Strength of Integral Pipe Flanges*  
(No. 2 Gasket Seating Stress and the Influence of Number of Bolts)

By Kunio NISHIOKA**, Yoshiyasu MORITA*** and Hisakazu KAWASHIMA****

The leakage of inner fluid through the gasket seat is influenced by the gasket material, inner fluid, the fastened force of the bolts, etc. Especially, the pressure distribution on the gasket seat is considered the most important factor for the leakage.

In this report, the effective seating width of the gasket was analyzed by FEM and was compared with the values given in JIS and ASME Standards.

Moreover, the minimum number of bolts required to get almost uniform pressure on the gasket seat was determined. The effect of the number of bolts on the flange stress was also analyzed and discussed.

1. Introduction

There are two practical problems with pipe flanges, that is (1) the security for the stress generating under making up and operating conditions and (2) the leakage of inner fluid along the gasketed joints. As for (1), in the previous report [1], we investigated in detail the stress distribution of flanges obtained by the finite element method in comparison with the results by stress formulas in the present Standards. Here, in this report the results from our investigations on (2) are described. In practice, most troubles with pipe flanges are caused by the leakage of inner fluid. To prevent the leakage, a sufficient compressive load by bolts should be applied on the contact surface between flange and gasket. In the Standards, the compressive stress (gasket seating pressure) on the contact surface is set as more than $m$ (gasket factor) times as large as inner fluid pressure.

These phenomena of the leakage are influenced by materials, dimensions and surface roughness of gaskets, shapes and surface roughness of flange facings, kinds of inner fluid and its temperature, etc. and these factors vary the gasket factor. Among these factors, the influence of gasket seating pressure is the largest and has been investigated by various experiments. [2][5] However, as far as we know, a theoretical investigation on it has not been carried out sufficiently. [4][6] Therefore, in this report we try to clarify the leakage phenomenon through analysis of gasket seating pressure by the finite element method.

In standards [7][10] the bolt load which gives gasket seating pressure is assumed to be axisymmetrical, that is, distributes uniformly on the circumference of the bolt center circle. In the case of a large number of bolts, this assumption is considered reasonable as reported in the previous paper. In practice, however, it is desired to reduce the number of bolts considering the actual operation. Therefore, from a practical point of view, it is very important to make clear how the number of bolts influences the distribution of gasket seating pressures.

First of all, the distribution of gasket seating pressures is analyzed on the assumption that the bolt load distributes axisymmetrically. In the next place, the gasket seating pressure under the limited number of bolts is investigated.

2. Flange contour and bolt load

In this report, the nominal diameter of the flange analyzed is 300 mm based on "Basic dimension of 100 kg/cm² steel pipe flanges, JIS (proposa)" [11] which has been described in detail in our first report. The gasket used is made of mild steel.

The bolt load $P_b$ required to prevent the leakage of inner fluid is given in the Standards as follows:

$$P_b = P_1 + P_2 + P_3$$

(1)

where

$$P_1 = \frac{P}{100} \cdot \frac{\pi}{4} \cdot d^2$$

(2)

$$P_2 = \frac{P}{100} \cdot \frac{\pi}{4} \cdot (d^2 - d^1)$$

(3)

$$P_3 = \frac{P}{100} \cdot 2\pi b Gm$$

(4)

Symbols used in these Eqs. (1) - (4) are shown in Fig. 1 and the details are given in our first paper.

* Received 27th June, 1977
** Assistant General Manager of Central Research Laboratories
*** Senior Research Engineer
**** Research Engineer
Central Research Laboratories, Sumitomo Metal Industries, Ltd., Amagasaki.
3. The gasket seating pressure

3.1 The analysis

On the assumption that the contour of the flange and the bolt load are axisymmetrical, the stresses on the gasket seating are studied by the finite element method. The finite element layout is shown in Fig. 2. Figure 3 shows an enlarged area near the gasket. As for the element, a triangle ring element of uniform strain is used on the condition that its minimum length is 2 mm at the stress concentrated portion of hub and 1.5 mm in the neighborhood of the gasket. For the boundary condition, because of the symmetry of a couple of flanges, the displacement at the middle plane of gasket is fixed in the axial direction, and free in the radial direction. On the other hand, as shown in our previous report, two load conditions below are studied.

