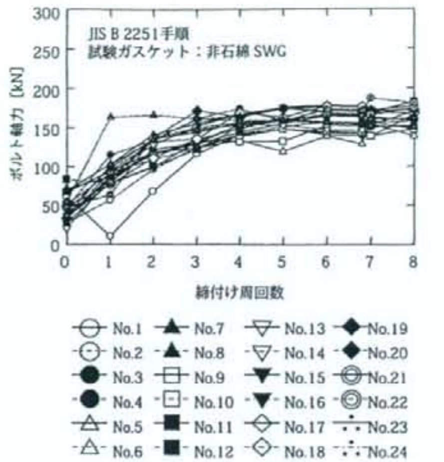

図3 JIS フランジ締付け方法による 500A フランジの締付け試験結果（うず巻形ガスケット）

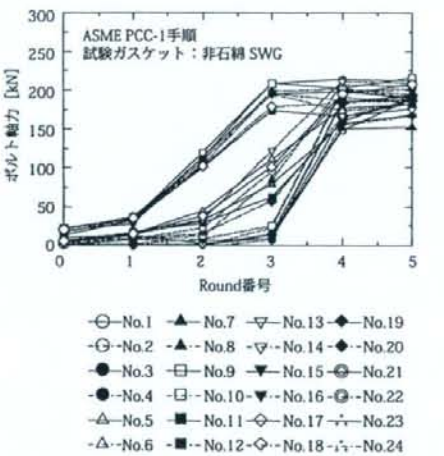

図4 ASME 締付け手順による 500A フランジの締付け試験結果（うず巻形ガスケット）

締付け手順の場合はパス数（締付け周回数）とし、ASME締付け手順の場合は複数のパスが含まれているRound数で表す。ただし、両手順の場合とも横軸の数値０はインストール完了段階を表す。ここで、締付け完了時の最終的なボルト軸力の平均値を比較するとJIS締付け手順による場合が低いが、締付けトルクの10%増加による補償をこの試験では加えていないためである。締付けトルクの補償分を考慮すれば、両手順による締付け完了時のボルト軸力の平均値は同等のレベルといえる。

多数本ボルト締結体であるフランジ継手の締付けにおいて、隣接するボルトが締め付けられるとボルト軸力が低下する弾性相互作用が発生する。対角締付けでは、この作用によりボルト総本数に依存してボルト軸力のレベルが２または３グループに分かれ、締付け周回を重ねても均一な締付け力を与えることは困難である。ASME締付け手順のボルト軸力変化を示している図２および図４が典型例であるが、Round 2およびRound 3において対角締付けによる弾性相互作用が顕著に現れている。ボルト総本数8本の100Aフランジではボルト軸力のレベルが2グループに分かれ、ボルト総本数24本の500Aフランジではボルト軸力のレベルが３グループに分かれる。一方、JIS締付け手順の場合は弾性相互作用によるボルト軸力のレベルのグループ化という現象が見られない。これはJIS締付け手順が採用している時計回り締付けの優位性の理由の一つである。厳密に言えば、時計回り締付けでも弾性相互作用が発生しているが、ボルト軸力のグループ化という現象にはつながらない。

また、JIS締付け手順では本締付けの最初から目標締付けトルクの100%による時計回り締付けを行っているが、即座に目標締付け力の100%という高いボルト軸力が発生することはなく、ボルト軸力は締付け周回数と共に漸増する。従って、ASME締付け手順のように締付けトルクを段階的に上昇させるRoundを設ける必要は全くない。

図５および図６は500A（20B）フランジを JIS締付け手順およびASME締付け手順に従い締め付けた場合のフランジ変位の変化をそれぞれ示す。フランジ内径側の変位はガスケット圧縮量をよく反映すると考えられるが、その均一性において両手順は同等、もしくはJIS締付け手順の方が良好である。JIS締付け手順が採用した大幅な締付け手順の簡略化（時計回り締付け、および目標締付けトルクの100%での締付け）により懸念されるフランジの片締めは発生しないことが確認された。

表５は、各条件における締付け完了時のボルト軸力のばらつきを比較して示す。JIS締付け手順の試験結果をASME締付け手順の場合と比較すると、フランジ呼び径、ガスケット種類によらず、ボルト軸

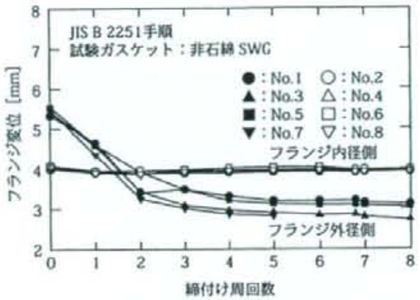

図5 JIS フランジ締付け方法による500A フランジのフランジ隙間の変化（うず巻形ガスケット）

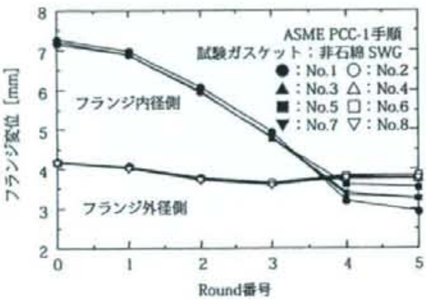

図6 ASME 締付け手順による500A フランジのフランジ隙間の変化（うず巻形ガスケット）

表5 ボルト軸力の均一性と締付け周回数（総パス数）に関する JIS 締付け手順と ASME 締付け手順の比較

		うず巻形ガスケット		ジョイントシートガスケット	
		締付け周回数[注]	軸力ばらつき	締付け周回数[注]	軸力ばらつき
100A フランジ	JIS B 2251	7	±10%	6	±13%
	ASME PCC-1	16	± 6%	9	±12%
500A フランジ	JIS B 2251	9	±19%	8	±20%
	ASME PCC-1	11	±20%	11	±19%

注）仮締付け（インストール）は、1回と数える

力に関して同等の均一性が得られており、同時に、ボルト軸力の目標軸力への収束は速い。表5に各条件における総締付け周回数の比較を併せて示す。ASME締付け手順は前述のように手順が複雑で多大な労力を要する。JIS締付け手順は締付け手順の簡略化・締付け周回数の短縮化が図られ、数周回以上の削減がなされている。さらにASME締付け手順では締付け工具の対角移動があることを考えると、JIS締付け手順は作業性において大きな優位性があることは明らかである。

7．今後の展望

7－1 ISO規格の動向との関連

金属製管フランジに関する国際規格ISO 7005の各パート（Part1～Part3）の改正作業が進められており、JISの管フランジ規格を追加させ国際市場の現状と整合させるための提案が進行中である。一方、ISO 7005の見直しにおいて、新たにフランジ継手締付け方法を規定に追加しようという動きがあり、これに対応するためにISO規格で引用できるJIS規格が必要となってきた。外国規格では、ASMEがフランジ継手締付け指針としてASME PCC-1を2000年に発行しており、この動きに対応できると見られる。ただし、経験則に基づき制定されたASME PCC-1は、日本の技術水準に適合しない。逆にJISフランジ締付け方法はASME関係者を始め欧米から注目されている。現実に、米国でもJISフランジ継手締付け方法に注目する技術者も多く、ASME PCC-1に従来の対角締付け法と並んでJISの一方向締付け法を導入しようとする動きもあるようである。そこでHPIS規格として制定されていたフランジ継手締付け指針をJISとして制定し、将来のISO化に備えることとした。

7－2 技術的課題

ガスケットは、ジョイントシートガスケット及びうず巻形ガスケットを対象としたが、適用範囲拡張の要望が強い。リングジョイントガスケットに適用できると実用性がいっそう高まると考えられる。延伸PTFEシートガスケットについては、JISフランジ継手締付け方法が適用できることが締付け試験により確認されている[8)]。延伸PTFEガスケットは圧縮率と応力緩和率が高いことによるガスケット片締めへの影響が懸念されたため適用されなかった経緯がある。

フランジの材質に対しても、バルブの材料として多く用いられている鋳鉄、銅合金など、適用範囲拡張の要望があった。これらの適用範囲については、科学的根拠が与えられれば順次拡張することとした。

8．おわりに

科学的な根拠に基づき開発されたJIS B 2251：2008「フランジ継手締付け方法」を紹介した。本JISはフランジ継手の組立てにおいて、従来方法に比べて、より正確な締め付け力が得られ、さらにより効率的な作業性を両立させた実用的な締付け手順となっている。最近のガスケット非石綿化への対応という視点からも、JISフランジ継手締付け方法の重要性は一層増している。圧力設備の安全性と信頼性向上のため、JISフランジ継手締付け方法が広く活用されることを期待している。

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July 27-31, 2008, Chicago, Illinois, USA

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APPLICATION OF BOLTED FLANGE JOINT ASSEMBLY GUIDELINES HPIS Z103 TR TO EPTFE SHEET GASKET

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ABSTRACT

Bolt tightening guidelines HPIS Z 103 TR for flange joint assemblies have been developed to provide a simple and effective procedure to tighten flange joint bolts. This assembly procedure is applicable to compressed fiber sheet gaskets and spiral wound gaskets, but is not applicable to expanded PTFE (ePTFE) sheet gaskets for the reason that the ePTFE material has lower modulus of elasticity and higher creep/relaxation rate. In this study, expansion of the application scope of HPIS Z103 TR to ePTFE sheet gaskets is investigated. Tightening tests are conducted using flange joint specimens of JPI 4 inch and 6 inch, and all bolt forces and flange gaps are measured at each tightening step to check for uneven tightening. Uniformity of the bolt forces and flange gaps are comparable to those obtained by other types of gaskets or by tightening procedure ASME PCC-1. The influences of gasket relaxation and elastic interaction on the bolt forces are also demonstrated. As a result, flange joint assembly guidelines HPIS Z 103 TR can be considered applicable to the high-density ePTFE sheet gasket, although a post-tightening step of 1 or 2 passes is necessary to compensate for the bolt force reduction induced by gasket relaxation.

