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Strength of Integral Pipe Flanges*

(No. 1 Stress Distribution in Flange and the
Criticism of the Conventional Standards)

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There are two points to which special attention should be paid in the design of flanges; the failure of the flange due to high stresses and the leakage of inner fluid through the gasket seat.

The formulas for calculating flange stresses given in the conventional Standards are so approximate as to require further investigation.

In this report, the stress distributions in a standard integral flange are analyzed by Finite Element Method, and the effects of the design factors (especially fillet radius and hub taper) on the stress concentration at the fillet are clarified. Then, the stress values obtained by Finite Element Method are compared with the values obtained by ASME and JIS Standards, and it is found that the maximum meridional stress is about one and a half times larger than the maximum stress obtained by ASME or JIS Standard.

1. Introduction

The flanges are the important parts of pipe joints and ones based on JIS, ANSI or JPI Standards are mainly used in Japan. In these standards, flanges are classified into integral type, loose type and so on, and the dimensions are specified in accordance with design pressure and flange facing. As flanges in these Standards are comparatively of small size and for low pressure, flanges of larger size and for higher pressure used in the piping system of the recent petrochemical industry should be newly designed to satisfy the operating conditions. The points to which attention should be paid in the design of these flanges are (1) the stresses in flanges from the standpoint of strength and (2) the leakage of inner fluid from the standpoint of the function.

The stress calculation methods of flanges are shown in JIS [1][2], ASME [3], BS [4], DIN [5] Standards, etc. and as shown in Fig. 1, these formulas are approximate method satisfying the condition of the equilibrium of forces and the continuity of deformations at the joints of cylindrical shell, conical shell and circular plate into which the flange is divided. (In this report, portions A and C shown in Fig. 1 are called the ring, portions B and D are the hub and the portion D is the pipe.) Though the applicability of these formulas has been investigated by using strain gages or photo elastic experiments [6][7], an investigation of flanges with large diameter

Fig. 1 Modeling of the flange contour

and for high pressure was made only by Hayashi, N., et al. [8] in 1969. The JIS (proposal) "Basic dimensions of 100 kg/cm² steel pipe flanges" [9] was published as the standard of flanges for high pressure in Japan. Although stress analysis of the flanges based on this standard is carried out by Hayashi, N., et al. by photo elastic experiment, it is difficult to say that the stress distribution has been clarified in detail.

In this report, an analysis of the stresses in the integral flange by means of

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Finite Element Method is made and the stresses obtained are compared with the values given by stress formulas in JIS and ASME Standards, and at the same time the influences of the taper of hub and the fillet radius at the end of hub on the stresses in the flange are clarified. Then, several problems which have to be solved in the application of the present Standards to the design of the flange are also discussed.

2. The method of stress analysis

2.1 The flange contour

The flanges which are investigated in this report are integral type based on JIS and JPI.

First of all, stresses in the flange shown in Fig. 2 (Type I) are analyzed. This flange has a maximum diameter and is to be applied to the highest pressure in JIS. Its dimensions are shown in Table 1 according to JIS (proposal) "Basic dimensions of 100 kg/cm² steel pipe flanges". The gasket for this flange is made of soft steel shown in Table 2.

Fig. 2 Flange contour (Type I)

Table 1 Dimensions of flange

Nominal size	mm	300
Inside dia. of flange	mm	d 251.9
Thickness of pipe	mm	g ₀ 33.3
Thickness of hub at large end	mm	g ₁ 75
Dia. of hub at large end	mm	d ₂ 402
Outside dia. of flange	mm	A 605
Thickness of flange	mm	t 106
Height of seat	mm	t _s 3
Outside dia. of seat	mm	g 345
Fillet radius	mm	r 20
Dia. of bolt circle	mm	C 515
Number of bolts	n	16
Dia. of bolt holes	mm	D 52
Size of bolt		M48

The flange shown in Fig. 3 (Type II) is one of the forged steel flanges for the petrochemical industry based on JPI and its nominal diameter is 24B used for 600 psi.