(1) when bolt load is acting (gasket seating condition)

(2) when bolt load and inner pressure are acting (operating condition)

In the case of (1), flange and gasket are calculated integrally, and the axial stress $\sigma_2$ of the gasket element on the gasket seating surface is regarded as the gasket seating pressure.

In the case of (2), it should be considered as the boundary condition that the penetration of inner fluid along the contact surface between flange and gasket reduces the effective width of gasket. First of all, the penetrating zone of inner fluid is determined as discussed in detail in our first report, and the gasket seating pressure is analyzed, considering the bolt load $P_b$, the inner pressure $p$ acting on the inner surface of flange, and the hydrostatic end force $P_1$ on area inside of flange.

3.2 The effective width of gasket

In Fig. 4, the solid line shows the pressure distribution at the gasket seat on the assumption that only the bolt load $P_b$ is acting. The acting point $Q$ of the concentrated joint-contact surface compression load $P_s$ is calculated by Eq. (5) from the balance of moment.

$$R = \int_{R_1}^{R_2} \frac{pg r dr}{P_s}$$

$$P_s = \int_{R_1}^{R_2} pg \frac{dr}{R}$$

$R$: radius at location of load $P_s$

$R_1$: inside radius of gasket

$R_2$: outside radius of gasket

$P_g$: gasket seating pressure

The effective width $b$ of gasket specified in the stress formulas of JIS and ASME is regarded as the width from the application point of $P$ to the gasket outside ($Q_T$ shown in Fig. 4) and is given as follows:

$$b = \frac{b_0}{2.52 \sqrt{b_0}}$$

where $b_0$ is the basic gasket width determined by the shape of gasket and flange facing, and in this case, is $N/2$, where $N$ is gasket width.

Figure 4 also shows the effective gasket width obtained from the formulas in the Standard and $Q_T/S_T$ becomes 0.46. The calcu-
Fig. 4 Gasket seating pressure and effective width (effect of hub taper)

Theoretical value $\frac{QT}{ST}$ by the finite element method becomes 0.355 and the effective width calculated in this case is smaller than the one in the Standards. The analysis by the finite element method is carried out under a complete elastic state, but the pressure in the neighborhood of gasket outside becomes large above the yielding point. Thus, in practice, it is considered that the gasket yields and its pressure is reduced slightly. In the Standards, though the yielding point of gasket is not specified, the minimum effective seating pressure is set at 12.66 kg/mm². The minimum effective seating pressure required to make tight seal between gasket and flange can be equal to a pressure to make the gasket surface follow the contour of flange facing. Therefore, it may be almost equal to the yielding point of gasket material. The gasket seating pressure obtained from the elasto-plastic stress analysis based on the assumption that the gasket of the complete elasto-plastic material yields at a stress of 12.66 kg/mm² is shown in Fig. 5.

It is clear from Fig. 5 that the gasket seating pressure obtained from the elasto-plastic analysis is smaller in the neighborhood of gasket outside than that from the elastic analysis, and becomes larger in the neighborhood of gasket inside. It is found that there is a slight difference between two values of $\frac{QT}{ST}$ which are 0.359 and 0.355 in the elasto-plastic and the elastic analysis, respectively. Consequently, the gasket seating pressure can be analyzed sufficiently by the elastic analysis, except the area in the neighborhood of gasket outside. Hence, the investigations as described below are performed by means of the elastic analysis.

In the next place, it is investigated how the gasket seating pressure is affected by the taper of hub, or the radius of curvature at the transition from hub to ring. As for the effect of radius, it is clear that the gasket seating pressure is not affected at all by a radius between 20 mm and 40 mm. With respect to the taper of hub, two cases of 1/1.25 and 1/5.0 besides 1/2.5 in the Standard are calculated and are shown by the broken and the dot-dash line in Fig. 4 respectively. This figure shows clearly that the gasket seating pressure at outside becomes larger when the taper is smaller. Since diameter $d_2$ of hub end is kept constant in this analysis, the smaller the taper becomes, the larger the rigidity of hub grows and the rotating angle of ring due to the bending moment becomes smaller. Hence, it is noticed that the effective gasket width increases with a decreasing taper of hub. However, judging from the absolute value, the width is not much influenced by the taper of hub.

3.3 The penetrating zone of inner fluid

As in the previous report, the penetrating zone of inner fluid is determined by the condition that inner fluid never penetrates into the area where the gasket seating pressure exceeds the inner pressure.

