**ABSTRACT** Gasketed bolted flange pipe joints are always found prone to leakage during operating conditions. Therefore performance of a gasketed flange joint is very much dependent on the proper joint assembly with proper gasket, proper gasket seating stress and proper preloading in the bolts of a joint. For a gasketed flange joint, the two main concerns are the joint strength and the sealing capability. To investigate these, a detailed three dimensional nonlinear finite element analysis of a gasketed joint is carried out using gasket as a solid plate. Bolt scatter, bolt bending and bolt relaxation are concluded the main factors affecting the joint’s performance. In addition, the importance of proper bolt tightening sequence, number of passes on joint performance are also presented. Summarizing, a dynamic mode in a gasketed joint is concluded, which is the main reason for its failure.

**INTRODUCTION**

Gasketed pipe flange joints are widely used in industry to connect pipe to pipe or pipe to equipment. These are used in a wide variety of different applications from water supply to high pressure and high temperature applications. In gasketed pipe joints, problem of bolt scatter, sealing and joint relaxation is observed and it is difficult to achieve uniform bolt stress during joint assembly as dynamic mode-of-load governs in the gasketed joint [1,2]; hence resulting in poor sealing and joint strength. Some experimental and numerical investigations [3-4] are performed to estimate bolt preload scatter due to the elastic interaction in the process of successive bolt tightening. These investigations are limited to the linear elastic material modeling. In addition, these do not consider bolt bending behaviour, flange rotation and flange stress variation. A detailed experimental studies are performed by Abid [1,2] to highlight bolt-bending behaviour, flange stress variation and flange rotation with special emphasis on joint strength and sealing capabilities. In the present study; a detailed three dimensional nonlinear finite element analysis of a gasketed joint is carried out using gasket configuration as a solid plate. This is considered as gasket sealing portion is considered to be completely compressed at the seating stress load applied during joint tightening. Bolt scatter, bolt bending and bolt relaxation are concluded the main factors affecting the joint’s performance. In addition, importance of proper bolt tightening sequence, number of passes on joint performance are also presented. Summarizing, a dynamic mode in a gasketed joint is concluded, which is the main reason for its failure. A Flange joint of four-inch 900° class is used in the present study.
FINITE ELEMENT ANALYSIS

Modeling

Abid et al [5-6] investigated joint strength and sealing capability under combined loading for an axi-symmetric 3-D model where the preload of each bolt was the same using a solid plate gasket. An angular portion (22.5° rotation of main profile or 1/16th part) of flange was modeled with a bolt hole at required position and then reflected symmetrically to complete 360 degree model. Gasket is modeled by rotating an area pattern about y-axis through 360 degrees in 16 numbers of volumes; it is possible to model half gasket with respect to thickness due to symmetry of geometry and loading conditions. Bolt is modeled by rotating an area pattern about axis defined by key points through 360 degrees in 4 numbers of volumes and then remaining 7 bolts are generated by virtue of symmetry in z-axis; the objective pipe flange connection is tightened by eight bolts. Half portion of bolt was modeled due to plane symmetry of bolt. Only a small portion of pipe is modeled to reduce computational time. The resulted flanged joint model is shown in Figure 1a. Commercial FEA software ANSYS [7] is used during the analysis. A four inch 900# class, ANSI flange joint is selected for this study.

ELEMENT SELECTION

Structural Solid Elements

Eight-nodded structural SOLID45 lower order isoperimetric element is used for modeling of flange, bolt, solid gasket and pipe.

Contact Elements:

Three-dimensional ‘surface-to-surface’ CONTA174 contact elements, in combination with TARGE170 target elements are used between the flange face and gasket, bolt shank and flange hole, the top of the flange and the bottom of the bolt, to simulate contact distribution. No friction was employed between any of the surfaces, since the forces normal to the contact surfaces would be far greater than the shear forces, therefore, this is a reasonable assumption.

