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New Development in Studies on the Characteristics of Bolted Pipe Flange Connections in JPVRC

This paper deals with some studies carried out in the bolted flanged connection committee (BFC) in Japan Pressure Vessel Council (JPVRC) on the stress analysis of a pipe flange connection using the elastoplastic finite element method. The characteristics of the connections with the different nominal diameters from 2 in. to 20 in. such as the contact gasket stress distribution, the hub stress, and the load factors were examined. The results from the finite element analyses were fairly consistent with the experimental results concerning the variation in the axial bolt force. By using the contact stress distributions and the results of the leakage test, the new gasket constants were evaluated. As a result, it was found that the variations in the contact stress distributions were substantial due to the flange rotation in the pipe flange connections with the larger nominal diameter. A method to determine the bolt preload for a given tightness parameter was demonstrated and the difference in the bolt preload between our research and PVRC was shown. In addition, the characteristics of pipe flange connection under a bending moment and internal pressure were also discussed and a newly developed bolt tightening method was demonstrated. [DOI: 10.1115/1.2140799]

1 Introduction

This paper describes some studies on the stress analysis of pipe flange connection and a bolt tightening method carried out in the bolted flanged connection committee (BFC) in Japan Pressure Vessel Council (JPVRC). Pipe flange connections with gaskets have been widely used in chemical, nuclear facilities and so on, and they are usually used under internal pressure as well as other loadings such as thermal, bending moments and so on. In an optimum design of pipe flange connections with gaskets, it is necessary to understand the characteristics of the connections under internal pressure. Important issues in designing pipe flange connections are the actual contact gasket stress distributions which govern the sealing performance, the hub stress from the flange design standpoint and a variation in the axial bolt force (the load factor) from bolt and sealing design standpoints when internal pressure is applied to the connections. Some researches [1–7] on pipe flange connections with gaskets have been carried out using the pipe flange connections with the smaller nominal diameter such as the sealing performance, the contact gasket stress distribution at the interfaces, hub stress and a variation in the axial bolt force. In practice, pipe flange connections with the larger nominal diameter have been often used, too. However, a question remains whether it is possible to apply the studied results obtained by the pipe flange connection with the smaller nominal diameter to the behavior of pipe flange connections with the larger nominal diameter, such as leakage evaluation and a method to determine the bolt preload.

PVRC [8–13] (Pressure Vessel Research Council) proposed the new gasket constants (G_b, a, G_s) and the tightness parameter T_p and it also proposed to evaluate the sealing performance and to determine the bolt preload by using the new gasket constants and the tightness parameter T_p. In the PVRC test procedure, the gasket constants and the tightness parameter T_p are obtained under uniform gasket stress in the gasket tightness tests. However, issues remain how to evaluate the sealing performance and the leakage in actual pipe flange connections with gaskets by using the new gasket constants proposed by PVRC. In actual pipe flange connections, it has been well known that the contact gasket stress distribution is not uniform and it is changed when an internal pressure is applied. In addition, another issue is how to evaluate the effect of a nonlinearity and a hysteresis in the stress-strain curve of a gasket on changes in the contact gasket stress distribution. It is well known that a change in the contact stress depends on the axial bolt force which changes as the internal pressure is changed. The contact gasket stress distribution is not taken into consideration in evaluating the tightness parameter T_p and the new gasket constants in the PVRC procedure.

Thus, in this paper, the contact gasket stress distributions in the pipe flange connections with the different nominal diameters from 2 in. to 20 in. under internal pressure are analyzed by the elastoplastic finite element method (EP-FEA) by taking account a nonlinearity and a hysteresis in the stress-strain curves of a spiral wound gasket, where two pipe flanges including the gasket are clamped together by bolts and nuts with an initial clamping force (preload) and an internal pressure is then applied. The effects of the nominal diameters of the pipe flange connections on the contact gasket stress distributions, the variations in the axial bolt force (the load factor) and the hub stress are analyzed by the EP-FEA [7,8].

Furthermore, the leakage tests and the measurements concerning a variation in an axial bolt force [7,8] were performed in the pipe flange connections with 3 in. and 20 in. nominal diameters (ASME/ANSI) [14] using helium gas. The EP-FEA results are compared with the measured results concerning the variation in axial bolt force and a amount of leakage. A method to determine a bolt preload is demonstrated for a given tightness parameter T_p. The values of the bolt preload for the pipe flange connection are compared with those by the PVRC procedure and discussion is made. In addition the characteristics of the connection subjected to a bending moment and internal pressure are also examined. Finally, a new bolt tightening method is demonstrated.

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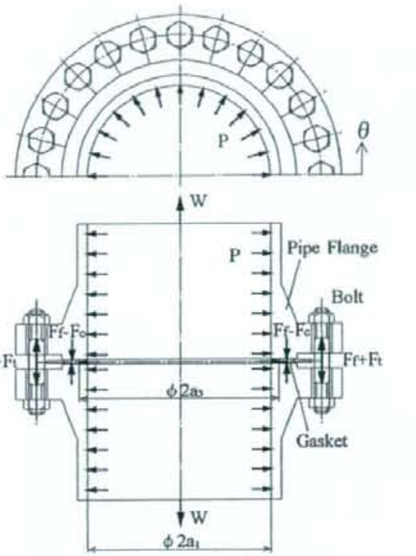

Fig. 1 A pipe flange connection with a spiral wound gasket subjected to an internal pressure

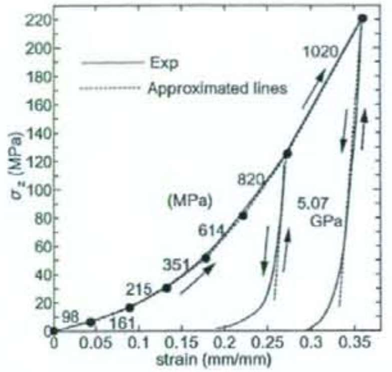

Fig. 2 Stress-strain lines of the spiral wound gasket used for the EP-FEA

2 Stress Analysis of Pipe Flange Connections Under Internal Pressure

2.1 Elasto-Plastic Finite Element Analysis. Figure 1 shows a pipe flange connection with a spiral wound gasket, in which two pipe flanges including the gasket are fastened with N bolts and nuts with a bolt preload F_f, subjected to internal pressure P. When the internal pressure P is applied to the pipe flange connection, a tensile load W $(=\pi a_1^2 P)$ acts on the end part of the connection in the axial direction, and an increment in axial bolt force F_t occurs in the bolts and the contact force F_c (per bolt) is eliminated from the gasket contact surfaces, that is, the total axial force W'/N $(=\pi a_3^2 P/N)$ (per bolt) due to the internal pressure P equals to the sum of F_t and F_c $(W'/N=F_t+F_c)$, where the inner diameter of the gasket is designated as $2a_3$ and that of the pipe as $2a_1$. Thus the contact gasket stress decreases as the internal pressure P increases. The actual gasket stress must be estimated exactly when the internal pressure P is applied to the connection for evaluating the sealing performance. The ratio of F_t to W'/N is called as the load factor [15,16] ϕ_g $[=F_t/(W'/N)]$. When the value of the load factor ϕ_g is obtained, the force F_c is determined by the equation $F_c=(1-\phi_g)W'/N$ and the actual average contact gasket stress is obtained by the equation $(F_f-F_c)/A$, where "A" is the gasket contact area.

The contact gasket stress distributions, the hub stresses and the load factor ϕ_g of the pipe flange connections with the different nominal diameters from 2 in. to 20 in. (2, 3, 4, 8, 12, 16, and 20 in.) are calculated by the elasto-plastic finite element method (EP-FEA). They are the class 300 in the API while the 3 in. pipe flange connections are the class 600 in the API standards (in the experiments, the connections with the 3 in. of the class 600 and the connections with the 20 in. of the class 300 were used).

In this study, a nonlinearity and a hysteresis in the stress-strain relationship of the spiral wound gasket are taken into consideration in the EP-FEA. Figure 2 shows a stress-strain curve of the spiral wound gasket used in this study. The ordinate is the contact gasket stress σ_z, and the abscissa is the strain. The dotted lines show approximated piecewise linear lines in the FEM analysis and the solid lines are obtained by the measurement. The slopes of the approximated lines in the loading are described as the numerals (MPa) in the figure. When the contact gasket stress is larger than 220 MPa, the slope of the stress-strain curve is 1100 MPa. The slope of the stress-strain in the unloading is held constant as 5.07 GPa.

2.2 Experimental Method. Experiments were carried out to measure the amount of gas leakage and variations in the axial bolt force in the pipe flange connections with 3 in. and 20 in. nominal diameter under internal pressure. The relationship between the actual contact stress and the tightness parameter T_p in the pipe flange connections is obtained by using the measured amount of the gas leakage and the calculated contact stress distributions. The mass leakage was measured from a variation in the pressure during some time interval.

2.3 Results of Elasto-Plastic Finite Element Analysis.

2.3.1 Contact Gasket Stress Distribution. Figure 3 shows the contact gasket stress distributions in the pipe flange connection with the 20 in. nominal diameter (under internal pressure) in the θ (circumferential)-direction at the distance $r=262.75$ mm (the inner radius of the gasket), 275.83 mm (the middle radius of the

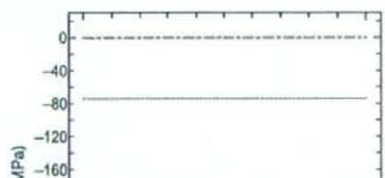

Fig. 3 The contact gasket stress distributions of the pipe flange connection with the 20 in. in the θ direction under internal pressure ($\theta=0°-7.5°$)

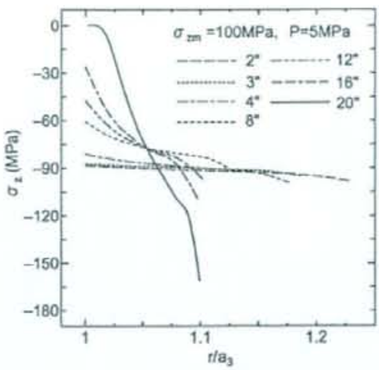

Fig. 4 The effects of the nominal diameter of the pipe flange connections on the contact gasket stress distributions in the r direction (the case where an internal pressure P=5 MPa is applied)

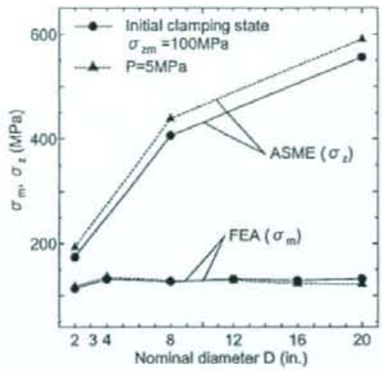

Fig. 5 The effects of the nominal diameter in the pipe flange connections on the hub stress (von Mises' equivalent stress σ_m) at the angle $\theta=0°$

gasket) and 288.9 mm (the outer radius of the gasket). The ordinate is the contact stress σ_z, the abscissa is the angle θ (=0°–7.5°). The bolt preload was chosen as F_f=200 kN and the internal pressure of 5 MPa was applied to the connection. It is observed that the variations in the contact gasket stress distributions in the θ direction are small. In the connections with the other different nominal diameter (from the 2 in. to the 16 in.), the variations in the contact stress distributions in the θ direction were small. Thus, hereinafter, the contact gasket stress distributions in the radial direction are shown at $\theta=0°$ (at the bolt axis).