INTRODUCTION

In the assembly of a bolted flange joint with a gasket, uniform bolt force and precise flange alignment are important to achieve uniform distributions of gasket stress as well as leak-free service of the joint. Bolt tightening guidelines HPIS Z 103 TR for the flange joint assemblies have been scientifically developed to provide a simple and effective procedure to tighten flange joint bolts [1]. In the HPIS procedure, bolts are tightened in a clockwise sequence and the tightening torque is 100% of the target torque in all passes or steps subsequent to the install step. The HPIS procedure also meets the requirement of practical assembly to reduce the assembly cost and avoid human error. With respect to the uniformity of the bolt forces and the flange alignment, it has been shown that HPIS is comparable to ASME PCC-1 [2], whereas the total number of passes/steps and time required can be reduced when using the HPIS procedure [3]. Thus, the proposal of JIS standardization (Japanese Industrial Standard, JIS B 2251) for the HPIS guidelines is now being discussed.

The HPIS procedure is applicable to compressed fiber sheet gaskets and spiral wound gaskets, but has not been considered applicable to expanded PTFE (ePTFE) sheet gaskets for the reason that the ePTFE material has a higher compressibility and creep/relaxation rate.

In this study, expansion of the application scope of HPIS Z103 TR to ePTFE sheet gaskets is investigated experimentally. To account for the higher creep/relaxation rate of the ePTFE gasket material, post-tightening characteristics are examined by a flange tightening test. As a result, an improved rule for the post-tightening step is proposed, and good uniformity of the bolt forces and flange alignment are achieved.

OVERVIEW OF HPIS Z103 TR GUIDELINES

Table 1 shows the tightening procedure given by the HPIS Z103 TR guidelines. The bolts are tightened to 100% of the target torque in all steps to decrease the total number of passes and to set an upper limit of tightening operation iterations. In the install step, 4 or 8 bolts are tightened with the aim of snugging and tentative tightening on a cross-pattern sequence to prevent flange misalignment. In the tightening step and post-tightening step, tightening in a clockwise sequence is employed with the aim of a simple procedure to avoid human error. The tightening step requires 4 or 6 passes with a tightening torque that is 100% of the target torque. Post-tightening may be conducted to compensate for the reduction in the bolt force resulting from the stress relaxation of the

Table 1 Tightening method according to HPIS Z 103 TR.

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% 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 4 hours and tighten by the previous step , but 1 or 2 passes.

*Target torque: Target value is increased by 10%.

gasket. Aging for a minimum of 4 hours is specified before the post-tightening step. Based on finite element analysis (FEA) simulations of the flange tightening process, the criteria for setting the upper limits of the tightening iterations is 85% of the target bolt force [4]. This might cause insufficient bolt force, so the target value of the tightening torque is increased by 10% in the HPIS guidelines. The HPIS guidelines are applicable to compressed fiber sheet gaskets and spiral wound gaskets, but are not applicable to other types of soft gaskets, metallic gaskets, or semi-metallic gaskets.

Figure 1 shows an example of a flange joint assembly test according to the HPIS guidelines. The test flange was an NPS 20, class 300 lb, raised face slip-on welding type flange specified in JPI (material: A105). The test bolts (studs) were 1¼–8 UNC with nuts (bolt material: A193 Gr. B7, nut material: A194 Gr. 2H). The test gasket was a compressed asbestos fiber sheet gasket (thickness 3mm). The lubricant used in the test was a dry coating type spray of MoS_2 (molybdenum disulfide). The effects of the elastic interaction were reduced by the clockwise sequence tightening so that the scatter range

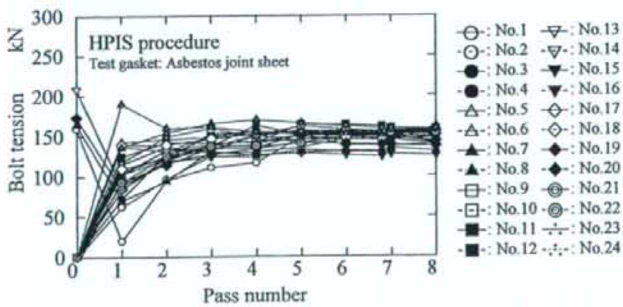

Fig. 1 Variations of bolt force of 20-inch flange tightened by HPIS Z103 TR procedure.

Table 2 Tightening method according to PCC-1.

Step	Loading
Install	Hand tighten, then "snug up" to 15 Nm to 30 N-m (not to exceed 20% of Target Torque) on a cross-pattern tightening sequence. Check flange gap around circumference for uniformity.
Round 1	Tighten to 20% to 30% of Target Torque on a cross-pattern tightening sequence.
Round 2	Tighten to 50% to 70% of Target Torque on a cross-pattern tightening sequence.
Round 3	Tighten to 100% of Target Torque on a cross-pattern tightening sequence.
Round 4	Continue tightening the bolts, but on a rotational clockwise pattern until no further nut rotation occurs at the Round 3 Target Torque value.
Round 5	Time permitting, wait a minimum of 4 hr and repeat Round 4; this will restore the short-term creep relaxation/ embedment losses.

Table 3 Total number of passes and scatter of bolt force.

		Spiral wound gasket		Compressed fiber sheet gasket	
		Total passes	Scatter of bolt force	Total passes	Scatter of bolt force
NPS 4 in.	HPIS	7	±10 %	6	±13 %
	PCC-1	16	± 6 %	9	±12 %
NPS 6 in.	HPIS	9	±19 %	8	±20 %
	PCC-1	11	±20 %	11	±19 %

*Install step corresponds to 1 pass.

of the bolt force was small; therefore, the bolt force became uniform with a small number of tightening passes. The final scatter range of the bolt force was ±20%. The total number of passes was 8, including the two passes of post- tightening; however, the post-tightening step was not needed in this case of compressed fiber sheet gaskets.

Table 2 shows the tightening procedure given by ASME PCC-1-2000 [2]. Bolts are tightened by the cross-pattern sequence and the tightening torque increases for several steps (Round 1 to Round 3). In Round 4 and Round 5, tightening in the clockwise sequence is applied, and the tightening operations are repeated until no further nut rotation occurs. Aging for a minimum of four hours is needed between Round 4 and Round 5, which compensates for the decrease of bolt tension by stress relaxation of the gasket. The above mentioned procedure is needed to prevent flange misalignment and to achieve uniform bolt preloads. However, this tightening method has

Table 4 Properties of gasket materials.

Gasket material	Specific gravity	Compressibility (JIS R 3453)	Percentage of creep relaxation (ASTM F38-00)
High-density ePTFE sheet (t=1.5 mm)	1.7	23 %	66 %
Low-density ePTFE sheet	0.62	56.8 %	35 %
Compressed fiber sheet (t=1.5 mm)	1.98	9 % (34.3 MPa)	31.0 %

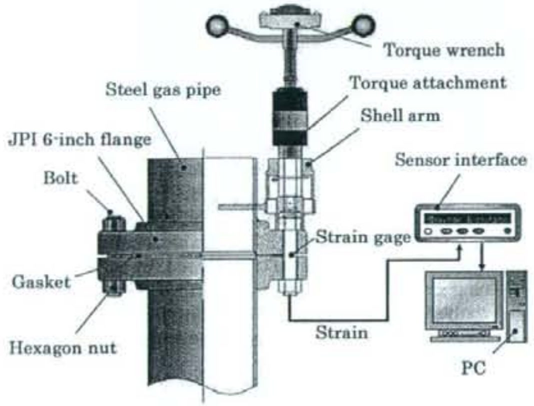

Fig. 2 Test equipment of 6-inch flange joint.

several disadvantages, such as the increase in tightening hours, cost, and the possibility of human error.

Table 3 lists the flange tightening test results for the total number of passes and the scatter of the bolt force, and thus provides a comparison of the HPIS and PCC-1 (PCC-1-2000) procedures. The HPIS procedure achieves comparable uniformity in the bolt force; in addition, the total number of passes required is significantly reduced.