Meshing

Before volume mesh generation area mesh is created on one side of the flange, bolt and solid plate gasket by specified number of divisions and space ratio for each line. Hub-flange fillet and raised face areas of flange are fine meshed due to high stress concentration. The areas of bolt head which makes contact with flange top is meshed with small size elements for fine mesh. Unmeshed volume of flange is then filled with elements by sweeping the mesh from adjacent area through the volume. Complete 360-degree flange model mesh is then generated from the angular portion of flange by symmetry reflection for 3-D finite elements as shown in Figure 1b. For bolt and solid plate gasket volumetric mesh is generated by sweeping the mesh from an adjacent area through the volume [Figure 1c,1d].
Material Properties

Allowable stresses and material properties for flange, pipe, and bolt and symmetry plate [8] are given in Table 1. An elastoplastic material model is used consists of two sections each having a linear gradient. The first section, which models the elastic material, is valid until the yield stress is reached. The gradient of this section is the Young’s Modulus of Elasticity. The second section which functions beyond the yield stress, and models the behavior of the plastic material, has a gradient of the plastic tangent modulus, which for this study was 10% of the Young’s Modulus of Elasticity previously [2].

Table 1: Material properties

<table>
<thead>
<tr>
<th>Parts</th>
<th>As per code</th>
<th>E (MPa)</th>
<th>ν</th>
<th>Allowable Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flange/ Pipe</td>
<td>ASTM A350 LF2</td>
<td>173058</td>
<td>0.3</td>
<td>248.2 (2/3rd σy)</td>
</tr>
<tr>
<td>Bolt</td>
<td>ASTM SA193 B7</td>
<td>168922</td>
<td>0.3</td>
<td>723.9</td>
</tr>
<tr>
<td>Gasket (Solid plate)</td>
<td>ASTM A182</td>
<td>164095</td>
<td>0.3</td>
<td>206.8 (2/3rd σy)</td>
</tr>
</tbody>
</table>

Contact generation

Between flange, bolt head and gasket

To define contact pair between flange and bolt head, flange face areas are taken as target surface, while bottom areas of the bolt head are taken as contact surface, same real constant number are assigned to both the target and contact elements [Figure 2(a)]. For contact pair generation between flange bottom surface and gasket [Figure 2(b)], flange bottom surface areas are taken as target surface while gasket top surface areas are taken as contact surface. The contact condition is applied and the friction is taken into consideration at the interfaces between the flange and the gasket, the chosen friction coefficient is varied from 0.1 to 0.2 and its effects on interface stress distributions are examined. As a result, the effect of friction coefficient is found to be very small.
Between bolt shank and bolt hole

To prevent rigid motion of flange during bolt up, contact is defined between bolt shanks and the bolt holes in flange [Figure 2(c)], as there is a gap present between the two surfaces, so contact surface offset (CNOF) is set to adjust initial contact conditions before contact generation, a positive value 1.61mm (gap between the two surfaces) is specified to offset the entire contact surface towards the target surface, while a negative value is used to offset the contact surface away from the target surface. In this case, as flange undergoes rigid motion and penetrates into bolt shank, so bolt hole areas in flange are set as contact surface, while bolt shank areas are set to be target surface.

![Figure 2: Contact pair between (a) flange face and bottom of bolt, (b) Flange bottom and Gasket, (c) bolt shank and bolthole](image)

Boundary Conditions

The flange and the gasket are free to move in the axial and the radial direction, providing flange rotation and the exact behaviour of stress in flange, bolt, and gasket. Symmetry conditions are applied to gasket lower portion. Bolts are constrained in radial and tangential direction by taking UX, UZ equal to zero on the neutral axis line of bolt. An axial displacement is applied on the bottom area of the bolt shank to get required pre-stress [Figure 3(a)].

Bolt preloading

To ensure a proper pre-load on the joint, the sequence in which bolts are tightened during a pass has a considerable importance in flange joint tightening as the joint relaxation mostly depends upon this factor. Bolts are tightened as per sequence-1 during the first four passes and as per sequence-2 during the last pass.

