On the Characteristics of Bolted Joints with Gaskets*

(stress analysis of full face metallic flat gaskets interposed between pipe flanges)

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This paper deals with a contact stress distribution in bolted joints when a full face metallic flat gasket is interposed between two pipe flanges. Replacing a gasket and pipe flanges by finite hollow cylinders the contact stress distribution is analyzed using three-dimensional theory of elasticity. By numerical computations, effects of the thickness and the stiffness of gaskets on the contact stress distribution are shown. Next, the force ratio of bolted joints with gaskets, which is the ratio of an increment of bolt axial force to an internal pressure and the sealing performance are analyzed. With an application of the internal pressure, a bending moment is produced in bolts. The maximum stress produced in bolts taking this bending moment into consideration is also analyzed. For verification, experiments are performed. Analytical results are in a fairly good agreement with experimental ones.

Key Words: Contact Stress, Metallic Gasket, Pipe Flange, Fixing Element, Elasticity

1. Introduction

Bolted joints with gaskets are widely used at connections of the mechanical devices in which the fluid is treated, especially pipe flanged connections with gaskets are used in piping systems. Many investigations^{(1)~(9)} on pipe flanged connections have been carried out and the results of them are standardized in the codes of ASME⁽¹⁰⁾, DIN⁽¹¹⁾, JIS⁽¹²⁾ and so on. Some investigations^{(13)~(20)} have been done with respect to gaskets themselves. But recently with the development of many other kinds of industries the working conditions have become increasingly severe, and it is necessary to make the design of pipe flanged connections with gaskets more reasonable from the viewpoints of the precision and the reliability of machines and the prevention of accidents. In order to make a rational design of these connections, it is necessary to examine the distribution of the contact stresses on the contact surfaces of gaskets, which governs the sealing performance, and the force ratio (the ratio of an increment of bolt axial force to an external load) in detail, when an initial clamping force and an external load caused by the fluid pressure are applied to bolted joints. But it seems that these problems have not been clarified sufficiently by investigations^{(1)~(9)} and the usage of gaskets is not made clear in the guideline of VDI 2230⁽²¹⁾. Therefore, in a series of our investigations^{(22)~(24)} on the characteristics of bolted joints with gaskets, effects of the stiffness and the thickness of gaskets on the sealing performance and the force ratio are studied by analytical theories and experiments in the case where two hollow cylinders⁽²²⁾ with a gasket are fastened with a set of bolt and nut, in the cases of T-flanged joints⁽²³⁾ and circular flanged joints⁽²⁴⁾.

In the present paper, discussion is made on the characteristics of the sealing performance and the

force ratio of a pipe flanged connection with a full face metallic flat gasket which is an important connection in practice. The purpose of the present paper is a contribution to the rationalization of the design of bolted joints with gaskets. In the analysis the distribution of contact stresses on gasket surfaces, in which a connection is joined with an initial clamping, is made clear using three-dimensional theory of elasticity. Furthermore, the force ratio is analyzed. Effects of the stiffness, the thickness of gasket and the pitch circle diameter on the characteristics of a joint are also discussed by numerical analyses and experiments.

2. Theoretical Analysis

2.1 Analysis of contact stress distribution

Figure 1 shows a pipe flanged connection in which a full face metallic flat gasket is interposed between pipe flanges and is fastened by N sets of bolts and nuts with an initial clamping force F_f . In order to analyze the contact stress on gasket surfaces the pipe flanges and the gasket are replaced by finite hollow cylinders as shown in Fig. 2. In the analysis it is assumed that the initial clamping force $N \times F_f$ acts on the bearing surface as an annular uniform compressive stress P_1 . The inner and the outer diameter of the hollow cylinders for the pipe flange [I] and the gasket [II] are designated by $2a$ and $2b$, the thickness by $2h_1$ and $2h_2$, Young's modulus and Poisson's ratio by E_1, ν_1 and E_2, ν_2 , respectively. The diameter of the bearing surface is designated by $2c$, and the pitch circle diameter by $2d$. Since bolt holes are neglected in the model for the analysis shown in Fig. 2, the distribution of contact stresses near the bolt holes are analyzed using the model of hollow cylinders demonstrated in the previous paper⁽²²⁾ as shown in Fig. 3. In Fig. 2, the inner diameter $2a_0$ of the hollow cylinder is equated to the diameter of a bolt hole, the thicknesses $2h_1$ and $2h_2$ of the hollow cylinders corresponds to those of a pipe flange and a gasket, respectively. In order to obtain the compressive spring constant for clamped parts which is treated in Section 2.2, the diameter of this hollow cylinder is determined in the way shown in Fig. 3(b). Namely, denote by $2b'_0$ the diameter of the circle when one circle with its center at the center of a bolt hole comes in contact with an adjacent circle with its center at the center of an adjacent bolt

