厚生労働科学研究研究費補助金

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非石綿ガスケットの高温密封性能の評価と試験方法の開発

平成１８年度　総括研究報告書

主任研究者　辻　裕一

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総括研究報告書

非石綿ガスケットの高温密封性能の評価と試験方法の開発

主任研究者　辻　裕一　東京電機大学工学部教授

研究要旨　本研究は，非石綿ガスケットへの代替化促進のため，プラント運転時の高温における長期間のガスケットの性能･品質に関する信頼性データの収集・提供を目的とする．初年度である平成１８年度は，標準的な高温ガスケット密封性能試験法の開発を進めた．常温試験（ROTT）と高温試験（HOTT）を組み合わせたJ-EHOT試験を提案した．高温での外乱条件が高温漏洩特性の評価方法の確立に重要であり，実機プラントのガスケットの使用条件を考慮し，数種類の外乱条件を比較検討した．フランジ継手中のガスケットの応力変化を推測しJ-EHOT試験に最適な外乱条件を決定するためにフランジ継手の有限要素解析を進め，フランジ継手組立て時および運転時のガスケット応力分布を求めた．次年度以降は，試験法の規格化を図ると共に，この成果に基づき，非石綿ガスケットの設計係数の決定，代替品選択のための指針の提供を中立研究機関の立場から行う．

分担研究者氏名・所属機関名及び所属機関における職名
　本田　尚・独立行政法人労働安全衛生総合研究所・産業安全研究所主任研究員

A. 研究目的

各種プラントの配管継手や機器フランジに多用されているガスケットは，高温でのアプリケーションに関して現時点での代替化が困難との理由から石綿含有製品製造等の禁止の対象となっていない．非石綿ガスケットは既に開発がある程度行われているが，プラント運転時における高温環境下での性能および品質の評価が定まっておらず，結果として石綿ガスケットの代替が進んでいない．非石綿ガスケットへの代替化促進の鍵は，プラント運転時の高温における長期間のガスケットの性能・品質に関する信頼性データの収集・提供である．

本研究では，標準的な高温ガスケット密封性能試験法の開発・規格化を進める．高温密封性能試験結果を利用し，非石綿ガスケットを用いたフランジ継手からの微少漏洩管理を目標に，最適なガスケット係数の表示方法を非線形有限要素解析に基づき検討する．この成果に基づき，非石綿ガスケットの設計係数の決定，代替品選択のための指針の提供を中立研究機関の立場から行う．さらに，統一的試験基準に基づく指針が提供されれば，ガスケットメーカにとっても性能目標となり，非石綿製品の開発を促す効果が期待される．

B. 研究方法

本研究は３年計画で，東京電機大学，および独立行政法人産業安全研究所で実施する．初年度である平成１８年度は，次に示す方法で研究を実施する．

まず東京電機大学では，常温のガスケット密封性能試験である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，非石綿うず巻形ガスケットを用いて試験を実施する．J-EHOT試験では，常温特性をROTT (pre-ROTT)，高温特性をHOTTにより評価し，その後，常温に戻し再度ROTT (post-ROTT) を行う．負荷シーケンスの前半のpre-ROTTがHPISガスケット試験法のシーケンスに相当する．この試験法は，常温における組立て／高温でのガスケットのエージングおよび外乱／シャ

ットダウンという実際のプラント運転状況を想定しているところに特徴がある．高温でのエージング前後のシール特性の変化を常温試験結果の比較により確認できる．さらに，pre-ROTT と post-ROTT の漏洩量測定結果の比較だけで，測定の難しい高温特性の評価が行える可能性がある．

産業安全研究所では，ガスケット付きフランジ継手におけるガスケット応力分布を明らかにするために ABAQUS を用いて有限要素解析を実行する．数種類の呼び径と圧力レーティングのフランジ継手について，ガスケットの非線形性，フランジ面での接触／分離を考慮できるモデルを作成し，初期締付け時と運転時のガスケット応力分布の変化を明らかにする．運転時のガスケット応力の除荷に着目して，J-EHOT 試験の高温での外乱シーケンスの根拠とする．同時に，非石綿ガスケットの設計係数を決定するための非線形有限要素解析モデルを作成する

（倫理面への配慮）

本研究の実施によって，生体及び環境へ影響を及ぼすことは無いので，倫理面への問題は無いと考える．

C. 研究結果

1：高温ガスケット密封性能試験法の検討・試験法の規格化

供試ガスケットは 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．作動流体は He ガスを使用し，試験温度は 300℃である．

pre-ROTT では，常温試験 Step 3 以降において漏洩量が石鹸膜流量計の測定限界以下になった．そこで，基本漏洩量 2×10^{-7} Pa・m³/s を下回る値を測定限界とした．

