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EFFECTS OF SCATTER IN BOLT PRELOAD ON THE SEALING PERFORMANCE IN BOLTED FLANGE CONNECTIONS WITH COVER OF PRESSURE VESSEL UNDER INTERNAL PRESSURE

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ABSTRACT

It has been well known that the scatter in axial bolt forces of bolted flange connections tightened by torque control methods is substantial. In evaluating the sealing performance of a bolted flange connection with a gasket subjected to internal pressure, it is necessary to know the contact gasket stress distributions due to the scatter of the axial bolt forces in the flange connections tightened by torque control methods. This paper deals with the leakage of a bolted flange connection with a cover of pressure vessel including a spiral wound gasket tightened by a torque wrench. The scatter in the axial bolt forces was measured using strain gauges attached at the shank of bolts. The amount of leakage from the bolted flange connection with cover of pressure vessel was measured by so-called pressure decay method. The gas employed was Helium. From the measured leakage, the actual assembly efficiency is examined. The eight bolts and nuts were tightened according to the ASME PCC-1 method and Japanese method developed by High Pressure Institute (HPI). The difference in the bolt preload was shown between the ASME method and the HPI method. The contact gasket stress distributions at the interface of the flange connection with the gasket were calculated under the measured axial bolt forces by means of finite element analysis. Using the calculated gasket contact stress distribution, the amount of gas leakage was estimated. The estimated gas leakage was compared with the measured results.

INTRODUCTION

Gasketed flange connections are widely used in chemical and power plants as well as other plants to connect pipes and equipment together. There are many man-holes and hand-holes as well, that are bolted flange connections with cover of pressure vessel. The sealing performance of those bolted connections is important in order to achieve the proper service and the safety operation of the plants. Furthermore, nowadays leakage from bolted connections is given attention from the environmental point of view [1,2]. Achieving good sealing performance is the main criteria for the design of the bolted connections. Lots of researches [3-5] reported sealing characteristics, gasket contact stress distributions, flange hub stress and bolt load changes in the bolted connections subjected to internal pressure.

Pressure Vessel Research Council (PVRC) proposed a leakage-based bolted connection design procedure which prescribed a bolt preload determination method. The design procedure is based on the new gasket constants obtained by the room temperature tightness test (ROTT). ROTT measures leak rates of Helium gas pressurized supposing uniform gasket contact stress. Contrary to this, authors previously reported that the gasket contact stress changes by applying internal pressure and it varies in the radial direction due to the flange rotation as well, for both the identical flange connection [6] and the flange connection with CPV [7]. It was found that the actual leak rates from the pressurized bolted connections can be estimated by using the actual gasket contact stress and the new gasket constants obtained by ROTT. It was also pointed out that further consideration should be necessary for the determination of bolt preload to achieve the required tightness parameter.

Assembling the bolted connections in the fields, torque methods are usually used. The tightening torque is controlled by such tools as a torque wrench, a hydraulic wrench and so on when the bolted connections need higher reliability in the leak tightness. However, there is a scatter in the axial bolt forces of the bolted connections assembled by the torque control method, due to the variety of the torque coefficients between bolts. The bolted connections are pressurized in operation the scatter of the bolt forces still remains. The gasket contact stress is obviously not uniform as ROTT supposed. Since it seems a few researches can be found that focus on the characteristics of leak tightness in the bolted connections considering the scatter of the axial bolt forces, it should be developed the researches on this subject. The gasket contact stress distribution due to the scatter of the axial bolt forces seems to make the actual leak rate to deviate the one estimated based on the uniform axial bolt forces. It is necessary to clarify the behavior of the bolted connections pressurized after assembled by the torque control method in order to establish the optimal design.

The design method proposed by PVRC defines the assembly efficiency according to the method of assembly. It seemed defined as the ratio of the minimum to average gasket stress which accounts for variations in the axial bolt forces and the gasket contact stress distribution. When a torque wrench is utilized to assemble the connection, the assembly efficiency is given to be 0.85 based on the fact that the required bolt preload is hardly achieved by the required torque. However, the assembly efficiency seems empirically determined based on the scatter of axial bolt forces, not taking the effect on leakage into account.