In the present work, following two sequences are used
Table 2: Target Stress calculated for each pass

<table>
<thead>
<tr>
<th>Applied Torque (Nm)</th>
<th>Bolt preload (KN)</th>
<th>Target Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>210</td>
<td>37</td>
<td>57</td>
</tr>
<tr>
<td>310</td>
<td>54</td>
<td>86</td>
</tr>
<tr>
<td>400</td>
<td>70</td>
<td>112</td>
</tr>
<tr>
<td>505</td>
<td>89</td>
<td>145</td>
</tr>
</tbody>
</table>

- Sequence-1: 1, 5, 3, 7, 2, 6, 4 and 8 [2] [Figure 3(b)]
- Sequence-2: 1, 2, 3, 4, 5, 6, 7 and 8 [2] [Figure 3(c)]

Bolts are tightened one by one with the torque control method [4] i.e. each bolt is tightened to a target stress for a given pass. In the experimental work [2], author tightened the joint in increments of torque 210, 310, 400 and 505 Nm as per sequence-1. Finally, all the bolts were tightened again to 505 Nm in one pass round as per sequence-2 to achieve uniform preload values. Target torques is converted to the bolt preloads for each pass. In simplified form, for lubricated fasteners the relationship of bolt preload achieved against a given torque with 0.2 as Factor of load loss due to friction is calculated as per [9]. Average bolt stress is then calculated by dividing the bolt preload by the nominal area of bolt shank, the joint is tightened to the target stress for each pass calculated as above. For this purpose an optimization routine is developed and used in manner that each time UY is applied on the bolt, the resulting stress on mid node of bolt shank (close to the strain gauge location) is compared with the target stress and in case of difference the UY is incremented and comparison is done again. Similarly the UY is incremented till it reaches an optimum value for which the target stress in bolt is achieved. Table 2 shows the bolt preloads and target stress calculated above against the applied torques.

![Figure 3](a) (b) (c)

Figure 3: (a) Boundary Conditions; Bolt tightening (b) Sequence-1 (c) Sequence-2

The magnitude of the axial displacement, UY applied to the bottom area of the bolt shank to pre stress each bolt to the target stress, is given in Table 3. Maximum displacement applied is to achieve 30% of the yield of the bolt, although this is considered very low but it avoids gasket crushing, based on this, maximum recommended applied torque by the gasket suppliers is 505 Nm [2].

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Table 3: Magnitude of UY for each pass

<table>
<thead>
<tr>
<th></th>
<th>Pass 1</th>
<th>Pass 2</th>
<th>Pass 3</th>
<th>Pass 4</th>
<th>Pass 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>B1</td>
<td>0.105</td>
<td>0.138</td>
<td>0.172</td>
<td>0.224</td>
<td>0.224</td>
</tr>
<tr>
<td>B5</td>
<td>0.095</td>
<td>0.138</td>
<td>0.170</td>
<td>0.222</td>
<td>0.225</td>
</tr>
<tr>
<td>B3</td>
<td>0.079</td>
<td>0.128</td>
<td>0.161</td>
<td>0.206</td>
<td>0.220</td>
</tr>
<tr>
<td>B7</td>
<td>0.069</td>
<td>0.123</td>
<td>0.156</td>
<td>0.204</td>
<td>0.220</td>
</tr>
<tr>
<td>B2</td>
<td>0.103</td>
<td>0.146</td>
<td>0.179</td>
<td>0.224</td>
<td>0.220</td>
</tr>
<tr>
<td>B6</td>
<td>0.094</td>
<td>0.142</td>
<td>0.175</td>
<td>0.221</td>
<td>0.222</td>
</tr>
<tr>
<td>B4</td>
<td>0.090</td>
<td>0.138</td>
<td>0.169</td>
<td>0.213</td>
<td>0.223</td>
</tr>
<tr>
<td>B8</td>
<td>0.088</td>
<td>0.136</td>
<td>0.168</td>
<td>0.212</td>
<td>0.222</td>
</tr>
</tbody>
</table>

