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A study on the sealing performance of bolted flange joints with gaskets using finite element analysis

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Abstract

Gaskets play an important role in the sealing performance of bolted flange joints, and their behaviour is complex due to nonlinear material properties combined with permanent deformation. The variation of contact stresses due to the rotation of the flange and the material properties of the gasket play important roles in achieving a leak proof joint. In this paper, a three-dimensional finite element analysis (FEA) of bolted flange joints has been carried out by taking experimentally obtained loading and unloading characteristics of the gaskets. Analysis shows that the distribution of contact stress has a more dominant effect on sealing performance than the limit on flange rotation specified by ASME.

Previous Next

Keywords

Bolted flange joints; Gasket characterization; Gasket contact stress; Flange rotation; Axial bolt force

1. Introduction

Flanged joints with gaskets are very common in pressure vessel and piping systems, and are designed mainly for internal pressure. These joints are also used in special applications such as in nuclear reactors and space vehicles. The connection of a fuel duct to a rocket engine is a typical application of these joints in space vehicles. Prevention of fluid leakage is the prime requirement of flanged joints. Many design variables affect joint performance and it is difficult to predict the behaviour of joints in service. A number of design codes and standards, which are principally based on the Taylor–Forge method [1], provide procedures for the design of flanged joints. Even joints designed with codes such as ASME, DIN, JIS and BS experience leakage and this problem is continuously faced by industry. All these codes are based on many simplifications and assumptions and hence may not predict the real behaviour of flanged joints with gaskets.

The complexities associated with the analysis of bolted flange joints with gaskets are due to the nonlinear behaviour of the gasket material combined with permanent deformation. The material undergoes permanent deformation under excessive stresses. The degree of elasticity (stiffness) is a function of the compressive stresses, which act on the gasket during assembly and after it is put into service. It is commonly recognized that gasket stiffness has a predominant effect on the behaviour of the joint because of its relatively low stiffness.

Another inherent problem with bolted joints is flange rotation and contact stresses. These are caused by the bolt pre-load and increase when the joint is subjected to internal pressure. The ASME code has made an attempt to correct this problem by adding a rigidity constraint ‘J’ based on the fixed rotation. This may not be adequate, as the rotation of the flange is not a unique value. Flange rotation causes variable compression across the gasket from the inner radius to the outer radius. Due to the variation in compression, the contact stresses also vary along the radius.

Sawa et al. [2] presented a mathematical model for determining the contact stress distribution in a pipe flange connection based on the theory of elasticity, treating it as an axisymmetric problem. The sealing performance, effective gasket width and the moments acting at the joint were discussed. Sawa et al. [3] extended their 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. In these investigations, neither the actual nonlinearity nor hysteresis of the gasket material was taken into consideration.

In the present work, a finite element (FE) model for finding the contact stresses in a gasket has been developed. Nonlinearity and hysteresis of the gasket under various loading and operating conditions are taken into account. Experiments have been carried out for finding the loading and unloading characteristics of the gasket materials, which are in turn used in the FEA. The influence of flange rotations on the sealing performance of different gasket models with varying loading and operating conditions has been studied. The increase in the axial bolt force when the joint is subjected to an internal pressure has also been analyzed. The distribution of contact stress on the gasket for different loading conditions has been studied. In the present analysis weld-neck (WN) raised-face (80 mm NPS, Class-600, ASME/ANSI B 16.5) flanges with selected spiral-wound ring gaskets are considered with material properties at room temperature.

2. Gasketed joint configuration and material properties

2.1. Geometry of the flange, gasket and bolt

All standard flanges are provided with a variety of machined faces. Raised-face flanges are preferable over full-face flanges for medium service conditions, and ring gaskets with small contact area reduce the bolt pre-load required to compress a gasket. Figs. 1(a) and (b) show the dimensions of the pipe flange (80 mm NPS, weld neck, Class 600, ASME/ANSI B 16.5) [5] and the spiral wound gasket used in the finite element analysis (FEA). M20 size bolts are considered in the analysis.