Figure 4 shows the effects of the nominal diameter in the pipe flange connections on the contact gasket stress distributions in the r direction. The ordinate is the contact stress σ_z, and the abscissa is the ratio of the distance r to the inner radius a_3 of the gasket. The nominal diameters of the pipe flange connections are chosen as 2, 3, 4, 8, 12, 16 and 20 in. where the average contact gasket stress is chosen as σ_{zm}=100 MPa and the internal pressure P =5 MPa. It is shown that the variations in the contact gasket stress distributions of the pipe flange connections with the larger nominal diameter are larger than those with the smaller nominal diameter. It is assumed that the main reason of this fact is due to the so-called "flange rotation." The flange rotation in the connections with the larger nominal diameter is larger than that with the smaller nominal diameter. From the comparison of the contact gasket stress distribution shown in Fig. 4 with that in the initial clamping state, it is observed that the reduction in the contact gasket stress of the pipe flange connections with the larger nominal diameter is larger than that with the smaller nominal diameter when the internal pressure is applied to the connections. This is because that the total tensile load W' $(=\pi a_3^2 P)$ per the gasket contact area which is caused due to the internal pressure in the pipe flange connections with the larger nominal diameter is larger than that with the smaller nominal diameter (for example; 3 in. → the reduction is 12.2 MPa, 20 in. →23.9 MPa). In addition, another reason is that the value of the load factor ϕ_g (described in 2.3.3) is different in the different pipe flange connections.

2.3.2 Hub Stress. Figure 5 shows the comparisons of the hub stress between the von Mises' stress σ_m obtained from the present FEA and the normal stress σ_z due to ASME at the angle $\theta=0°$, where the initial average contact stress is held constant at σ_{zm} =100 MPa and the internal pressure as P=5 MPa. The ordinate is the von Mises' equivalent stress σ_m and the normal stress σ_z, while the abscissa is the nominal diameter D (in.) of the pipe flange. The difference in the hub stress between the initial clamping state and the state where the internal pressure is applied is small. However, the difference is substantial between the results from the present FEA and the normal stress obtained from the ASME code. As the nominal diameter D is increased, the difference in the hub stress increases. Thus the bolt preload cannot be increased taking into account the hub stress due to the ASME code. Actually, the hub stress is smaller than that from the ASME code, thus, the bolt preload can be increased. It is safer for preventing the leakage to increase the bolt preload in pipe flange connections.

2.3.3 Load Factor. Table 1 shows the load factor ϕ_g of the pipe flange connections with the different nominal diameters from 2 in. to 20 in. obtained by the FEM analyses. The load factor ϕ_g of the connections with 2 in. nominal diameter is the biggest, and as the nominal diameter of the pipe flange is increased, the value of the load factor ϕ_g of the connections decreases. The force F_c, which is eliminated from the contact surface, is obtained by the equation $F_c=(1-\phi_g)W'/N$. Thus, the force F_c increases as the value of the load factor ϕ_g of the connections decreases. This means that the sealing performance is getting worse because the gasket stress decreases. In Table 1, it is observed that the value of the load factor becomes negative from the nominal diameter 8 in. This is assumed that the flange rotation occurs in the pipe flange connection. This result corresponds to the contact gasket stress distributions shown in Fig. 4. In determining the bolt preload F_f of the pipe flange connections with the larger nominal diameter, it is necessary to take into account that the value of the load factor ϕ_g becomes negative.

Table 1 Load factor ϕ_g of the pipe flange connections

Nominal diameter of the pipe flange connections	Load factor ϕ_g
2"	0.251
3"	0.161
4"	0.108
8"	-0.0599
12"	-0.126
16"	-0.197
20"	-0.226

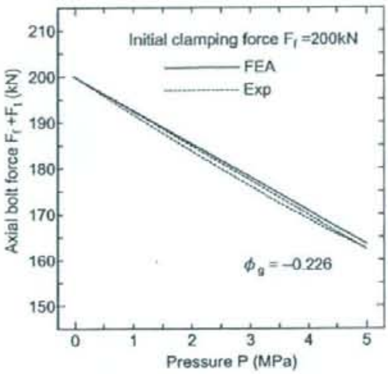

Fig. 6 Comparison of an increment in axial bolt force between the experimental and the FEA results

2.4 Experimental Results and Comparisons

2.4.1 Results of Leakage Tests for Pipe Flange Connections. The leakage tests for the pipe flange connections with the spiral wound were carried out (Nominal diameter of pipe flanges is 3 in. and 20 in.). The values of gasket constants "G_b" and "a" obtained by the present experiments using the actual reduced contact gasket stress when the internal pressure is applied are $G_b=16.5$ and $a=0.305$ for the 3 in. pipe flange connection, and 19.0 and 0.209 for the 20 in. pipe flange connection while they are 19.1 and 0.273 due to the PVRC data. A difference in the values between the present study and PVRC is small. However, a difference in the values G_b and a obtained using the initial clamping stress with those by PVRC is substantial. Thus it can be concluded that the actual reduced gasket stress in the pipe flange connections under internal pressure must be employed in estimating an amount of leakage using the PVRC data.

2.4.2 Comparisons of the Load Factor. Figure 6 shows the comparisons of a variation in axial bolt force (load factor ϕ_g). The ordinate is the axial bolt force F_f+F_t and the abscissa is the internal pressure P. The solid line shows the results by the EP-FEA. The dotted line shows the experimental results. The initial clamping force (bolt preload) F_f is determined as 200 kN for the 20 in. pipe flange connections. Figure 6 shows that the axial bolt force decreases linearly with increasing of internal pressure. Fairly good agreements are observed between the results in the EP-FEA for the 3 in. and the 20 in. pipe flange connections and the experimental results.

2.5 Determinations of Bolt Preload for a Given Tightness Parameter T_p. For a given tightness parameter T_p (point "A" in Fig. 7, this is denoted as T_{pa}) under the internal pressure P, the initial clamping force F_f (preload) must be determined. Figure 7 shows the method to determine the initial clamping force F_f (preload) for a given tightness parameter. For a given tightness parameter T_{pa} [point "A" in Fig. 7(*a*)] under the internal pressure, the contact gasket stress of the connections [point "B" in Fig. 7(*a*)] is determined using the results of leakage tests, and the required initial contact stress [point "C" in Fig. 7(*a*)] and the initial clamping force (preload) F_f are determined using the FEM calculations. In the PVRC procedure, (1) the tightness parameter T_{pa} [point "A" in Fig. 7(*b*)] is given from the design condition, (2) the tightness parameter T_p [point D in Fig. 7(*b*)] is determined empirically such as $T_p=1.5T_{pa}$, (3) the required initial contact gasket stress [point "C" in Fig. 7(*b*)] is obtained from point D through point F as shown in Fig. 7(*b*).

The initial clamping force (preload) F_f was obtained as 155 kN

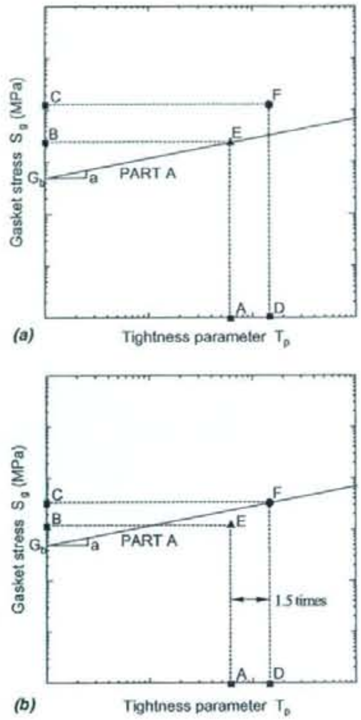

Fig. 7 A method for determining bolt preload F_f. (*a*) The case where the results of the FEA are used. (*b*) The case of the PVRC procedure.

($T_p=100, P=5$ MPa) by the PVRC procedure for the 20 in. pipe flange connection while it was 209 kN by the present study. The differences in the value of the bolt preload F_f between the present calculations and the PVRC results are substantial. Thus, in designing the initial clamping force (preload) F_f of the pipe flange connections with the larger nominal diameter, the difference must be taken into consideration. When the internal pressure of 5 MP is applied to the pipe flange connections with the larger nominal diameter (20 in.), the ratio of T_p in the case of initial clamping state to that in the case where an internal pressure is applied is obtained as about 2.5 in the PVRC procedure, while the ratio is chosen as 1.5 (at room temperature) in the PVRC procedure described above. On the other hand, in the pipe flange connections with the smaller nominal diameter when the internal pressure of 5 MPa is applied to the connections, a fairly good agreement is found between the present calculations and the PVRC results. However, it is not fully elucidated that the tightness parameter T_p in the case of initial clamping state at room temperature is 1.5 times larger than the T_p.

In the case where an internal pressure is 5 MPa and the tightness parameter is $T_p=1000$, the required initial clamping force (preload) F_f in the connections with the 3 in. is calculated as 60 kN from the present study while it is 58 kN from PVRC. For the connection with 20 in. nominal diameter, it is 332 kN from the present study while it is 289 kN from the PVRC procedure. In addition, in the case of the tightness parameter $T_p=1800$ and the internal pressure $P=10$ MPa, a difference in the initial clamping force (preload) is about 10% between the present calculations and the PVRC procedure. Thus, in designing the initial clamping force

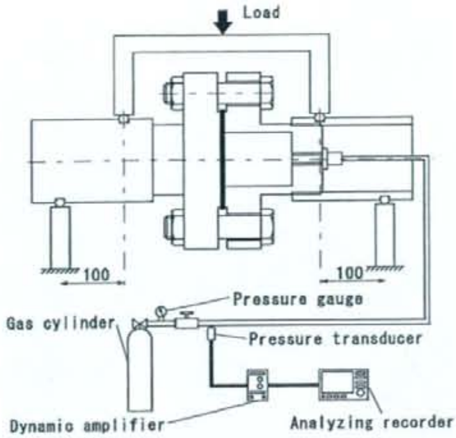

Fig. 8 Schematic of the pipe flange connection subjected to a bending moment and internal pressure

(preload), the difference must be taken into consideration when higher internal pressure ($P=10$ MPa) is applied to the connections. When the internal pressure of 10 MP is applied to the pipe flange connections with the smaller nominal diameter, the ratio of T_p in the case of initial clamping state to that in the case where an internal pressure is applied is obtained as about 2.1 in the PVRC procedure. For better sealing, the actual reduced contact gasket stress must be taken into consideration.