FLANGE TIGHTENING TEST WITH EPTFE SHEET GASKET

Test Gasket

The test gaskets are high-density expanded PTFE (ePTFE) sheet gaskets (i.e., a high-density version of ePTFE at approximately 1.7 g/cc). Table 4 shows the material properties for the high-density ePTFE sheet, low-density ePTFE sheet, and compressed fiber sheet gasket. The test

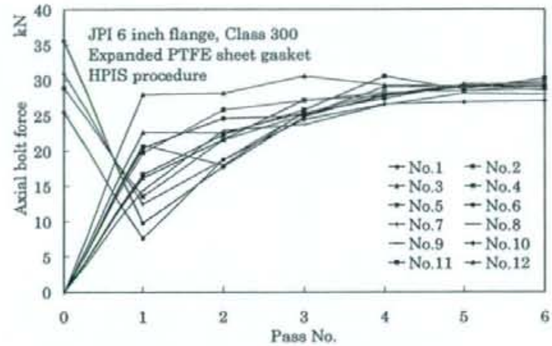

Fig. 3 HPIS Z103 TR result of tightening test for 6-inch flange with ePTFE sheet gasket (aging time after tightening step: 4 hours).

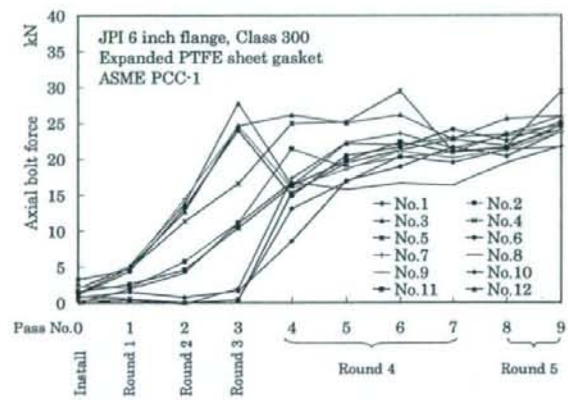

Fig. 4 PCC-1 result of tightening test for 6-inch flange with ePTFE sheet gasket.

gaskets dimensions are a thickness of 3 mm and either NPS 4 or NPS 6, both of which are ASME/ANSI Class 300 Group I. Gasket constants used in the flange calculations are m = 2.5 and y =19.6 MPa.

Test Flange Joint Model

Test flange joint specimens of NPS 4 and NPS 6 were prepared. Figure 2 shows the setup of the tightening test for a 6-inch bolted flange joint with gasket. Test flanges are NPS 4 or NPS 6, class 300 lb (material: A105), raised face slip-on welding type flanges specified in JPI (Japan Petroleum Industry). Test bolts (studs) have a nominal diameter of M20 with nuts (bolt material: A193 Gr. B7, nut material: A194 Gr. 2H).

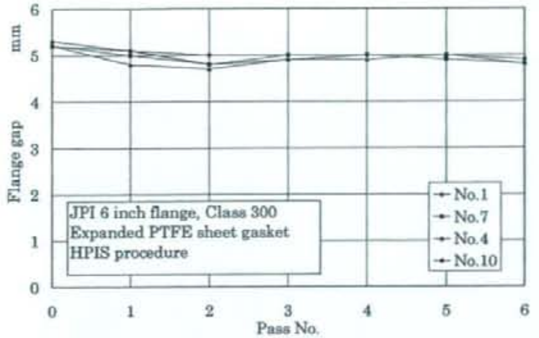

Fig. 5 Flange gap obtained by tightening test of 6-inch flange with ePTFE sheet gasket.

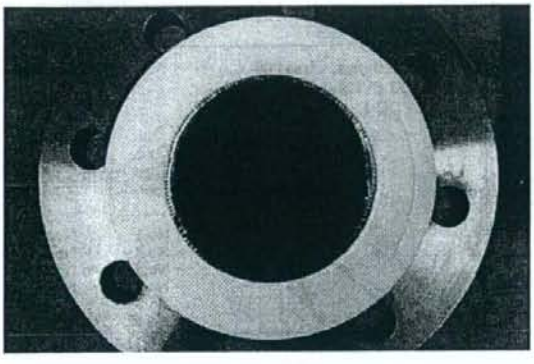

Fig. 6 Expanded PTFE sheet gasket of NPS 4 after tightening test according to HPIS procedure.

The bolt force is controlled by a torque control method in which the tightening torque is controlled by a manual torque wrench. The axial bolt force of each bolt is measured using a strain gauge embedded in a drilled hole of the bolt. The measurement data are recorded by a PC through a digital multimeter. The flange displacement is measured by a vernier caliper as the flange gaps at 4 points of the outer perimeter of the flange. To clarify the influence of gasket relaxation, the bolt force of all bolts is measured every 5 min for 4 hours after the tightening step and every 5 min for 4 hours after the post-tightening step.

Test Conditions

The target bolt forces are 33.2 kN for the 4-inch flange and 35.5 kN for the 6-inch flange; these forces were determined based on the gasket stress 30 MPa. The lubricant used in the test is a dry coating type spray of MoS_2.

Table 5 Total number of passes and scatter of bolt force for ePTFE sheet gasket.

		ePTFE sheet gasket	
		Total passes	Scatter of bolt preload
NPS 4 in.	HPIS	7	±3 ～ ±18 %
	PCC-1	13	± 3 %
NPS 6 in.	HPIS	7	±5 ～± 10 %
	PCC-1	10	±15 %

*Install step corresponds to 1 pass.

APPLICABILITY OF HPIS GUIDELINES TO EPTFE GASKET

Figure 3 shows the HPIS Z 103 result of the tightening test of the 6-inch flange. Pass 0 is the "Install step," passes 1 – 4 are the "Tightening step," and passes 5 and 6 are the "Post-tightening step." In pass 1, there is relatively large dispersion in the bolt force caused by the elastic interaction between adjacent bolts; this result is generally observed under the cross-pattern tightening sequence. The influence of the elastic interaction on the bolt force is reduced by the clockwise tightening sequence; therefore, the bolt force becomes uniform in a relatively small number of tightening passes. After pass 6, the average bolt force compared to the target is 92%, and the dispersion of the bolt force is ±5%.

Figure 4 shows the PCC-1 results of the tightening tests of the 6-inch flange by the tightening methods. In the PCC-1 method, the 12 bolts divide into 3 groups of bolt forces from Round 1 to Round 3. This phenomenon is caused by the elastic interaction between adjacent bolts due to the cross-pattern tightening sequence. In the HPIS method, since the effects of the elastic interaction on the bolt force is small, a uniform bolt force may be achieved in a reduced number of tightening passes in both the 6-inch flange with 12 bolts as well as the 4-inch flange with 8 bolts.

Figure 5 shows the relationship between the number of tightening passes and the flange displacement in the 6-inch flange tightening test. The flange displacement is distributed in a significantly small range, such as 0.2 mm after pass 4 and 0.1 mm after pass 6. Generally, with uneven tightening of the flange in the clockwise tightening sequence, there is a concern that the compression strain of the gasket becomes large around the starting point of the bolt tightening sequence; this phenomenon reflects the non-uniformity of the flange displacement. However, uneven tightening is not observed here. In addition, the ePTFE sheet gasket shown in Fig. 6 indicates neither uneven tightening nor mechanical failures, such as wrinkles, flaws, or extrusions.

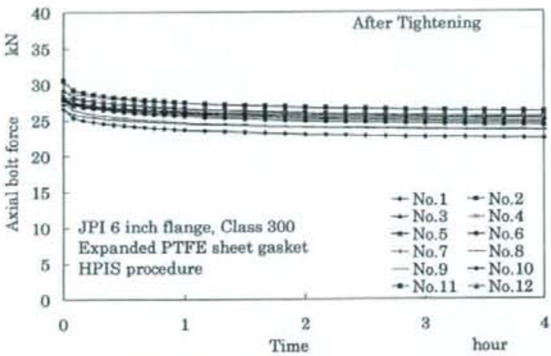

Fig. 7 Relaxation of axial bolt force after tightening step of 6-inch flange with ePTFE sheet gasket (aging time after tightening step: 4 hours).

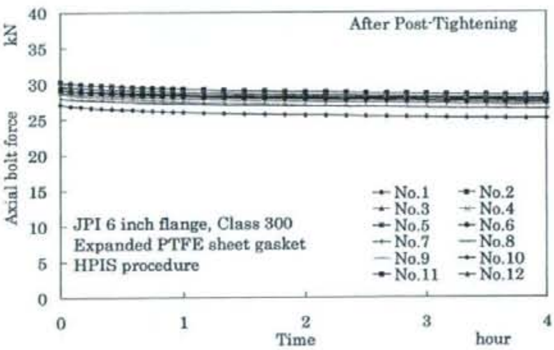

Fig. 8 Relaxation of axial bolt force after post-tightening step of 6-inch flange with ePTFE sheet gasket (aging time after tightening step: 4 hours).