RESULTS AND DISCUSSIONS

Bolt stress variation with bolt up

To determine bolt relaxation or bending behaviour during tightening the bolts as per sequence-1 and 2, four nodes are selected at an angle of 90 degree on shank of each bolt. B1/1 and B1/2 represents inner and outer nodes respectively, B1/3 and B1/4 represents side nodes and B1/M represents the mid node on bolt shank. Similar nomenclature is used for all other bolts. For average bolt stress, mid node on the shank of the bolt is selected [Figure 4(a,b)].

Figure 4c shows how the preload variation of bolt-1, which is tightened first, varies with tightening other bolts during pass-1. Bolt stress reduces when neighboring bolts, bolt-2 and bolt-8. This is concluded due to the elastic interaction [3] of flange which deforms in axial direction during to the bolt load application, thus relaxing the bolt closest to the bolt being tightened, Figure 5(a) clarifies the phenomena of bolt-2 and bolt-4 relaxation when bolt-1, bolt-3 and bolt-5 are tightened. Flange areas beneath bolt-2 and bolt-4 are in compression resulting in relaxation of
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b bolt-2 and bolt-4. Tightening all other bolts, bolt preload of bolt-1 increase and is concluded due to the flange opening phenomena, stretching the bolt in tension, Figure 5(b) shows how bolt-5 is made in tension when bolt-1 is tightened during Pass-1. It is obvious that with bolt-1 tightening, flange areas beneath bolt-1 goes in compression due to the downward flange rotation whereas flange areas at the other end at 180 degree goes up, resulting bolt-5 in tension. Due to this axial stress of bolt-5 is increased. Similarly slight increase in stress is observed in case of tightening all other bolts except neighboring bolts.

![Exaggerated Deformation Plots](image)

Figure 5: Exaggerated Deformation Plots (a) Bolt Relaxation phenomena (b) Flange Opening Phenomena

Figure 6: Individual Bolt up Effect (Pass-1)

Figure 6 shows the effect of tightening of one bolt on all other the bolts for 1st pass. It is clear that tightening any bolt relaxes its neighboring bolts while slight rise in stress is observed in the remaining bolts. At the end of 1st pass, i.e. when bolt-8 is being tightened, 27 to 38% preload relaxation is observed in bolt-3 and bolt-7 respectively. Bolt 2,4,6 and 8 remains completely relaxed during tightening bolt 3,5 and bolt-7. Figure 7 represents the bolt preload variation with the bolt up procedure. For all the bolts, almost same stress variation pattern is observed for the first four passes. However a slight variation in the preloads is observed during the last pass as per sequence-2. This ensures more uniform preload distributions at the last pass. Maximum preload variations are observed for bolt-5. For first four passes, the maximum stress at bolt-5 is observed when bolt-2 is tightened and minimum when bolt-4 is tightened. The preload variations in bolt-5 during bolt-5 tightening are 29 MPa for the 1st pass, 20 MPa for 2nd, 17 MPa for 3rd, 15 MPa for 4th and 4 MPa for the last pass. Bolt-8 remains relaxed till the end of first pass until it is tightened.