This paper deals with the effects of the scatter in bolt preload on the sealing performance of a bolted connection with a cover of pressure vessel (CPV) subjected to internal pressure. An actual scatter in bolt preloads is measured using the bolted connection assembled by a torque wrench with a target torque. The target torque is determined by the PVRC method using new gasket constants G_b, a and G_s. Two different tightening procedures are adopted to assemble the connections, one is ASME PCC-1 [8], the other is HPIS Z 103 [9], and to compare them. Then internal pressure is applied to the connection and monitored. The actual tightness parameter accounting the scattered bolt preload is compared to the required tightness parameter. Then the assembly efficiency based on leakage is examined. Finite element analysis of the bolted connection with CPV is also performed to examine the gasket contact stress distribution. By using the calculated gasket contact stress distribution, tightness parameter is estimated.

FINITE ELEMENT ANALYSIS

Finite element analysis on the bolted flange connection with CPV subjected to internal pressure is performed. Figure 1 shows a bolted flange connection with a CPV, a gasket inserted in between them, internal pressure applied. After assembled by a torque wrench with a target torque to achieve the required bolt preload, the actual bolt preload for each bolt becomes F_{fi}, respectively. Here, i denotes a serial number of bolts, i=1, 2, 3, … , N. When internal pressure P is applied to the connection, the bolt force increases by F_{ti} and the gasket contact force is reduced by F_{ci} representing each area near the bolt i. Pressure thrust force W ($=\pi r_i^2 P$, r_i is inner radius of flange) is also applied on the flange end.

Figure 2 plots the finite element model used in the analysis. A full 360°, 3-dimensional model is required to simulate the scattered bolt preloads in the bolted connection with CPV. For the analysis, ANSYS Ver.9.0, general purpose finite element analysis software, is utilized. 20-noded hexahedral elements are used for metal parts and gasket elements for the gasket. Though the gasket element accounts for its stiffness in the thickness direction only, a nonlinear stress-strain relationship of the gasket is taken into account as shown in Fig.3.

EXPERIMENTAL METHOD

Leakage tests on the bolted flange connection with a CPV subjected to internal pressure is carried out. A flange, 3B, Class 600, raised face, welding neck and a companion cover flange are specified in ASME B 16.5 [10], both made out of stainless steel 304. The flange and the cover plate are clamped together by 8 bolts made of low alloy steel B7 with a spiral wound gasket inserted made of stainless steel 304. Figure 4 shows the

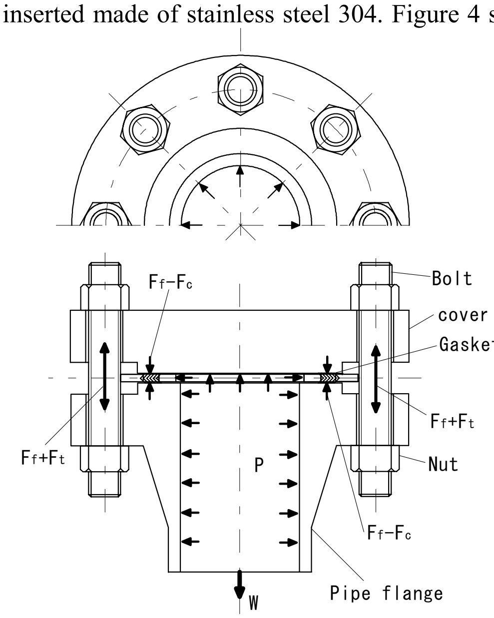


Fig.1 Bolted Flange Connection with Cover of Pressure Vessel subjected to Internal Pressure