3 Stress Analysis of Pipe Flange Connections Under Bending Moments

Figure 8 shows a pipe flange connection with a gasket under a bending moment M and an internal pressure. The pipe flange connection was analyzed by FEA calculations concerning the contact gasket stress distributions and the hub stress. In addition, leakage test for the pipe flange connection with the nominal diameter of 50 mm in JIS were carried out. The types of gaskets employed were the spiral wound and the joint sheet gaskets. The new gasket constants G_b and a were calculated from the amount of gas leakage (helium) and the contact gasket stress distributions obtained from the FEA calculations. The bending moments M applied were $M=1.5$ and 2.5 kNm. The values of the new baskets constants (spiral wound gaskets) G_b and a were obtained as $Gb=3.76$ (MPa), and $a=0.5245$ for $M=1.5$ kNm, $Gb=2.35$ (MPa), and $a=0.6883$ for $M=2.5$ kNm, and $Gb=16.55$ (MPa) and $a=0.3077$ for $M=0$ while they were $Gb=19.10$ (MPa) and $a=0.273$ from the PVRC data. It is easily found that an amount of leakage is increased as the bending moment M increases. When a pure bending moment is applied to a pipe flange connection, a change in the integration of the contact gasket stress is zero. However, the leakage increases. The leakage increases from the contact gasket surfaces at the tension side of connection under the bending moment. The leakage is sensitive to the reduction of the contact gasket stress in the connection. In addition, the initial clamping force F_f was determined under the condition of $T_p=1202$ and $P=6.67$ MPa. The gasket used was the joint sheet. The initial clamping force F_f was calculated as 53.65 kN for $M=1.5$ kNm, 82.14 kN for $M=2.5$ kNm and 21.41 kN for $M=0$ (internal pressure only) while 18.97 kN from the PVRC procedure. Thus, it is important to take into account the effect of bending moment.

Table 2 Newly developed tightening procedure by JPVRC

Step	Loading
Install	Hand tighten all bolts, then tighten 4 or 8 equally spaced bolts with gradually increased tightening torque to 100% of target torque on a cross-pattern tightening sequence. Check flange gap around circumference for uniformity.
Tightening	Tighten all bolts with tightening torque to 100% of target torque on a rotational clockwise pattern for specified iterations six passes for 10 inch and greater flange, 4 passes for others).
Post-tightening	If necessary, wait a minimum of four hours and tighten by the previous step, but 1 or 2 passes.

4 New Tightening Procedure Proposed by Japanese Committee (BFC) in JPVRC

For reducing tightening turns and increasing the tightening accuracy, a new tightening procedure has just proposed by BFC in JPVRC [17]. Table 2 shows the new procedures for tightening while the star sequence bolt tightening procedure in ASME PCC-1 [18] have been already published. In the new procedure, bolts are tightened in one way (clockwise or anti-clockwise) after 4 or 8 bolts are tightened by hands. The new procedure [specified in Japan Industrial Standards (JIS) in near future] is excellent in tightening time and very simple when a lot of bolts must be tightened.

5 Conclusions

This paper has described the characteristics of the pipe flange connections with gaskets under internal pressure and a bending moment studied in BFC of JPVRC. In addition, the new bolt tightening method developed by the BFC in JPVRC was also demonstrated. The following results were obtained.

(1) The variations in the contact gasket stress distributions in the pipe flange connections with the larger nominal diameter subjected to internal pressure were found to be larger than those with the smaller nominal diameter. It was observed that the reductions in the contact gasket stress in the pipe flange connections with the larger nominal diameter were larger than those with the smaller nominal diameter.

(2) A variation in von Mises' hub stress obtained from the present FEA calculations was small while the hub stress σ_z obtained from the ASME code increased as the nominal diameter increased. It can be assumed that the hub stress is actually smaller than expected and thus the bolt preload can be increased for a much safer design.

(3) The values of the load factor ϕ_g of the pipe flange connections with the larger nominal diameter were negative. Thus, it was found that the reduction in the contact gasket stress increased as the nominal diameter of pipe flange connections increased, that is, a leakage easily occurs for pipe flange connections with larger nominal diameter. A fairly good agreement was observed between the results of the EP-FEA and the experimental results in the connections with 3 in. and 20 in. nominal diameters.

(4) In estimating the new gasket constants, it was demonstrated that the actual reduced gasket stress had to be used taking into account the value of the load factor. In addition, it was necessary to take into account the flange rotation in determining the bolt preload.

(5) It was shown that the effect of external bending moment on the leakage was substantial in designing pipe flange connection.

(6) A new bolt tightening procedure developed by the BFC in JPVRC was demonstrated.

Nomenclature

a = new gasket constant; the slope of the gasket-loading line in "PART A"

$2a_1$ = inner diameter of pipe
$2a_3$ = inner diameter of gasket
A = gasket contact area in the analysis
$2b_1$ = outer diameter of pipe
$2b_3$ = outer diameter of gasket
F_c = force eliminated from the contact surfaces [$=(1-\phi_g)W'/N$]
F_f = initial clamping force (preload)
F_t = increment in axial bolt force
G_b = new gasket constant; contact gasket stress at $T_p=1$ in "PART A"
N = bolt numbers
P = internal pressure
T_p = tightness parameter
T_{pa} = assembly tightness parameter
W = axial force due to internal pressure ($=\pi a_1{}^2 P$)
W' = total axial force due to internal pressure ($=\pi a_3{}^2 P$)
ϕ_g = load factor ($=F_t/W'$)

References

[1] Water, E. O., and Schneider, R. W., 1969, Trans. ASME, Ser. B, **91-3**, 615.
[2] Sawa, T., Ogata, N., and Nishida, T., 2002, ASME J. Pressure Vessel Technol., **124**(4), pp. 385–396.
[3] Ando, F., Sawa, T., Ikeda, M., and Furuya, T., 1998, "Assessing Leakage of Bolted Flanged Joints Under Internal Pressure and External Bending Moment," Int. J. Pressure Vessels Piping **376**, pp. 39–44.
[4] Kumano, H., Sawa, T., and Hirose, T., 1994, "Mechanical Behavior of Bolted Joints under Steady Heat Conduction," ASME J. Pressure Vessel Technol., **116**, pp. 42–48.
[5] Morohoshi, T., and Sawa, T., 1994, "On the Characteristics of Rectangular Bolted Flanged Connections with Gaskets Subjected to External Tensile Loads and Bending Moments," ASME J. Pressure Vessel Technol., **116**, pp. 207–215.
[6] Sawa, T., Higurashi, N., and Akagawa, H., 1991, "A Stress Analysis of Pipe Flange Connections," ASME J. Pressure Vessel Technol., **113**, pp. 497–503.
[7] Sawa, T., Hirose, T., and Nakagomi, Y., 1996, "Behavior of a Tapered Hub Flange with a Bolted Flat Cover in Transient Temperature Field," ASME J. Pressure Vessel Technol., **118**, pp. 115–120.
[8] Hsu, K. H., Payne, J. R., Bickford, J. H., and Leon, G. F., 1990, *Proceedings of the 2nd International Symposium on Fluid Sealing*, CETIM La Baule, 9.
[9] Payne, J. R., and Bickford, John, 1998, "*Gaskets and gasketed joints*," Marcel Dekker Inc., New York.
[10] Leon, G. F., and Payne, J. R., 1989, "An Overview of the PVRC Research Program on Bolted Flanged Connections," Press. Vessel Technology, **1**, 217.
[11] Payne, J. R., Bazergui, A., and Leon, G. F., 1984, "A New Look at Gasket Factors," *Proceedings of the 10th International Conference on Fluid Sealing*, BHRA, Innsbruck, 345.
[12] Payne, J. R., Bazergui, A., and Leon, G. F., 1985, "New Gasket Factors, A Proposed Procedure," PVP (Am. Soc. Mech. Eng.) **98-2**, 85.
[13] Payne, J. R., Leon, G. F., and Bazergui, A., 1987, "Obtaining New Gasket Design Constants from Gasket Tightness Data," *Proc '87 Spring Conf. Exp. Mech.*, SEM, 298.
[14] ASME/ANSI B16.5, 1988, Sec. VIII, Div. I.
[15] Sawa, T., Morohoshi, T., and Kumano, H., 1991, "A New Calculation Method of the Spring Constant in a Bolted Connections with a Gasket," PVP (Am. Soc. Mech. Eng.) **217**, pp. 119–128.
[16] Sawa, T., and Shiraishi, H., 1983, "A Simple Method to Calculate the Force Ratio of Bolted Joints," Bull. JSME, **26**(216), pp. 1088–1096.
[17] Tsuji, H., and Nakano, M., 2002, "Bolt Preload Control for Bolted Flange Joint," PVP (Am. Soc. Mech. Eng.) **433**, pp. 163–170.
[18] ASME PCC-1-Guidelines for Pressure Boundary Bolted Flange Joints Assembly.

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EVALUATION OF SEALING BEHAVIOR OF GASKETS BASED ON THE TEST METHOD HPIS Z104 PROPOSED IN JAPAN

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ABSTRACT

This paper discusses the gasket testing procedure HPIS Z104 to obtain fundamental sealing behavior of gasket established in Japan. The testing procedure consists of 11 combinations of gasket stresses and a constant internal pressure. It takes about 3 hours to complete one test, which is acceptable for gasket manufacturers. In order to demonstrate the validity of the testing procedure, measurements of leak rates of compressed fiber sheet gaskets were carried out. It has shown that the fundamental sealing behavior can be well characterized using the proposed testing procedure with reasonable time and cost.