![Bolt Preload Variation](image)
Bolt bending behaviour is studied along the four locations along inner, outer and side locations on each bolt shank at each pass end [Figure 8]. Almost similar bending behaviour is observed for all the eight bolts. Tensile stresses in all the bolts are observed at all bolt locations during the bolt up. The difference in axial stresses between the side nodes is negligible for all the bolts, indicating a slight sidewise bolt bending. However the difference in axial stress between the inner/outer nodes is obvious. Inner nodes for all the bolts remain in maximum tension while the outer nodes in minimum tension indicating bolt bending. Bending is increased remarkably with increasing torque and is the maximum during the last two passes. It is also observed that during tightening bolts at 90 and 180-degree locations to the target bolt, the difference in axial stresses between side nodes is little but tightening bolts other than at 90 and 180-degree locations, the difference becomes dominant; this might be due to flange displacement causing the bolt to bend sidewise. For example, axial stress variations at the four locations of bolt-8 during tightening each bolt are shown in Figure 9. Tightening the first four bolts (bolt-1, 3, 5 & 7) in a pass, the difference in the axial stress between side nodes is observed. However the difference is negligible with last four bolts tightening.

Bolt Bending Behaviour

Figure 7: Individual Bolt up Effect (Pass-1)

Figure 8: Bolt bending behaviour of all 8 bolts in the joint

Figure 9: Axial stress variations at four locations of bolt-8 during tightening each bolt
**Scatter in Bolt Stresses**

Preload of bolts 1, 3, 5 & 7 in the first four passes are less than the target stress. At the end of first pass, 27-38% preload relaxations are observed for bolt-3 and bolt-7 respectively. This is due to the reason that these bolts are tightened in advance of the neighboring two bolts, so tightening the last four bolts (bolts 2, 4, 6 & 8), they undergoes relaxation. For example, Bolt-7 is found with minimum stress during the first four passes, reason is that tightening the last two bolts (bolt-6 & 8) in a given pass, bolt-7 is relaxed. Bolt- 2, 4, 6 & 8 are found at stress level greater than the target stress. The reason being that these bolts are tightened as last four bolts in a pass, their neighboring bolts are tightened already, so there are greater stresses than the target stresses in these bolts. The maximum preload difference is observed between bolt-2 and bolt-7, because bolt-2 is first bolt after tightening first four bolts. Tightening bolts 4, 6 & 8 its preload increases, whereas bolt-7 is last bolt in tightening first four bolts. Tightening bolt-6 and bolt-8 at the end of pass, it relaxes. Scatter is greater in bolts 1, 3, 5 & 7 as compared to bolts 2, 4, 6 & 8 because the gasket is seated to its minimum thickness with 1st four bolts tightening. Figure 10a shows the bolt scatter obtained at the completion of each pass after tightening each bolt to the target stress. It is observed that the bolt scatter is the maximum for the 1st pass. For 2nd pass, scatter reduces and its magnitude remains almost the same for pass 3 and pass 4 and is minimized/reduced at last pass with sequence-2. Figure 10b represents variations of bolt stresses at the completion of each pass. The preload variations are minimized during the pass-5. This concludes the importance of last pass with clockwise tightening.

Table 4 illustrates difference between the maximum and the minimum axial bolt stress. Maximum preload difference at the last pass is only 9 MPa, ensuring uniform bolt stress distribution.

**CONCLUSIONS**

From the results of study it is concluded that the joint’s integrity and sealing performance is very much dependent on the material properties of the gasket used. The joint with solid plate gasket experiences almost uniform bolt stress and bending behaviour and shows almost same variations at all passes. Bolt scatter, bolt bending and bolt relaxation are concluded the main factors affecting the joint’s performance. To control these the use of proper bolt tightening sequence,
number of passes is concluded important. Summarizing, a dynamic mode in a gasketed joint is concluded, which is the main reason for its failure.

![Diagram](image)

Figure 10: (a) Scatter in Bolt Stress at Completion of Each Pass, (b) Bolt stress variation at the end of each pass

<table>
<thead>
<tr>
<th>Pass Number</th>
<th>Maximum Difference (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>27</td>
</tr>
<tr>
<td>2</td>
<td>21</td>
</tr>
<tr>
<td>3</td>
<td>21</td>
</tr>
<tr>
<td>4</td>
<td>19</td>
</tr>
<tr>
<td>5</td>
<td>9</td>
</tr>
</tbody>
</table>

**REFERENCES**