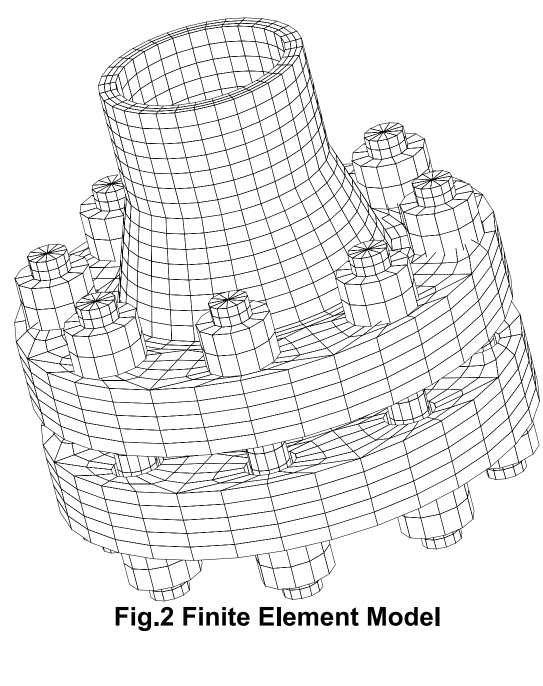


Fig.2 Finite Element Model

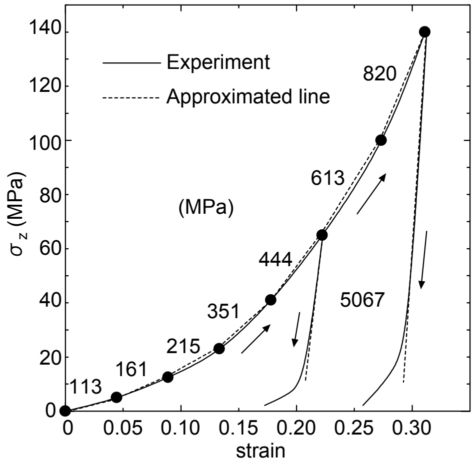


Fig.3 Loading-Unloading Curves of Gasket Used in Finite Element Analysis

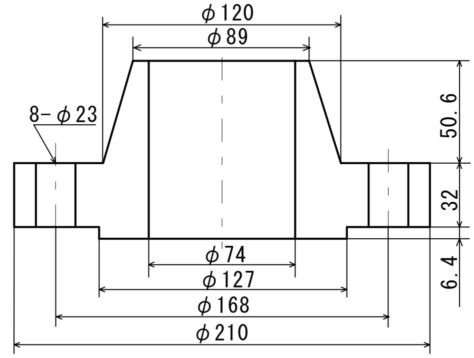
dimensions of the members of the bolted connection. Material properties, such as the modulus of elasticity, Poisson's ratio for the flange, cover plate and bolts according to ASME B&PV Code Sec. II Part D [11] are used. During the bolting up, bolt axial force is measured by using two strain gauges attached to the shank of bolt calibrated in advance. Threads of bolts and nuts and the baring faces are lubricated by a dry coat of MoS_2. The torque coefficient is obtained by single bolt tightening tests after stabilization of the friction coefficient, before the leakage tests. The target torque T is calculated by the following equation.

$$T = k \cdot F_f \cdot d \tag{1}$$

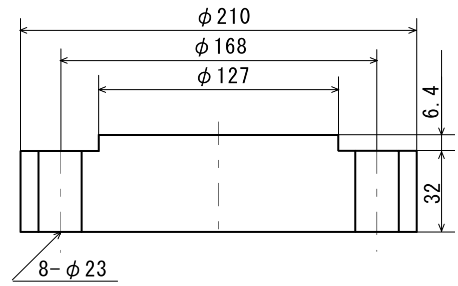
Here, k denotes the torque coefficient, F_f is the target axial bolt force and d is the nominal bolt diameter. The torque coefficient is obtained to be 0.1094 by the single bolt tightening tests. By using the torque coefficient, the target torque is determined according to the target bolt force. The target bolt force F_f is calculated according to the PVRC design method using new gasket constants G_b=19.1MPa, a=0.273 and G_s=2.3×10^{-8}MPa based on the desired tightness class and an assembly efficiency. The assembly efficiency is 0.85 supposing a tightening by a torque wrench. Table 1 shows the loading conditions for the leak tests. Bolts are tightened in accordance with the two different procedures of ASME PCC-1 so-called star-pattern method and HPIS Z103 so-called circular-pattern method, respectively. Internal pressure is applied by Helium gas and monitored by a pressure transducer. Leak rates are calculated by pressure decay method. The leak test is carried out at room temperature.

VARIATION OF BOLT FORCE DURING TIGHTENING

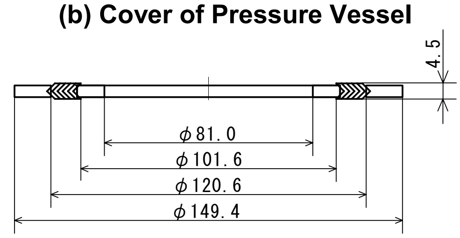
Figure 5(a) shows a series of data for variations of bolt forces during the bolting up by ASME star-pattern method in the Case No.9 of the loading conditions. Contrary, Fig.5(b) shows the ones by HPIS circular-pattern method in the same loading condition. In the ASME method, the ways of bolt force increasing clearly separate in two groups, as reported [12]. One consists of bolts No. 1, 3, 5 and 7, the other consists of bolts No.2, 4, 6 and 8, though the numbering of bolts is shown in Fig.6. In the HPIS method, the bolt forces increase much promptly compared to the ASME method. Only 5 turns almost