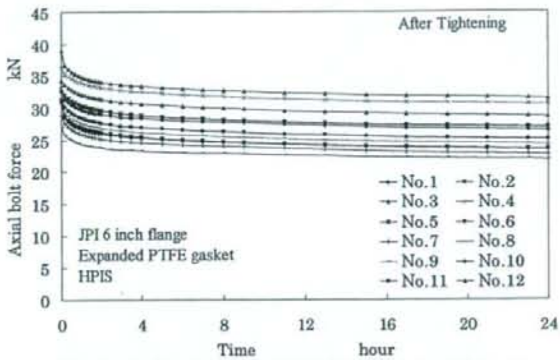

Fig. 9 Relaxation of axial bolt force after Tightening step of 6-inch flange with ePTFE sheet gasket (aging time after tightening step: 24 hours).

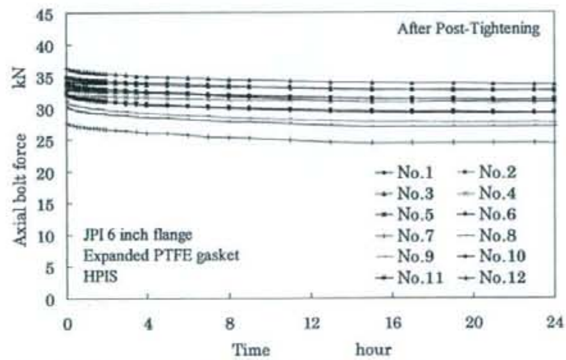

Fig. 10 Relaxation of axial bolt force after post-tightening step of 6-inch flange with ePTFE sheet gasket (aging time after tightening step: 24 hours).

Table 5 shows the total number of passes and the scatter of the bolt force obtained by the flange tightening test with the ePTFE sheet gasket. The total number of passes, including the install step and the post-tightening step, and the scatter of the bolt force, are measured after the post-tightening step. In comparison with the PCC-1 procedure, the HPIS procedure achieves comparable uniformity in the bolt force, and the total passes required are significantly reduced. These characteristics are the same for other types of gaskets, such as the compressed fiber sheet and the spiral wound gasket.

The results with the two sizes of flanges demonstrate that an accurate and uniform bolt force is achieved and uneven tightening does not occur; therefore, the HPIS guidelines can be considered applicable to the ePTFE sheet gasket.

CONSIDERATION OF POST-TIGHTENING

Time-dependent Behavior of the Bolt Force

The HPIS procedure specifies a post-tightening step of 1 or 2 passes to compensate for the reduction in the bolt force resulting from stress relaxation of the gasket. The aging time between the tightening step and the post-tightening step is 4 hours or more.

Figure 7 shows the relaxation behavior of the bolt force during the 4 hours after the tightening step, and Fig. 8 shows the relaxation behavior during the 4 hours after the post-tightening step. The reduction of bolt force over 4 hours is 12% after the tightening step (Fig. 7), and 7% after the post-tightening step (Fig. 8). By applying the post-tightening step, the influence of gasket relaxation on the bolt force is decreased.

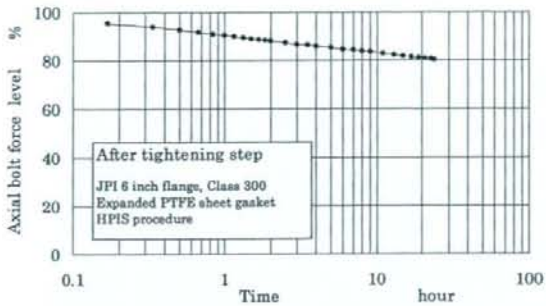

Fig. 11 Relaxation of axial bolt force after Tightening step of 6-inch flange with ePTFE gasket (aging time after tightening step: 24 hours).

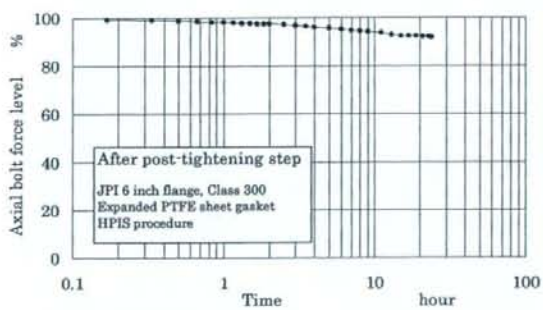

Fig. 12 Relaxation of axial bolt force after post-tightening step of 6-inch flange with ePTFE gasket (aging time after tightening step: 24 hours).

The post-tightening step is necessary to apply the HPIS guidelines to the ePTFE sheet gasket due to its higher creep/relaxation rate. The number of passes needed for post-tightening is 1, because the bolt force of pass 6 is equivalent to that of pass 7, as shown in Figs. 4 and 5.

Influence of Aging Time before Post-tightening Step on Relaxation Behavior

By increasing the aging time from 4 hours to 24 hours, the influence of aging time on relaxation behavior of the bolt force is examined. Figure 9 shows the time-dependent behavior of the bolt force over 24 hours after the tightening step, and Fig. 10 shows the behavior after the post-tightening step. These figures represent the HPIS procedure. The Scatter of the bolt force shown in Figs. 9 and 10 are greater than that shown in Figs. of 7 and 8 obtained by the same tightening conditions; however, this difference is considered a normal variation resulting from the variation of friction coefficients of contact surfaces of the bolt and nut.

The relaxation rate of the bolt force is relatively high over 4 hours after the tightening step, as shown in Fig. 9. After 12 hours, the relaxation rate is significantly lowered so that the bolt force reduction is negligible. The reduction of the bolt force over 24 hours is 20% after the tightening step (Fig. 9), and 8% after the post-tightening step (Fig. 10). By increasing the aging time to 24 hours, the influence of gasket relaxation on the bolt force is reduced. Consequently, it is recommended that post-tightening is conducted after a minimum aging time of 12 hours.

Figure 11 shows the semi-log plot for the time-dependent behavior of the average bolt force over 24 hours after the tightening step, and Fig. 12 shows the behavior after the post-tightening step. Since the semi-log plot yields straight lines, the bolt force reduction is expressed by an exponential function. The straight line after the post-tightening step has a smaller slope than that after the tightening step; therefore, the post-tightening step is effective against gasket relaxation and the subsequent reduction of bolt force.

ESTIMATION OF RELAXATION BEHAVIOR OF BOLT FORCE BY VISCOELASTICITY MODEL OF THE FLANGE JOINT

A viscoelasticity model is applied to the flange joint consisting of the gasket, bolts and flanges. The three-parameter solid [5] shown in Fig. 13 is a simplified model for describing the time-dependent behavior of the flange joint, such as the reduction of axial bolt force after the tightening step or after the post-tightening step. Spring K_1 and damper C_1 represent the viscoelastic behavior of the gasket, and spring K_2 represents the elastic behavior of the flanges, bolts, and gasket. Parameters K_1, C_1 and K_2 are determined as $K_1 = 4.07$ kN/L and $C_1 = 2.04$ kN·hr/L for $K_2 = 1.0$kN/L, by fitting the experimental data for the axial bolt force reduction after the tightening step (0–24 hours) shown in Fig. 14. The axial bolt force reduction curve is formulated as follows:

$$F = 26.0 + 6.4\,e^{-0.49\,t} \quad \text{[kN]} \qquad (1)$$

where F (kN) is the axial bolt force and t (hour) is the elapsed time. Figure 14 shows the bolt force reduction curve over 24 hours after the tightening step. The axial bolt force decreases to 80% of the initial level over 24 hours.

The axial bolt force increases again to the initial level at the post-tightening step at 24 hours. Assuming that the same parameters K_1, C_1 and K_2 are also applicable to the 24 hours after the post-tightening step, the bolt force reduction curve after the post-tightening step is estimated (24–48 hours). The axial bolt force reduction curve is calculated successively as follows:

$$F = 31.1 + 1.3\,e^{-0.49(t-24)} \quad \text{[kN]} \qquad (2)$$

The curve of Eq. (2) is also plotted in Fig. 14. The axial bolt force decreases to 96% of the initial level over 24 hours after the post-tightening step. The axial bolt force reduction and relaxation rate are lowered by applying the post-tightening step, in agreement with the experimental results.

This simple three-parameter solid model satisfactorily describes the time-dependent behavior of the axial bolt force reduction and helps to explain the benefit of the post-tightening step.

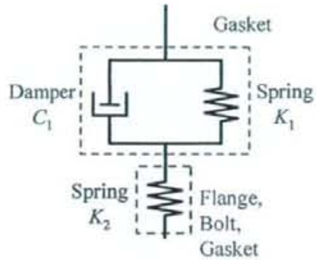

Fig. 13 Three-parameter solid for viscoelasticity behavior of gasketed bolted flange joint.