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Fig. 1 Model of a pipe flanged connection with a full face metallic flat gasket fastened with an initial clamping force

Fig. 2 Model for analysis and dimension of a pipe flanged connection

hole. And denote by $2b'_0$ and $2b''_0$ the diameters of the circles which come in contact with the inner and the outer circumferences of the pipe flange. Then, comparing the values of $2b'_0$, $2b''_0$ and $2b_0$, the smallest one is taken as the diameter $2b_0$ of the hollow cylinders.

In the analyses of pipe flanges and a gasket shown in Fig. 2, Michell's stress function Φ is used. The stresses and the displacements are expressed as follows:

$$\left. \begin{aligned} \sigma_r &= \frac{\partial}{\partial z} \left(\nu \nabla^2 \Phi - \frac{\partial^2 \Phi}{\partial r^2} \right) \\ \sigma_\theta &= \frac{\partial}{\partial z} \left(\nu \nabla^2 \Phi - \frac{1}{r} \frac{\partial \Phi}{\partial r} \right) \\ \sigma_z &= \frac{\partial}{\partial z} \left\{ (2-\nu) \nabla^2 \Phi - \frac{\partial^2 \Phi}{\partial z^2} \right\} \\ \tau_{rz} &= \frac{\partial}{\partial r} \left\{ (1-\nu) \nabla^2 \Phi - \frac{\partial^2 \Phi}{\partial z^2} \right\} \end{aligned} \right\} \dots\dots\dots (1)$$

$$\left. \begin{aligned} u &= -\frac{1+\nu}{E} \frac{\partial^2 \Phi}{\partial r \partial z} \\ w &= \frac{1+\nu}{E} \left\{ (1-2\nu) \nabla^2 \Phi + \frac{\partial^2 \Phi}{\partial r^2} + \frac{1}{r} \frac{\partial \Phi}{\partial r} \right\} \end{aligned} \right\} \dots\dots\dots (2)$$

where the displacement is denoted by u in r -direction and by w in z -direction. The shearing modulus is denoted by G and Poisson's ratio by ν . Then, the following equation must be satisfied.

$$\nabla^2 \nabla^2 \Phi = 0, \quad \left(\nabla^2 = \frac{\partial^2}{\partial r^2} + \frac{1}{r} \frac{\partial}{\partial r} + \frac{\partial^2}{\partial z^2} \right)$$

Expanding the stress distribution on the bearing surface into Bessel series, the boundary condition of Fig. 2 is expressed as follows:

$$\left. \begin{aligned} &\text{For the finite hollow cylinder [I] (pipe flange),} \\ &\quad r=a, r=b: \sigma_r = \tau_{rz} = 0 \\ &\quad z=h_1: \sigma_z = e_0 + \sum_{s=1}^{\infty} e_s C_0(\gamma_s r), \\ &\quad \tau_{rz} = 0 \\ &\text{For the finite hollow cylinder [II] (gasket),} \\ &\quad r=a, r=b: \sigma_r = \tau_{rz} = 0 \\ &\text{On the contact surface,} \\ &\quad \left(\frac{\partial w}{\partial r} \right)_{z=-h_1} = \left(\frac{\partial w}{\partial r} \right)_{z=h_1}, (\sigma_z)_{z=-h_1} = (\sigma_z)_{z=h_1} \\ &\quad (\tau_{rz})_{z=-h_1} = (\tau_{rz})_{z=h_1} = 0 \end{aligned} \right\} \dots\dots\dots (3)$$

where

$$\begin{aligned} e_0 &= -\frac{d_2^2 - d_1^2}{b^2 - a^2} P_1 \\ e_s &= -\frac{2\{d_2 C_1(\gamma_s d_2) - d_1 C_1(\gamma_s d_1)\}}{\gamma_s \{b^2 C_0^2(\gamma_s b) - a^2 C_0^2(\gamma_s a)\}} P_1 \\ d_2 &= d + c, \quad d_1 = d - c \end{aligned}$$

The method of analysis for Eq. (3) and the nomenclature are the same as shown in the previous paper⁽²²⁾. The shearing force between contact surfaces is neglected.