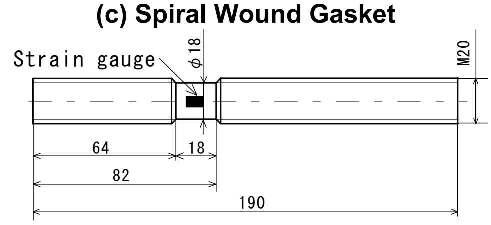
(a) Flange



(b) Cover of Pressure Vessel



(c) Spiral Wound Gasket

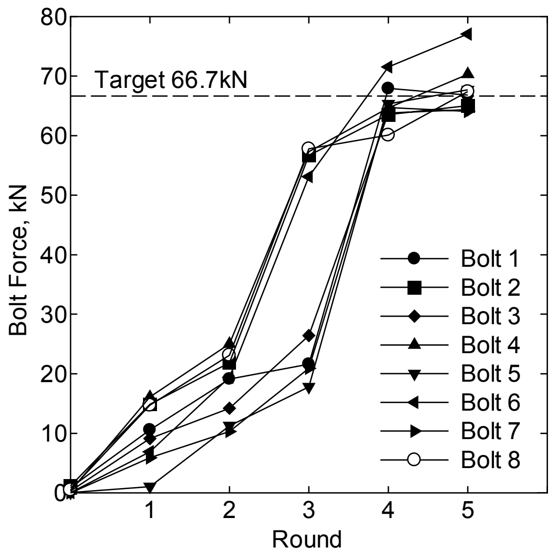


(d) Bolt

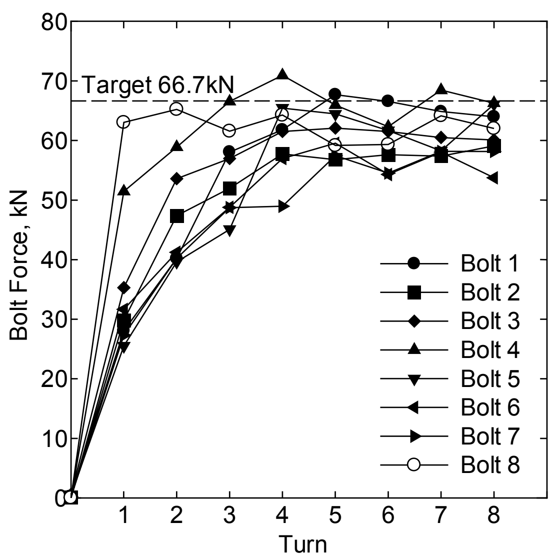
Fig.4 Dimensions of Parts for Bolted Connection

Table 1 Loading Conditions in Leak Tests

Case No.	Tight-ness Class	Target Bolt Force [kN]	Target Torque [N-m]	Internal Press. [MPa]	Tpmin	Tpa
1	0.1	12.2	20	1	1.80	2.70
2	0.1	16.5	27	1	5.41	8.11
3	0.1	19.0	31	1	9.01	13.52
4	1	22.9	38	3	18.02	27.03
5	1	30.9	51	3	54.06	81.09
6	1	35.6	58	3	90.10	135.2
7	10	43.0	71	5	180.2	270.3
8	10	58.0	95	5	540.6	810.9
9	10	66.7	110	5	901.0	1351.5



(a) ASME PCC-1 (Star-Pattern)



(b) HPIS Z 103 (Circular-Pattern)

Fig.5 Variations of Bolt Forces during Bolting-up

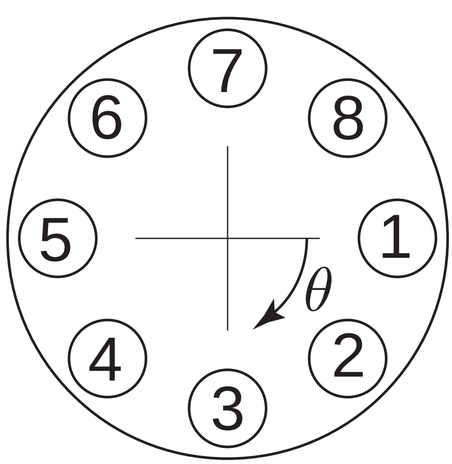


Fig. 6 Numbering of Bolts

achieve a convergence of bolt forces by the HPI method, though the ASME method requires total 7 turns to complete the round 4. Table 2 shows the differences in the scatter of bolt preloads. Two kinds of scatter are shown. One is the ratio of the difference between the target and the minimum to the target bolt force, $(F_f-F_{f\min})/F_f$, the other is the ratio of the difference between the maximum and the minimum to the target bolt force, $(F_{f\max}-F_{f\min})/F_f$. The data show that the difference in the scatter of bolt forces by the two methods is small, on the whole.