Keywords: Gasket testing, Joint sheet gasket, Leak rate, Leakage

INTRODUCTION

With the recent increase of a safety and environmental concern, the tightness of gasketed flanged connections becomes an important issue[1]. In order to estimate the tightness of gasketed flanged connections, the sealing behaviors of gaskets must be available. Currently, two methods to test the sealing behaviors have been established in the North America and Europe independently[2]. One of the problems of these methods is that gasket tests take a long time to perform and are costly. This is mainly due to the facts that the targeted leak rates in both the methods are very small compared with those considered in the design of flanged connections and that they are only measurable using a mass spectrometry.

A method to test the tightness of gaskets, HPIS Z104 2005, was published in Japan[3]. The testing procedures to obtain a fundamental sealing behavior of gasket are discussed in this paper. The targeted minimum leak rate is about $1.69 \times 10^{-4} \sim 1.69 \times 10^{-2}$ Pa·m^3/s (0.1~10 atm cc/min), which is measurable using a burette. The testing procedure includes 11 steps of gasket stresses. It takes about 3 hours to complete one test, which is reasonable and acceptable for gasket manufacturers. In order to clarify the validity of the testing procedure, measurements of leak rates of compressed fiber sheet gaskets were carried out. The sealing behavior of the gasket is discussed based on the test results. It was shown that the fundamental sealing behavior can be well characterized using the proposed testing procedure with reasonable time and cost.

NOMENCLATURE

Dimensions

d_i : ID of contact surface of gasket (mm)

d_o : OD of contact surface of gasket (mm)

Area

A_g : Contact area of gasket (mm^2)

Gasket stress and thickness change of gasket

δ_a : Arithmetic mean value of thickness changes of gasket (mm)

σ : Gasket stress (N/mm^2)

σ_e : Effective gasket stress (N/mm^2)

σ_{max} : Maximum gasket stress (40 and 100 N/mm^2 for non-metallic gasket and spiral wound gasket, respectively)

Internal pressure

P : Test pressure (MPa)

Leak rate

L : Measured leak rate of gasket ($Pa \cdot m^3/s$)

L_s : Fundamental leak rate ($Pa \cdot m^3/s$)

Load

W : Compressive load (N)

W_0 : Preload (N)

Others

k : Shape factor of gasket

EQUATION REPRESENTING LEAK RATE OF GASKET

The gasket testing procedure HPIS Z104 to obtain fundamental sealing behavior of gasket is explained below.

An equation representing a leak rate of gasket is defined under the following assumption: A leak rate is proportional to the inner radius of gasket d_i and is inversely proportional to the width $(d_o - d_i)/2$ of gasket (see Fig. 1).

$$L \propto \frac{d_i}{(d_o - d_i)/2} \qquad (1)$$

On the basis of the assumption, the leak rate of a gasket L can be expressed by the following equation:

$$L = \frac{1}{d_o/d_i - 1} L_s = k\, L_s \qquad (2)$$

where, L_s is the fundamental leak rate whose physical meaning is a leak rate of a gasket where the outer diameter d_o is twice as large as the inner diameter d_i. The value k is the shape factor which is expressed as

$$k = \frac{1}{d_o/d_i - 1} \qquad (3)$$

As understood from Eq.(2), the leak rate L stays constant when the ratio d_o / d_i is kept constant even if the gasket size is changed. The shape factor is 1.0 when the outer diameter d_o

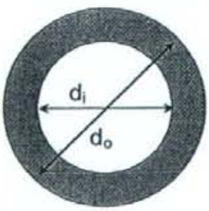

Fig. 1 Dimensions of gasket

is twice as large as the inner diameter d_i as shown in Fig. 1.

Thus, measured leak rates are converted to the fundamental leak rate by the following equation:

$$L_s = \frac{L}{k} \qquad (4)$$

TEST METHOD

Test rig and gaskets

An example of test rig used in the leak rate measurements is shown in Photo 1. A gasket is compressed between the upper and lower platens. Test conditions are as follows:

- The test gasket: 20K50A(JIS B2404)
- Pretreatment of gasket: 23±2℃ and 50±5% environment for 48 hours
- Platens: with raised face of 96 mm OD
- Surface roughness of the platens: 1.6~3.2 μmRa
- Test medium: Helium gas

Procedure to test sealing behavior

A preload W_0 calculated from the following equation is applied to a gasket before a test.

$$W_0 = 0.05\, A_g\, \sigma_{max} \qquad (5)$$

The gasket contact area A_g is defined by the following equation:

$$A_g = \frac{\pi}{4}(d_o^{\,2} - d_i^{\,2}). \qquad (6)$$

The gasket stresses and test pressures are summarized in Tables 1 and 2. Different gasket stresses are used depending on gasket type. The maximum gasket stresses are 40 and 100

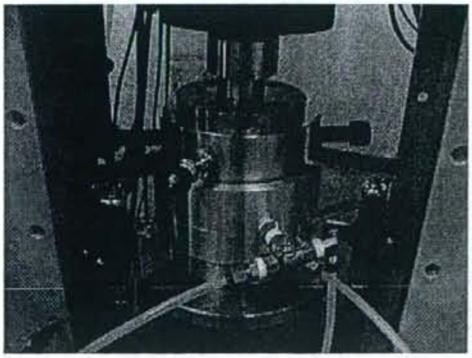

Photo 1 Test rig

Table 1 Test sequence I (For non-metallic gaskets)

Step	S1	S2	S3	S4	S5	S6	S7	S8	S9	S10	S11
σ N/mm^2	5	10	20	10	5	20	30	40	20	10	5
P MPa	2										

Table 2 Test sequence II (For spiral wound gasket)

Step	S1	S2	S3	S4	S5	S6	S7	S8	S9	S10	S11
σ N/mm^2	12.5	25	50	25	12.5	50	75	100	50	25	12.5
P MPa	4										

MPa for sheet gaskets and spiral wound gaskets, respectively. Compressive load of gasket is calculated by

$$W = A_g \sigma . \tag{7}$$

The testing procedure is also shown in Fig. 2. In the testing procedure, the gasket is loaded to half of the maximum gasket stress, then, unloaded until one eighth of the maximum gasket stress. The gasket is loaded again to the maximum stress, then, unloaded again. This testing procedure simulates the assembly process and the working condition of gaskets.

The internal pressures are 2 and 4 MPa for sheet gaskets and spiral wound gaskets, respectively. The gasket is left for 5 minutes when the gasket stress is changed. Then, the leak rate is measured using a burette. It takes about 15 minutes for each measurement. Leak rate measurements for the 11 steps can be completed approximately within 3 hours.

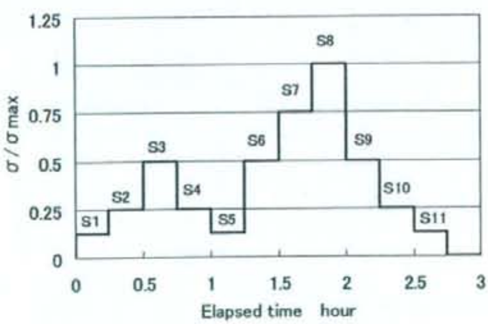

Figure 2 Testing sequence of gasket

Data processing and graphical representations of test data

Effective gasket stress is calculated using the following equation:

$$\sigma_e = \frac{W - \pi\, d_i^{\,2} P/4}{A_g}. \tag{8}$$

where, the load W and the internal pressure P are measured values.

The test results are summarized in the following graphs:

- Effective gasket stress σ_e – deformation of gasket δ_g
- Fundamental leak rate L_s – Effective gasket stress σ_e
- Fundamental leak rate L_s – deformation of gasket δ_g

TEST RESULTS

Figure 3 shows the fundamental leak rate L_s as a function of the gasket stress obtained using the loading-unloading sequence shown in Fig. 2. The tested gasket is compressed non-asbestos fiber sheet gasket #1995(Nichias Corporation), of which thickness is 1.5 mm. The inner and outer diameters of contact surface of gasket, d_i and d_o, are 61 and 96 mm respectively.

The leak rate is expressed by the unit Pa · m^3/sec throughout this paper. The leak rate decreases with increasing the gasket stress in the assembly process. When the gasket is unloaded at about 20 MPa, which corresponds to a working condition, the leak rate stays less than that in the assembly condition. Leak rates for S5 through S10 could not be measured because the leak rates were well under the measurable limit of a burette. It can be seen that the leak rates for loading and unloading cycles are different each other. As shown in this result, the sealing behavior is complex and is difficult to formulate.

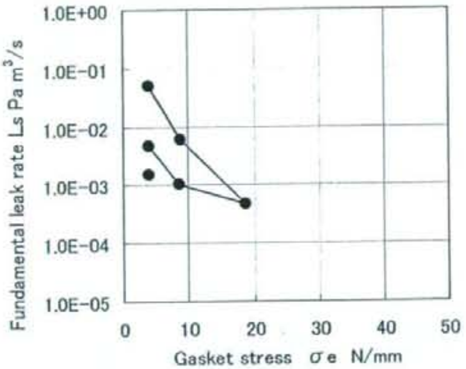

Figure 3 Sealing behavior as a function of gasket stress

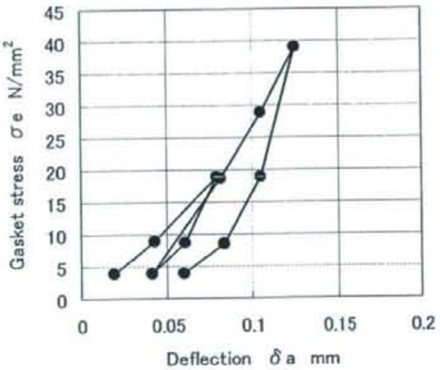

Figure 4 Stress-deflection curve of gasket

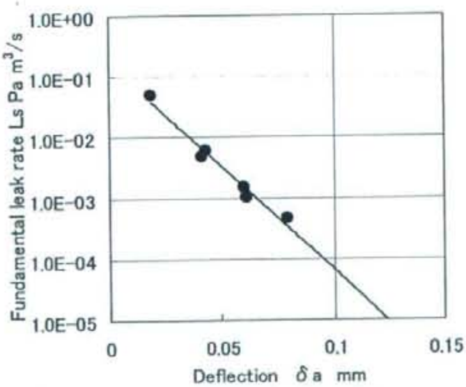

Figure 5 Sealing behavior as a function of gasket deflection

Figure 4 shows the stress-deflection diagram of the tested gasket during the seal test. As shown in the figure, the inclination increases as the gasket is compressed. This means that the gasket becomes stiffer. When the gasket is unloaded at about 20 MPa, a large plastic strain is observed compared with the recovery of the elastic strain. The reason for this is thought that the compressed fiber sheet gaskets are porous in nature. Once they are compressed, the material cannot be recovered due to its porous structure.