In order to analyze the finite hollow cylinders having the boundary condition (3), referring to the theory of finite hollow cylinder reported by Shibahara, et al.⁽²³⁾, the stress function Φ for hollow cylinders [I], and [II] is put as follows: For the hollow cylinder [I], Φ is $\Phi_I + \Phi_{II}$. In Eqs. (4) and (5), $A_0, B_0, C_0, \bar{A}_n, \bar{B}_n, \bar{C}_s, \bar{A}_n, \bar{B}_n, \bar{C}_s, A_{II0}, B_{II0}, C_{II0}, \bar{A}_{II n}, \bar{B}_{II n}$ and $\bar{C}_{II s}$ are undetermined coefficients to be determined from the boundary conditions.

The values of $\bar{P}_{sn}$, $\bar{Q}_{ns}$, $\bar{R}_{ns}$, $\bar{S}_{ns}$ are obtained by replacing Ω_s , h_1 , β_n and Δ_n in P_{sn} , Q_{sn} , R_{sn} and S_{sn} with Ω_s , h_2 , β_n and Δ_n , respectively. By solving Eqs. (7)~(9), the undertermined coefficients are determined and by using these coefficients, stresses and displacements are obtained.

2.2 Analysis of force ratio

As shown in Fig. 4 when an internal pressure P is applied to a pipe flanged connection which is fastened with an initial clamping force F_f as shown in Fig. 1, an increment F_t of the bolt axial force is produced in each bolt and a compressive force F_c is eliminated from the contact surfaces. Then, the relationship between P and F_t , that is the force ratio, is obtained by the following equation^{(22)~(24)}.

$$\phi_s = \frac{F_t}{(W/N)} = \frac{K_t}{K_t + K_{cg}} \left(\frac{K'_{cg}}{K_{cg}} \right) \dots\dots\dots (10)$$

where K_t is the spring constant⁽²⁶⁾ of a bolt-nut system. K_{cg} is the compressive one⁽²²⁾ of clamped parts which is obtained from the equation $K_{cg} = F_f/2e'_0$ analyzing the case shown in Fig. 3. Using the mean displacement $2e_1$ on the bearing surfaces, K'_{cg} is obtained from the equation $K'_{cg} = F_f/2e_1$ in the model shown in Fig. 2. K_{tg} is a tensile spring constant which is obtained from the equation $K_{tg} = (W/N)/2e_2$ using the mean displacement $2e_2$ on the bearing surfaces as shown in Fig. 4(b). This method may be applicable under the condition $(\pi d/N)/(c + 2h_1) \leq 1$.

(a) model of pipe flanged connection

(b) model for analysis of the tensile spring constant K_{tg}

Fig. 4 Pipe flanged connection to which internal pressure P is applied

In an analysis of K_{tg} , assuming that the load W ($W = P\pi a^2$) acts on the hub as a uniform tensile stress P_2 in the axial direction, the boundary conditions are given as follows:

For the hollow cylinder [I] (pipe flange)

$$r = a : \sigma_r = P, \quad \tau_{rz} = 0$$

$$r = b : \sigma_r = \tau_{rz} = 0$$

$$z = h_1 :$$

$$\sigma_z = \frac{e^2 - a^2}{b^2 - a^2} P_2 + 2P_2$$

$$\times \sum_{s=1}^{\infty} \frac{e C_1(\gamma_s e)}{\gamma_s \{b^2 C_0^2(\gamma_s b) - a^2 C_0^2(\gamma_s a)\}} C_0(\gamma_s r),$$

$$\tau_{rz} = 0$$

For the hollow cylinder [II] (gasket)

$$r = a : \sigma_r = P, \quad \tau_{rz} = 0$$

$$r = b : \sigma_r = \tau_{rz} = 0$$

.....(11)

where the boundary condition on the contact surfaces is the same as shown in Eq. (3). The method of the analysis in Fig. 4(b) is the same as mentioned in Section 2.1. When Φ_g is obtained by Eq. (10), the compressive force F_c which is eliminated from the contact surfaces is obtained from the equation $F_c = (1 - \Phi_g) W$.