The stresses in this flange are measured, in order to confirm the applicability of the results obtained by Finite Element Method.

Table 2 Characteristics of gasket

Nominal size		300
Material, Type		Soft flat metal gasket of mild steel
Gasket width	mm	N 15
Inside dia. of gasket	mm	d 315
Effective width	mm	b 6.9
Gasket factor	m	5.50

Fig. 3 Flange contour (Type II)

2.2 Bolt load

In the practical piping system, the bolt load, P_b required to prevent the leakage of inner fluid along the contact surface of the gasket is given by Eq. (1) considering the load condition in Fig. 4, although the calculating method is established under the relevant Standards.

Fig. 4 Load condition

$$P_b = P_1 + P_2 + P_3 \quad (1)$$

Where:

- P_b : bolt load kg
- P_1 : the hydrostatic end force on area inside of flange kg
- P_2 : the hydrostatic end force on area of flange facing kg
- P_3 : the joint-contact surface compression load to prevent the leakage kg

The force P_1 in Eq (2) is determined by the inner pressure. The force P_2 is obtained from Eq. (4) that is used in all Standards. The load P_3 determined by the inner pressure depends on the Reference Standards as shown in Eq. (3).

$$P_1 = \frac{p}{100} \cdot \frac{\pi}{4} \cdot d^2 \quad (2)$$

$$P_2 = \frac{p}{100} \cdot \frac{\pi}{4} \cdot (d_3^2 - d^2) \text{ (JIS)} \quad (3)$$

$$\frac{p}{100} \cdot \frac{\pi}{4} \cdot (G^2 - d^2) \text{ (ASME)}$$

$$P_3 = \frac{p}{100} \cdot 2\pi b G m \quad (4)$$

where:

- p : inner pressure kg/cm²
- d : inside diameter of pipe mm
- d_3 : outside diameter or penetrating zone of inner fluid along the contact surface of the gasket

In JIS it is calculated as follows:

- $d_3 = d_{gi} + 2N/3$ mm
- d_{gi} : inside diameter of the gasket mm
- N : gasket width mm
- G : diameter at location of concentrated gasket load P_3
- $G = d_{gi} + 2(N - b)$ mm
- b : effective gasket width mm
- $b = \begin{cases} b_0 & (b_0 \leq 6.5 \text{ mm}) \\ 2.52\sqrt{b_0} & (b_0 > 6.5 \text{ mm}) \end{cases}$
- b_0 : basic gasket seating width mm

This value is determined according to the shape of flange facing and is shown as follows for the flange investigated here.

$$b_0 = N/2$$

- m : gasket factor $m = \sigma_g / p_0$
- σ_g : mean gasket compressive stress at the leakage of inner fluid kg/mm²
- p_0 : inner pressure at the leakage of inner fluid kg/mm²

As shown in Eq. (3), the assumption on the width of penetrating zone of inner fluid along the contact surface depends on the JIS or the ASME. Regarding this point, a detailed investigation will be made in the next paper. As the flange studied in this paper is based on JIS, the bolt load P_b is calculated by JIS formula.

2.3 The method of analysis

As stated in the introduction, the formulas in the Standards are based on the approximate analysis and the stress concentration at the point between hub and ring is not taken into consideration at all in those formulas. In this report, the bolt load is assumed to be a distributed load which is acting uniformly on the circumference as assumed in the Standards. We intend to calculate the stresses in the flanges in detail by Finite Element Method as an axisymmetrical problem and compare the obtained results with those obtained by the conventional stress calculation method in the Standards.