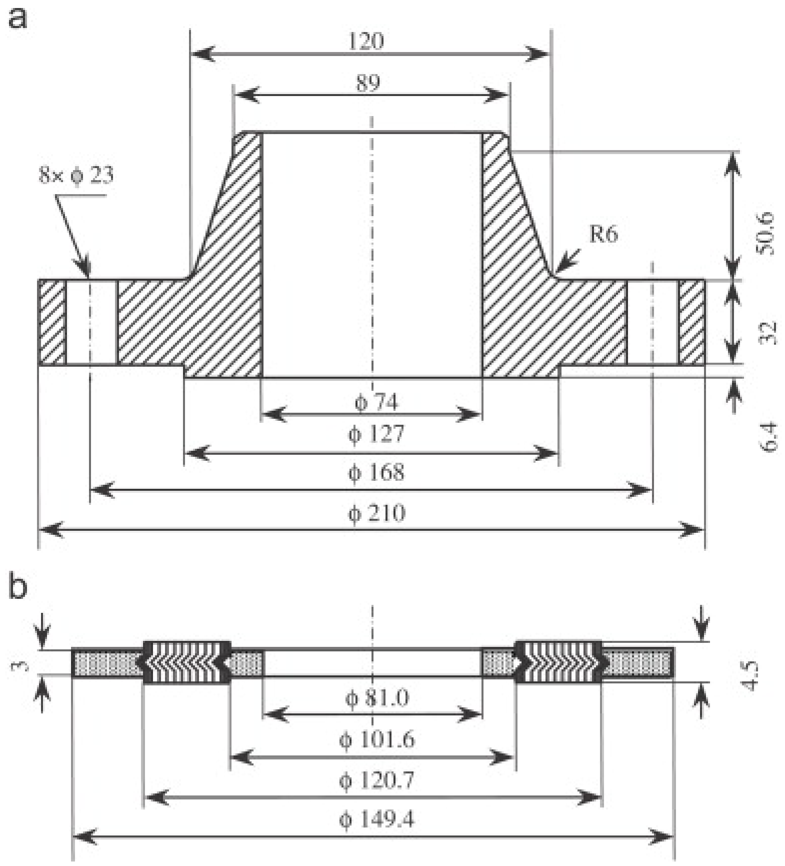

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Fig. 1. Dimensions of the flange and gasket used in the FEA: (a) flange and (b) spiral-wound gasket.

2.2. Material properties of the flange and the bolt

The flange and the bolt material properties are assumed to be homogenous, isotropic and linearly elastic. Materials for flanges and bolts are chosen as forged carbon steel (A105, Young's modulus, E=195 GPa, Poisson's ratio, v=0.3) and chromium steel (A193-B7, E=203 GPa, v=0.3).

2.3. Gasket characterization

Gaskets are often multilayered materials, exhibiting nonlinear behaviour in loading and unloading conditions. The modulus of elasticity for the bolting-up or compression (loading) stage is different from the decompression (unloading) stage of the gasket due to the internal pressure. When the gasket is decompressed, it shows strong hysteresis which is nonlinear and leads to permanent deformation usually confined to through-thickness. The contribution to the stiffness from membrane (in plane) and transverse shear are much smaller and hence neglected. As input data for FEA, each reloading curve is assumed to be identical with the unloading one for simplicity.

The FE program ANSYS offers a number of elements to model gaskets. These elements consider geometric and material nonlinearities and membrane and transverse shear are neglected. Thus the pressure–versus–closure behaviour can be directly applied to characterize the gasket material. A load compressive mechanical test (LCMT) [6], [7] has been carried out for finding the mechanical characteristics of gasket material which are in turn used in the FEA. ANSYS software has a provision to input the LCMT data points. Fig. 2 shows the material properties for different spiral-wound gaskets. The nonlinear compression behaviour with two different unloading curves for spiral-wound gaskets is considered to predict the nonlinear behaviour of the gasket in the analysis.

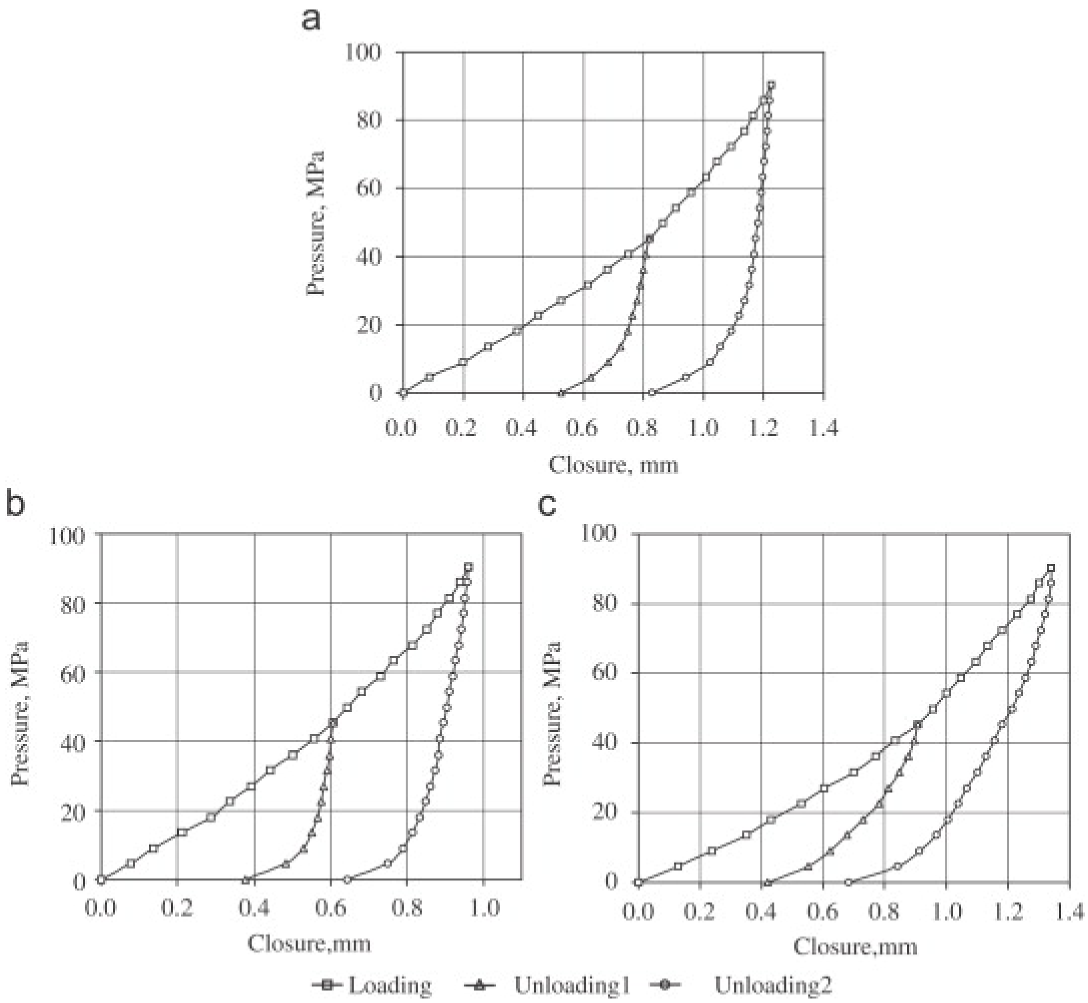

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Fig. 2. Characteristics of different types of spiral-wound gaskets obtained experimentally: (a) asbestos filled (b) graphite filled and (c) PTFE filled.