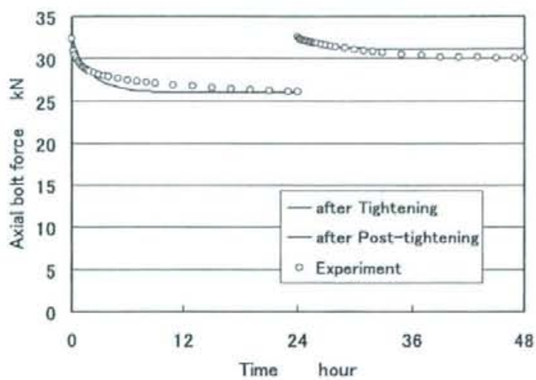

Fig. 14 Estimation of bolt force reduction curve by viscoelasticity model of three-parameter solid.

CONCLUSIONS

Expansion of the application scope of flange joint assembly guidelines HPIS Z103 TR to ePTFE sheet gaskets is investigated to account for the higher creep/relaxation rate of the ePTFE gasket material. The results obtained are summarized as follows:

(1) Practical applicability of the HPIS guidelines to the high-density ePTFE sheet gasket in a flange joint assembly is demonstrated because accurate and uniform bolt force without uneven tightening is achieved.
(2) Guidelines HPIS Z 103 TR are applicable to the high-density ePTFE sheet gasket in a flange joint assembly provided that a post-tightening step of 1 or 2 passes is carried out to compensate for the bolt force reduction induced by gasket relaxation.
(3) It is recommended that the post-tightening is conducted after a minimum of 12 hours.
(4) The bolt force reduction and relaxation rate are lowered by applying the post-tightening step.

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APPLICATION OF PLASTIC REGION TIGHTENING BOLT TO FLANGE JOINT ASSEMBLY

-Downsizing of Flange Joint and Behavior of Bolt Force under Internal Pressure-

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ABSTRACT

The present paper describes the behavior of plastic region tightening of a bolt in a downsized flange joint subjected to internal pressure. An API 4-inch flange joint is downsized for plastic region tightening. The bolt is reduced from M16 to M8, and the bolt pitch circle diameter and the outer diameter of the flange are decreased by 11%. The flange rigidity and the stresses of the compact flange joint are calculated and are superior to the original API flange joint. The bolt spacing is also examined, and the correction factor for bolt spacing is acceptable. Internal pressure is applied to a compact flange joint, and the behavior of additional bolt force is demonstrated. Load factor depends on the type of gasket, such that the load factor is positive for a flexible graphite sheet gasket. The load factor is in agreement with the value calculated by the Load Factor Method (LFM). When the external force is applied to the bolted joint under plastic region tightening, the allowable limit of the additional bolt force is approximately 10% of the bolt yield force. In the present experiment, the additional bolt force is as small as 1% of the bolt yield force. Therefore, the additional bolt force has sufficient margin for the allowable limit.

INTRODUCTION

Flange joints with gaskets are widely used for the joints of piping and pressure vessels in various plants. The pressurized fluids contained in such piping and vessels are often under high temperature and are harmful. A number of studies have been carried out to establish a design method and an assembly procedure for the completely leak-free flange joint [1]-[3]. It is difficult to completely prevent leakage, which may cause an accident.

Plastic region tightening of the bolt is attractive as a practical tightening method in which the target of the tightening force is the yield point or the plastic region of the bolt [4]. Plastic region tightening has the advantages of a higher axial force and reduced scattering of the tightening force. The bolt yield force is determined by the mechanical properties of the bolt material and the effect of the friction coefficient of the threads is small. Plastic region tightening prevents fatigue fracture and relaxation of the joint, which increases the reliability of the joint. Provided that equivalent joint performance is obtained, the diameter of the bolt can be reduced, the number of bolts can be decreased, or the strength class of the bolt can be lowered. Therefore, for critical applications, high performance and reduced cost are achieved.

Plastic region tightening has been applied successfully in various fields. In the automotive industry, plastic region tightening is applied to the cylinder head bolt and the connecting rod bolt in the engine assembly. In the architectural field, plastic region tightening is applied to friction grip bolts.

In a previous study [5], plastic region tightening was applied to an API 4-inch flange joint using a bolt downsized from M16 to M8. An internal pressure was then loaded on the flange joint and an additional axial bolt force was demonstrated. A compressed sheet gasket was used in the flange joint, and the additional axial bolt force decreased with the increase of the internal pressure.

It is possible for both the bolt and the flange joint to be downsized by applying plastic region tightening. In the present paper, a compact flange joint for plastic region tightening is designed and the flange rigidity and flange stresses are confirmed. The load factor between the API flange joint and the compact flange joint is then compared, and the fatigue strength of the bolt is investigated.

NOMENCLATURE

A_r : Contact area of the gasket
A_r : Cross sectional area of the root diameter
C : Bolt circle diameter
CF: Correction factor
d : Bolt diameter
d_2 : Pitch diameter of the bolt
d_s : Diameter of net cross sectional area
E_{yo} : Elastic modulus of the flange joint
F_{bn} : Additional axial bolt force
F_{fy} : Bolt yield force
g_0 : Thickness of the hub at the small end
h_o : Hub length parameter
J : Flange rigidity index
L : Flange stress factor
m : Factor for the gasket operating condition
M_o : Flange design moment for the operating condition
N : Number of bolts
P : Pitch of the bolt
p : Internal pressure
S_H : Flange hub stress
S_R : Flange radial stress
S_T : Flange tangential stress
t : Flange thickness
V : Flange stress factor
W : Thrust force
σ_A : Fatigue limit of the bolt
σ_a : Stress amplitude of the bolt
σ_{ys} : Tensile yield stress
Φ_g : Load factor
μ_s : Friction coefficient of the thread

INVESTIGATION OF DOWNSIZING OF THE BOLT

Figure 1 shows the relation between axial bolt force and bolt elongation in tightening. Beyond the yield point of the bolt, the minimum tightening bolt force obtained by plastic region tightening as evaluated by the bolt yield F_{fy} is expressed as follows [6], [7]:

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

where d_2 is the pitch diameter, d_s is the diameter of the net cross-sectional area, P is the pitch, σ_{ys} is the yield stress, and μ_s is the friction coefficient of the thread. Using the value of σ_{ys} = 993 MPa, as obtained by a uniaxial tension test, in Eq. (1), F_{fy} is obtained. For example, the M8 bolt with σ_{ys} = 993 MPa and μ_s = 0.056, F_{fy} is estimated from Eq. (1) to be 32 kN, and the target axial force requirement is satisfied [5]. It is possible for the flange joint to be downsized in accordance with the downsizing of the bolt. The API 4-inch flange joint, the original bolt of which is M8, is actually downsized to correspond to the M8 bolt. The, compact flange joint is therefore tightened by the plastic region tightening method, and the internal pressure is applied.

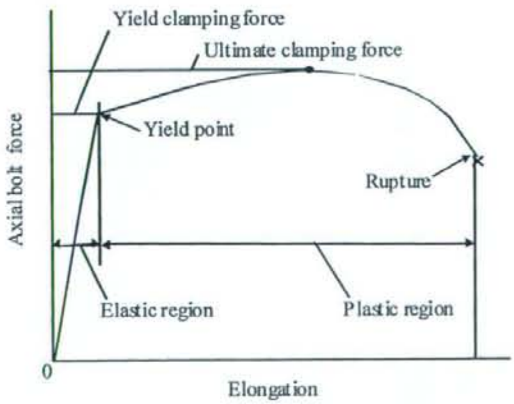
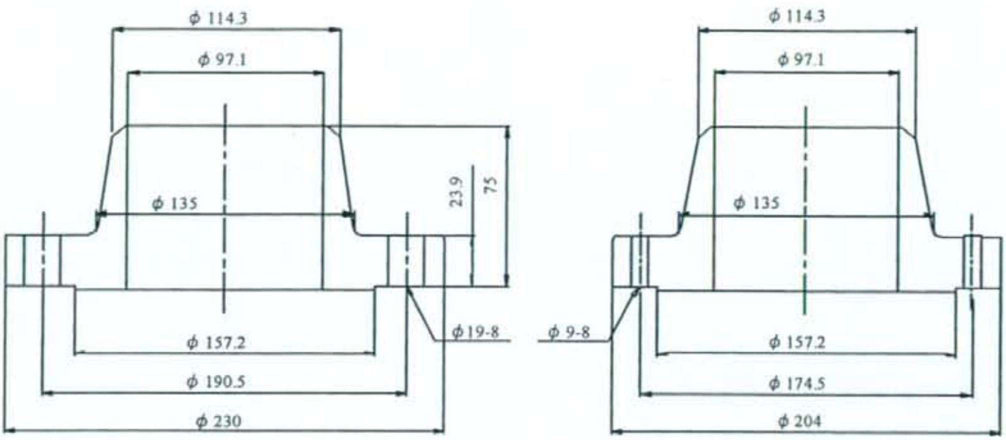


Fig.1 Relation between axial bolt force and bolt elongation in tightening.



