# Analysis of Gap Behaviour Between Contact Surfaces of a Bolted Flange

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***Abstract* — Analyzing a bolted joint is inherently a nonlinear analysis. Hence, it is difficult to predict the max. values of stress induced in the bolts and the corresponding gap induced between the contact surfaces of the flanges using the conventional static stress equations from solid mechanics. Therefore, the finite element method is employed to find out the behavior of bolted flange joint. This project is aimed at observing the max. gap induced as a function of applied bolt pretension and approximating the characteristic gap curve for predicting the flange behavior at different values of bolt pretension.**

***Keywords—Von mises stress, skewness, jacobian ratio***

## I. INTRODUCTION

Analysis of bolted flanges is one of the most common applications of the finite element analysis. The resultant region of contact after application of external force is not known beforehand. This gives rise to mathematical model in which the contact forces at a particular point are a function of displacement of that point from the contact region. Due to this inherent property exhibited by these type of components, analytical method proves inadequate for estimating the contact stress values and induced gap value at a particular location at the interface between the two flanges.

It is important to note here that not all nodes in the mesh are nonlinear but only the nodes at the interface between the two flanges exhibit this behavior. In order to reduce the computational time required for solving the mathematical model, we employ symmetry of the computational model and consider only $1/12^{th}$ of the actual model. This can be done without affecting the actual result since the boundary conditions and geometry of the model are periodic.

## II. GEOMETRY SPECIFICATIONS

*Bolt Designation:*

The specifications for bolt used for this analysis are as follows:

Bolt Selected: M12 (Coarse series) with Iso-Metric Threads

Pitch: 1.75 mm

Core diameter ($d_c$) = 9.853 mm

Tensile Stress Area = 84.3 mm$^2$

No. of bolts: 6

Angular pitch = $60^0$

The geometry parameters are given in table 1.

The completer flange model is depicted in fig. 1. This model is sliced to get the required radially symmetrical body.

TABLE 1. FLANGE GEOMETRY PARAMETERS

| Sr. No. | Parameter | Value | Unit |
|---|---|---|---|
| 1 | Flange outer diameter | 75 | mm |
| 2 | Shaft diameter | 30 | mm |
| 3 | Bolt pitch circle diameter | 100 | mm |
| 4 | Bolt diameter | 12 | mm |
| 5 | Angular pitch of the bolt | 60 | deg |
| 6 | Shaft length protrusion | 50 | mm |

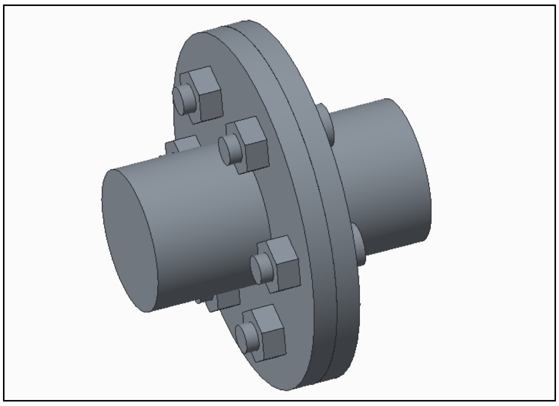

Fig. 1. CAD Model of Bolted Flange Assembly

The fig. 1 shows the bolted flange geometry used for analysis purpose. The dimensions are given in the table 1.

The primary focus of this analysis is to evaluate the maximum gap induced between the flange contact surfaces. To minimize the geometry creation and analysis time, the threads in the bolts and nut are not physically modelled. This doesn't affect the accuracy of the results since the effect of threads is accounted for by the 'bonded' type of contact specified between the bolt surface and the flange hole surface.

When the bolt pretension is applied on the bolt surface, in effect a specified length is reduced from the bolt body – except bolt head – to account for the required pretension. This length is equal to,

$$\delta L = (P_{pretension} * L_{shank}) / (A_{bolt} * E_{bolt}) \text{ [1]}$$

Where, $P_{pretension}$ = Specified bolt pretension (N)

$L_{shank}$ = Bolt shank length (m)

$A_{bolt}$ = Bolt cross section area (m$^2$)

$E_{bolt}$ = Elastic modulus of bolt (N/m$^2$)

The geometry created was imported in STP file format for meshing and further analysis in the solver workbench.