2.2.1 Estimation of internal pressure when the separation of the contact surfaces begins

With an increase of the internal pressure P , a separation of the contact surfaces begins from the inner circumference. By analyzing both cases of Figs. 2 and 4(b) and by superposing the results at the contact surfaces, the internal pressure P is determined such that the stress σ_z at $r = a$ becomes zero when the separation of the contact surfaces begins.

2.2.2 Analysis in the case where contact surfaces are separated

With further increase of the internal pressure P , the separation of the contact surfaces proceeds. It is supposed that the contact surfaces are separated from the inner circumference to $r = R_c$ as shown in Fig. 5(a). Analyzing both the case where the force F_b ($= F_f + F_t$) acts on the bearing surface of the hollow cylinder having the inner diameter $2R_c$ as shown in Fig. 5(b) and the case where the internal pressure P is applied as shown in Fig. 5(c), and superposing both results, the value of W/F_b is determined such that the stress σ_z at $r = R_c$ becomes zero. Where W is $P\pi R_c^2$. At the same time, the value of K_{cg}/K_{tg} in Eq. (10) changes as the separation proceeds. A model for the analysis shown in Fig. 5(b) and (c) is assumed. Namely, K'_{cg} is obtained from the equation $K'_{cg} = F_b/2e'_1$ in Fig. 5(b), and K_{tg} from the equation $K_{tg} = (W/N)/2e'_2$ in Fig. 5(c). As mentioned above, using Eq. (10) it is possible to obtain the force ratio in the case where the contact surfaces are separated. When the contact surfaces are separated under the position $r = (R_c)_{\max}$ in Fig. 5(a), namely under the bearing surface, it is assumed that the leakage of fluid is caused. The leakage is influenced by the other factors such as the characteristics of internal fluid and the surface roughness on gasket and so on, but in this investigation the effect of stress at the contact surfaces on the leakage is discussed.

2.3 Analysis of a bending moment produced in bolts

With an application of the internal pressure P a bending moment is produced in bolts besides an increment F_t of bolt axial force. In order to evaluate this bending moment, the bending compliance γ_p of clamped parts for internal pressure is defined by the equation $\gamma_p = (W/N)/\theta_p$ as shown in Fig. 6(a). The bending compliance γ_c is defined by the equation

$\gamma_c' = M_b/\theta_c$ using the inclination θ_c of the bearing surface caused by the moment M_b shown in Fig. 5(b). The bending compliance γ_b of a bolt is defined by the equation $\gamma_b = M_b/\theta_b$ using the inclination θ_b shown in Fig. 6(c). By using the bending compliances mentioned above, the following equation is obtained^{(22)~(24)}.

(a) model of pipe flanged connection

(b) $K'cg$

(c) $Ktgc$

Fig. 5 Pipe flanged connection in the case where the contact surfaces are separated

Fig. 6 Analysis of bending moment caused in bolts

$$\psi_p = \frac{M_b}{W/N} = \frac{\gamma_b}{\gamma_b + \gamma_c} \left(\frac{\gamma_c}{\gamma_p} \right) \quad (12)$$

Since γ_c in $\gamma_b/(\gamma_b + \gamma_c)$ is the bending compliance in the neighbourhood of the bolt holes, it is obtained by using the model of the hollow cylinder shown in Fig. 3(b). The value of γ_c'/γ_p is obtained by analyzing the cases shown in Fig. 6(a) and (b). The method of analysis is the same as the one mentioned in Section 2.1. Using Eqs. (10) and (12) the maximum increment σ_{tmax} of stress produced in the bolt is expressed as follows:

$$\sigma_{tmax} = \left(\frac{\phi_p}{A} + \frac{\psi_p}{Z} \right) \left(\frac{W}{N} \right) \quad (13)$$

where A is the cross sectional area of a bolt axis and Z is the section modulus. The analysis in the case where contact surfaces are separated is done in the same way.