(a) API 4-inch flange.

(b) Compact flange joint.

Fig.2 Geometry and dimensions of flange joints.

DESIGN AND FABRICATION OF THE COMPACT FLANGE JOINT

Geometry and Dimensions

Figure 2 shows the geometry and dimensions of the flange joint. Figure 2(a) shows the API 4-inch flange joint, and Fig. 2(b) shows the compact flange joint. The bolthole diameter is downsized from ϕ 19 to ϕ 9 for the M8 bolt. The bolt pitch circle diameter of the bolt and the outer diameter of the flange joint are downsized as much as possible. It is possible to reduce the flange rotation due to the downsizing of the bolt pitch diameter. The flange thickness is not modified in order to maintain the flange rigidity.

Comparison of Flange Rigidity

The flange rigidity is confirmed according to ASME Section VIII-Div.1 App.2 [8], flange rigidity index J, which is expressed as follows:

$$J = \frac{52.14\ VM_O}{LE_{yo}g_0^{\ 2}h_O} \leq 1.0 \qquad (2)$$

where V and L are the flange stress factors, M_O is the flange design moment for the operating condition, g_0 is the thickness of the hub at the small end, h_o is the hub length parameter and E_{yo} is the elastic modulus of the flange joint. The flange rigidity index of the original API flange joint calculated by Eq. (2) is J = 0.156. The rigidity of the compact flange joint is J = 0.127. Compared to the original API flange joint, the value of J of the compact flange joint is decreased by 18%. The flange rigidity is improved by the downsizing of the flange joint. Since $J \geqq$ 1.0 corresponds to an inclination angle of the flange of 0.3 degrees or less, the flange rotation is also sufficiently small.

Consideration of Bolt Spacing

The bolt spacing is calculated according to TEMA [9]. The correction factor CF is expressed as follows:

$$CF = \sqrt{\frac{\pi C / N}{2d + 6t/(m+0.5)}} \qquad (3)$$

where C is the bolt pitch diameter, N is the number of bolts, d is the bolt diameter, t is the flange thickness, and m is the factor for the gasket operating condition. The denominator of Eq. (3) is the maximum bolt spacing, and the numerator is the bolt spacing. Generally speaking, given $CF \leqq 1.0$, it is not necessary to correct the stress. The values of CF with respect to the original API flange and the compact flange are calculated by Eq. (3). The CF of the original API flange is 0.91, and that of the compact flange is 0.97. The CF of the compact flange is 7% larger than that of the original API flange. Although the CF of the compact flange is high, the ratio to the maximum bolt spacing is 1.0 or less, which appears not to be a problem.

Table 1 Flange stresses calculated according to ASME Section VIII-Div.1 App.2 [8].

Type of flange	Flange stresses MPa		
	S_H	S_R	S_T
Original API 4-inch flange	112	121	56
Compact flange	91	98	44

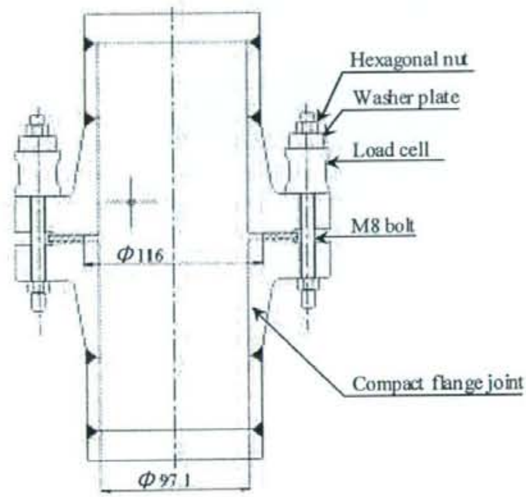


Fig.3 Compact flange joint subjected to internal pressure.

Flange Stresses

Table 1 shows the flange stress calculated according to ASME Section VIII-Div.1 App.2. Here, S_H is the flange hub stress, S_R is the flange radial stress, and S_T is the flange tangential stress. The stresses are smaller in the compact flange, as compared with the original flange. The bending moment can be reduced by downsizing the flange, and flange stresses have a sufficient margin. In other words, the design is conservative.

PLASTIC REGION TIGHTENING TEST OF A COMPACT FLANGE JOINT USING A DOWNSIZED BOLT

Figure 3 shows the compact flange joint subjected to internal pressure. The test flange is the downsized 4-inch class 150 lb (material: A-105), raised face, slip-on welding neck type flange. The test bolt (stud) has a nominal diameter of M8 with nuts

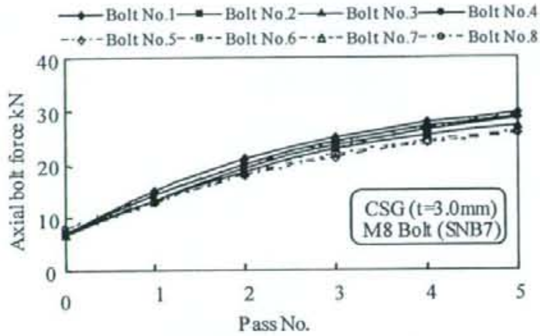


Fig.4 Relation between Pass No. and axial bolt force.

(bolt material: A193 Gr.B7, nut material: A194 Gr.2H). The outputs of the load cells used to measure the bolt axial force are detected with a strain amplifier and a digital multimeter controlled by a PC. The lubricant used in the test is a dry coating spray of MoS_2 (molybdenum disulfide).

The tightening test procedure follows the modified HPIS procedure [10], which employs a rotational clockwise pattern tightening sequence, to achieve the assembly efficiency and joint reliability. The angle control method is used for tightening rather than the torque control method. The test gasket is a compressed sheet gasket constructed of a non-asbestos material (No. 6500, Nippon Valqua Co.).

Figure 4 shows the results of the tightening test of the flange joint. The axial bolt force increases in proportion to the pass number from pass No. 1 to pass No. 2. The rate of the bolt elongation increment changes when the bolts enter the plastic region. In pass No. 3, the axial bolt force becomes 25 kN and reaches the yield point. In pass No. 5, all of the bolts enter the plastic region. The average value of the final axial force of all bolts is 28 kN, which is near the target axial bolt force of 30 kN. The scatter of the axial bolt force is 10% or less after the tightening process was completed. This value is somewhat small compared to 13% in the HPIS procedure for elastic region tightening [10]. The uniformity of the axial bolt forces obtained by the test is equivalent or superior to that of elastic region tightening [10].

Since a higher axial bolt force was obtained under plastic region tightening, as compared with elastic region tightening, the M8 bolt downsized from the M16 bolt, was used in the present study. The target axial bolt force for the combination of the test flange and the gasket was obtained. Assuming that the number of bolts and the strength class of the 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 can downsize the bolts and the flange due to higher tightening forces and uniformity.

BEHAVIOR OF AN ADDITIONAL AXIAL BOLT FORCE ON A FLANGE JOINT SUBJECTED TO INTERNAL PRESSURE

The integrity of the flange joint subjected to internal pressure must be ensured when designing the flange joint under plastic region tightening. Thus, the behavior of the additional axial bolt force of the flange joint subjected to the internal pressure is demonstrated, as compared with the behavior of the additional axial bolt force under elastic region tightening.

Test Method and Test Conditions

The test flange is the downsized 4-inch class 150 lb (material: A105), raised-face slip-on welding neck type flange. The test gaskets are a non-asbestos compressed sheet gasket and a flexible graphite sheet gasket. The test bolt (stud) has a nominal diameter of M8 with nuts (bolt material: A193 Gr. B7, nut material: A194 Gr.2H). An internal pressure of from 1 MPa to 5 MPa is applied by a hydraulic pump to the flange joint under plastic region tightening.

Difference in Load Factor between the Original Flange and the Compact Flange

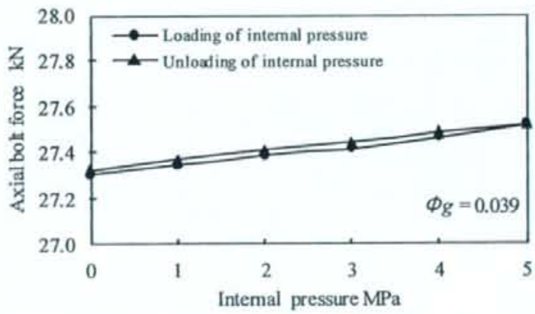
Figure 5 shows the behavior of the axial bolt force of the flange joint under plastic region tightening subjected to internal pressure. The test gasket is the compressed sheet gasket. Figure 5(a) shows the result of the original flange, and Fig. 5(b) shows that of the compact flange. Load factor ϕ_g is expressed as follows:

$$\phi_g = \frac{F_{bn}}{W} \qquad (4)$$

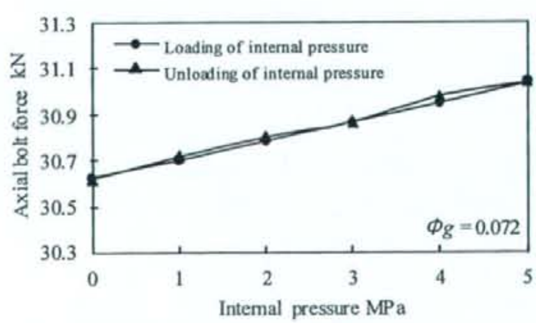
where F_{bn} is the additional axial bolt force. The thrust force generated by applying the internal pressure calculated as $W = A_g \cdot p/N$, where A_g is the contact area of the gasket, P is the internal pressure, and N is the number of bolts. Figure 5(a) shows that the value of F_t of the original flange is approximately 0.22 kN and the load factor ϕ_g calculated from the experimental data is 0.039. In addition, as shown in Fig. 5(b), the value of F_t of the compact flange is 0.42 kN, and the load factor ϕ_g is 0.072. The load factor of the compact flange is higher than that of the original flange. As the bolt pitch circle diameter shifts inside the flange as the flange is downsized, the moment arm decreases. Although less stress variation on the gasket is desirable, the bolt is subjected to a severe condition. This problem is examined in the subsection "Investigation of Bolt Fatigue Strength" later herein.