外乱条件については 2 種類 (Type I, Type II) を検討した．Type I シーケンスの外乱では，試験で漏洩量の変化が見られなかったので，HOTT での除荷負荷の幅を大きくし，内圧負荷により生じるガスケットの除荷レベルに合わせると共に，除荷回数も増加させる Type II シーケンスを提案した．Type I シーケンス，外乱を想定したガスケット締付圧の 75 N/mm² までの除荷を 2 回行った．Type II シーケンスでは，75, 50, 25, 12,5 N/mm² まで 5 回ずつ除荷した．共に漏洩量に大きな変化が見られず，約 10^{-5} Pa・m³/s で一定のシール性能を示す．

2：ガスケットの非線形有限要素解析・ガスケット係数の指針化

管フランジ締結体の軸対称有限要素解析モデルを作成し，初期締付け時のガスケットの非線形挙動，および内圧によるフランジローテーションとそれに伴うガスケットの非線形挙動を解析した．その結果，初期締付け状態ではガスケット中央に最大圧縮応力が生じていたが，内圧が掛かるにつれて，最大圧縮応力が減少するとともに外周側にシフトし，規定内圧ではガスケット内周部がフランジと分離する挙動が確認された．

D. 考察

1：高温ガスケット密封性能試験法の検討・試験法の規格化

post-ROTT により，シャットダウン時，再起動時のシール性能評価が行える．高温時と比較してガスケットの基本漏洩量のレベルが 2 桁～3 桁上昇していることが確認できた．ガスケットがエージングにより硬化していることから，高温から常温への温度変化がガスケット締付圧の変動より漏洩量に大きい影響を与える．

他の実験結果では HOTT での外乱により，漏洩レベルが大きく変化したケースもある．これはガスケットの個体差とも考えられる．今後さらに実験を繰り返してデータの再現性を調査するとともに，高温の外乱のシーケンス（除荷・負荷の幅，回数）について検討する必要がある．

2：ガスケットの非線形有限要素解析・ガスケット係数の指針化

今回作製した有限要素モデルでは，初期

締付け状態から内圧を負荷していくと，ガスケット内周部に作用する圧縮応力が減少，ガスケットとフランジが分離する現象をシミュレートできたが，フランジローテーションによって本来減少するはずのボルト軸力は変化しなかった．これは，ボルト軸力としてフランジ上面の節点に集中荷重中 F を与えているためである．このため，今後はボルト軸力に相当する初期変位をボルトに与えるなど，境界条件の見直しが必要である．

E. 結論

平成１８年度の研究により，以下の結論を得た．

HPIS ガスケット試験法を拡張した高温ガスケット試験法（J-EHOT 試験法）を提案した．常温における組立て／高温でのガスケットのエージングおよび外乱／シャットダウンというプラント運転状況を想定して，ガスケットの高温漏洩特性を明らかにした．試験法規格化までには試験結果の再現性の確認と，高温の外乱のシーケンス（除荷・負荷の幅，回数）について検討する必要がある．

ガスケットの非線形有限要素解析では，ボルトによる初期締付け後に内圧が負荷されると，ガスケット内周部の圧縮応力が減少・分離する現象をシミュレートすることができた．ただし，同時にボルト軸力が低下する現象を再現することができなかったことから，今後解析モデルの改良が必要である．

F. 健康危険情報

無し

G. 研究発表

1. 論文発表

Toshiyuki Sawa, Satoshi Nagata, Hirokazu Tsuji : New Development in Studies on the Characteristics of Bolted Pipe Flange Connections in JPVRC, Transaction of the ASME・Journal of Pressure Vessel Technology, Vol. 128, 103-108, 2006.

2. 学会発表

T. Kobayashi, H. Tsuji: Evaluation of Sealing Behavior of Gaskets Based on the Test Method HPIS Z104 Proposed in Japan, Proc. ASME PVP 2006/ICPVT 11 Conference, PVP2006-ICPVT-11-93512, 1-5, 2006.

S. Kaneda, H. Tsuji: Application of Plastic Region Tightening Bolt to Flange Joint Assembly, Proc. ASME PVP 2006/ICPVT 11 Conference, 2006, PVP2006-ICPVT11- 93553, 1-7, 2006.

長谷川聡，沖長徹，中島聡宏，山口篤志，齋藤暁洋，辻 裕一：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)，山梨講演会講演論文集，No.060-4，217-218，2006.