3. Finite element modelling

3.1. Discretization

A three-dimensional FE model has been developed for bolted flange connections with gaskets using ANSYS [8]. These joints possess geometric characteristics which are symmetrical about an axis. They can be defined in terms of a primary segment which is repeated at equally spaced intervals about the axis of symmetry. Taking into account the rotational symmetry, a quarter (90° segment) model of the joint is considered in the analysis for an eight bolt model. Similarly, one-sixth (60°) and one-fifth (72°) segments are considered for six and 10 bolt models, respectively. Fig. 3 shows the mesh division used in the FE analysis for a bolted flange joint with a spiral-wound gasket and eight bolts (1/4 model). Solid elements (SOLID185) [9] are used to model the geometry of the flange.

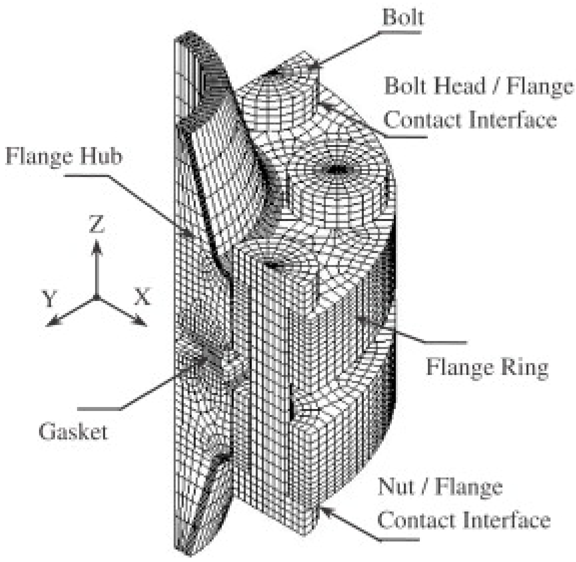

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Fig. 3. Finite element mesh of bolted flange joint with spiral-wound gasket with eight bolts.

3.2. Modeling of the gasket

The gasket is modeled with interface elements (INTER195) [9]. These elements are based on the relative deformation of the top and bottom surfaces and offer a direct means to quantify the through-thickness deformation of the gasket joints. An element midplane is created by averaging the coordinates of node pairs from the bottom and top surfaces of the elements. The stress normal to the midsurface of an element in the gasket layer is the same as the gasket pressure. As in-plane deformation and transverse shear are neglected, there is only one component which is normal to the gasket. Thus complete gasket behaviour (through-thickness deformation of the gasket) is characterized by a pressure–versus–closure (relative displacement of top and bottom gasket surfaces) relationship.

3.3. Pretension in the bolts

Pretension elements (PRETS179) [9] shown in Fig. 4 are used to model the load in a bolted joint due to tightening at the time of assembly. All pretension elements will have a common pretension node (K). This node is the third node for the pretension element whereas nodes I and J are on the sectioned mid-surface of the bolt. Sides A and B on the pretension section are connected by one or more pretension elements, one for each coincident node pair. A pretension node (K) is used to control and monitor the total tension loads. In the first stage (bolting-up), load was applied to the pretension node as a force. The force “locks” on the second stage (pressurized), allowing additional loads. The effect of the initial load is preserved as a displacement after it is locked. The bolt and nut threads are modeled as part of an unthreaded shank with the minor diameter of the bolts, and the bolt head and nut are assumed cylindrical to avoid meshing difficulties.

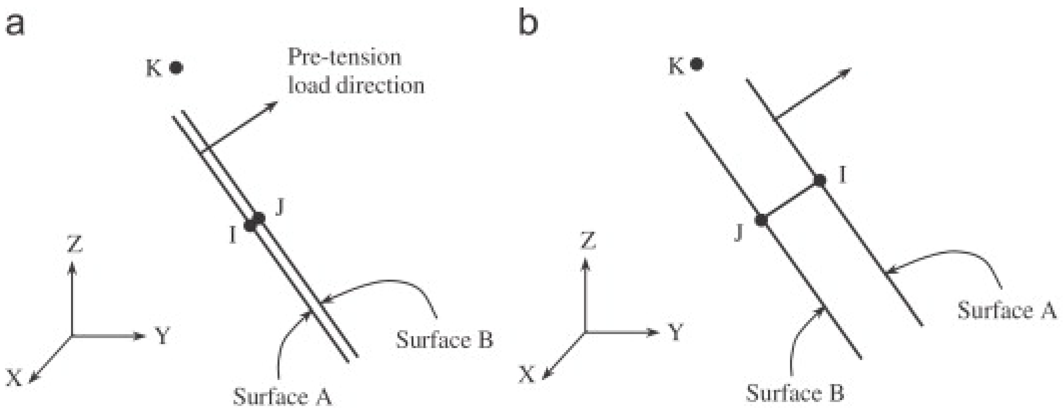

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Fig. 4. PRETS179 element: (a) before adjustment and (b) after adjustment.