(a) pipe flange

(b) bolt

Fig. 7 Pipe flange and bolt used in experiments

3. Experimental Method

Figure 7 shows dimensions of pipe flanges and bolts⁽²⁷⁾ used in experiments in order to obtain the force ratio. The material of pipe flanges is steel (S45C, JIS) and three types of flanges of which the pitch circle diameters $2d$ are 105, 120, 135mm are prepared as shown in Fig. 7(a). The size of bolts shown in Fig. 7(b) is M16 and the material is SCM3 (JIS). On the shank of the bolt, two strain gauges (KYOWA Electronic Instrument Co., Ltd., KFC-2-C1-11) are attached in 180° phase to each other. The thickness $2h_2$ of gaskets is taken as 5, 10, 20mm and the material is mild steel (SS34, JIS) and aluminium (A2-H, JIS). The maximum surface roughness $R_{\max}$ of gaskets is below $5\mu\text{m}$ for mild steel and below $7\mu\text{m}$ for aluminium. After two pipe flanges with a gasket are fastened by eight sets of bolts and nuts with the initial clamping force $F_f = 20\text{kN}$, the internal pressure is applied to the assembly by using an oil pressure pump (RIKEN Precision Machine Co., Ltd., MP-5C) as shown in Fig. 8. Then, the magnitude of the internal pressure is measured with a pressure transducer (KYOWA Electronic Instrument Co., Ltd., PG-500KU) which is attached on the pipe flange, and the increment of bolt axial force and the maximum stress are picked up by the strain gauges. The output signals are recorded by an X-Y recorder (YOKOGAWA Electric Works Co., Ltd., 3036) through dynamic amplifiers (KYOWA Electronic Instruments Co., Ltd., DPM-110B). The fluid used in experiments is spindle oil (#150) which is used at the room temperature.

4. Results of Analyses and Comparisons with Experimental Ones

4.1 Results of analyses

The numerical computations are done with the same dimensions of pipe flanges and gaskets as used in the experiments shown in Fig. 7 in order to compare with them. The terms of series are 20 in the computations. It is assumed that Poisson's ratio of pipe flanges and gaskets is 0.3. Young's modulus E_2 of gaskets is 206GPa for mild steel, 68.7GPa for aluminium, and that of pipe flanges is $E_1 = E_3 = 206\text{GPa}$. Also for the number of bolts, $N = 8$.

Figure 9 shows the effects of the ratio E_1/E_2 of Young's modulus of pipe flanges to that of the gaskets on the contact stress caused by the initial clamping force. It is seen that the compressive stress becomes large on the inner circumference of contact surface with an increase of the ratio E_1/E_2 . Figure 10 shows the effects of the thickness h_2/a of gaskets

Fig. 8 Experimental sketch

on the contact stress when the ratio of E_1/E_2 is held constant. With an increase of the thickness h_2/a of the gaskets, it is seen that the compressive stress becomes uniform. Figure 11 shows the effects of the pitch circle diameter d/a . The smaller the pitch circle diameter d/a is, the larger the compressive stress becomes on the inner circumference ($r/a = 1.0$). Figure 12 shows the analytical results of the contact stress distribution for Fig. 4. The contact stress distribution and the internal pressure when the separation of the contact surfaces is caused are estimated by superposing the results of Figs. 10 upon those of Fig. 12 when

(case of $d/a = 2.1$, $h_1/a = 0.6$, $h_2/a = 0.2$, $c/a = 0.48$)

Fig. 9 Effects of the ratio of Young's modulus of pipe flange to that of gaskets on distribution of the contact stresses

(case of $d/a = 2.1$, $h_1/a = 0.6$, $E_1/E_2 = 3$, $c/a = 0.48$)

Fig. 10 Effects of the thickness of gaskets on distribution of the contact stresses

the internal pressure is applied to the connection.

4.2 Comparisons with respect to the force ratio

Figure 13 shows comparisons of numerical results with experimental ones in respect to the force ratio. The abscissa is the internal pressure P and the ordinate is the bolt axial force $F_f + F_t$. The initial clamping force F_f is 20kN. With an application of the internal pressure P the increment F_t of bolt axial force increases linearly, and yields a curve when the separation of the contact surfaces begins from the inner circumference. Furthermore, the separation of the contact surfaces proceeds with the increase of the internal pressure P and then, the leakage of internal

fluid is caused from bolt holes. Usually, the force ratio is obtained by using the straight line which indicates a relationship [Eq. (10)] between the internal pressure P and the increment F_t of bolt axial force. The numerical value of the force ratio is 0.080

(case of $h_1/a = 0.6$, $h_2/a = 0.2$, $E_1/E_2 = 3$, $c/a = 0.48$)

Fig. 11 Effects of the pitch circle diameter on distribution of the contact stresses

(case of $h_1/a = 0.6$, $E_1/E_2 = 3$, $e/a = 1.36$)

Fig. 12 Analytical results for the case of Fig. 4 (the distribution of the contact stresses)