澤 俊行，辻 裕一：ガスケット非石綿化の動向と密封特性試験方法の検討，日本高圧力技術協会，平成１８年度春季講演会・講演概要集，32-33，2006.

H. 知的財産権の出願・登録状況（予定を含む）

1．特許取得

無し

2．実用新案登録

無し

3．その他

無し

厚生労働科学研究費補助金（労働安全衛生総合研究事業）
分担研究報告書

高温ガスケット密封性能試験法の検討・試験法の規格化

主任研究者　辻　裕一　東京電機大学工学部教授

研究要旨　本研究では，高温ガスケット密封性能試験方法としてHPISガスケット試験法を高温に拡張した試験方法J-EHOTを提案する．常温における組立て／高温でのガスケットのエージングおよび外乱／シャットダウンという実際のプラント運転状況を想定しているところに特徴がある．今後の課題として，高温での外乱シーケンスの検討が挙げられる．

A. 研究目的

各種プラントの配管継手や機器フランジに多用されているガスケットは，高温でのアプリケーションに関して現時点での代替化が困難との理由から石綿含有製品製造等の禁止の対象となっていない．非石綿ガスケットは既に開発がある程度行われているが，プラント運転時における高温環境下での性能および品質の評価が定まっておらず，結果として石綿ガスケットの代替が進んでいない．非石綿ガスケットへの代替化促進の鍵は，プラント運転時の高温における長期間のガスケットの性能・品質に関する信頼性データの収集・提供である．

本研究では，標準的な高温ガスケット密封性能試験法の開発・規格化を進めることを目的とする．統一的試験基準に基づく指針が提供されれば，ガスケットメーカにとっても性能目標となり，非石綿製品の開発を促す効果が期待される．

B. 研究方法

常温のガスケット密封性能試験である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．

$$L_s = \frac{L}{k} \qquad (1)$$

ここで，L (Pa・m³/s) は漏洩量，kはガスケット形状係数であり，次式で表される．

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

HPIS ガスケット試験法を高温に拡張した試験方法J-EHOTを提案する．J-EHOT試験では，常温特性をROTT (pre-ROTT)，高温特性をHOTTにより評価し，その後，常温に戻し再度 ROTT (post-ROTT) を行う．図1にJ-EHOTのうず巻形ガスケットに対するガスケット締付圧の負荷シーケンスを示す．負荷シーケンスの前半のpre-ROTTは，HPISガスケット試験法のシーケンスに相当する．ガスケットを常温状態でPre-ROTTに従って負荷する．その後，ガスケットを試験温度まで昇温させ，試験内圧一定で90時間放置(エージング)する．エージング終了後，高温のまま外乱に相当する除荷・負荷を2回繰り返す．最後にガスケット温度を常温に戻し，除荷過程Post-ROTTを行う．

この試験法は，常温における組立て／高温でのガスケットのエージングおよび外乱／シャットダウンという実際のプラント運転状況を想定しているところに特徴がある．高温でのエージング前後のシール特性の変化を常温試験結果の比較により確認できる．さらに，pre-ROTTとpost-ROTTの漏洩量測定結果の比較だけで，測定の難しい高温特性の評価が行える可能性がある．

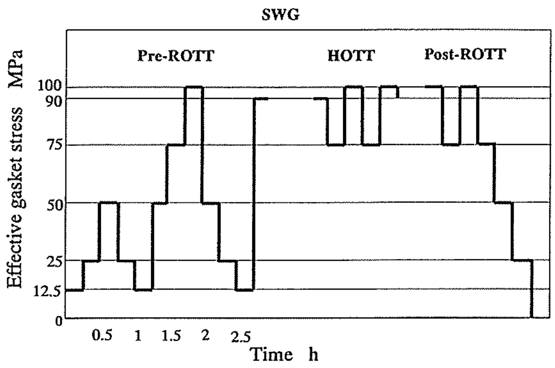

Fig. 1 Loading sequence of a gasket (Type I)

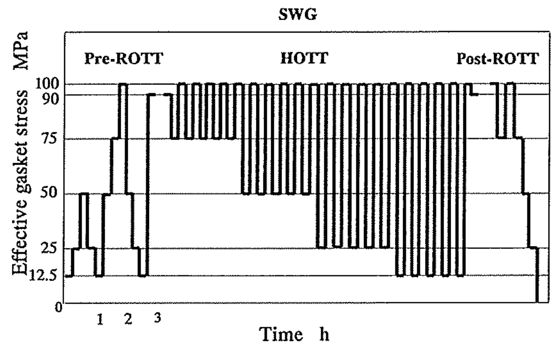

Fig. 2 Loading sequence of a gasket (Type II)