Difference in Load Factor Depending on Gasket Type

Figure 6 shows the behavior of the axial bolt force of the flange joint by using the compressed sheet gasket subjected to internal pressure. The test flange is the compact flange joint. As shown in Fig. 6, the axial bolt force decreases slightly with increasing internal pressure when the compressed sheet gasket is used. In addition, as shown in Fig. 5(b), the axial bolt force increases slightly with the internal pressure when the flexible graphite sheet gasket is used. Table 2 shows the load factors



(a) Case of the original API flange joint.



(b) Case of the compact flange joint.

Fig.5 Behavior of the axial bolt force of the flange joint subjected to internal pressure.

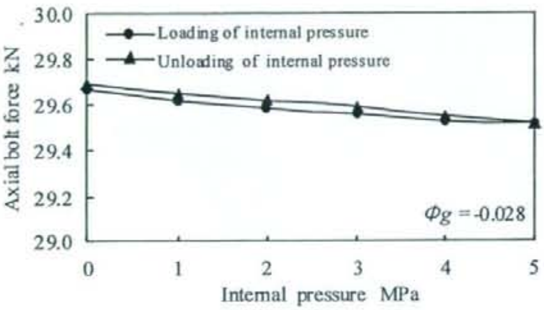


Fig.6 Behavior of the axial bolt force of the flange joint by using the compressed sheet gasket subjected to internal pressure.

Table 2 Comparison of the load factor for all test conditions.

Type of flange	Type of gasket	Load factor Φ_g
Original API 4-inch flange	CSG	-0.017 *
	FGSG	0.024
Compact flange	CSG	-0.028
	FGSG	0.072

*Refer to [5].

for all of the test conditions. Even though the bolts are tightened with equal axial force for the compressed sheet gasket and the flexible graphite sheet gasket, the load factor varies. When the elastic modulus of the gasket is different, variation in the load factor may occur.

Allowable Limit of Additional Bolt Force

When the external force is applied to the bolted joint under plastic region tightening, the allowable limit of the additional axial bolt force is approximately 10% of the bolt yield force. In the present experiment, the additional axial bolt force is positive. However, the additional axial bolt force is approximately 1% of the bolt yield force at maximum and has a sufficient margin for the allowable limit of the bolt tightened to the plastic region. Upon unloading the internal pressure, the bolt force returns to original value before internal pressure loading. Additional plastic elongation is not generated by the internal pressure.

Investigation of Bolt Fatigue Strength

Nest, the fatigue strength of the bolt is examined. Figure 7 shows relation between bolt force and thrust force generated by the internal pressure. The fatigue strength of the bolt, σ_A, is

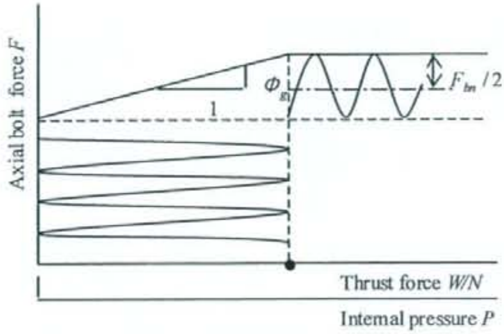


Fig.7 Relation between the axial bolt force and thrust force.

expressed as follows [4]:

$$\sigma_A \approx 0.75\left(\frac{180}{d} + 52\right) \qquad (5)$$

where σ_A is the stress amplitude of the bolt (nominal stress amplitude), and d is the diameter of the bolt. If the value of σ_a is less than or equal to the value calculated from Eq. (5), the fatigue strength of a bolt is sufficient. When the internal pressure is applied to a flange joint under plastic region tightening, the additional bolt force F_{bn} of $\phi_g \cdot W/N$ is generated, where N is the number of bolts. The stress amplitude of the bolt is calculated as $\sigma_a = F_{bn}/2/A_r$, where A_r is the cross-sectional area of core diameter. Therefore, determination of the fatigue strength of the bolt is expressed as follows:

$$\frac{\phi_g W}{2A_r N} \leq \sigma_A \qquad (6)$$

In the case of Fig. 5(b), the additional bolt force F_{bn} is 0.42 kN, and the stress amplitude of the bolt σ_A is 3 MPa. Since the value of the fatigue limit of the bolt calculated by Eq. (5) is 56 MPa. The bolt has a sufficient margin for fatigue strength.

CONCLUSIONS

Plastic region tightening was applied to the compact flange joint assembly using a downsized bolt, and internal pressure was applied to the flange joint. The following results were obtained:

(1) The API 4-inch flange joint was designed for plastic region tightening. The required tightening force of 32 kN is satisfied by the M8 bolt, rather than the M16 bolt specified in the API standard. As a result, the flange rigidity was improved by downsizing the flange joint. In addition, the bolt spacing has no effect on the flange rigidity, and the flange stresses have a sufficient margin.

(2) The behavior of the additional axial bolt force of the compact flange subjected to the internal pressure was demonstrated. The load factor of the compact flange was larger than that of the original API flange. In addition, the load factor varied according to the gasket type. When the elastic modulus of the gasket was different, it was possible for the load factor to vary.

(3) The fatigue strength of the bolt was examined. The stress amplitude of the bolt was approximately 5% of the fatigue limit. The bolt had a sufficient margin for the fatigue strength.

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EVALUATION OF CREEP PROPERTIES OF NON-ASBESTOS JOINT SHEET GASKETS AT ELEVATED TEMPERATURE BY THREE-DIMENSIONAL VISCOELASTICITY MODEL

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ABSTRACT

When the gasket in a flange joint is used over the long term at an elevated temperature, the gaskets experience creep/relaxation. The creep of the gaskets may cause leakage of the internal fluid. Many gaskets are used at elevated temperatures, so the clarification of their creep properties at elevated temperatures is urgently needed. The creep of non-asbestos gaskets at an elevated temperature was tested using four-inch flanges and compressed non-asbestos joint sheet gaskets. The test conditions are 180°C and 500 hours.

A three-dimensional viscoelasticity model that yields more accurate results compared to the viscoelasticity model, which uses the conventional single axis, was applied to the elevated temperature creep properties. Using the three-dimensional viscoelasticity model, the gasket creep is divided into the viscoelasticity component that converges on a certain strain and the volume change component that increases with time. The gasket strain is evaluated by the three-dimensional viscoelasticity model that considers the stress reduction. It is shown that the gasket strain is divided into the pure creep component of the gasket and the volume change due to the weight loss and chemical factor.

INTRODUCTION

The axial bolt force decreases when the gasket has undergone creep/relaxation, and to make matters worse, the internal fluid may leak as a result. Recently, the creep property of gaskets was evaluated by the viscoelasticity model of single axis [1, 2, 3]. Kauer defined the PTFE gasket as a visco-elastic-plastic model. The gasket stress distribution was calculated by the FE model when the creep property of the visco-elastic-plastic model was provided [1]. Bouzid defined the gasket as a viscoelasticity model and the bolt as a spring, and obtained the equation relating the bolt creep and the flange creep. Those equations agree well with the results of FE analysis [2]. Kobayashi also defined the gasket as a viscoelasticity model, and estimated the decremental axial bolt force in the gasketed joint fastened by the bolt. The estimated results well accord with the experimental results [3]. These references present the creep property and the relaxation of the gaskets as a function of time. It is anticipated that the life of the gasket could be predicted by relating the creep property to the leakage. The creep property of gaskets occurs markedly at elevated temperatures. Therefore, there is an urgent need to clarify the creep properties at an elevated temperature. There have been reports on the creep property at room temperature, but the clarification of creep property under elevated temperatures is insufficient. In this research, the three-dimensional viscoelasticity model is applied to the creep property of a non-asbestos gasket at the elevated temperature, and the changes in the stress and strain occur in the gasket are evaluated. Also, the result calculated from the three-dimensional viscoelasticity model is compared with the experimental results at the elevated temperature. In this way, the creep mechanism of the gasket is investigated.