The gasket seating pressure distribution at the flange specified in the Standard is shown in Fig. 6 by a solid line. From the deformations of the flange and gasket shown in Fig. 7, it is found that there is a clearance in a portion between flange and gasket. Consequently, inner fluid soaks into the clearance, and the inner pressure acts on both gasket and flange in the penetrating zone.

In Fig. 6, besides 1/2.5 taper in the
pressure is affected by the number of bolts. In the same manner as the gasket seating pressure, the stresses in the flange are affected by the number of bolts and the maximum stress generates in a cross section through the bolt center. Therefore, it is very important to clarify the influence of the number of bolts from a standpoint of the flange strength.

4.1 Analysis
In order to investigate the effect of the number of bolts, the gasket seating pressure and the stresses in the flange under only one bolt are analyzed. By superposing these results, it is possible to obtain the stress distribution in the case of an arbitrary number of bolts. The flange is assumed to be an axisymmetrical body. The solution for one bolt is obtained by expanding the bolt load into a Fourier series in the circumferential direction and superposing them. Considering the convergency of the solution and the whole contour of the flange, the load by one bolt is assumed to act on the center of bolt in the radial direction and distribute circumferentially in cosine form with a narrow width of 0.01 rad. With these assumptions, the bolt load is treated as an almost concentrated one. The distribution of bolt loads, q, is shown in Fig. 8 and given as a function of the maximum value \( q_c \) and the angle \( \theta \).

4. The effect of the number of bolts
In the above analysis, the bolt load \( P_b \) has been regarded as an axisymmetrical load. However, as flanges are connected to each other by a limited number of bolts, the pressure on the gasket seat becomes higher in a cross section through the bolt center but becomes lower at the middle of each bolt. As the reduction of the gasket seating pressure causes a leakage of inner fluid; it is desirable that the pressure distribution be as uniform as possible. On the other hand, in the case of making up flanges it is desired to reduce the number of bolts. In order to find out the proper number of bolts, it is investigated how the gasket seating pressure is affected by the number of bolts.
Equation (9) can be expanded into a Fourier series in the range from \( -\phi \) to \( \phi \) as follows:

\[
q = \sum_{k=0}^{\infty} q_k \cos k\theta
\]

where

\[
q_k = \frac{\pi B}{4\phi R} \left( \frac{\pi}{2\phi} \right)^2 - k^2 \cos k\phi
\]

\( k = 1, 2, 3, \ldots \)

With respect to the flange whose nominal diameter is 300 mm, the convergency of the material stress \( \sigma_m \) at the position where it becomes maximum is shown in Fig. 9. In the case that \( k \) is over 20, \( \sigma_m \) converges sufficiently, and hence, \( k \) is set at 20.

![Graph showing convergency of stress values](image)

Fig. 9 Convergency of stress values

If the stress by one bolt in a cross section with the angle \( \theta \) is expressed as \( \psi \), the stress by \( n \) number of bolts load, \( \psi(\theta, n) \), at the cross section with the angle \( \theta \) is obtained as Eq. (11) by superposing \( \psi \).

\[
\psi(\theta, n) = \psi(\theta) + \psi(\theta - \alpha) + \psi(\theta - 2\alpha) + \ldots + \psi(\theta - (n-1)\alpha)
\]

where \( \alpha = \frac{2\pi}{n} \), the angle between the bolts.

### 4.2 Effect on the gasket seating pressure

Figure 10 shows the gasket seating pressure at the inner and outer edges of the gasket changes in the circumferential direction with the number of bolts. The pressure at the gasket seat becomes largest in a section through the bolt center, and smallest in one through the center between bolts.

In Fig. 11, the gasket seating pressures in a section through the bolt center and midway between the bolts are shown with a solid line and a broken line respectively. Furthermore, the pressure at the inner edge of the gasket is designed with mark \( \alpha \) and that at the outer edge, with mark \( \phi \).

Though the gasket seating pressure in a cross section through the bolt center is much different from that at midpoint between bolts, the difference between them gets smaller with an increasing number of bolts and the stress distributes almost uniformly in the circumferential direction in the case of more than 12 bolts. Therefore, it is clear that 16 bolts in the Standard give a sufficiently uniform gasket seating pressure on the circumference.