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## III. MESH

The values of stress induced in the structural analysis using computational methods are highly mesh sensitive. If the mesh is not sufficiently refined, the values of stress induced as shown in the results will be wrong. Therefore, in order to get sufficiently good results, it is absolutely essential that the mesh metric values are above a particular standard values.

In order to get accurate results, the mesh at the contact interface and at the bolt contact surface can be refined progressively.

To account for the sharp curvature and the edge locations effectively without increasing the no. of nodes to a great extent, a hex dominant mesh was used[2]. The table 2, gives the mesh parameter values used for computing the stress values in this analysis.

To get sufficiently accurate results, a preliminary analysis was first carried out with coarse hexahedral mesh and then refined to get the more accurate values.

The type of elements used to carry out the analysis was SOLID186. It is a 3-D, 20 nodes solid element that exhibits quadratic displacement behavior. The element has three degrees of freedom per node and hence total degrees of freedom per element is 60. The structure of the element with the node configuration is depicted in fig. 2.

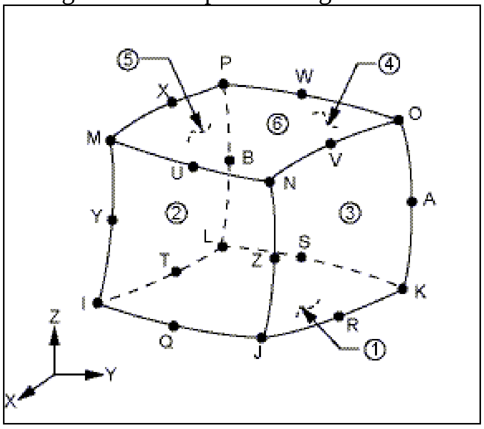


Fig. 2. SOLID186 Element with midside nodes[4]

The element displays quadratic displacement behavior due to presence of midsize nodes. This property is important to capture the deformation of the nodes with fewer number of nodes and elements.

TABLE 2: MESH PARAMETER VALUES FOR PRELIMINARY ANALYSIS

| Sr. No. | Parameter | Value |
|---|---|---|
| 1 | Nodes | 66442 |
| 2 | Elements | 15628 |
| 3 | Avg. Orthogonal Quality | 0.843 |
| 4 | Avg. Skewness | 0.283 |
| 5 | Min. Edge Length | 3 mm |
| 6 | Jacobian Ratio | 1.5715 |

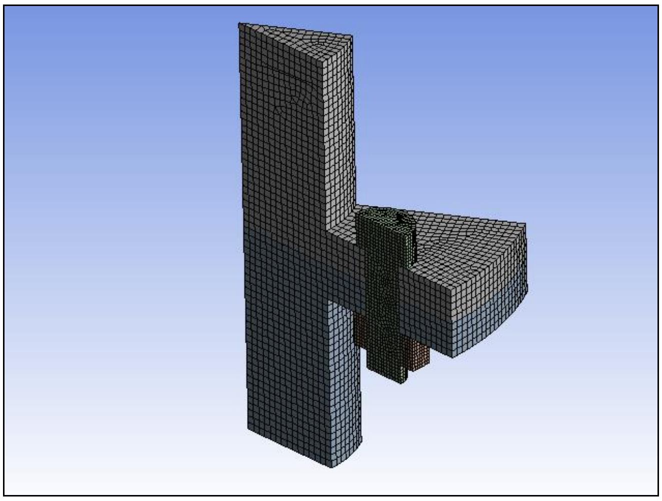

Fig. 3. Symmetrical flange model with coarse mesh

## IV. ANALYSIS SETUP

For analysis purpose, we apply the loads in two load-steps. In first load-step, we'll apply the required bolt pretension and then in the second, we apply the deforming force on the shaft connected to the flange. As we are considering only 1/12th of the geometry for computational efficiency, we impose frictionless support boundary conditions on the free vertical faces of the two flanges. This prevents the displacement of nodes at these surfaces in direction normal to the surface as well as the in-plane translational motion of the nodes on the same surface. The fig. 4 shows the boundary conditions applied on the bolted flange geometry.