The method of the analysis is as follows. There are two load conditions:

- (1) only bolt load is acting (gasket seating condition)
- (2) bolt load and inner pressure are acting (operating condition)

Under the boundary condition that inner pressure is acting, it should be especially considered that the effective zone of the gasket is reduced by the penetration of inner fluid along the contact surface between gasket and flange. In order to determine the area of the penetrating zone of inner fluid, the pressure on the contact surface is analyzed with Finite Element Method placing the nodal points of the finite elements on the contact surface between the gasket and the flange. On the area where the gasket pressure is above inner pressure p , inner fluid never penetrates. On the other hand, inner fluid penetrates into the area which the pressure p acts on. Therefore, the area of the inner fluid penetration is determined by iterative calculations performed until the pressure of the contact surface coincides with inner pressure p , changing the area of the surface on which p acts.

(1) Finite element layout of flange

(2) Finite element layout in the neighbourhood of gasket

Fig. 5 Finite element layout and boundary condition

Figure 5 shows a finite element model and the boundary conditions which are considered in this calculation. A triangle ring element of uniform strain is used under the condition that its minimum length is 2 mm at the stress concentrated portion of hub end and 1.5 mm at gasket joints. As for the boundary condition, because of the symmetry of a couple of flanges about the middle plane of gasket, the displacement there is fixed in the axial direction and free in the radial direction. Further, as for the applied loads, the bolt load P_b , inner pressure p and hydrostatic end force due to inner pressure on area inside of flange, P_1 are given. Gasket seating pressure is then given as the reacting pressure under these boundary conditions above mentioned. The area on which inner pressure is acting is determined in comparison with the gasket pressure obtained and the inner pressure p , and at the same time, the stresses in the whole flange are determined.

3. Results

3.1 Comparison between the results of analysis and experiment

In order to verify the results obtained by Finite Element Method, the results are compared with the values given by measuring stresses in the flange, type II as shown in Fig. 3, where only bolt load P_b acts. Figure 6 shows where wire strain gages are attached:

- (1) on the line through a bolt center A-A
- (2) On the line through the center between each bolt, B-B

Fig. 6 Positions of stress measurement

Strain gages are attached to five lines in all and strains are measured in the circumferential and the meridional direction. In this measurement, the bi-axial gage whose length is 2 mm is used at

the middle of hub and at pipe, and at the joints between hub and ring, and hub and pipe, a five-continuous gage with length of 1 mm is used in consideration of stress concentration.

Each tightening load of eight bolts in the neighbourhood of stress-measured area of the flange is measured precisely by the gages attached to the body of the bolts 90° apart from each other circumferentially. The other sixteen bolts are coated with MoS₂ so that a uniform coefficient of friction can be given. These bolts are carefully fastened by a torque-wrench with same torque, and all bolt loads are made equal.

From the results of stress measurement, it is found that the stress at the section B-B is at most 7 percent smaller than that at the section A-A, and the difference between both the stresses is slight. At the same time, it is clear that the stresses on the flange are almost uniform in the circumferential direction. This is due to such construction that the number of bolts is large and the pitch between bolts is relatively small in comparison with flange thickness. In the finite element analysis, it is assumed that the bolts are fastened uniformly in the circumferential direction, and there is no difference between the values of stresses at every section of the flange. Figure 7 shows comparison between the calculated values and mean values obtained by the measurements at the section A-A and B-B. In the analysis, it is assumed that the flange has no holes, and its shape is axisymmetrical, and the bolt load is distributed uniformly on the circumference along the bolt center. Due to the difference between the assumption and the real condition, maximum stresses at the joint between hub and ring are generated nearer to the hub by the influence of the bolt head. However, on the whole, the calculated values agree with the experimental data.

Fig. 7 Experimental results of stresses

After the applicability of this analysis is confirmed as well as the size of elements, the same analysis as above mentioned is carried out on the flange, Type I.

3.2 The stress distribution in flange

3.2.1. The stress distribution under the action of bolt load only

The flange is fastened with bolts before inner pressure acts. Figure 8 shows the results of the stress distributions in flanges fastened with only bolt load $p_b = 161.9 \times 10^3 \text{ kg}$ by Finite Element Method. In this figure, the solid and the broken line show the stresses at the outer and the inner surface, respectively. The stress σ_t is a circumferential stress and σ_m is a meridional one.

