} = 0 \dots\dots\dots (7)$$

where $G^0 = 2r_g^0$

In the case of $r_i^0 < r_0 - b_n$, r_i^0 and r_g^0 are determined as follows:

$$r_i^0 = \frac{2}{3} \frac{\{r_0^3 - (r_0 - b_n)^3\}}{\{r_0^2 - (r_0 - b_n)^2\}} + \frac{2P_g^0/K_0}{\{r_0^2 - (r_0 - b_n)^2\}} \dots\dots\dots (8)$$

$$(r_g^0)^3 - 1.5r_i^0(r_g^0)^2 - 0.5\{(r_0 - b_n)^3 + r_0^3\} + 0.75r_i^0\{(r_0 - b_n)^2 + r_0^2\} = 0 \dots\dots (9)$$

The contact area A_g^0 of the gasket and the flange facing is calculated by radial width $r_0 - r_i^0$.

Figure 9 Recovery from deflection of gasket in pressurized condition

6. Relationship between leak properties and gasket loads

The relationship between leak properties and gasket loads is derived from the experimental results and the calculations. The values of E_g and E_{g0} are determined through compression tests with specimens of 50 mm in diameter. They are,

$E_{g0} = 1$ GPa (at the highest compressive load)

$E_g = 2$ to 1 GPa (at around 0 - 0.015 mm in unloading deflection)

The values of E_g are taken as 1.1 GPa for aluminium flanges and 1.7 GPa for steel flanges by taking into consideration the experimental results, shown in Figure 4. It is expected that the differences in the values of E_g result from the differences in unloading deflections between them. Figure 10 shows the calculated results about bolt loads and gasket loads together with some of the experimental results.

Figure 10 Calculated results about bolt load and gasket load

Figure 11 shows the relationship between leak detector outputs and calculated gasket loads. The leak detector outputs increase as the gasket loads decrease, that is, the pressure increases. Some difference is recognized in the results between No. 1 and No. 2 gaskets. Leakage of No. 2 gasket increases particularly sharply. It is found that the leak detector outputs correlate with the gasket loads in spite of the flange flexibility, although a relatively wide scatter band has to be assumed. The experiment No. 6 Al in which the gasket seating load is 75 percent of the others, is also contained in the correlations.

Assuming the limit of leak detector outputs in Figure 11, the gasket load corresponding to the limit can be obtained. The highest operating pressure can be determined as the highest pressure capable of maintaining a required gasket load by calculating the gasket load as a function of pressure. It is, however, difficult to

define the limit of leak detector output. It is desirable that the limit be determined based on flanges which have experienced service with reliability.

Figure 11 Correlation of leak detector output to gasket load

As the steel flanges used here have long been in service, without leakage, up to 1 MPa, a gasket load at 1 MPa can be considered a reference. As shown in Figure 10, the highest pressure capable of maintaining the gasket load is 0.8 MPa in the case of aluminium flanges. Consequently, it is necessary to limit the highest operating pressure of aluminium flanges to 80 percent of that of steel flanges.

These calculations can be useful to determine the thickness of the flange ring of an aluminium flange capable of maintaining the same gasket load as that of a steel flange.

7. Conclusions

The effects of flange flexibility on the changes in bolt loads and leak properties were confirmed by pressure tests on aluminium and steel flanges having almost identical dimensions.

It is found that the leak detector outputs correlate with the calculated gasket loads.

The highest operating pressure for aluminium flanges is determined on the basis of the equivalent gasket load on steel flanges at the same pressure. Such an approach may be taken to the design of flanges made of materials having a higher ratio of allowable stress to modulus of elasticity than the conventional steel. It seems that the highest operating pressure for the aluminium flanges discussed here should be limited to 0.8 MPa even when the flange satisfies the stress code up to 1 MPa of pressure.

In the case of aluminium flanges, the highest operating pressure should be determined by:

1. Calculating design pressure according to J.I.S. design codes.

2. Calculating the highest operating pressure based on gasket loading.
3. Choosing the lower pressure.

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