It has reported that the leak rate has a close relation with the gasket strain or deflection [4, 5]. The leak rate is arranged by the deflection of the gasket and the relation between them is shown in Fig. 5. The leak rate is indicated using a log scale. The solid line is an approximated result mentioned later. As shown in the figure, experimental results in the assembly process and those in the working condition almost coincide each other and fall onto one straight line on a semi-log graph. This fact strongly suggests that the leak rate of gasket is directly related to the gasket deflection. The reason for this is thought to be as follows: many micro leak paths through the section of gasket exist because the gaskets used in this study are porous. It is thought that the cross sectional areas of the micro leak paths govern the leak rate and the strain or deflection of gaskets is a direct measure of the cross sectional area of the micro leak paths. Thus, the leak rate has a good correlation with the gasket strain or deflection. As the relation between the leak rate and the gasket deflection is linear on a semi-log plot, the leak rate can be expressed by the following equation:

$$L_n = 0.1675 \cdot e^{-78.22\delta_a} \qquad [\mathrm{Pa \cdot m^3/s}]. \qquad (9)$$

The calculated result by Eq. (9) is also plotted in Fig. 5. The experimental data are well approximated by Eq. (9).

CONCLUSIONS

This paper discusses the gasket testing procedure HPIS Z104 to obtain fundamental sealing behavior of gasket established in Japan. It is shown that the fundamental sealing behavior can be well characterized using the proposed testing procedure.

The future project is to develop experimental formulas to indicate the sealing behavior of gasket for various kind of gaskets based on measured data obtained using the testing procedure HPIS Z104.

REFERENCES

[1] H. D. Raut, A. Bazergui and L. Marchand, 1981, "Gasket Leakage Behavior Trends", WRC Bulletin 271, pp.16-42.
[2] M. Derenne, J. C. Vignaud and J. R. Payne, 1997, "Bolted flanged Gasketed Joints Technology: Comparison of North American and European Approaches", Current Topics in the

Design and Analysis of Pressure Vessels and Piping, ASME PVP-Vol. 354, pp.209-244.
[3] HPIS Z104 2005, "The Test Method for Sealing Behavior of Gaskets for Pipe Flanges", (in Japanese), High Pressure Institute of Japan.
[4] T. Kobayashi, T. Nishida and Y. Yamanaka, 2001, "Mathematical Model for Sealing Behavior of Gaskets Based on Compressive Strain", Analysis of Bolted Joints, ASME PVP-Vol.416, pp.105-109.
[5] T. Kobayashi, T. Nishida and Y. Yamanaka, 2002, "Simplified Sealing Test Procedure of Gaskets Based on Compressive Strain", Analysis of Bolted Joints, ASME PVP-Vol.433, pp.29-34.

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APPLICATION OF PLASTIC REGION TIGHTENING BOLT TO FLANGE JOINT ASSEMBLY

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ABSTRACT

Elastic region tightening based on torque control method is conventional method of tightening a bolt. Axial force of the bolt is controlled by a torque wrench, however, it is not easy to achieve uniform bolt tightening force. When torque method is applied to flange joint assembly, the scatter of the bolt tightening foeces are large. They might cause the leakage of the internal fluid from a flange joint. Recently, plastic region tightening is remarked for critical applications, which provides good uniformity in bolt preloads and high preloads compared with the elastic region tightening.

In this research, the plastic region tightening is applied to flange joint assembly and its superiority in uniformity of the bolt tightening force is demonstrated. For the tightening tests, JPI-4inch flange, spiral wound gaskets and M16 bolts were used. Axial force and elongation of all bolts in the flange were measured. Bolts were tightened by modified HPIS flange tightening procedure which incorporates the angle control method into the clockwise tightening sequence.

Experimental results show that variations of the axial force in the plastic region was smaller than those in the elastic region. The influence of the elastic interaction on the axial force in the plastic region is also small. It is concluded that the application of plastic region tightening to flange joint assembly is effective for the leak-free joint and that the nominal diameter of the bolt can be reduced.

INTRODUCTION

A flange joint with a gasket is widely used for the joint of the piping and the pressure vessel in various plants. The pressurized fluid contained are often under high temperatures and harmful. It is difficult to prevent the leakage completely, therefore there is a risk that the leakage causes an accident. Many researches have been carried out to establish a method of design and assembly procedure of the leak-free flange joint, though it has not been reached.

Conventional tightening method of the flange joint tightened with multi-bolts is elastic region tightening by the torque control method with torque wrench. In the elastic region tightening method, variation of the friction coefficient under tightening affect the uniformity of the tightening force. Additionally, flange joint assembly brings some problems and difficulties; the tightening force variation due to elastic interaction, uneven flange gaps and the relaxation of the gasket. They may cause ununiformity in the gasket stress and result in the leakage of an internal fluid from the flange joint.

In the automotive industry, the plastic region tightening is in attracted as practical tightening method in which the target of tightening force is yield point or plastic region. The plastic region tightening has advantages as follows; higher tightening force, less variation of the tightening force. The yield tightening force is determined by the mechanical properties of the material and the effect of the friction coefficient of the bolt is small. The plastic region tightening prevents advantageously the fatigue fracture and the relaxation of the joint and may increase the reliability of the joint. The diameter of the bolt is reduced, the number of the bolt is decreased or the strength class of the bolt is lowered. For the critical application, therefore, high performance and cost reduction are achieved compatibly.

The plastic region tightening has been applied to various fields. In the automotive industry, it is applied to the parts where frequent disassembly and reassembly are not required such as the cylinder head bolt and the connecting rod bolt in the engine assembly, and its superiority is demonstrated. In an architectural field, it is applied to friction grip bolts. Many researches have been done to clarify the superiority of the plastic region tightening and to establish a strength design of the joint.

Table 1 HPIS tightening procedure [2].

Step	Loading
Install	Hand tighten all bolts, then tighten 4 or 8 equally spaced bolts with gradually increased tightening torque to 100% of target torque on a cross-pattern tightening sequence. Check flange gap around circumference for uniformity.
Tightening	Tighten all bolts with tightening torque to 100% of target torque on a rotational clockwise pattern for specified iterations (6 passes for 10 inch and greater flange, 4 passes for others).
Post-tightening	If necessary, wait a minimum of four hours and tighten by the previous step, but 1 or 2 passes.

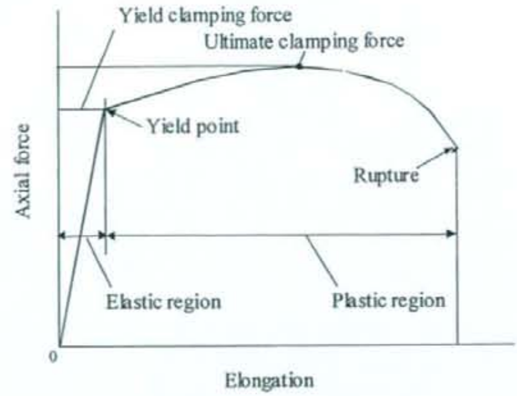


Fig.1 Relation between axial force and bolt elongation under tightening.

Table 2 Tightening control method.

Tightening control method	Index	Tightening area	Tightening coefficient Q
Torque control method	Tightening torque	elastic region	1.4-3
Angle control method	Tightening angle	elastic region	1.5-3
		plastic region	1.2
Torque gradient control method	Tightening torque gradient for tightening angle	elastic limit	1.2

Today, the advantages of the plastic region tightening has been noticed. However, the characteristics of joints with multi-bolts tightened to the plastic region such as the flange joint is not yet shown.

In this research, the plastic region tightening is applied to flange joint assembly. Tightening procedure for the multi-bolt joint according to the plastic region tightening method is proposed and the superiority in the uniformity of the bolt tightening force is demonstrated. The uniformity of the bolt tightening force contributes to the leak-free service of the flange joint. Since the plastic region tightening obtains higher tightening force compared with the elastic region tightening, it enables the bolt and the flange joint to be downsized.

OVERVIEW OF THE BOLT TIGHTENING PROCEDURE FOR FLANGE JOINT ASSEMBLY

Although the importance of bolt tightening force control is recognized and the torque control method is applied widely, it is difficult to achieve accurate tightening force. The friction coefficient may varies even if the torque wrench is carefully used, and it follows that the scatter in tightening force becomes at least ±17% [1].

JPVRC BFC committee has proposed a new tightening method (HPIS procedure, namely clockwise pattern tightening method[2]), with the aim of a simple and effective procedure which achieves uniform bolt preloads and accurate flange alignment [2].

Table 1 shows the steps of the HPIS tightening procedure. The bolts are tightened to 100% of the target torque in all steps to decrease the number of tightening steps, except for the install step to prevent the flange misalignment by snugigng and tentative tightening. In all steps, the clockwise-pattern bolt tightening sequence is employed with the aim of a simple procedure to prevent human errors.

HPIS tightening procedure (Clockwise-pattern tightening method) achieves a comparable uniformity of the bolt tightening force, while the total tightening steps/rounds required for the joint assembly is decreased, compared with ASME method which employes the torque increment steps and a cross-pattern bolt tightening sequence[3]. HPIS procedure reduces both the cost and the human errors to avoid the complicated specification on the torque increment rounds and the tightening sequence, so that the method is practical with effectiveness and simplicity [2],[4].

THE CONCEPT OF PLASTIC REGION TIGHTENING

Table 2 shows various tightening control methods. The torque control method is applied to the elastic region tightening. The angle control method and the torque gradient method are applied to the plastic region tightening. The tightening factor

$Q = F_{f\,max}/F_{f\,min}$ shown in Table 2 means a scatter in the tightening force. The value of Q for the plastic region tightening including the yield point tightening is smaller than that for the elastic region tightening.

Figure 1 shows the relation between axial force and elongation of bolt under tightening to rupture.

Plastic region tightening is a method of tightening to the plastic region where the tightening force is greater than the yielding load (yield clamping force in Fig. 1) of the bolt. The tightening force of the bolt, which depends on its mechanical properties and the friction coefficient, is 80-90% or more of the tensile strength of the bolt. Since the control of strength of the bolt is easier than the control of friction coefficient, the scatter of tightening force by the plastic region tightening is smaller than the elastic region tightening. The plastic region tightening method has higher efficiency of the strength utilization of the bolt.

For the plastic region tightening, the tightening force is not controlled by the torque control method, therefore, an angle control method, i.e. the control of the rotational angle of the nut, is employed. Since the slope of curve at plastic region shown in Fig.1 is small, effect of the error of the rotational angle on the scatter of the tightening force is small.