Fig. 8 Stress distribution in flange (in the case of bolt load only)

It is found that the stress σ_m is very large at the outer surface between hub and ring, and obviously a large stress concentration occurs at that portion. On the contrary, σ_m becomes smaller at the joints between hub and pipe, and approaches zero in the pipe. At the inner surface between hub and ring, σ_m changes from tensile to compressive due to the bending moment generated by bolt load.

The circumferential stress σ_t concentrates at the outer surface between hub and ring just like σ_m .

3.2.2 The stress distribution under the action of inner pressure

The stress distribution is analyzed in the case that not only bolt load $p_b = 161.9 \times 10^3 \text{ kg}$ but also inner pressure $p = 100 \text{ kg/cm}^2$ acts on the flange. Figure 9 shows the stress components σ_m and σ_t at the inner and outer surfaces of the flange. Comparing Fig. 9 with Fig. 8, it is found that in the pipe, σ_m and σ_t are generated by the inner pressure p and the hydrostatic end force P_1 , while, at the joints between hub and ring, there is little influence of P_1 and p ; especially the maximum value of σ_m is almost unchanged.

As discussed above, it is the stresses σ_m and σ_t that are important in considering the strength of the flange. When the inner pressure p acts besides the bolt load, the position of the maximum stress moves a little nearer to hub but its absolute value is almost unchanged. From these results it may be pointed out that only the bolt load should be considered for the calculation of the strength of the flange from the practical point of view.

Fig. 9 Stress distribution in flange (in the case of inner pressure acting)

3.3 Comparison with stress values obtained by Standard formulas

With respect to the maximum stress in the flange of nominal diameter 300 where both bolt load p_b and inner pressure p act, the stress values obtained by Finite Element Method in this paper are compared with those obtained by the formulas in JIS and ASME in Table 3. The symbols, A and B, in this table correspond to those shown in Fig. 10.

Table 3 Stresses at points A and B obtained by FEM and Standard Formulas

kg/mm ²						
	FEM		JIS		ASME	
σ_r	A	6.6	A	4.6		
	B	-1.0			B	6.2
σ_z	A	6.3	A	6.5	A	5.9
σ_t	A	5.5				
	B	3.1	B	3.6	B	4.3
σ_m	A	9.3				

It is clear that the stress values given by Finite Element Method do not always agree with those calculated by the Standards.

Fig. 10 Shape factor of flange

The meridional stress σ_m becomes maximum at point A. However, a formula related with this σ_m has never been given and the contour at the point A has not been taken into consideration at all in the Standards.

The stress σ_r at a point B calculated by Finite Element Method is -1.0 kg/mm^2 when the inner pressure p acts on the inner surface perpendicularly. In the formula of ASME, σ_r is regarded as a stress on the inner edge of circular plate (C) (on the side of the conical shell (D)) and therefore, is considered to be generated on the point B at the joint of the inner surface between hub and ring.

Here, the stress σ_m is greater than one half of the maximum stress calculated from the Standards and it is a very important factor affecting the strength of the flange. This fact means that it is necessary to calculate the exact stress distribution from the standpoint of prevention of a brittle fracture and a fatigue of the flange. Although the standards have taken an ample factor of safety for such stresses in the flange, rigorous studies on the actual stress would be more important.

3.4 Effect of the shape factor of the flange

Whereas the taper of hub is considered in some formulas in the Standards, the radius at the joint between hub and ring is not taken into consideration. Here, the effect of these factors on flange stresses is investigated in detail.

3.4.1 The effect of the radius at the joint between hub and ring

In the present Standards, the radius r at the transition from hub to ring is specified in accordance with inner pressure and size of flanges. As for the flange discussed in this paper, as shown in Table 1, r is 20 mm in JIS, and for almost the same size of the flange in ANSI, r is 12.7 mm. As the magnitude of r will have an effect on the stress concentration at that portion, it is necessary to clarify the effect of the radius of curvature on the safety of flange.