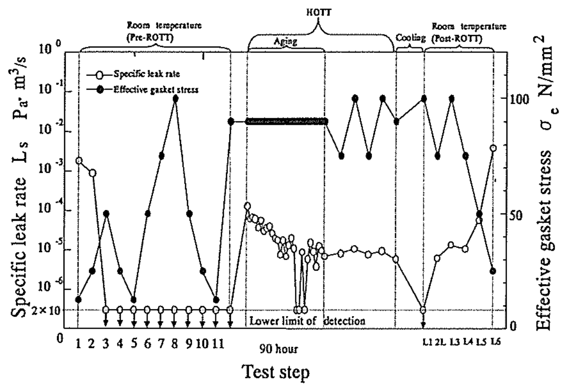

Fig. 3 Result of J-EHOT for loading sequence type I

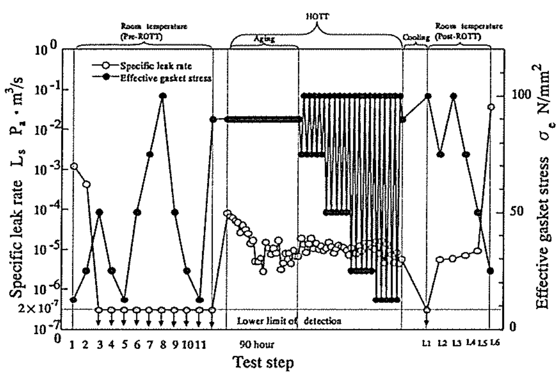

Fig. 4 Result of J-EHOT for loading sequence type II

C. 研究結果

供試ガスケットは 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．作動流体は He ガスを使用し，試験温度は 300℃である．

図 1 に示す HOTT の外乱では，試験で漏洩量の変化が見られなかったので，図 2 に示すように，HOTT での除荷負荷の幅を大きくし，内圧負荷により生じるガスケットの除荷レベルに合わせると共に，除荷回数も増加させる負荷シーケンスを提案している．図 4 及び図 5 にそれぞれ図 1 及び図 2 の試験シーケンスに対応する J-EHOT 試験の結果を示す．縦軸に基本漏洩量を対数として，横軸に時間経過 (Step) として示す．pre-ROTT では，常温試験 Step 3 以降において漏洩量が石鹸膜流量計の測定限界以下になった．以下，基本漏洩量 2×10^{-7} Pa・m^3/s を下回る値を測定限界とする．

HOTT では，外乱を想定したガスケット締付圧の 75 N/mm^2 までの除荷を 2 回行った（図 4)．次に外乱のシーケンスを変更し 75, 50, 25, 12,5 N/mm^2 まで 5 回ずつ除荷した（図 5)．図 4 及び図 5 共に漏洩量に大きな変化が見られず，約 10^{-5} Pa・m^3/s で一定のシール性能を示す．

D. 考察

post-ROTT により，シャットダウン時，再起動時のシール性能評価が行える．高温時と比較してガスケットの基本漏洩量のレベルが 2 桁～3 桁上昇していることが確認できた．ガスケットがエージングにより硬化していることから，高温から常温への温度変化がガスケット締付圧の変動より漏洩量に大きい影響を与える．

他の実験結果では HOTT での外乱により，漏洩レベルが大きく変化したケースもある．これはガスケットの個体差とも考えられる．今後さらに実験を繰り返してデータ数を増

やすとともに，高温の外乱のシーケンス（除荷・負荷の幅，回数）について検討する必要がある．

E. 結論

フランジ継手用ガスケットの高温密封性能試験方法を提案すると共に試験を実施した．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ガスケット試験法（J-EHOT 試験法）を提案した．

(2) 常温における組立て／高温でのガスケットのエージングおよび外乱／シャットダウンというプラント運転状況を想定して，ガスケットの高温密封性能を明らかにした．

F. 健康危険情報

無し

G. 研究発表

1. 論文発表

Toshiyuki Sawa, Satoshi Nagata, Hirokazu Tsuji : New Development in Studies on the Characteristics of Bolted Pipe Flange Connections in JPVRC, Transaction of the ASME・Journal of Pressure Vessel Technology, Vol. 128, 103-108, 2006.