### 4.3 Effect on the stress of the flange

The stress in the flange change between bolts in the same manner as the gasket seating pressure. The circumferential distribution of the maximum meridional stresses \( \sigma_m \) under only one bolt load \( P_0 \) is shown in Fig. 12, and \( \sigma_m \) changes with the number of bolts. As the number of bolts is reduced, the required force for one bolt becomes larger since the total bolt load must be a required constant value. Consequently, as shown in Fig. 12, in the case of a small number of bolts, \( \sigma_m \) becomes largest in a cross section through the bolt center and smallest at midpoint between bolts. On the contrary, it is pointed out that with an
increasing number of bolts, the difference diminishes between the stress values at each section and an almost uniform stress distribution is obtained.

![Graph showing stress distribution](image)

**Fig. 12** Effect of number of bolts on stress distribution

Figure 13 shows the relation between the maximum stress $\sigma_m$ in a cross section of the bolt center and the number of bolts. As the number of bolts increases, the stress distributes uniformly and $\sigma_m$ becomes smaller. On the other hand, the effect of the number of bolts is considered only in British Standard in which the stress value in the case of the axisymmetrical bolt load is multiplied by the modification coefficient $C_F$ given by Eq. (12). From this calculation, the maximum stress value in the case of an arbitrary number of bolts is obtained.

$$C_F = \sqrt{\frac{2 \times \text{diameter of bolt} \times \text{flange thickness}}{\text{bolt pitch}}}$$

**Fig. 13** Relation between number of bolts and the maximum meridional stress

The results obtained from these calculations are shown with a broken line in Fig. 15. According to BS Standard, more than 8 bolts in the flanges are assumed to give uniform circumferential stress distribution. According to results by the finite element analysis, the effect of the number of bolts is different from that given by BS Standard. Since the stress calculated by the modification coefficient for the bolt pitch is lower than the actual one in the whole range of numbers of bolts, attention should be paid to the application of the Standard for the design of flanges.

5. Experiments

In order to confirm the applicability of the method and the results of the analysis, the stresses of the following flange are measured by experiments with a changing number of bolts. The tested flange is a forged steel flange for the petrochemical industry, with nominal diameter 24B and inner pressure 600 psi (Type II). Figure 14 shows the distribution of the maximum meridional stresses $\sigma_m$ on the circumference at the joint between hub and ring under only the bolt load and the measured values are compared with the calculated ones. As $\sigma_m$ in the cross section through the bolt center becomes maximum and changes in accordance with the number of bolts, the comparison of the measured values in that section with the calculated ones is made in Fig. 15. From these figures, it is proved that the measured values are consistent with the calculated ones and this analysis can be applied sufficiently.

![Graph showing stress distribution](image)

**Fig. 14** Effect of number of bolts on stress distribution

![Graph showing stress distribution](image)

**Fig. 15** Effect of number of bolts on the maximum meridional stress
6. Conclusions

With respect to "the integral pipe flange, nominal diameter 300, inner pressure 100 kg/cm²" based on JIS, the gasket seating pressure much related to the leakage of inner fluid was analyzed by the finite element method. Further, the influences of the number of bolts on the gasket seating pressure and the stress of the flange were investigated and the following results are obtained:

(1) The effective gasket width obtained from the gasket seating pressure under only the bolt load is smaller than that specified in the present Standard.

(2) The penetrating zone of inner fluid along the contact surface of gasket where inner pressure acts, increases as the taper of hub becomes larger and the rigidity of hub reduces.

(3) The penetrating zone is intermediate between the values assumed in JIS and ASME.

(4) Although, in the case of a limited number of bolts, the gasket seating pressure becomes minimum at midway between the bolts, it is pointed out that the gasket seating pressure distributes almost uniformly on the circumference in the case of the number of bolts specified in the present Standard.

(5) In the case that the flange is fastened by a limited number of bolts, the maximum meridional stress at the hub end is found to become maximum in the cross section through the bolt center. Only in British Standard the effect of the number of bolts is considered and the modification coefficient for the bolt pitch is applied. However, it is noticed that the stress by BS is a little lower than the actual one.

In this report, only one kind of gaskets is investigated, although many kinds of gaskets are classified according to the shapes and the materials. Hence, it is necessary to clarify the effect of them sufficiently, which is a subject for the future study.

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References

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(7) ASME Boiler and Pressure Vessel Code, Sec VIII, Appendix II.
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