GASKET STRESS DISTRIBUTION

Figure 7 shows averaged gasket contact stress distributions in the circumferential direction obtained by FEA simulating an ideal uniform tightening and the PCC-1 torque control tightening in the Case No.9 of the loading conditions, respectively. The gasket stress by the torque control tightening is less than that by the ideal uniform tightening. But the difference becomes smaller in the pressurized state than the initial clamping state.

Table 2 Differences in the scatter of the bolt preloads

Case No.	ASME PCC-1		HPIS Z 103	
	$\frac{F_f - F_{f\min}}{F_f}$ [%]	$\frac{F_{f\max} - F_{f\min}}{F_{f\min}}$ [%]	$\frac{F_f - F_{f\min}}{F_f}$ [%]	$\frac{F_{f\max} - F_{f\min}}{F_{f\min}}$ [%]
1	31.6	86.6	36.2	25.7
2	14.2	27.0	26.8	16.8
3	45.2	39.3	36.8	22.1
4	33.0	13.9	31.7	33.6
5	24.8	18.4	28.1	39.2
6	18.9	22.7	24.1	11.7
7	9.4	21.3	24.0	36.7
8	12.4	25.1	19.5	20.8
9	3.9	20.3	19.4	23.3

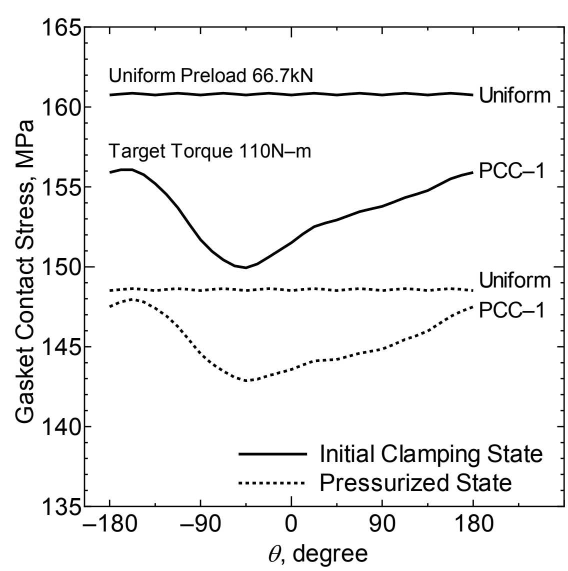
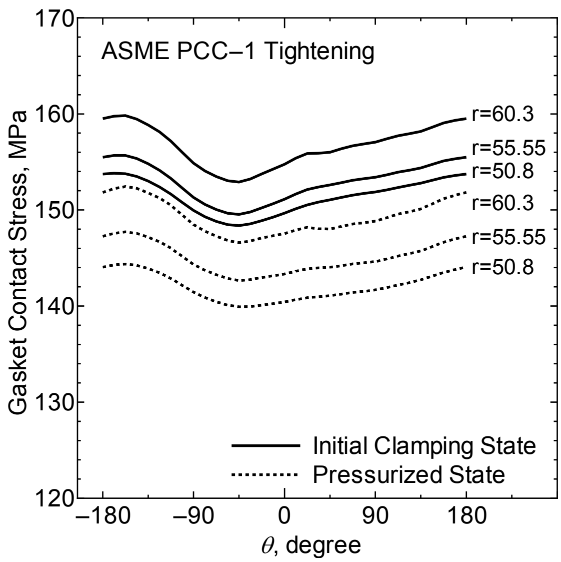
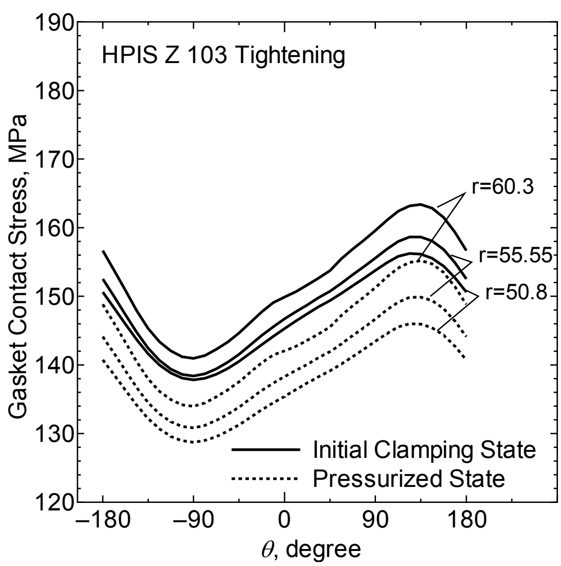