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H. 知的財産権の出願・登録状況（予定を含む）

1．特許取得

無し

2．実用新案登録

無し

3．その他

無し

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分担研究報告書

ガスケットの非線形有限要素解析・ガスケット係数の指針化

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研究要旨　石綿ガスケットの代替品として開発された新材料，新形式のガスケットに対して，科学的根拠のあるガスケット係数を決定し石綿代替化の促進を図ることを目的とし，管フランジ締結体の非線形接触有限要素解析を行った．

A．研究目的

ガスケットは各種プラントの配管継手や機器フランジに多用されているが，石綿のもたらす重篤な健康障害のリスクから，代替品によるガスケットの非石綿化が急務である．しかしながら，現行のガスケット係数の根拠が明確でないために，石綿代替品として開発されたガスケットのガスケット係数が決定できないことが，代替化の足かせとなっている．

そこで，非石綿ガスケットの設計係数の決定を目的として，管フランジ締結体の非線形接触解析を行い，ボルトによるフランジ初期締結後に，内圧が負荷されることでフランジローテーションが生じ，ガスケットの面圧およびボルト締結力が低下する現象を解析した．

B．研究方法

有限要素解析は，汎用の有限要素解析コードABAQUSを使用して行った．管フランジ締結体はJPI class300（4インチ）とし，図1に示すボルト穴を考慮しない軸対称モデルを作製した．ボルトには軸方向のみに剛性を有する線形一次元ばね要素を用いた．フランジの材料定数はヤング率206GPa，ポアソン比0.3とし，ガスケットの材料特性は圧縮試験に基づいて同定した応力－ひずみ関係を与えた．なお，管フランジ締結体は上下対称であることから，ガスケット厚は1/2とし，図1のようにガスケット下面に変位境界条件を設定した．また，フランジとガスケットの間には滑りが生じるものとし，両者の間に接触面を設定した．

荷重境界条件は，フランジ上面のボルトの頭が接する位置に集中荷重 F を負荷した．また，フランジ内面に内圧 p を，管上面には内圧により生じる軸方向の引張り力に相当する圧力 p' を負荷した．

有限要素解析は，まずボルトの初期締付けによるガスケットの非線形挙動を解析し，初期締付けによる変形を保持した状態で内圧を付与し，フランジローテーションとそ

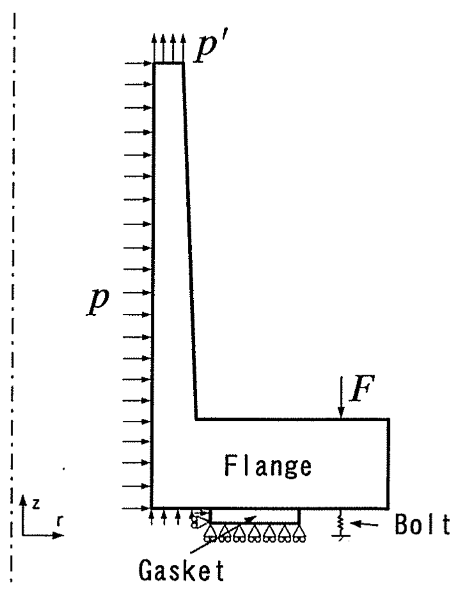
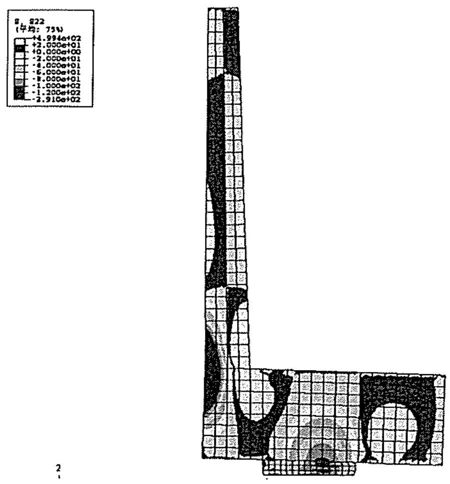
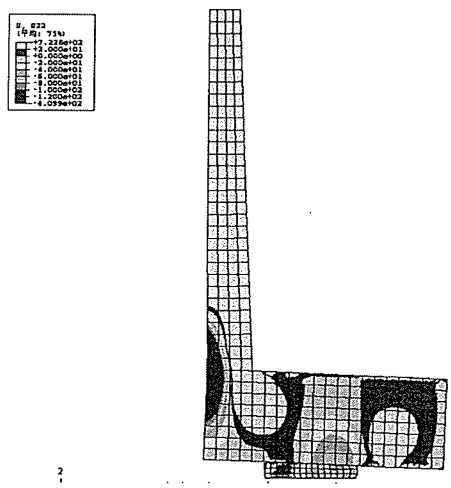


Fig. 1 有限要素モデル

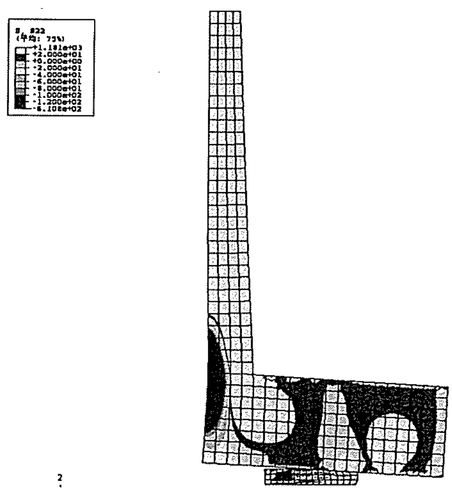
れに伴うガスケットの非線形挙動を解析した．



(a)ボルト初期締付け時の応力分布



(b) 規定内圧の 1/2 での応力分布



(c) 規定内圧における応力分布

Fig. 2 非線形有限要素解析結果

C．解析結果

解析結果を図 2 に示す．なお，図では解析結果を変形率 7 倍で表示している．図 2(a)はボルトに初期締付け力を付与した際の軸方向（z 方向）の応力分布である．ガスケットは圧縮され，中央より若干外周側に高い圧縮応力が発生している．また，フランジとガスケットの間の摩擦により，ガスケット外周にバレリングが生じている．この状態から内圧を徐々に負荷し，規定の 1/2 の内圧を負荷した状態が図 2(b)である．内圧によってフランジローテーションが生じ，ガスケット内周の応力が低下するとともに，最大圧縮応力がガスケット外周にシフトしている．また，最大圧縮応力も小さくなっている．図 2(c)は規定の内圧が作用した場合の応力分布であるが，ガスケットに作用する圧縮応力は，ほぼ最外周で最大となり，内周部は完全にフランジから分離している．

D．考察

内圧によってフランジローテーションが生じると，通常ボルト軸力は低下する．しかしながら，図 2 を見ると，内圧を負荷してもボルト軸力により生じるフランジ内の応力分布は変化しておらず，内圧によってボルト軸力は変化していないことになる．これは，図 1 のように節点力 F をボルト締付け力として与えていることに起因している．そこで，今後はボルト軸力に相当する初期変位をボルトに与えるなど，境界条件を見直す必要がある．

E．結論

非石綿ガスケットの設計係数の決定を目的として，管フランジ締結体の非線形接触解析を行ったところ，ボルトによるフランジ初期締付け後に内圧が負荷されると，フランジローテーションによってガスケット内周部の圧縮応力が減少し，ガスケットがフランジと分離する挙動をよくシミュレートすることができた．しかしながら，同時にボルト軸力が低下する現象は再現することはできなかったことから，今後解析モデルの改良が必要である．

Ｆ．健康危険情報

なし

Ｇ．研究発表

1.　論文発表

なし

2.　学会発表

なし

（発表誌名巻号・頁・発行年等も記入）

Ｈ．知的財産権の出願・登録状況（予定を含む。）

1. 特許取得

なし

2. 実用新案登録

なし

3.その他

なし

研究成果の刊行に関する一覧表

雑誌

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Toshiyuki Sawa, Satoshi Nagata, Hirokazu Tsuji	New Development in Studies on the Characteristics of Bolted Pipe Flange Connections in JPVRC	Transaction of the ASME ・Journal of Pressure Vessel Technology	Vol. 128	103-108	2006
Takashi Kobayashi, Toshiyuki Sawa, Hirokazu Tsuji, Shoichi Shigetome	Evaluation of Sealing Behavior of Gaskets Based on the Test Method HPIS Z104 Proposed in Japan	ASME PVP 2006/ICPVT 11 Conference, 2006	PVP2006-ICPVT-11-93512	1-5	2006
Sinobu Kaneda, Hirokazu Tsuji	Application of Plastic Region Tightening Bolt to Flange Joint Assembly	ASME PVP 2006/ICPVT 11 Conference, 2006	PVP2006-ICPVT11-93553	1-7	2006
長谷川聡，沖長徹，中島聡宏，山口篤志，齋藤暁洋，辻裕一	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）	山梨講演会講演論文集	No.060-4	217-218	2006
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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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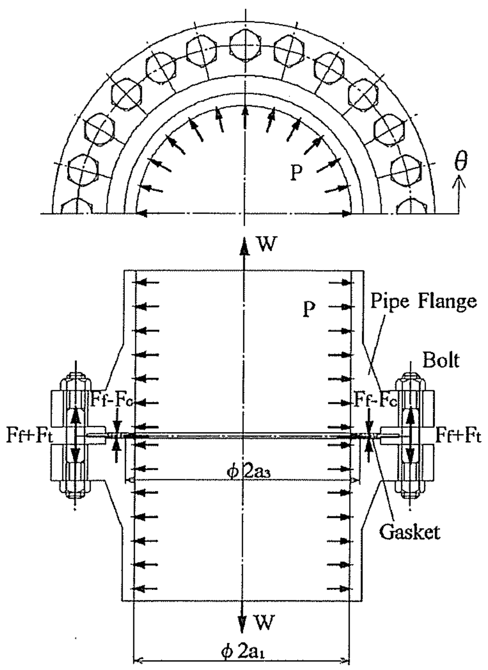

Fig. 1 A pipe flange connection with a spiral wound gasket subjected to an internal pressure

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

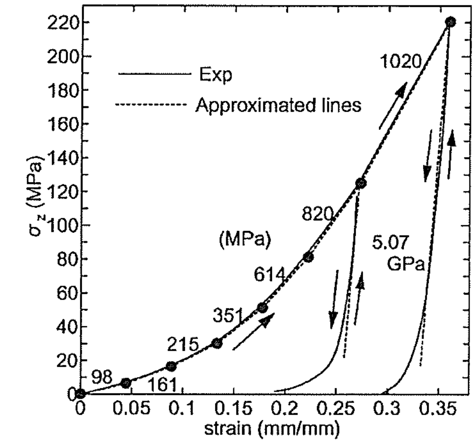

Fig. 2 Stress-strain lines of the spiral wound gasket used for the EP-FEA

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

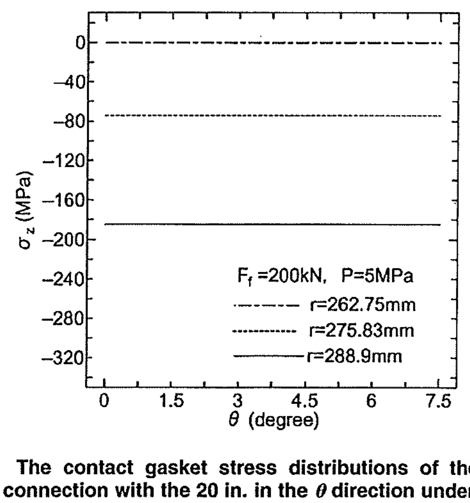

Fig. 3 The contact gasket stress distributions of the pipe flange connection with the 20 in. in the θ direction under internal pressure (θ=0°–7.5°)

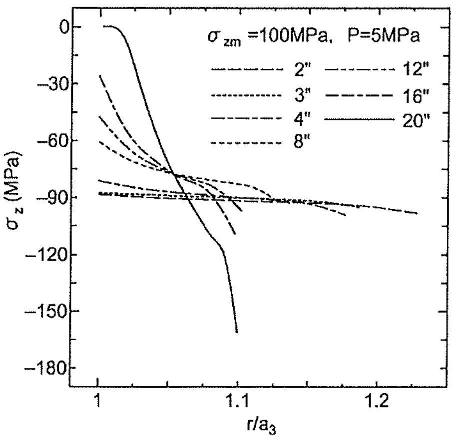

Fig. 4 The effects of the nominal diameter of the pipe flange connections on the contact gasket stress distributions in the *r* direction (the case where an internal pressure P=5 MPa is applied)

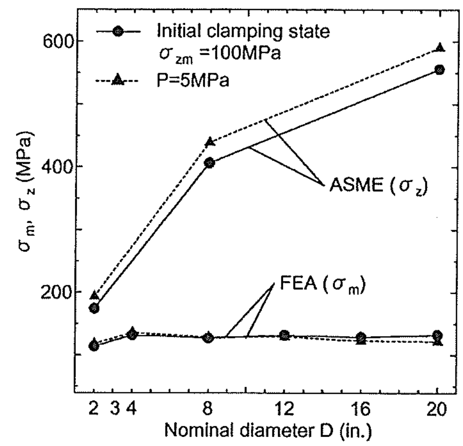

Fig. 5 The effects of the nominal diameter in the pipe flange connections on the hub stress (von Mises' equivalent stress σ_m) at the angle θ=0°

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 θ=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 θ=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