(a) the case of $2h_2 = 10\text{mm}$, and $2d = 105\text{mm}$, and gasket is aluminium ($E_1/E_2 = 3$)

(b) the case of $2h_2 = 10\text{mm}$, $2d = 105\text{mm}$, and gasket is mild steel ($E_1/E_2 = 1$)

(c) the case of $2h_2 = 10\text{mm}$, and $2d = 120\text{mm}$, and gasket is aluminium ($E_1/E_2 = 3$)

Fig. 13 Comparisons of numerical results with experimental ones with respect to force ratio

[the case where $2h_2 = 10\text{mm}$, $2d = 105\text{mm}$ and gasket is aluminium ($E_1/E_2 = 3$)]

Fig. 14 Comparisons in the case where bending moment is taken into consideration

in Fig. 13(a), 0.039 in (b) and 0.037 in (c). Numerical results are in a fairly good agreement with experimental ones. It seems that numerical results are in a fairly good agreement with experimental ones with respect to the estimation of the internal pressure. The internal pressure when the leakage of fluid is caused is 18.0MPa for the case of Fig. 13(a) and 17.0MPa for the case of (b). It is noticed that the internal pressure when the leakage of fluid is caused becomes large with an increase of the ratio E_1/E_2 . The internal pressure when the leakage of fluid is caused is 11.5MPa for Fig. 13(c). The larger the pitch circle diameter is, the smaller it becomes. Namely, the force ratio becomes small, but the sealing performance declines. When the other conditions are held constant and the thickness $2h_2$ of the gasket is varied as 5 and 20mm corresponding to the case of Fig. 13(a) the numerical values of the force ratio are 0.070 and 0.098 and the internal pressures when the leakage of fluid is caused are 17.4 and 18.9MPa, respectively. From the results mentioned above, it is confirmed that the force ratio becomes large and the internal pressure when the leakage of fluid is caused also becomes large with an increase of the thickness $2h_2$ of the gaskets. It is also seen that the increment F_t of axial force becomes pretty large when the separation of contact surfaces is caused. The numerical value of the force ratio in the case of a joint without a gasket is 0.032 and the internal pressure when the leakage of fluid is caused is 12.2MPa for the case where the pitch circle diameter $2d$ of a joint is 105mm. Figure 14 shows the comparisons of the maximum stress taking account of the bending moment produced in bolts. The numerical result is in a fairly good agreement with the experimental one. Also the increment F_t of bolt axial force obtained from the force ratio is shown by an alternate long and short dash line. The maximum stress is pretty larger than the stress obtained from the increment of a bolt axial force. Accordingly, in the design of bolted joints, it is noticed that the maximum stress produced in bolts must be used. It is shown that the maximum stress becomes rapidly large when the separation of contact surfaces is caused. This fact is undesirable from viewpoints of the fatigue strength of bolts.

5. Conclusions

The purpose of the present paper is a contribution to a reasonable design of pipe flanged connections with gaskets by analyzing the characteristics of joints. Discussion is made on a distribution of the contact stresses and the force ratio of a pipe flanged connections in which a full face metallic flat gasket is interposed between pipe flanges. The following results are obtained.

(1) Replacing pipe flanges and a gasket by finite hollow cylinders, a method of analysis of the contact stress distribution, which governs the sealing performance of pipe flanged connections, is demonstrated by using three-dimensional theory of elasticity.

(2) A method of analysis is demonstrated with respect to the force ratio when an internal pressure is applied to the assembly and a method of analysis of each spring constants is also demonstrated. A method to obtain an increment of bolt axial force and a method to estimate an internal pressure when the leakage of internal fluid is caused are shown in the case where the contact surfaces are separated.

(3) A method of analysis to obtain a bending moment caused in the bolts is shown with an application of an internal pressure to connections and a calculation method of the maximum stress taking account of the bending moment is also shown.

(4) Based on the method mentioned (1), the

effects of the thickness, the stiffness of a gasket and the pitch circle diameter on the contact stress are made clear by numerical computations.

(5) For verification of the results obtained from (2) and (3), experiments are performed. It is shown that numerical results are in a fairly good agreement with experimental ones. It is shown that the sealing performance becomes better with an increase of the ratio of Young's modulus and the thickness of a gasket and with a decrease of the pitch circle diameter. But it is shown that conversely the force ratio becomes large.

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