3.4. Contact interfaces

In the present model, the bolt and nut are treated as a single entity and the flange ring is a separate entity. Since the behaviour of these two are different in terms of load–deformation characteristics, surface-to-surface contact elements are used to model contact interfaces to carry out an analysis as a three-dimensional contact problem. As the members (flanges, gasket) and bolts are deformable both can be treated as in the flexible-to-flexible category. Contact has been considered between the bolt head and flange ring interface as well as the nut face and flange ring interface. The option with no slipping friction is used to simulate interaction of the contact surfaces. Since the flange ring is stiffer than the bolt, it is modeled as a target surface (TARGE170) [9] and the bolt head/nut face is modeled as the contact surface (CONTA173) [9]. The contact elements themselves overlay the solid elements describing the boundary of a deformable body. These elements connect the nodes at the interfaces or gap with a high stiffness when the interface is in contact and with very small or zero stiffness when the interface separates.

After carrying out convergence studies, the model is discretized with 24,396 solid, 192 interface, 160 pretension and 612 contact elements and a total of 32,642 nodes.

3.5. Loading and boundary conditions

FE analysis consists of bolt pre-loading and pressure loading conditions.

3.5.1. Bolt pre-loading condition

A gasketed flange in the bolting-up stage is analysed to obtain the initial stresses and deformations in the flanges and gasket due to the clamping forces during assembly. Due to rotational symmetry, the displacements (boundary conditions) and loads are applied on a single segment in the analysis. For the model created in a cylindrical coordinate system (r, θ, z), the circumferential displacements (U_θ=0) are constrained in the 0° and 90° planes for an eight-bolt model. Similarly, circumferential displacements in the 60° and 72° planes are arrested along with the 0° plane for six and 10 bolt models, respectively. In the bolting-up stage, only the bolt pre-load (*F*) has been applied uniformly on the pretension element section through the pretension node.

3.5.2. Pressure loading condition

The gasketed flange is analysed by considering internal pressure in addition to the bolt pre-load. Due to the internal pressure, a hydrostatic end force and pressure force are induced on the joint. The hydrostatic end force exerted on the closed end of a pipe system is calculated based on the inner diameter of the pipe and the total hydrostatic end force is calculated based on the inner diameter of the gasket. The pressure force is the difference between the total hydrostatic end force and the hydrostatic end force on the area inside the flange. The hydrostatic end force has been applied uniformly in the axial direction at one end of the pipe and the other end has been fixed in the axial direction (U_z=0). The force due to internal pressure has been applied uniformly in the axial direction on the inner faces of both the flange surfaces. The same boundary conditions of constraining the circumferential displacements (U_θ=0) as applied in the bolting-up condition are maintained in this stage also.

Twenty load steps for each of the bolting-up and pressurized stages have been considered. It is found that 20 load steps adequately represent the nonlinear behaviour and give convergence of the stress values in both stages.

4. Flange rotation with the ASME code

Angular rotation of a flange under the influence of bolt pre-load and reaction forces is called flange rotation. This is measured with respect to the centre of the cross-section of the flange. The ASME code has a rigidity index ‘J’ to check the flange rotation. The rigidity factor for integral flanges (weld-neck) is equivalent to a flange rotation limit of 0.3°. This may not be adequate, as the rotation of the flange is not a unique value as it changes with internal pressure.

When the gasket joint is subjected to internal pressure, the stress on the gasket is reduced as the pressure tries to move the joint members apart. As stress is reduced, the gasket expands due to its elastic nature. However, under the loads used in practical conditions it does not recover to the initial geometry. Experience has shown that the reduced stress on the gasket under pressurized conditions is as important as the initial stress. When the gasket stress is too small, the joint may leak or the gasket may be blown out of the joint. The ASME code rules define this reduced stress (referred to as residual stress) in terms of a “maintenance” (“mollifying”) gasket factor “*m*” which is dimensionless. The desired minimum residual stress is said to be the product of “*m*” and the contained pressure (*P*). The code also specifies a “*y*” factor which is the initial gasket stress or surface pressure required to pre-load or seal the gasket to prevent leaks in the joint as the system is pressurized [6], [7].

5. FEA results

5.1. Flange rotation

When bolts are tightened to achieve the desired surface pressure, the sealing material (gaskets) is deformed. Due to the eccentricity of the bolt pre-load, the hydrostatic end force exerted on the flanges, the seating force (gasket reaction load) and the internal pressure on the inner faces of the flange, a bending load acts on the flange, thereby causing a rotation of the flange. The gasket is subjected to a non-uniform contact stress due to this rotation of the flange and difficulties occur in joint sealing. Large rotation may cause buckling of the inner windings or separation of the sealing element in the case of spiral-wound gaskets due to possible contact of the flange with the raised face outside diameter which will act as a pivot point.

Fig. 5 shows the rotation of the flange in terms of the axial displacements at the bottom surface of the ring portion along the radial direction for an asbestos filled spiral-wound gasket model for a bolt pre-load F=30 kN with different internal pressures (P=0, 3, 5 and 10 MPa). It is observed that axial displacements vary non-linearly in the radial direction. While calculating the flange rotations, only the axial displacements at the inner and outer radius of the flange are considered for simplicity. Fig. 6 shows the calculated flange rotation with an asbestos filled spiral wound gasket at different bolt pre-loads and internal pressures. The rotation is large when the joint is subjected to internal pressure. Table 1 shows the flange rotation for different spiral-wound gaskets with eight bolts. It is observed that the calculated flange rotations are well below 0.3°, as specified by ASME [10].

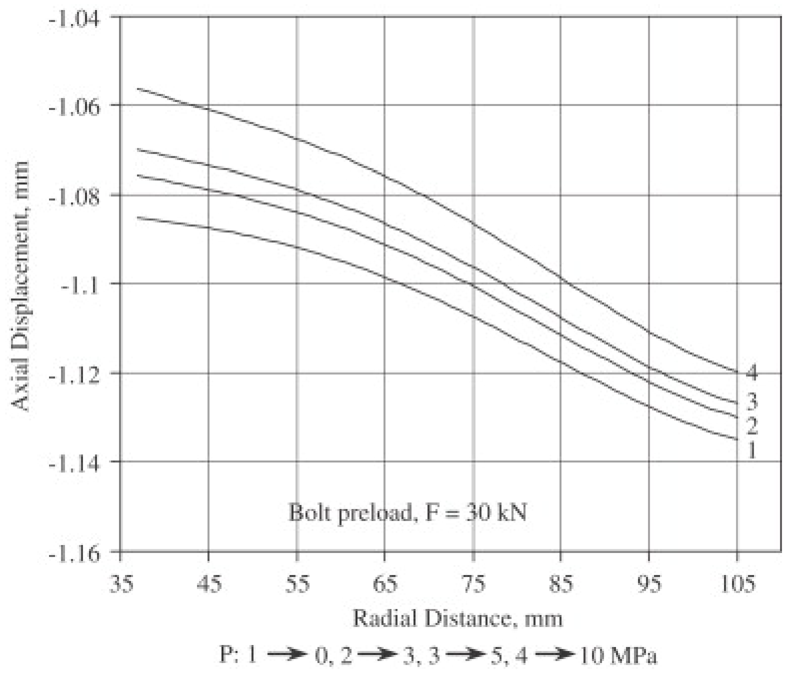

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Fig. 5. Flange rotation in terms of axial displacements along the radial direction.

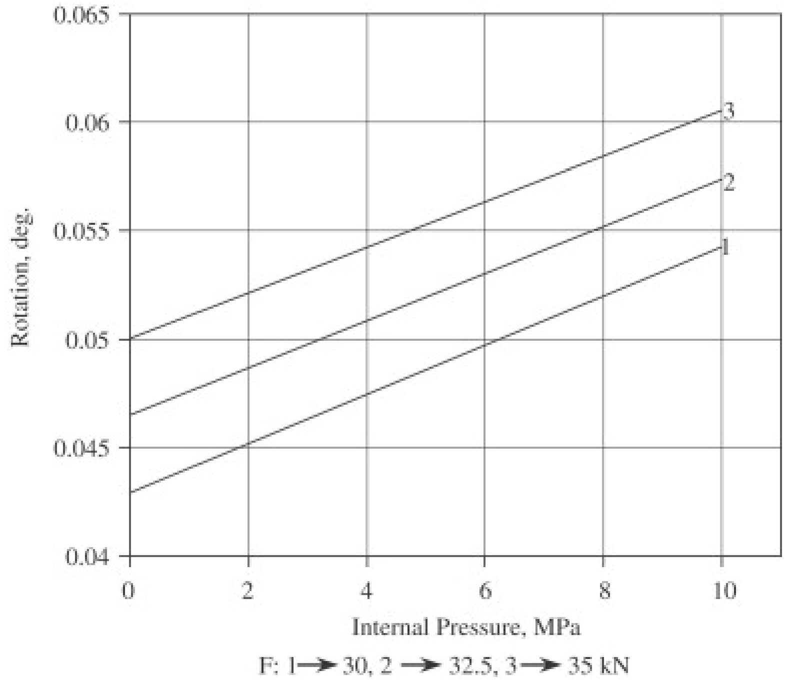

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Fig. 6. Variation of flange rotation with internal pressure.

Table 1. Flange rotation for different spiral-wound gaskets with eight bolts

Spiral-wound gaskets	Bolt preload, *F* (kN)	Flange rotation (deg)			
		Internal pressure, *P* (MPa)			
		0	3	5	10
Asbestos filled	30.0	0.042734	0.046402	0.048715	0.054146
	32.5	0.046286	0.049829	0.052054	0.057253
	35.0	0.049850	0.053294	0.055454	0.060463
Graphite filled	30.0	0.042723	0.045741	0.047826	0.053238
	32.5	0.046290	0.049251	0.051336	0.056684
	35.0	0.049844	0.052781	0.054867	0.060143
PTFE filled	30.0	0.042734	0.045830	0.048627	0.055818
	32.5	0.046295	0.049194	0.051901	0.058851
	35.0	0.049837	0.052577	0.055205	0.061906

5.2. Variation of axial bolt force

When the joint is tightened with a bolt pre-load, an initial tensile force in the bolt and an initial compressive force in the connected members (flanges and gasket) are introduced.

When an internal pressure is applied to the joint, the bolt is subjected to increased tensile load and the connected members are subjected to a reduced compressive load. The variation of axial bolt force depends on the relative stiffness of the bolts and the connected members. Gaskets are provided for preventing leakage and are relatively soft compared to other joint members. As a result, the stiffness of a gasketed joint is nearly equal to the stiffness of the gasket, and the variation of axial bolt force is influenced by the stiffness of the gasket.

Fig. 7 shows the variation of axial bolt force with internal pressure for different spiral-wound gaskets with eight bolts for an initial bolt pre-load of F=30 kN. It is observed that the increase in axial bolt force is highest in the PTFE-filled spiral-wound gasket due to its lower stiffness and lowest for the graphite filled spiral wound gasket. Table 2 shows the increase in axial bolt force for various spiral wound gaskets with different loading and operating conditions.

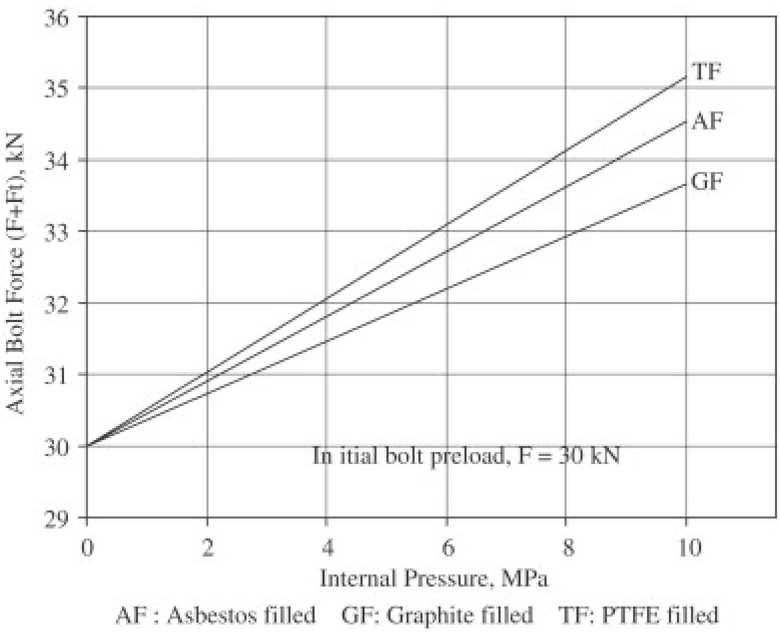

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Fig. 7. Increase in axial bolt force with internal pressure for different spiral-wound gaskets.

Table 2. Increase in axial bolt force for different gaskets for eight bolt model

Spiral wound gaskets	Bolt preload, F (kN)	Axial bolt force (F+F_t), (kN)		
		Internal pressure, P (MPa)		
		3	5	10
Asbestos filled	30.0	31.48	32.38	34.37
	32.5	33.90	34.73	36.56
	35.0	36.33	37.12	38.81

Spiral wound gaskets	Bolt preload, *F* (kN)	Axial bolt force (*F*+*F*), (kN)		
		Internal pressure, *P* (MPa)		
		3	5	10
Graphite filled	30.0	31.03	31.76	33.74
	32.5	33.49	34.23	36.16
	35.0	35.97	36.71	38.60
PTFE filled	30.0	31.08	32.31	35.53
	32.5	33.44	34.61	37.66
	35.0	35.82	36.94	39.82

5.3. Contact stress on gaskets

The results of FEA carried out with an asbestos filled spiral-wound gasket and eight bolts for internal pressures of 0, 5 and 10 MPa and 30 kN bolt pre-load are shown in Fig 8. The figure shows the effect of bolt pre-load and internal pressure on the gasket contact stress distribution in the radial direction. It is observed that the contact stress is compressive and decreases as the internal pressure increases. The reduction in the gasket contact stress for lower internal pressures (5 MPa) is found to be 15–21% and 30–40% for higher internal pressures (10 MPa). It is also found that the contact stress varies from the inner to the outer radius of the gasket with 1–4% higher values at the outer radius. This is due to the flange rotation discussed in Section 5.1. The contact stresses decrease by 8–10% when the system is subjected to an internal pressure of 5 MPa.

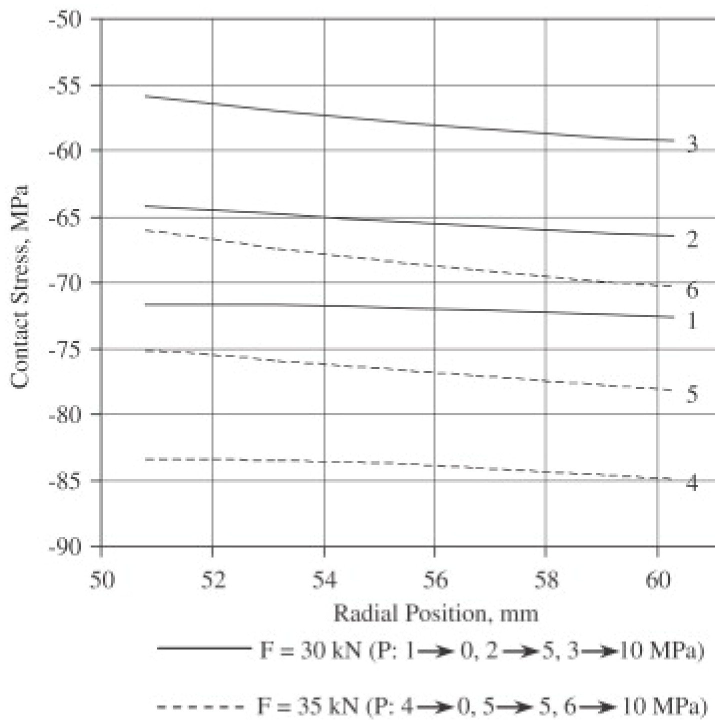

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Fig. 8. Variation of the contact stress on an asbestos filled spiral-wound gasket in the radial direction.

Fig. 9 shows a comparison of the contact stress distribution on the gasket in the radial direction under the bolt pre-load (F=30 kN) and pressurized (P=5 MPa) condition for different spiral wound gaskets. It is observed that the variation of contact stress on the gasket from the inner to the outer diameter is higher in graphite filled (GF); followed by asbestos (AF) and PTFE filled (TF) spiral-wound gaskets. Table 3 shows the contact stress on various spiral-wound gaskets at the inner and outer radial positions for different bolt pre-loads and internal pressures.

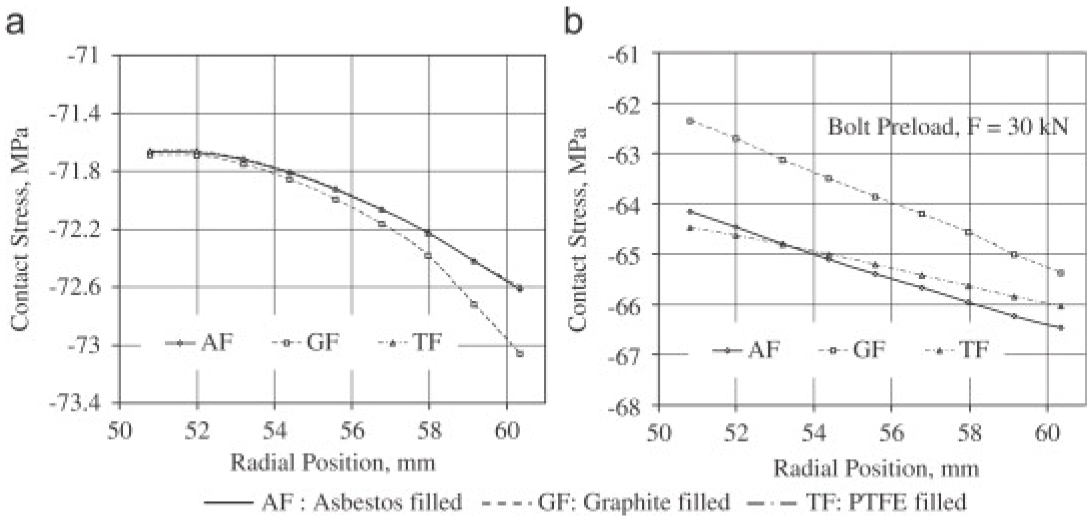

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Fig. 9. Distribution of contact stress on different spiral-wound gaskets: (a) bolt preload, F=30 kN and (b) internal pressure, P=5 MPa.

Table 3. Contact stress variation on different spiral-wound gaskets for eight bolt model

Spiral wound gasket	Bolt preload, F (kN)	Radial position of gasket	Contact stress on gasket (MPa)			
			Internal pressure (MPa)			
			0	3	5	10
Asbestos filled	30.0	Inner	−71.67	−67.41	−64.16	−55.91
		Outer	−72.61	−69.00	−66.46	−59.27
	32.5	Inner	−77.39	−72.88	−69.46	−60.74
		Outer	−79.06	−75.25	−72.56	−64.95
	35.0	Inner	−83.45	−78.70	−75.19	−66.06
		Outer	−84.98	−81.00	−78.19	−70.27
Graphite filled	30.0	Inner	−71.68	−65.94	−62.36	−53.88

Spiral wound gasket	Bolt preload, *F* (kN)	Radial position of gasket	Contact stress on gasket (MPa)			
			Internal pressure (MPa)			
			0	3	5	10
		Outer	−73.0	−68.31	−65.37	−58.24
	32.5	Inner	−77.50	−71.82	−68.22	−59.57
		Outer	−78.89	−74.17	−71.2	−64.01
	35.0	Inner	−83.30	−77.57	−73.97	−65.14
		Outer	−85.23	−80.47	−77.53	−70.23
PTFE filled	30.0	Inner	−71.66	−66.63	−64.47	−59.61
		Outer	−72.60	−67.90	−66.04	−61.65
	32.5	Inner	−77.63	−72.24	−69.92	−64.57
		Outer	−78.68	−73.63	−71.62	−66.85
	35.0	Inner	−83.32	−77.60	−75.14	−69.42
		Outer	−85.26	−79.89	−77.77	−72.57

5.4. Sealing performance

The contact stresses have been considered at the inner radius of the gasket, as these values are lower than those at the outer radius and a leak may occur at a lower stress region. Fig. 10(a) shows the variation of contact stress with internal pressure for an asbestos filled spiral-wound gasket with eight bolts at different bolt pre-loads. The minimum residual stress for different internal pressures as per ASME is also plotted in this figure. The intersection of the contact stress line and residual stress (ASME) line gives the limiting internal pressure for a given bolt pre-load.

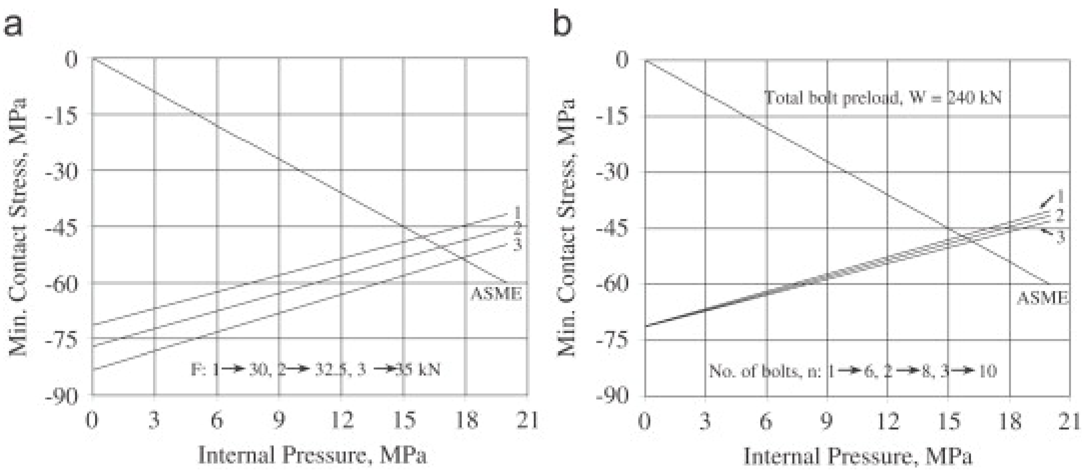

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Fig. 10. Variation of the minimum contact stress at the inner radius of an asbestos filled spiral-wound gasket with internal pressure: (a) eight bolts model and (b) different number of bolts.

For example, the proposed '*m*' and '*y*' gasket factors [10] for the asbestos filled spiral-wound gasket are 3 and 68.95 MPa (10000 psi), respectively. If the bolt pre-load is F=30 kN for an eight bolt model (Fig. 10(a)) the intersection point with the ASME residual stress line, will show that leakage would occur for a minimum contact stress of 47.70 MPa (3×15.9) when the internal pressure reaches 15.9 MPa. Fig. 10(b) shows the variation of contact stress with internal pressure for six, eight and 10 bolt models. It is observed that leakage will occur at a lower internal pressure and contact stress when the number of bolts is less for a given magnitude of total bolt pre-load (W=30×8=240 kN).

Table 4 shows maximum allowable internal pressures to maintain the minimum contact stress to avoid leakage for six, eight and 10 bolt models, respectively, on various spiral wound gaskets with different bolt pre-loads. These tables show the corresponding flange rotation for different loading conditions and the values are well below 0.3° as specified by the ASME code [10]. For comparison, the bolt pre-load obtained from the ASME code is also given at the corresponding internal pressure. The values given by the ASME code are based on empirical relations which include effective gasket seating width (b) in two different regions, namely basic gasket seating width $b_0 \leqslant 6.35$ and $b_0 > 6.35$ mm. As per ASME, the basic gasket seating width bo depends on a set of combinations of flanges, flange facings and gaskets whereas the present approach is applicable for any range of flanges, gasket types, sizes and bolt pre-loads.

Table 4. Allowable internal pressures for different spiral-wound gaskets

Spiral-wound gaskets	Total bolt preload, *W* (kN)	Internal pressure (MPa)	Gasket contact stress (MPa)	Flange rotation (deg)	ASME total bolt preload (kN)
(a) Six bolt model					
Asbestos filled	240	15.67	−47.02	0.061339	262.42
	260	16.60	−49.80	0.064400	271.59
	280	17.60	−52.80	0.069199	281.44
Graphite filled	240	15.24	−45.72	0.060362	258.19
	260	16.49	−49.48	0.063794	270.50
	280	17.65	−52.95	0.068651	281.93
PTFE filled	240	17.44	−52.33	0.067321	279.86
	260	18.57	−55.72	0.072505	291.00
	280	19.64	−58.93	0.075634	301.54

Spiral-wound gaskets	Total bolt preload, *W* (kN)	Internal pressure (MPa)	Gasket contact stress (MPa)	Flange rotation (deg)	ASME total bolt preload (kN)
(b) Eight bolt model					
Asbestos filled	240	15.90	−47.70	0.061880	264.69
	260	16.82	−50.46	0.064802	273.75
	280	17.81	−53.43	0.069491	283.51
Graphite filled	240	15.43	−46.28	0.059273	260.06
	260	16.70	−50.09	0.064270	272.57
	280	17.87	−53.61	0.069057	284.10
PTFE filled	240	17.72	−53.17	0.068183	282.62
	260	18.87	−56.60	0.071513	293.95
	280	19.95	−59.85	0.076428	304.60
(c) Ten bolt model					
Asbestos filled	240	16.18	−48.55	0.062321	267.45
	260	17.13	−51.38	0.065341	276.81
	280	18.15	−54.45	0.070065	286.86
Graphite filled	240	15.68	−47.04	0.059648	262.52
	260	16.98	−50.93	0.064673	275.33
	280	18.17	−54.52	0.069489	287.06
PTFE filled	240	17.97	−53.90	0.066855	285.09
	260	19.13	−57.39	0.071983	296.52
	280	20.23	−60.69	0.077022	307.36

6. Conclusions

Three types of gaskets namely AF, GF and TF spiral-wound gaskets were considered to determine the sealing performance of these gaskets. The distribution of gasket contact stress is observed to be non-uniform across the gasket width with higher values at the outer radius. The difference in the contact stress between the inner and outer radii depends on the gasket type and flange flexibility. The ASME code does not consider these factors. From the studies carried out in the present work, the following conclusions are made:

i. Results show that leakage in the flanged joint may occur even if the flange rotation is well below the value of 0.3° specified by the ASME code. It is important to maintain the minimum contact stresses to avoid leakage. This influence the contact stress distribution in the radial direction.

ii. The increase in axial bolt force with increase in internal pressure is found to be highest in TF spiral-wound gaskets (due to low stiffness) and the least for GF spiral-wound gaskets. The gasket characteristics play a significant role in determining the pre-load of the bolt.

iii. Variation in contact stress distribution in the radial direction is found to be highest in GF spiral-wound gaskets and the least for TF spiral-wound gaskets. The FE method is very useful during the design process for the selection of gasket, pretension of bolts and number of bolts.

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