Fig.7 Comparison of Average Gasket Contact Stress Distributions in Circumferential Direction between Uniform Bolt Preload and Torque Control Tightening

Figures 8(a) and 8(b) also plot the gasket contact stress distributions in the circumferential direction along the inner perimeter (r=50.8mm), the center of width (r=55.55mm) and the outer perimeter (r=60.3mm) of the gasket, obtained by FEA simulating the ASME tightening and the HPIS tightening in the Case No.9 of the loading conditions. The gasket stress varies in the circumferential direction as well as the radial direction because of the scatter of bolt forces. The variation in the radial direction becomes wider in the pressurized condition than in the initial clamping condition, for both tightening methods. The variation in the circumferential direction, however, does not look to change significantly. In this case, the range of scatter in the bolt preload is smaller in the ASME tightening than the HPIS tightening as shown in Figs.5(a) and 5(b). The variation of gasket stress in the circumferential direction is slightly smaller in the ASME tightening than in the HPIS tightening.



(a) According to ASME PCC-1



(b) According to HPIS Z 103

Fig.8 Gasket Contact Stress Distributions in Bolted Connections Tightened by Torque Control Method

Considering the results for all the cases, it is found that there is no significant difference in the range of variation of gasket contact stress between two tightening procedures.

LEAK TEST RESULT

Figure 9 shows the comparison of tightness parameters in case of tightness class T_c=10, obtained by the experimental leak tests, estimated by FEA and the design figure in PVRC method. The thin solid line indicates the design figure in T_c=10 determined by PVRC method and should be achieved. Actual tightness parameters measured by the experimental tests almost comply with the design figure. FEA results seem to give somehow unconservative estimations. The way of estimation using FEA results is as follows. The gasket is divided into n segments in the circumferential direction and into m segments in the radial direction. For each circumferential segment, the average gasket stress S_n along the radial direction is calculated by Eq.(2). By using S_n, each segment tightness parameter ΔT_{pn} is obtained by Eq.(3). Then, the estimated tightness parameter is the sum of those segment tightness parameters, as Eq.(4). Here, the present analysis adopts m=12, n=32. $\Delta r=(r_{go}-r_{gi})/m$. S_{mn} is the averaged gasket stress at the segment of m and n. r_{go} and r_{gi} denote outer and inner radius of gasket, respectively. T_{pa} is an assembly tightness parameter. G_b, G_s and a are new gasket constants obtained by ROTT.

$$S_n = \frac{\sum_{m=1}^{12}\left[2r_{gi} + 2\left(2m-1\right)\Delta r\right]\Delta r \cdot S_{mn}}{r_{go}^2 - r_{gi}^2} \tag{2}$$

$$\Delta T_{pn} = \frac{1}{32}\left[\frac{S_n}{G_s}\right]^{\frac{1}{b}}, \quad b = \frac{\log\left[\left(G_b/G_s\right)T_{pa}^a\right]}{\log T_{pa}} \tag{3}$$

$$T_p = \sum_{n=1}^{32}\Delta T_{pn} \tag{4}$$

Figure 10 shows the actual assembly efficiency based on leakage obtained by the tests. The actual assembly efficiency η_{act} is calculated by the following equation.