ESTIMATION OF YIELD TIGHTENING FORCE BASED ON RIGID-PLASTIC MODEL

Estimation method of yield tightening force of bolts based on rigid-plastic model was proposed by Tsuji et al. [5], [6]. It is possible to estimate the yield tightening force in the plastic region tightening by the use of the material constant obtained by uniaxial tension of the bolt. The yield tightening force F_{fy} of the bolt is expressed as

$$F_{fy} = \frac{\pi d_S^2 \sigma_{ys}}{4\sqrt{1+3\left\{\frac{3}{2}\frac{d_2}{d_S}\left(\frac{P}{\pi d_2}+1.155\mu_s\right)\right\}^2}} \qquad (1)$$

where d_2 is pitch diameter, d_S is diameter of net cross-sectional area, P is pitch, σ_{ys} is yield stress and μ_s is friction coefficient of thread.

Bolts with reduced shank of hollow cylinder were used for the tightening test in order to adjust the yield tightening force. The yield tightening force of the bolt with reduced shank is also calculated based on the solution extended to a hollow cylinder.

For the rigid-plastic hollow cylinder, the yield criterion under the combined loads of axial tension and torque is expressed as follows :

$$\left(\frac{F_{fy}}{F_y}\right)^2 + \left(\frac{T_{sy}}{T_y}\right)^2 = 1 \qquad (2)$$

where F_{fy} is axial tension and T_{sy} is torque. The value of F_y and T_y are yield point loads when the hollow cylinder is subjected to axial tension only and torque only, and expressed as follows :

$$F_y = \frac{\pi}{4} d_E^2 (1-k^2)\sigma_{ys} \qquad (3)$$

$$T_y = \frac{\pi}{12} d_E^3 (1-k^3)\tau_{ys} \qquad (4)$$

where σ_{ys} is the yield stress (under uniaxial tension), d_E is outer diameter of reduced shank body and $k = d_i/d_E$ is diameter ratio of hollow cylinder. Shearing yield stress $\tau_{ys} = \sigma_{ys}/\sqrt{3}$ is derived from von Mises yield criterion.

Since thread torque T_s is proportional to the axial force of the bolt F_f, the relation between yield tightening force F_{fy} and yield thread torque T_{sy} is expressed as

$$T_{sy} = F_{fy}\frac{1}{2}\left(\frac{P}{\pi}+1.155\mu_s d_2\right) \qquad (5)$$

Knowing μ_s, F_{fy} is obtained from Eqs. (2)-(5) as follows :

$$F_{fy} = \frac{\pi d_E^2 (1-k^2)\sigma_{ys}}{4\sqrt{1+3\left\{\frac{3}{2}\frac{d_2}{d_E}\frac{(1-k^2)}{(1-k^3)}\right\}\left(\frac{P}{\pi d_2}+1.155\mu_s\right)^2}} \qquad (6)$$

When σ_{ys} = 900MPa obtained from uniaxial tension test is substituted for Eq.(6), F_{fy}=42.8kN. Calculated yield tightening force is decreased by 8% compared with the yielding load 45.1kN obtained by the uniaxial tension.

PRELIMINARY TIGHTENING TEST OF BOLT

INDICATOR TYPE OF BOLT

Characteristic of plastic region tightening of the bolt used is examined before the tightening test of the flange joint.

Figure 2 shows the indicator type of bolt for the measurement of the bolt elongation. Nominal diameter of the bolt is M16. The material of the bolt is SNB7, and of the nut is S45C. The middle part of the bolt was machined to a reduced shank body. A through hole of the diameter 6mm is drilled and a indicator rod of diameter 4mm is passed through the bolt to mea-

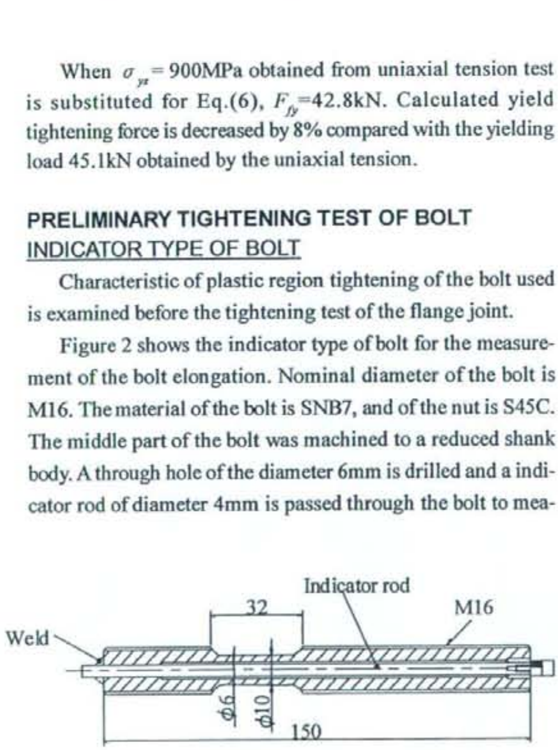


Fig.2 Indicator-type of bolt.

sure the bolt elongation. An edge of the rod is fixed by welding, and a flat gage head is set on the other edge.

TIGHTENING TEST CONDITIONS

Figure 3 shows a assembly of a bolt and a elongation detector and Fig. 4 shows a setup for the plastic region tightening test of the bolt. Voltage change of a potentiometer by the displacement of the indicator rod is measured with a digital multimeter and recorded by a PC. A load cell which measures the tightening force is placed between nuts. The lubricant used in the test is a dry coating spray of MoS_2 (molybdenum disulfide) used in many plants.

Firstly, the bolt is tightened by hand lightly. After that, the bolt is tightened untill the bolt breaks by the angle control method using a torque wrench with a ten times torque amplifying device.

CHARACTERISTIC OF BOLT UNDER THE PLASTIC REGION TIGHTENING

Figure 5 shows the test result of the plastic region tightening of a set of the SNB7 bolt and the nut. Figure 5 (a) shows the relation between the nut rotational angle and the bolt tightening force and (b) shows the relation between the bolt elongation and the bolt tightening force. The bolt yielded when its elongation reached to 0.3 mm, and ruptured at 2.1 mm in elongation. The yield tightening force is 47.9kN, determined by 0.2% permanent set of the reduced shank body length. These behaviors are similar to those of general high-strength bolt tightened to the plastic region.

To obtain the relation between the nut rotational angle and the bolt elongation, the compression compliance of the joint should be taken into account, besides the pitch of the bolt. Since the compression compliance of the flange joint is large due to the gasket and the flange rotation in the case of the following flange tightening test, the target value of the tightening angle should be chosen to make sure that bolt is settled into the plastic region.

TARGET VALUE OF NUT ROTATIONAL ANGLE

Figure 6 shows the method to determine the target value of the nut rotational angle θ_N. In order to minimize the influence of the variation of the nut rotation angle on the tightening

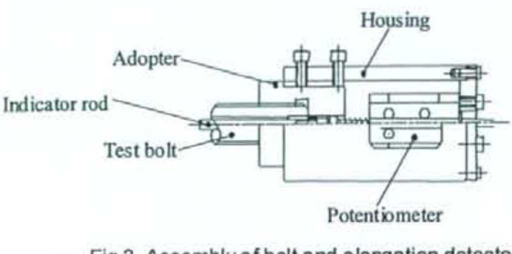
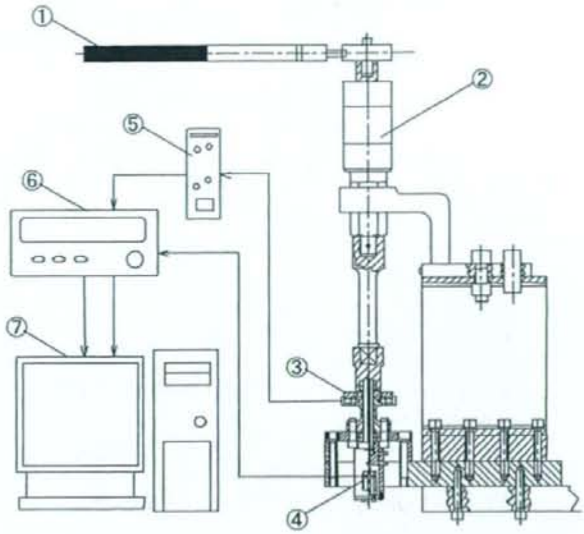
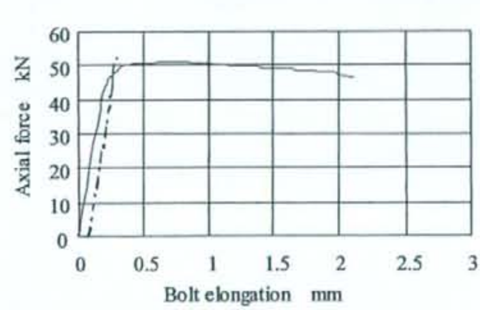


Fig.3 Assembly of bolt and elongation detector.

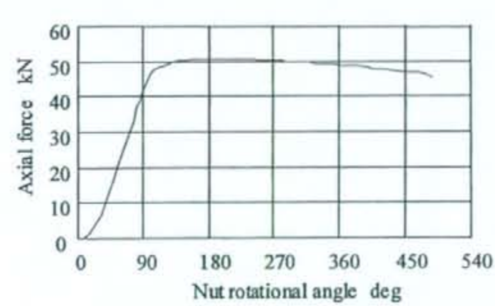


① Torque wrench ② Ten times torque amplifying device
③ Load cell ④ Potentiometer ⑤ Strain amplifier
⑥ Digital multi-meter ⑦ Personal computer

Fig.4 Setup for plastic region tightening test.



(a) Relation between axial force and bolt elongation.



(b) Relation between axial force and nut rotational angle.

Fig.5 Result of plastic region tightening test.

force, θ_U shown in Fig.6 seems to be suitable for the target value. Necking of the bolt starts just after the maximum point θ_U, so that the weakening effect of the reduction in the cross-sectional area is dominant and the tightening force decreases until the bolt ruptures. It might be undesirable to generate necking immediately after tightening. Considering the assembly efficiency in the site and the reuse of the bolt, nut rotational angle should be as small as possible. Then, the mean value of θ_Y and θ_U, is considered as the optimum target value of the nut rotational angle θ_N,

$$\theta_N = \frac{1}{2}(\theta_Y + \theta_U) \qquad (7)$$

The value of θ_N is set to 420 degrees by the preliminary flange tightening test, taking into account of the compliance of the actual flange joint.