NOMENCLATURE

σ_s — Recommended gasket stress (MPa)
σ_g — Gasket stress (MPa)
σ_m — Mean normal stress (MPa)
σ'_{ij} — Stress deviation tensor (MPa)
σ_{zz} — Stress in thickness direction (MPa)
σ_{rr} — Radial stress (MPa)
$\sigma_{\theta\theta}$ — Circumferential stress (MPa)
ε_g — Creep property of the gasket
ε_m — Mean normal strain
ε'_{ij} — Strain deviation tensor

ε_{zz}—Strain in thickness direction

ε_{rr}—Radial strain

$\varepsilon_{\theta\theta}$—Circumferential strain

ε_g^{*}—Gasket strain which considers relaxation of thickness direction

a, b—Coefficient that consists of spring constant and viscosity

A, B, A', B'—Differential operators

s—Laplace variable

t—Time (h)

K—Module of volume elasticity of gasket (MPa)

E—Young's module of gasket (MPa)

μ—Viscosity coefficient of gasket (GPa·h)

ν—Poisson's ratio

CREEP TEST AT ELEVATED TEMPERATURE

Test device and test gasket

Figure 1 shows a test device for conducting the creep test at an elevated temperature. The flange is raised face, Class 600, blind type and four inches. The gasket temperature is measured by a thermo-couple that is mounted between the flange and the test gasket. The gasket displacement is measured by a differential transformer for high-temperatures. Copper pipe for cooling is installed in the upper part of the flange. The bolt that tightens the flange is a 7/8-9UNC. The material of the bolt and flange is SUS304.

The non-asbestos joint sheet gasket No. 6502 that is made by Nippon Valqua Corporation was used as a test gasket. The dimensions of the test gasket are 180 mm outer diameter, 116 mm inside diameter, and 3 mm thickness. The recommended gasket stress σ_s of the test gasket is 40 MPa when the internal fluid is a gas. The usable temperature range of the test gasket is from -50°C to 214°C. The gasket factor and the minimum tightening pressure of the test gasket are shown in Table 1.

Test method and test result

The test gasket is placed in the flange of the test device and then loaded by eight bolts so that the gasket stress becomes 40 MPa. The tightening procedure follows the "Bolt Tightening Guidelines for Pressure Boundary Flanged Joint Assembly," HPIS Z103 TR2004 [4], but the post-tightening isn't carried out. After tightening, the test device is put in the furnace, as shown in Figure 2. The temperature on the inside of the furnace is heated to 180°C, and the gasket displacement is measured from the start time of the test to 500 hours.

Figure 3 shows the results of compression creep test of the gasket under an elevated temperature. The gasket temperature rises to 165°C in 11 hours from the start time of the test, and the gasket temperature after 11 hours remains constant. After the temperature is stabilized, the gasket displacement gradually increases. In addition, the gasket displacement under the elevated temperature has reached approximately 0.083 mm in 500 hours.

Figure 4 shows the gasket strain and gasket stress under the elevated temperature. An each measured initial value is the time when the temperature of the gasket became 165°C. The gasket strain increases with the time, but the increment of the gasket strain decreases with time. The strain in the test end was approximately 0.028. The gasket stress decreased to 20.7 MPa by the end of the test.

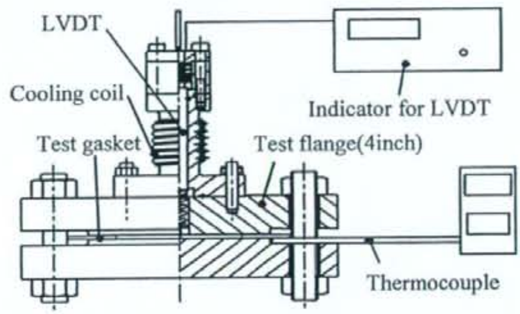


Fig. 1 Configuration of test device

Table 1 Gasket factors m and y for compressed sheet gaskets No.6502

Thickness of gaskets t (mm)	Gaskets factor m	Minimum tightening pressure y (MPa)
1.5	2.75	25.50
3.0	2.00	10.98

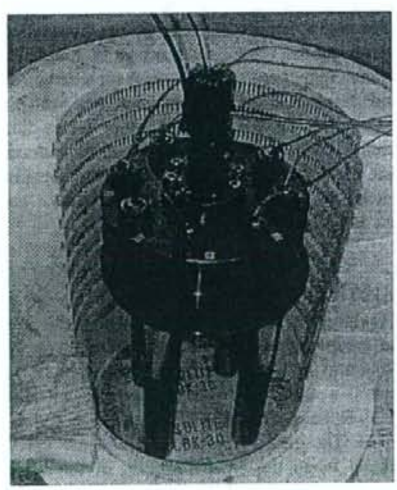


Fig. 2 Elevated temperature creep test

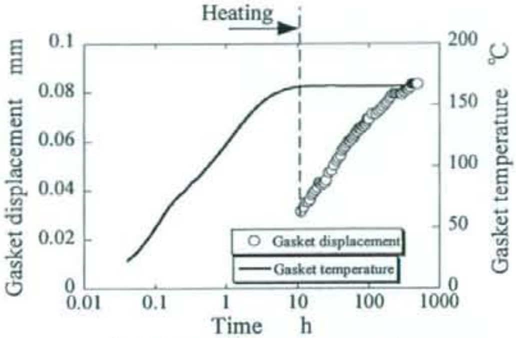


Fig. 3 Result of compression creep test

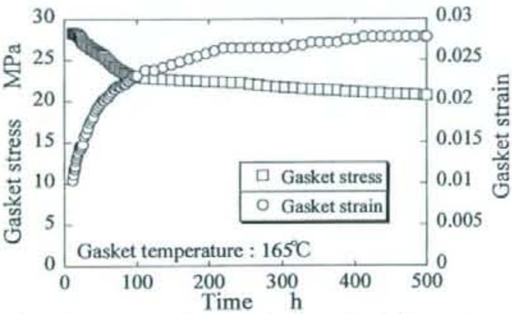


Fig. 4 Creep curves and stress reduction at elevated temperature

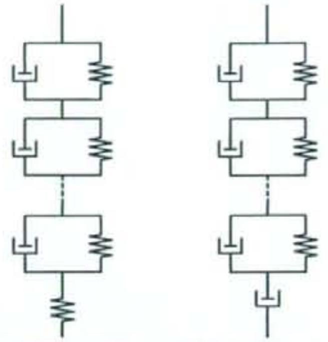
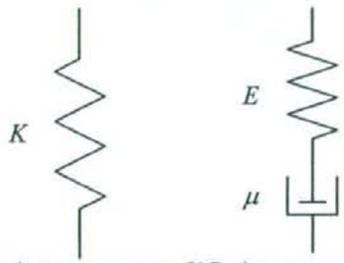

(a) Kelvin chain and a spring (b) Kelvin chain and a damper

Fig. 5 Viscoelasticity model of single axis



(a) Hydrostatic stress component (b) Deviator stress component

Fig. 6 Three-dimensional viscoelasticity model

EVALUATION OF CREEP CHARACTERISTIC AND RADIAL/CIRCUMFERENTIAL STRESSES BY THREE-DIMENSIONAL VISCOELASTICITY MODEL [5]

Three-dimensional viscoelasticity model

The gasket is bound by the friction between the gasket and the flange in both the radial direction and the circumferential direction and is subjected to a stress other than that in the thickness direction. Since the viscoelasticity model shown in Figure 5 is a single axis model, it can't evaluate the stress and strain in directions other than the thickness direction.

In this research, a three-dimensional viscoelasticity model that is capable of evaluating the triaxial stress state was applied to the creep property of the test gasket. The three-dimensional viscoelasticity model divides the triaxial stress state into the hydrostatic stress component and the deviator stress component, and the creep property is evaluated. Figure 6 shows the three-dimensional viscoelasticity model that is used in this research. The hydrostatic stress component and the deviator stress component are applied to the elastic model and the Maxwell model, respectively.

Three-dimensional viscoelasticity property

As shown in Figure 5, the viscoelasticity model is usually defined as the combination of the Kelvin and Maxwell model or in the multi-model. Regardless of series or parallel, the relation between the strain and stress is calculated using the following differential equation for any kind of combination using the single axis model.

$$\sum_{k=0}^{m} a_k \frac{d^k \sigma}{dt^k} = \sum_{k=0}^{n} b_k \frac{d^k \varepsilon}{dt^k} \tag{1}$$

Equation (1) may also be written in terms of the hydrostatic stress component and deviator stress component as follows:

$$A\sigma_m = B\varepsilon_m \tag{2}$$

$$A'\sigma'_{ij} = B'\varepsilon'_{ij} \tag{3}$$

where A and B are differential operators as follows: