A study on the sealing performance of flange joints with gaskets under external bending using finite-element analysis

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The manuscript was received on 6 July 2007 and was accepted after revision for publication on 13 November 2007.

DOI: 10.1243/09544089JPME155

Abstract: The effect of external bending on flange joints in pipelines, pressure vessels, and pressure vessel nozzles is significant and difficult to assess. In the current work, the sealing performance of a gasketed flange joint under external bending has been studied by using a three-dimensional finite element analysis, considering the non-linear material behaviour of the gaskets. Spiral wound gaskets and non-metallic gaskets with various sealing materials are considered. Results show that the allowable bending moment on a flange joint depends on the tightness required, type of gasket, and its deformation characteristics. Gaskets with elastomeric sealing materials show good performance against leakage under external bending.

Keywords: flange joint, bending load, gasket, finite-element analysis

1 INTRODUCTION

A number of design codes and standards such as ASME, DIN, JIS, and BS, which are principally based on the Taylor–Forge method [**1**], provide procedures for the design of flanged joints. Considerable work on the performance of gasketed flange joint for normal operating conditions (bolt load and internal pressure) has been carried out by many authors using analytical and numerical methods. Sawa *et al.* [**2**] presented a mathematical model to determine the contact stress distribution in a pipe flange connection based on the theory of elasticity by treating it as an axisymmetric problem. Sawa *et al.* [**3**] extended the earlier investigation by a numerical method considering the stress–strain curve of the gasket as piecewise linear. Bouzid and Derenne [**4**] developed an analytical method considering the rotational flexibility of the flange for determining the contact stresses in order to predict the joint tightness. The finite-element (FE) simulation for bolt-up process with spiral wound gasket was carried out by Fukuoka and Takaki [**5**]. Murali Krishna *et al.* [**6**] showed that the distribution of contact stress has a dominant effect on sealing performance than the limit on flange rotation specified by ASME. Siddique and Abid [**7**] have provided theoretical insight into mechanical stress, relieving modes for attenuation of residual stresses in flange joint through finite-element analysis (FEA).

In many operating conditions, external bending loads such as dead weight, wind-induced moments, and thermal expansion arise on the flange joint. These external loads cause the flange joints to bending deformation and the gasket is compressed additionally over the initial compression on one side and the initial compression is released at the opposite side. The joint must maintain the seal under this condition to prevent any leakage on the tension side. The ASME [**8**] procedure for the design of flange joint with bending is based on the calculation of equivalent internal pressure that replaces the bending moment. The available codes and standards at present do not account directly for the effect of external loads in the flange joint design. The results are conservative because the design is based on an artificially high pressure, and the required bolt preload to prevent gasket leakage is proportional to this pressure. It is also important to note that the gasket characteristics and joint leakage are not taken into consideration in the equivalent

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pressure method. Koves [9] evaluated the effect of external bending moment on the flange joint and developed a correction factor using energy method. His approach assumes a uniformly distributed average gasket stress, but in reality, stresses vary across the width of the gasket because of the flange rotation. Nash and Abid [10] made an experimental study on ANSI and compact VCF-bolted flange joints under pressure, external force, and moment and compared the results. They established a superposition approach for estimating the load capacities of these joints. Their study show that sufficient amount of bending loads can be allowed on a flange joint. In the investigations mentioned earlier, the non-linear behaviour of the gasket, gasket characteristics, type of gasket, and the changes in bolt axial force during loading were not considered while calculating the allowable bending moment on a flange joint.

In the present work, the influence of external bending on a gasketed flange joint was studied by FEA for spiral wound gaskets and non-metallic gaskets. A three-dimensional FE model was created with contact elements between the interfaces. Surface-to-surface contact elements are used to model contact interfaces to carry out the analysis as a three-dimensional contact problem. The non-linear material property of gaskets is used in the analysis. Three different sealing materials for both types of gaskets were considered. The sealing performance of the joint with each gasket under different bolt preloads and internal pressures is studied at the room temperature and the results are compared. The allowable bending moments are calculated for the joint with each gasket for different bolt preloads.

2 FE ANALYSIS

The flange joint assembly consists of flange welded with pipe, bolting, and the gasket. The analysis of the joint with bolting, pressure, and external bending loads are considered as a three-dimensional contact problem. A commercial FE code ANSYS® 8.1 [11] has been used. Taking the geometric symmetry into consideration, one-quarter of the joint is modelled.

2.1 Geometry

Weld neck raised-face flanges are widely used for medium service conditions. In the present study, an 80 mm NPS weld neck, Class 600 with dimensions as per ASME/ANSI B 16.5 [12] with M20 size bolts is considered.

2.2 Discretization

The flange is divided into three mesh density regions, viz, hub portion, fillet, and the flange ring. Extremely fine mesh was created at the fillet region to handle the high stress gradients. Other regions had gradual coarser mesh at increasing distance from the fillet region. The flange, pipe, and bolts are meshed with eight noded brick elements (SOLID 45), and the gasket is modelled with gasket element (INTER 195). There are two interfaces in the assembly: the interface between the flange ring and the bolt head and that between gasket and raised face. Target elements (TARGE 170) and contact elements (CONTA 173) are created in the interfaces. As the members (flanges and gasket) and bolts are deformable, both can be treated as in the flexible-to-flexible category. As the flange ring is stiffer than the bolt, it is modelled as a target surface and the bolt head/nut face is modelled as the contact surface. The pretension elements (PRETS 179) are used to model the bolt preloads. The bolt has been meshed with eight noded brick elements (SOLID 45), and the pretension elements are inserted at the mid-section of the bolt with specified normal direction. A pretension node (K) is used to control and monitor the total tension load across the bolt shank. The FE model consists of 85 627 elements and 96 545 nodes, as shown in Fig. 1.

2.3 Material properties

Materials for flanges and bolts are considered as forged carbon steel (A 105) and chromium steel (A 193-B7). Young's modulus and Poisson's ratio for the flange material is 195 GPa and 0.3, respectively, and for bolt material, these are 203 GPa and 0.3. The deformation behaviour of gasket material is non-linear, and a typical pressure-closure relation of a

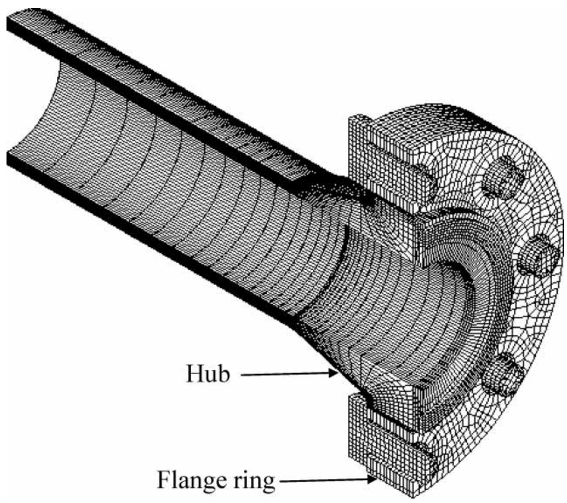

Fig. 1 FE model

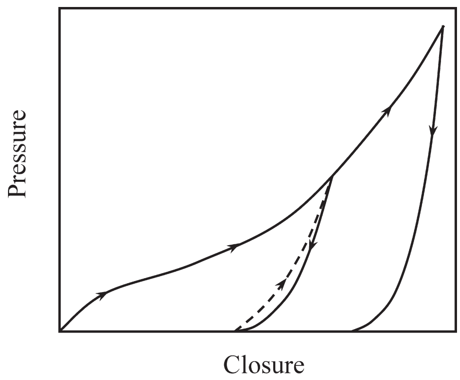

Fig. 2 Typical pressure-closure relationship of gasket

gasket under loading, unloading, and reloading is as shown in Fig. 2.

ANSYS offers a number of elements to model gaskets. These elements consider geometric and material non-linearities, and membrane and transverse shear are neglected. Thus, the pressure versus closure behaviour can be directly applied to characterize the gasket material. Mechanical characteristics of the gasket material carried out by Murali Krishna [**13**] by a load compressive mechanical test (LCMT) [**14**, **15**] has been used in the present study. ANSYS software has a provision to input the LCMT data points. Figure 3 shows the material properties for different spiral wound gaskets and non-metallic gaskets [**13**]. The non-linear compression behaviour with two different unloading curves is considered in the present analysis. If unloading occurs at an intermediate point, the unloading curve is interpolated from neighbouring curves. For repeated load cycles the reloading is assumed to take the same path as the unloading curve.

The following gaskets are considered for their mechanical properties (Fig. 3) and used in the analysis:

(a) spiral wound gasket-asbestos-filled (SWG-AF);
(b) spiral wound gasket-graphite-filled (SWG-GF);
(c) spiral wound gasket-teflon-filled (SWG-TF);
(d) non-metallic gasket-compressed asbestos-filled (NMG-CAF);
(e) non-metallic gasket-compressed asbestos with wire reinforcement (NMG-CAFWR);
(f) non-metallic gasket-rubber-filled (NMG-RUBBER).

2.4 Loading and boundary conditions

A typical geometry of a flanged joint with all the loads is shown in Fig. 4.

The following loads were applied in the FE model:

(a) bolt preload (F),
(b) pressure load (P),
(c) bending moment (M).

To consider the non-linearities, the loads are applied in steps to trace the exact non-linear behaviour of the assembly under bending loading. In the bolting-up stage, only the bolt preload (F) has been applied uniformly on the pretension element section through the pretension node. In the pressurizing stage, in addition to the bolt preload, a design internal pressure of 10 MPa is applied at the inner surface of the flange and pipe. The hydrostatic end-force exerted on the closed end of the pipe system is calculated on the basis of the inner diameter of the pipe and has been applied uniformly in the axial direction at one end of the pipe. In the third load step, the bending moment is applied as a couple at the pipe end, in addition to the preload and internal pressure. Sufficient length of the pipe is modelled to avoid local effects. The displacement boundary conditions are applied as shown in Fig. 5.

3 FEA RESULTS

3.1 Validation of the present method

The present model is validated with the available analytical and experimental [**3**] works for the bolting-up and pressurizing loading, and the analysis is extended for bending load conditions. Figure 6 shows a comparison of gasket stresses during bolting-up and pressurizing stages between the present model and published values on an 80 mm NPS weld neck, Class 600 flange with SNG-AF. The results obtained in the present analysis are in good agreement with the published results.

Figure 7 shows a comparison of variation in axial bolt force with the experimental results for an initial bolt preload of $F = 50$ kN. The increment in the axial bolt force varies linearly as the internal pressure increases. A difference of 1.5 per cent in the axial bolt force at higher internal pressure ($P = 6$ MPa) is observed between the experimental results and the present work.

3.2 Sealing performance

In bolted flange joints, external bending varies significantly with the bolt axial force and the gasket contact stress. During the bolting-up process, the required gasket stress is established by tightening the bolts. When the joint is pressurized, the gasket initial compression is released. The corresponding decrease in the gasket contact stress depends on the hydrostatic end-force, bolt elastic properties, and gasket unloading characteristics. Under the bending load, the gasket is compressed at one side and the initial compression is further reduced at the opposite side. Figure 8 shows the contour plot of a typical gasket

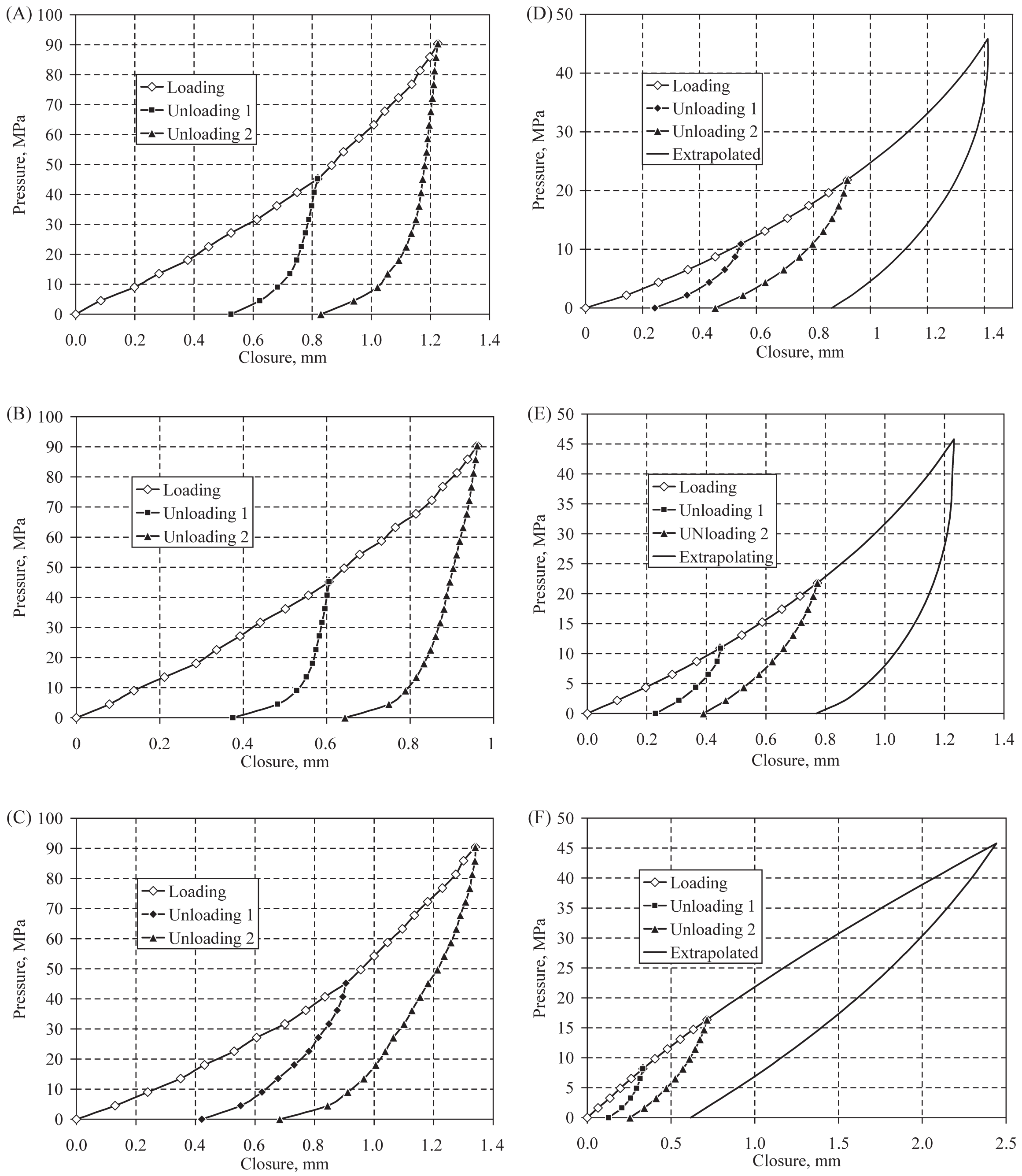

Fig. 3 Characteristics of different types of gaskets: (a) SWG-asbestos filled, (b) SWG-graphite filled, (c) SWG-Teflon filled, (d) NMG-compressed asbestos, (e) NMG-compressed asbestos with wire reinforcement, and (f) NMG-rubber

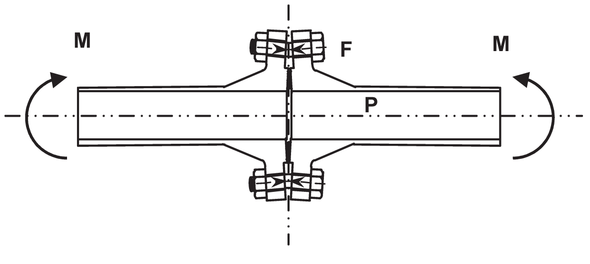

Fig. 4 Flange joint with loads

contact stress distribution under different stages of loading on one half of the symmetry.

When the gasket stress is very small, the joint may leak or the gasket may blow out of the joint. The ASME code defines this reduced stress (referred to as residual stress) in terms of a maintenance gasket factor *m*, which is dimensionless. The desired minimum residual stress is the product of *m* and the contained pressure (*P*). The code also specifies a *y* factor,

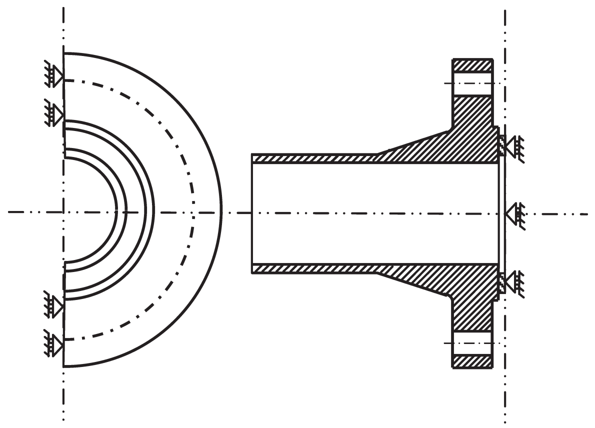

Fig. 5 Boundary conditions

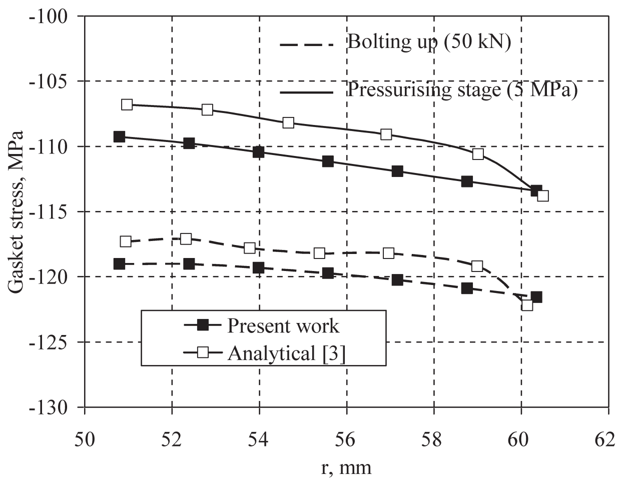

Fig. 6 Comparison of gasket stress in SWG-AF

which is the initial gasket stress or surface pressure required to preload or seal the gasket to prevent leaks in the joint as the system is pressurized [**14**, **15**]. The factor m is 3 and 2.5 for spiral wound and

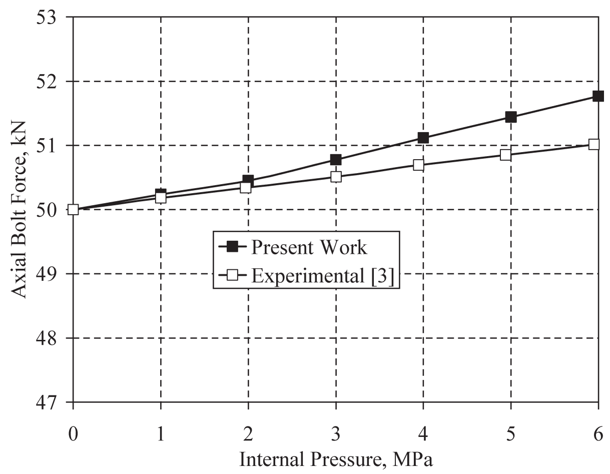

Fig. 7 Comparison of axial bolt force in SWG-AF

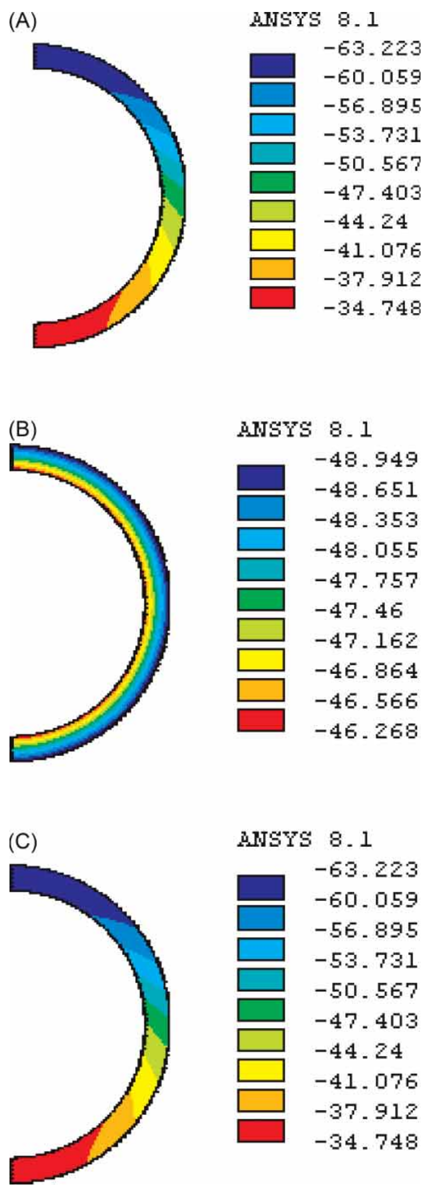

Fig. 8 Non-uniform gasket stress distribution (MPa): (a) bolt-up stage, (b) pressurizing stage, and (c) external bending

non-metallic gaskets, respectively. Theoretically, the leakage will occur when the minimum gasket contact stress is lower than the internal pressure. The required minimum gasket stress to maintain the design internal pressure of 10 MPa for spiral wound gasket is 30 MPa and for non-metallic gasket is 25 MPa. For the problem under study, the gasket contact stress is minimum at the tension side and further it varies across the width of the gasket. The minimum gasket stress specified by ASME has to be maintained under bending load conditions to avoid any leakage.

Figures 9 to 11 show the gasket contact stresses across the width of the gasket on the tension side of

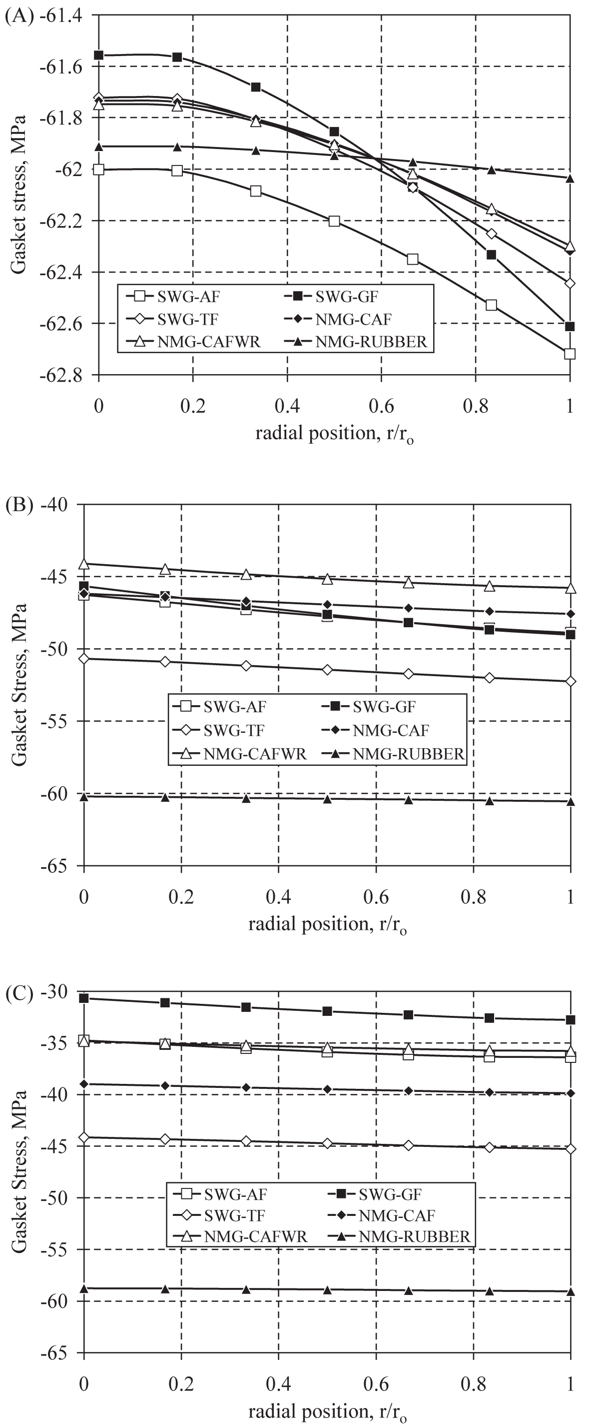

Fig. 9 Stresses across the width of the gasket for a bolt preload of 25 kN at the tension side of the joint (r = radius and r_o = outer radius): (a) bolt-up stage, (b) pressurizing stage, and (c) external bending

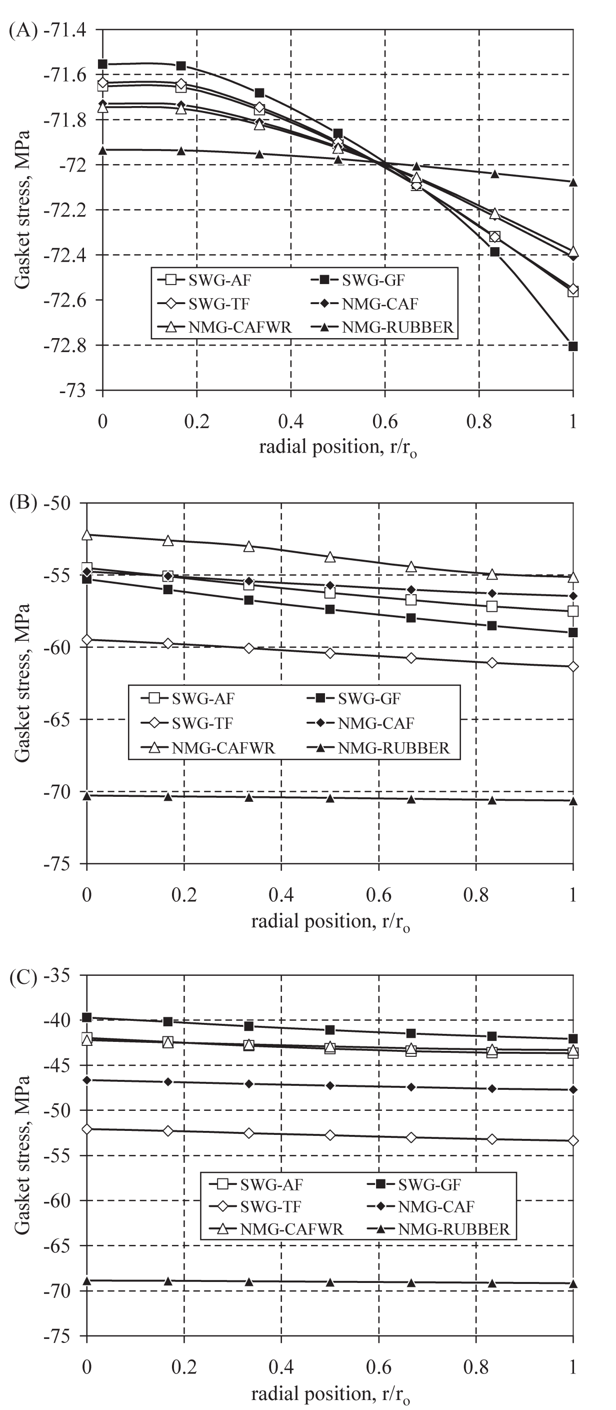

Fig. 10 Stresses across the width of the gasket for a bolt preload of 30 kN at the tension side of the joint (r = radius and r_o = outer radius): (a) bolt-up stage, (b) pressurizing stage, and (c) external bending

the assembly for bolt preloads of 25, 30, and 40 kN for spiral wound and non-metallic gaskets. In order to compare to the sealing performance, a bending moment of 5 kN m was taken at an internal pressure of 10 MPa. The increase in gasket stress towards outer radius is more at the bolting-up stage because of the flange rotation. When the internal pressure is applied, the gasket contact stress is reduced because of the hydrostatic end-force, and in the external bending loading conditions, the difference is further reduced.

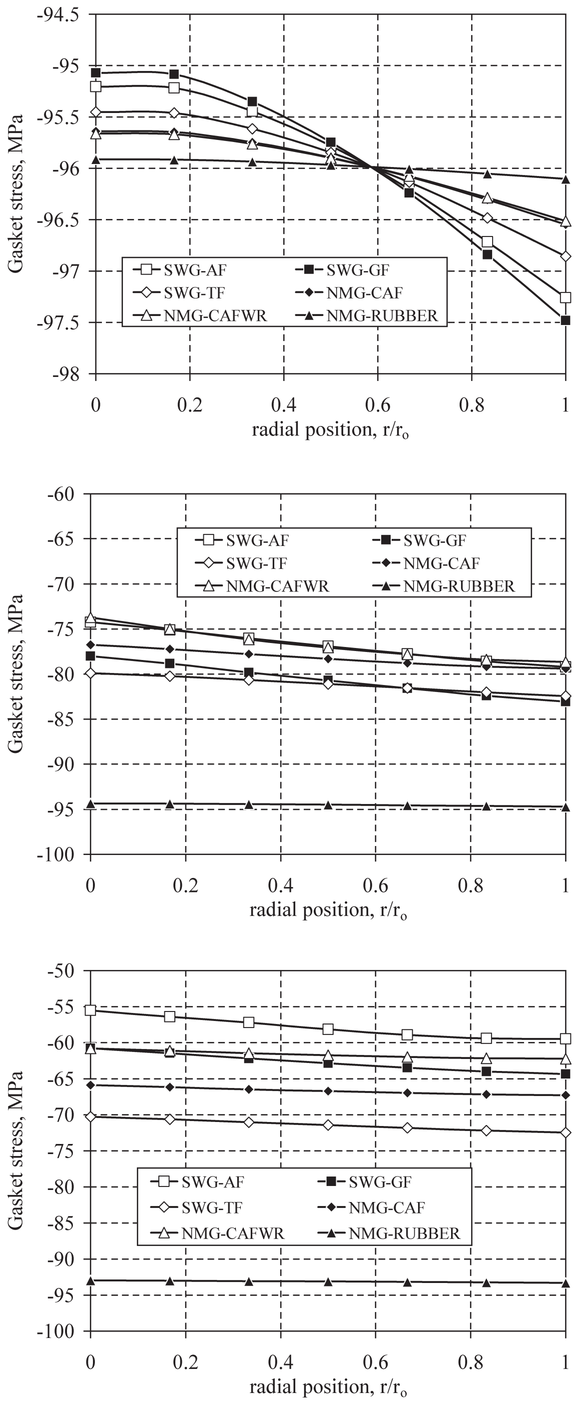

Fig. 11 Stresses across the width of the gasket for a bolt preload of 40 kN at the tension side of the joint (r = radius and r_o = outer radius): (a) bolt-up stage, (b) pressurizing stage, and (c) external bending

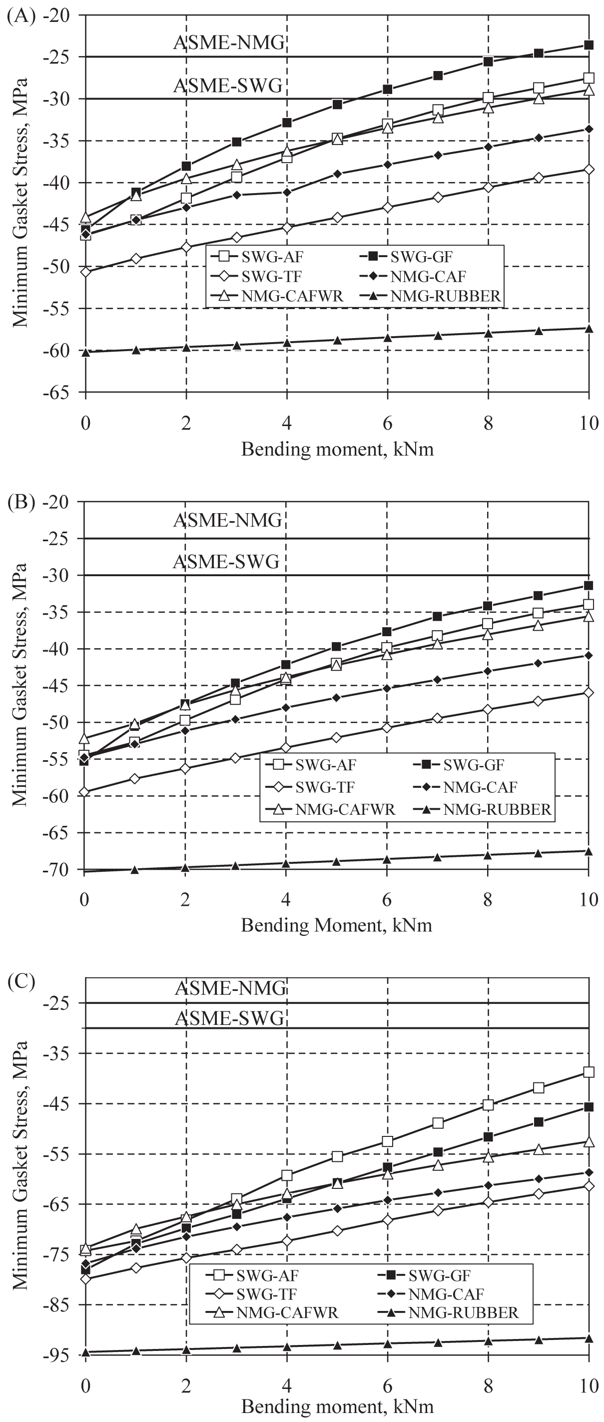

Fig. 12 Effect of bending moment of the pipe flange connection on the minimum gasket stress (internal pressure P = 10 MPa): (a) bolt-preload 25 kN, (b) 30 kN, and (c) 40 kN

The decrease in gasket contact stress for SWG-GF is more when compared with other gaskets. This was very low for rubber gasket and SWG-TF. At higher bolt preloads, the change is further increased in the gaskets other than with elastomeric sealing materials (SWG-TF and NMG-RUBBER).

The changes in the minimum gasket contact stress are shown in Fig. 12. The bolt preload is balanced by the gasket reaction force before internal pressure and bending loads are applied. Under the internal pressure and bending load, there is an increase in the bolt axial force and a decrease in the gasket

Table 1 Allowable bending moment for different gaskets

Gaskets	Bolt preload (kN)	Allowable bending moment (kN m)	Minimum gasket stress (MPa)
Spiral wound gasket-asbestos filled	25	4.8	35
	30	4	44
	40	2.4	66.5
Spiral wound gasket-graphite filled	25	5	31
	30	4.3	41.5
	40	2.4	68.5
Spiral wound gasket-Teflon filled	25	4.3	45
	30	3.6	54
	40	2	75.5
Non-metallic gasket-compressed asbestos filled	25	4.7	39.5
	30	4	48
	40	2.4	70.5
Non-metallic gasket-compressed asbestos filled with wire reinforcement	25	4.9	35
	30	4.3	43.5
	40	2.7	66
Non-metallic gasket-rubber	25	3.4	59
	30	2.6	69.5
	40	1	94

contact stress. From the results it can be observed that the reduction in gasket contact stress is greater for SWG-GF when compared with other gaskets and reaches a minimum stress of 30 MPa at a bending moment of 5 kN m with 25 kN bolt preload, which is the ASME code recommended minimum residual stress for spiral wound gaskets. At a higher bolt preload of 40 kN for the same gasket, the minimum gasket contact stress is higher with corresponding reduction in the bending load-carrying capacity. The reduction in the gasket contact stress is more for materials with high unloading slope curves. Because of this, a similar amount of release in the initial compression at the tension side will result in larger reduction of gasket contact stress for AF and GF spiral wound gaskets and CAF and CAFWR non-metallic gaskets than TF spiral wound gasket and rubber gasket.

3.3 Allowable bending moment

FEA is carried for six types of gaskets to determine allowable bending moment on the flange joint, considering the von-Mises stress at the fillet region (hub and flange ring junction). For the flange under study, the maximum allowable stress is 138 MPa as per ASME SEC II Part A [**16**]. The maximum allowable bending moment with this stress limit for the joint with each gasket is shown in Table 1. Both SWG-AF and SWG-GF show similar bending load-carrying capacity and experience similar minimum gasket contact stress. SWG-AF has a higher minimum gasket contact stress than the above two gaskets, SWG-TF and rubber gasket show higher minimum gasket contact stress with a comparatively good bending load-carrying capacity than other gaskets.

The allowable bending moment depends on the selection of bolt preload and the gasket characteristics. For higher tightness, the bolt preload is set even up to ~90 per cent of the yield stress for non-frequently disassembled joints or 75 per cent of the yield stress for frequently disassembled joints of the bolts. In the present work, the maximum bolt preload of 40 kN (25 per cent of bolt yield stress) is considered. As higher bolt preloads give rise to both bolt and flange stresses, the allowable bending moment on a flange joint may be reduced considerably at higher bolt preloads.

4 CONCLUSIONS

A flange joint is analysed under external bending using FEA by considering the non-linear properties of six types of gaskets. The following conclusions are drawn from the present work.

1. From the FEA results, it can be seen that considerable bending moment can be allowed for a flange joint even when the joint is working under the design internal pressure. This allowable bending moment depends on the bolt preload and the gasket characteristics. If higher tightness is required, the allowable bending moment will reduce significantly because of the higher bolt preloads and the resulting high stresses.
2. The sealing ability of gaskets with elastomeric materials shows good performance against leakage under external bending loads.

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