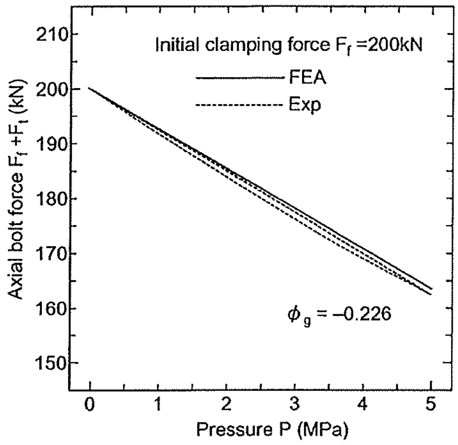

Fig. 6 Comparison of an increment in axial bolt force between the experimental and the FEA results

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

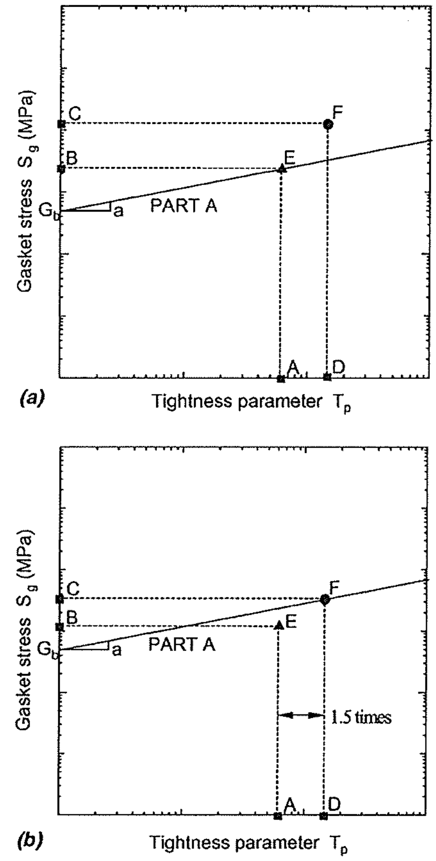

Fig. 7 A method for determining bolt preload F_f. (*a*) The case where the results of the FEA are used. (*b*) The case of the PVRC procedure.

(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

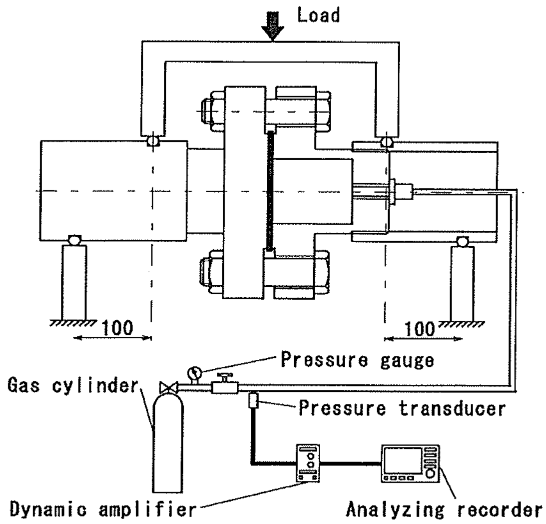

Fig. 8 Schematic of the pipe flange connection subjected to a bending moment and internal pressure

(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 $(=\pi a_1{}^2P)$
W' = total axial force due to internal pressure $(=\pi a_3{}^2P)$
ϕ_g = load factor $(=F_t/W')$

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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).

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

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

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

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

$$k = \frac{1}{d_o/d_i - 1} \tag{3}$$

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

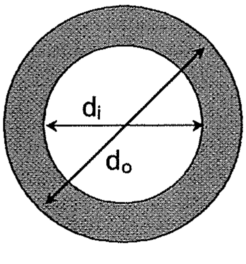

Fig. 1 Dimensions of gasket

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

$$L_s = \frac{L}{k} \tag{4}$$

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.

$$W_0 = 0.05\, A_g\, \sigma_{max} \tag{5}$$

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

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

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

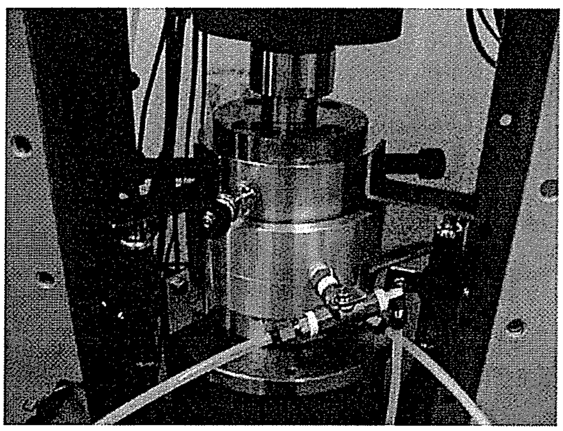

Photo 1 Test rig