Figure 11 shows the stress components, σ_m and σ_t , at the outer surface of the flange in relation to the curvature and in this figure the position of the maximum stress is indicated with the mark o and it changes with the magnitude of r . It is evident from this figure that the greater stress concentration is generated locally as the radius becomes the smaller.

Fig. 11 Fillet radius and stress distribution at flange outer surface

Summarizing the foregoing results, the influence of the radius on the maximum stress values, σ_r , σ_z , σ_t and σ_m , is shown in Fig. 12. Not only σ_t and σ_m but also σ_r and σ_z become greater as the radius becomes smaller. This figure also shows the maximum stress values calculated according to JIS and ASME, σ_z in JIS and σ_r in ASME, for reference. Both values are small in comparison with σ_m .

Fig. 12 Effect of fillet radius on stresses

3.4.2 Effect of the taper of hub

The influence of the taper of hub is given in the ASME Standard. However, in JIS, the hub is replaced with a cylinder with the average diameter of hub and the influence of the taper of hub is not taken into consideration. Hence, a comparison of the results with those calculated by ASME is made in this section and for three cases in which the taper of hub is 1/2.5 specified in JIS, 1/1.25 and 1/5.0, the stresses in the flanges are analyzed respectively.

Figure 13 shows the effect of the taper of hub on the distribution of σ_m and σ_t at the outer surface of the flange. The stress concentrates at two points, A and B (referring to Fig. 10). It is found that the stress concentration at the portion C, is very small in the case of the taper of 1/5.0, while for 1/1.25 taper, a high stress concentration arises at the portions, A and C.

Then, the stresses σ_z and σ_m at portions A and C are shown in Fig. 14 in relation to the taper of hub. As the taper of hub becomes larger, the stress σ_z becomes smaller at the portion A, and on the

Fig. 13 Relation between hub taper and stress distribution

contrary, increases at the portion C. Consequently, the stress σ_z at the portion A is equal to that at the portion C when the taper of hub is about 1/1.6. Furthermore, in this figure the stress σ_z obtained according to ASME is also shown and it agrees with the results calculated by Finite Element Method.

On the other hand, the meridional stress σ_m , which is the most important factor affecting the strength of the flange as mentioned above, has always a maximum value at the portion A when the taper of hub is 1/5.0 $\sim$ 1/1.25, and it is clear that σ_m does not decrease greatly even if the taper of the hub is over 1/2.5 specified in JIS, though it becomes a little smaller as the taper of hub becomes larger.

Fig. 14 Effect of hub taper on stresses

4. Conclusions

The stress distribution in the integral pipe flange based on JIS, of the nominal diameter 300 mm and for inner pressure 100 kg/cm² is analyzed by Finite Element Method, and the results obtained are compared with those calculated according to JIS and ASME. The following results are obtained:

- (1) Assuming that the contour of the flange and the bolt load are entirely axisymmetrical, the stresses of the flange are analyzed by Finite Element Method. The calculated values agree with the measured stresses. Thus it is proved that the maximum stress in the flange can be analyzed as a solution of an axisymmetrical problem.
- (2) The results of calculation according to the conventional standards are not always in good agreement with the results obtained by

Finite Element Method and experiments. Especially the maximum stress value calculated by the Standards is about 2/3 of that by Finite Element Method.

(3) The maximum stress is a meridional stress σ_m which is an important factor affecting the strength of the flange. However, in the Standards this point has not been considered sufficiently.

(4) The stress is affected by bolt load and is influenced only a little by inner pressure p .

(5) As the radius of curvature at the joint between hub and ring becomes smaller, the concentrated stress at that portion becomes larger.

(6) As the taper of hub becomes larger, the location of the maximum stress σ_z in the axial direction moves from the joint between hub and ring to the joint between hub and pipe, which is in good agreement with the results according to ASME. However, the maximum meridional stress σ_m , in the case of the taper of hub being 1/5.0 1/1.25, always arises at the joint of hub and ring. It should be noted that it is necessary to estimate σ_m exactly and investigate it from the standpoint of flange strength.

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