PLASTIC REGION TIGHTENING TEST OF THE FLANGE JOINT

SETUP FOR TIGHTENING TEST

Figure 7 shows a setup for the plastic region tightening test of the flange joint. Bolt tension is controlled by the angle control method where a torque wrench and a ten times torque amplifying device are used for tightening operation. Tightening force of each bolt is measured using a load-cell with strain gages and the flange gaps are measured by a vernier caliper. Elongation of the bolt is measured using the potentiometer. Measured data are recorded by a personal computer through a digital multimeter. The test flange is the 4 inch class 150 lb (material: SFVC2A), raised face slip-on welding type flange specified in JPI (Japan Petroleum Industry). The test bolt (stud) is nominal diameter of M16 with nuts (bolt material: SNB7, nut material: S45C). Figure 8 shows a test gasket, spiral wound gasket (SWG) made of non-asbestos filler with an outer ring (No.591, Nippon Valqua Co.).

TEST CONDITIONS

Tightening test procedure follows the slightly modified HPIS procedure which employs a rotational clockwise pattern tightening sequence, to achieve an assembly efficiency and the joint reliability. The angle control method is used for tightening instead of the torque control method. As an install step, bolts are hand tightened, and are tightened by a cross-pattern tightening sequence with a snug torque corresponding to 10kN of the bolt axial force. As a main step, bolts are tightened in a rotational clockwise pattern sequence and the nut rotational angle is 60 degrees. Since the tightening force is 49kN when the bolt completely reaches the plastic region by the preliminary test, target value of the nut rotational angle is set to 420 degrees, corresponding to 7 passes. Lubricant used in the test is a dry coating type spray of MoS_2 (molybdenum disulfide) applied to many plants.

RESULTS AND DISCUSSIONS

Figure 8 shows the result of the tightening test of the flange joint: Figure 8 (a) shows the relation between the pass number and the bolt elongation and (b) shows the relation between the pass number and the axial force. The axial forces of all bolts are measured during the tightening process. The bolt elongation increases in proportion to the pass number from pass No.1 to pass No.4. The rate of the bolt elongation increment changes when the bolts reaches the plastic region (Fig. 8 (a)). In pass No.5, the bolt axial force becomes 40kN and reaches the yield

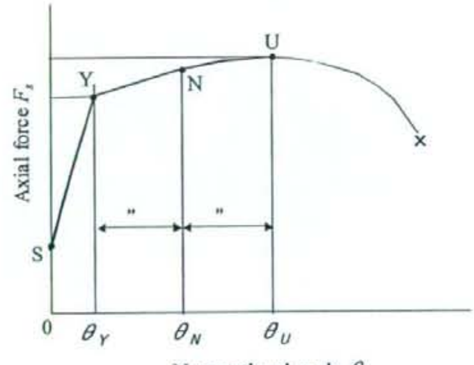
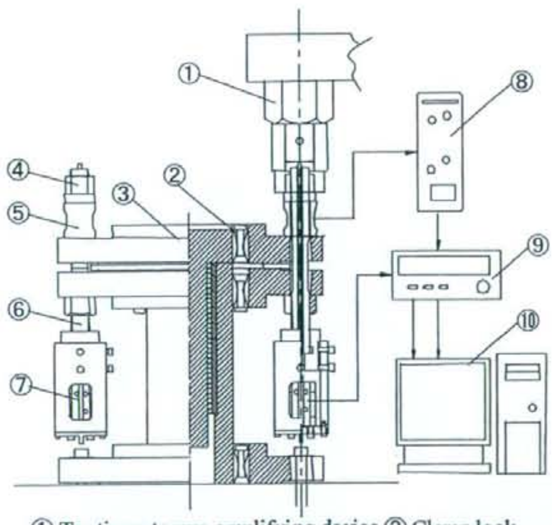


Fig.6 Determination of target of nut rotational angle.



① Ten times torque amplifying device ② Clamp lock
③ Flange joint ④ Hexagon nut ⑤ Load-cell
⑥ Bolt ⑦ Potentiometer ⑧ Strain amplifier
⑨ Digital multimeter ⑩ Personal Computer

Fig.7 Setup for flange tightening test.

point. In pass No.6, all bolts reach the plastic region. Their final axial force, namely tightening force are about 45 kN at θ_N. Scatter of axial force is 10% or less when tightening was completed. The uniformity of bolt tightening forces obtained by the test is equivalent or superior to that of the elastic region tightening. A favorable result is obtained compared with the data of HPIS procedure for the elastic region tightening. In the case of the bolted joint tightened by many bolts such as the flange joint, the scatter of the bolt axial force is small by the plastic region tightening.

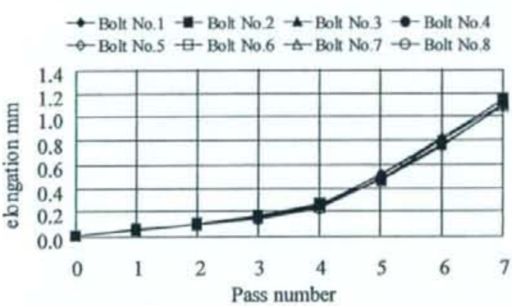
Figure 9 shows the elastic interaction in the tightening processes: Fig. 9 (a) shows variation of the bolt axial force in the pass of the elastic region and (b) shows variation of the bolt axial force in the pass of the plastic region.

When two adjoined bolts have been arranged relatively closely, the influence of the elastic interaction on the variation of the bolt axial force is significant under the elastic region tightening (torque control method). The influence of the elastic interaction appears in the elastic region under the plastic region tightening by the angle control method as well as the elastic region tightening by the torque control method. When bolts reach the plastic region, tightening of a bolt does not influence the axial force of the neighboring bolts. An increase in the axial force is very small in the plastic region, so that the influence of the elastic interaction is also small. It is effective in the uniformity of the bolt tightening force.

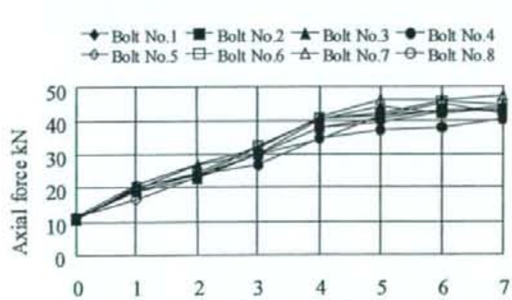
The flange gaps are measured at four points of the flange circumference in order to check the uneven clamping. Ununiformity of flange gaps measured are from 0 mm to 0.4 mm.

These data are the comparable to the data of the elastic region tightening by HPIS procedure. Uneven clamping does not appear in the plastic region tightening.

In this research, bolts with the reduced shank of cross-sectional area $A_g = 50.3$ mm^2 are used to adjust the tightening force and their yield clamping forces are 50 kN. Standard tightening force for the combination of the test flange and the gasket is 35kN. Assuming that the number of bolts and the strength class of bolts are constant, equal tightening force can be achieved by using M8 bolts (stress area $A_s = 36.6$ mm^2). Application of plastic region tightening to the flange joint is able to downsize the bolts and the flange due to higher tightening forces and their uniformity.

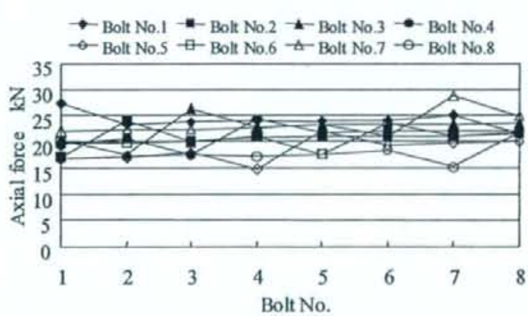


(a) Relation between pass number and bolt elongation.

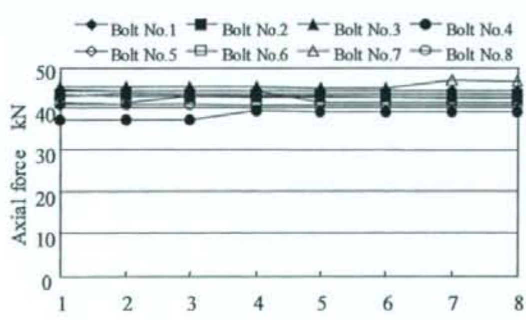


(b) Relation between pass number and bolt elongation.

Fig.8 Result of tightening test of flange joint.



(a) Pass at elastic region. (Pass No.2)



(b) Pass at plastic region. (Pass No.6)

Fig.9 Variation of axial force during one pass.

CONCLUSIONS

This paper described the application of plastic tightening bolt to the flange assembly. The following results were obtained.

(1) The behavior of the plastic region tightening of the bolt made of SNB7 was examined. There was no difference between the SNB7 bolt and the high strength bolt for the automotive application.

(2) The estimation method of the yield clamping force of the bolt with reduced shank of hollow cylinder was proposed.

(3) The flange joint was tightened to the plastic region of the bolt by the modified HPIS tightening procedure with the angle control method.

(4) Under the plastic region tightening, the influence of the elastic interaction appeared in the elastic region as well as the elastic region tightening by the torque method. When bolts reached the plastic region, tightening the bolt does not influence the axial force of the neighboring bolts. Uniform tightening force is achieved, so that scatter of final tightening force was 10% and less.

(5) When the plastic region tightening is applied to the 4 inches standard flange joint, equivalent tightening force can be achieved by using M8 bolts in place of M16 bolts.

REFERENCES

[1] VDI 2230 Blatt 1, 2003, Systematische Berechnung hoch beanspruchter Schraubenverbindungen Zylindrische Einschraubenverbindungen.

[2] HPIS Z103 TR2004, 2004, Bolt Tightening Guidelines for Pressure Boundary Flanged Joint Assembly, High Pressure Institute of Japan.

[3] ASME PCC-1 2000, 2000, Guidelines for Pressure Boundary Bolted Flange joint Assembly, ASME.

[4] Nakano, M. and Tsuji, H. 2002, Bolt Preload Control for Bolted Flange Joint, ASME PVP-Vol.433, pp.163-170.

[5] Tsuji, H. and Maruyama, K., 1999, Estimation of Yield Clamping Force Based on Rigid-Plastic Model, 1999 ASME IMECE DE-Vol. 105, pp.157-162.

[6] Sawa, T., Nagata, S, and Tsuji, H., 2006, New Development in Studies on the Characteristics of Bolted Pipe Flange Connection in JPVRC, ASME Journal of Pressure Vessel Techonology, Vol.128, pp.103-108.