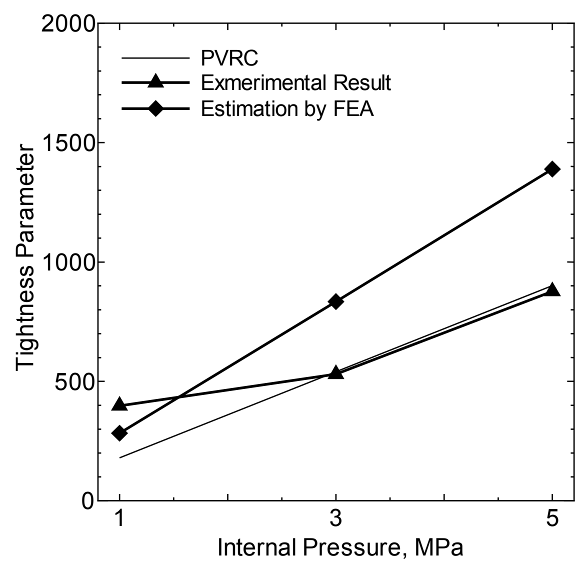


Fig.9 Comparison of Tightness Parameter

$$\eta_{act} = \frac{S_a^{ex}}{S_{ya}} = \left(\frac{T_{pa}^{ex}}{T_{pya}}\right) = \eta\left(\frac{T_p^{ex}}{T_{p\min}}\right)^a \tag{5}$$

$$S_{ya} = \frac{G_b}{\eta}\left(T_{pa}\right)^a, \quad T_{pa} = 1.5T_{p\min} \tag{6}$$

$$S_a^{ex} = G_b\left(T_{pa}^{ex}\right)^a, \quad T_{pa}^{ex} = 1.5T_p^{ex} \tag{7}$$

Here, T_p^{ex} is the measured tightness parameter by the test. $T_{p\min}$ is the required tightness parameter which should be achieved in operating condition. Subscript a means bolted up condition. Super script ex indicates based on actual conditions. The PVRC method assumes constant assembly efficiency independent of the tightness class. However, as shown in Fig.10, it is not a constant value that the actual assembly efficiency considering the achievement of tightness parameter to the one required by design. Though η=0.85 is appropriate up to T_c=10, it seems to be insufficient in the condition that the higher leak tightness is required. Therefore, the assembly efficiency should be determined by taking into account the achievement of tightness parameter to the required one, not by the achievement of bolt force to the target force.

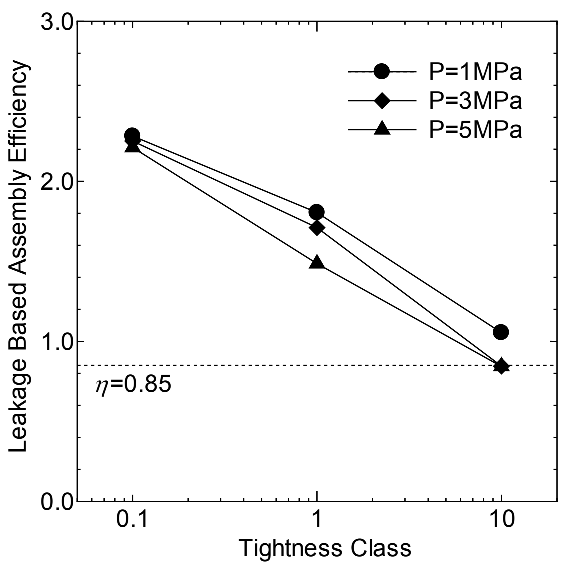


Fig.10 Actual Assembly Efficiency based on Leakage accounting Scatter of Bolt Forces

CONCLUSION

The following conclusions are obtained from the present study.

1) By using the 3B Class600 WN-RF flange connection with CPV, the scatter of bolt preload and the variation of bolt forces during the bolting up with a torque wrench is examined usung two different tightening procedures, ASME PCC-1 and HPIS Z 103. Though there is no significant difference in the final scatter of the bolt preload by two procedures, HPIS method promptly gives a convergence of bolt forces compared to ASME method.
2) Leakage tests are carried out using the bolted flange connection with CPV tightened up by a torque wrench with a target torque given by PVRC design method considering the assembly efficiency. Tightness parameters are obtained from the measured leak rates and compared to the required tightness parameter. It is found that the measured tightness parameters almost comply with the required ones.
3) Finite element analysis is performed to obtain the actual gasket contact stress simulating the tightening process. The calculated gasket stress varies in the circumferential direction as well as the radial direction due to the scatter of bolt preload.
4) Actual assembly efficiency based on leakage is calculated using the leak test results. Though the assembly efficiency adopted in PVRC design is constant, the actual assembly efficiency is not constant but depending on the tightness class. The assembly efficiency should be determined by considering the actual leak tightness in the bolted connections assembled with scattered bolt preloads.

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