フランジ継手用ガスケットの常温・高温下における漏洩量評価

(J-EHOT 試験方法の提案)

Evaluation of Leakage of Gasket for Flanged Joint
under Room Temperature and Elevated Temperature
(Proposal of J-EHOT Test Procedure)

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Key Words: Gasket, Flanged joint, Leakage test, HOTT,ROTT, EHOT, Tightness,

1. 緒言

管用ガスケットのシール性能をまとめ米国機械学会(ASME)内の圧力容器研究委員会(PVRC)は、新ガスケット係数(G_b, a, G_s)を提案している。その係数を導き出す常温試験であるROTT(Room Temperature Tightness test)[1]試験及び高温試験であるHOTT(Hot Operational Tightness Test)[2]試験)及びEHOT(Emission Hot Tightness Test)試験も公表されている。

一方、日本高圧力技術協会(HPI)の圧力設備のシーリング技術研究委員会(STOP)で独自の試験方法として常温でのガスケットの基本密封特性試験方法[3](以下、HPISガスケット試験法と呼ぶ)を規定した。

本研究ではHPISガスケット試験法を高温に拡張した試験方法J-EHOTを提案した。300℃でJ-EHOT試験を実施し、うず巻形非石綿ガスケットの漏洩量を測定している。

2. HPIS ガスケット試験法

HPIS ガスケット試験法では、ガスケット寸法に依存しないものとして定義された基本漏洩量L_s (Pa・m³/s)を有効締付圧σ_e(N/mm²)と圧縮変形量δ_a (mm)で評価する。L_sは次式で表される。

$$L_s = \frac{L}{k} \tag{1}$$

ここで、漏洩量L (Pa・m³/s)、ガスケット形状係数kである。基本漏洩量を用いることにより試験・実用面での汎用性が増す。kは次式で表される。

$$k = \frac{1}{d_o / d_i - 1} \tag{2}$$

HPIS ガスケット試験法では試験手順の簡略化及び時間の短縮が試されている。図1及び2にうず巻形ガスケットにおけるガスケット締付圧の負荷シーケンスを示す。現在HPISでは常温試験のみの規定である。

3. J-EHOT の提案

HPIS ガスケット試験法を高温に拡張した試験方法J-EHOT(Emission Hot Tightness Test)を提案する。EHOT試験では常温特性をROTT、高温特性をHOTTで評価していたものをHOTTの後にROTTを行うことにより高温でのエージング後のシール性能の変化によって劣化したガスケットがシール性能に及ぼす影響を確認できる。この試験法は実際のプラント運転状況を想定しているところに特長がある。

また、HOTT試験だけでは高温下における評価が難しいため、高温下での評価にE-HOTT試験の結果を加えることを提案する。J-EHOT試験ではプラントにおいてのスタートアップ時とシャットダウン後の再起動時とのガスケットのシール性能評価を行うことができる。これにより漏洩レベルがどれくらい増えているか評価ができる。

試験はガスケットを試験装置に装着し、室温状態で図1のPre-ROTTに従ってStepを行う。その後、ガスケット温度を300℃まで上昇させる。昇温後、試験内圧一定で90時間放置(エージング)する。エージングの間、漏洩量は6時間おきに測定する。測定には石鹸膜流量計を用いる。エージング終了後、除荷・負荷過程を繰り返し終了する。図1のStepではHOTT試験において漏洩量の変化が見られなかったので、図2のStepにおいて荷重除荷負荷の幅を大きくし、除荷回数も増加させた。高温試験終了後にガスケット温度を常温に戻し、Post-ROTTとして除荷過程を行う。

4. 試験装置及び試験ガスケット

図3に試験装置の構成を示す.試験ガスケットは油圧シリンダによって均一に圧縮される.プラテンには計8個のカートリッジヒータが埋め込まれており450℃まで昇温が可能である。温度はプラテンに埋め込まれたシース形熱電対により測定している.また,冷却装置によりロードセル,油圧シリンダは熱による影響を受けない。作動流体はHeガスを使用する。試験ガスケット・プラテン周りはメタルベローズとメタル中空Oリングにより密封されており、漏洩したHeガスはメタルチューブを通り石鹸膜流量計へと導かれる。この流量計はガラス体積管内の所定距離内を石鹸膜が移動する時間からHeガスの漏洩量を測定するものである。ガスケットひずみの測定にはダイヤルゲージ及び高温用クリップゲージにて測定する。

試験ガスケットは3インチ内外輪付非石綿うず巻形ガスケット(ASME/ANSI Class 300 NPS3/No.8596,日本バルカー工業製)である。

本試験装置の優位性は、高温状態においてガスケット応力、ガスケット変位、Heガスの漏洩量、ガス内圧を同時に測定できる点にある。これにより実際のプラントの運転サイクル(スタートアップ、シャットダウン)に近づけた理想的な条件で試験でき、従来は不可能だったガスケットの長期劣化化の様子を測定可能にしている。

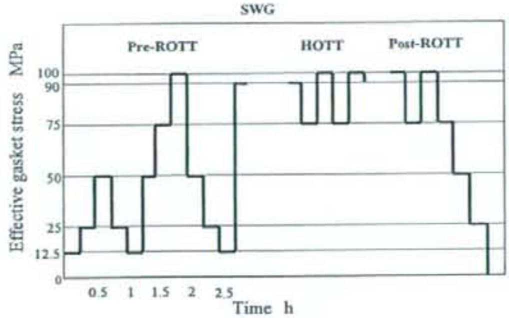

Fig. 1 Loading sequence of a gasket

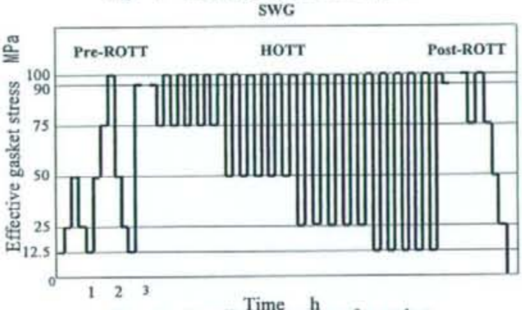

Fig. 2 Loading sequence of a gasket

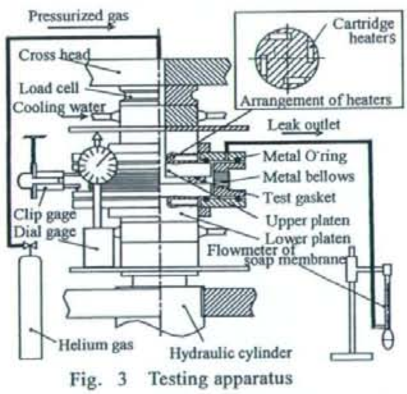

Fig. 3 Testing apparatus

5. 試験結果

図 4 及び 5 にHPISガスケット試験/高温試験/常温試験の流れを基本漏洩量を対数とし，横軸に時間経過(Step)として示す．図 4 及び 5 から，常温試験Step3 において漏洩量を測定することが出来ない．これは石鹸膜流量計の測定限界により測定不可能になったためである．また，その後のStepでも同様に測定限界に達している．そこで本研究では基本漏洩量 2×10^{-7} (Pa・m³/s)を下回る値を測定限界として位置づける．石鹸膜流量計を用いた非石綿うず巻き形ガスケットの常温試験では，このことを考慮に入れる必要がある．

プラント運転時の外乱を想定したガスケット応力の除荷過程を図 4 では 2 回，100，75(N/mm²)の幅で，図 5 では 5 回づつ 100,75,50,25,12,5 でそれぞれ行ったが漏洩量はあまり変化が見られず図 4 及び図 5 共に約 10^{-5}(Pa・m³/s)で一定のシール性能が見られる．HOTT試験後のJ-EHOT試験によりスタートアップ時とシャットダウン後再起動時とのシール性能評価が得られた．これによりガスケットの漏洩レベルが 2 桁～3 桁上昇していることが確認できた．これは実際のプラントにおけるガスケットの使用条件を考慮したもので，初期締付からのシール性能評価が得られた．これにより，HPISガスケット試験方法を用いて，高温後常温におけるシール性能評価が可能であることが認められた．図 4 及び 5 から各試験の結果を比較すると途中のばらつきはあるが，基本漏洩量は高いレベルで最終的な収束を見せている．

以前の実験結果では HOTT 試験において漏洩レベルが大きく変化したものもある．これはガスケットの個体差であると考えられる．よって今後さらに実験を繰り返して高温の外乱のパターンの数はデータ数を増やす必要がある．

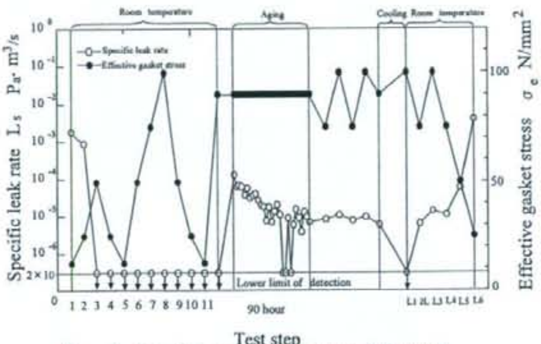

Fig. 4 Loading sequence diagram of a gasket

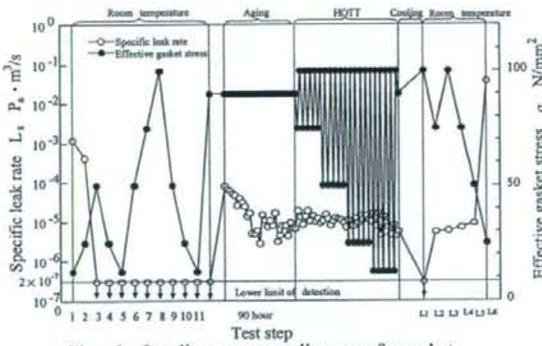

Fig. 5 Loading sequence diagram of a gasket

6. 結言

高温後における管フランジ用ガスケットの基本密封特性試験を行った．以下に得られた成果を示す．

(1) HPIS ガスケット試験法を拡張した高温後のガスケット試験法（J-EHOT 試験法）を提案した．

(2) 提案した試験方法により高温後常温における評価をガスケットの漏洩レベルの比較により，可能であることを明らかにした．

参考文献

(1) Bazergui,A., Payne , J.R.,and Marchand, L, "Effect of Fluid on Seayneling Behavior of Gaskets," Proc.10th International Conference on Fluid Sealing, GHRA, Innsbruck, Austria,April(1984)

(2) Derenne,M., Marcand, L., Payne,J.R; " Elevated Temperature Testing of Gaskets for Bolted Flange Connections,"

(3) 日本高圧力技術協会，管フランジ用ガスケットの基本密封特性試験方法 HPIS Z104, 1-15, (2005)