The numerical solution for this model can be transformed to visualize the result for the entire flange. But this comes at a cost of greater graphics power usage.

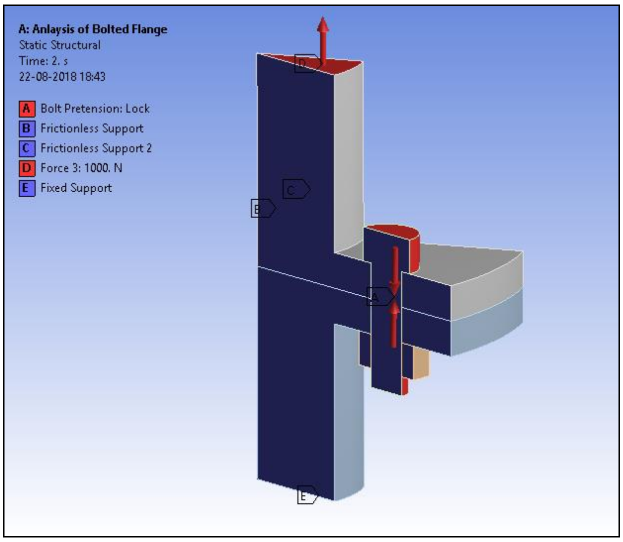


Fig. 4. Boundary conditions applied to the model assembly

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## V. ANALYSIS RESULTS FOR COARSE MESH

For the coarse mesh as depicted in fig. 3, the maximum gap induced is observed to be 7.6059e-5 m. This value of gap converges to the true solution as the mesh is refined. As can be seen in fig. 6, for refined mesh this value of maximum gap induced changes to 6.7646e-5 m.

Comparison between the flange contact surfaces in both the figures reveals the change in displacement at various locations due to mesh refinement. With more refined mesh, there is significant rise in the minimum displacement region. That is, the region in the vicinity of the bolt shows zero displacement which was not the observed in case of analysis for the refined mesh.

## VI. NUMERICAL ANALYSIS WITH REFINED MESH

Table shows the mesh parameter values for the analysis carried out using refined mesh. The element type is kept same whereas the refinement factor and the minimum element size is varied to get more accurate numerical results. The mesh statistics and mesh parameters are listed in table.

TABLE 3. REFINED MESH PARAMETER VALUES

| Sr. No. | Parameter | Value |
|---|---|---|
| 1 | Nodes | 195520 |
| 2 | Elements | 49419 |
| 3 | Avg. Orthogonal Quality | 0.8518 |
| 4 | Avg. Skewness | 0.2625 |
| 5 | Min. Edge Length | 3 mm |
| 6 | Jacobian Ratio | 1.5592 |
| 7 | Standard Deviation | 0.8775 |

To obtain better results for the same computational cost, the mesh refinement is carried out locally in the bolt only. The mesh for the two flanges remains the same even in the refined model.

## VII. NUMERICAL SOLUTION AND VERIFICATION

When the bolt is preloaded, a very small gap is induced between the flange contact surfaces as a result of contact forces between the bolt head and the flange surface. But this is of the order of $10^{-3}$ mm and is negligible when compared to the actual value observed under the deforming force. Thus, it can be neglected without loss of fidelity. The fig. 5, shows the gap induced between the contact surfaces when the deforming force is applied to the flange.

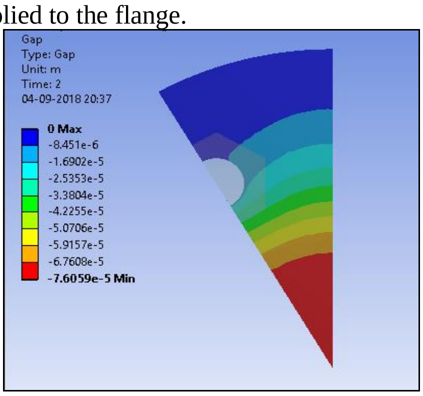


Fig. 5. Gap between the flanges under the deforming force with coarse mesh

As expected, the gap is maximum on the inner portion of the flange section where the deforming force is close the region and acts directly along the axis of the flange.

The bolt resists further deformation along the flange section by acting as the connecting link between the two flanges and this induces the stress along its length. The variation in stress is depicted in the fig. 7.

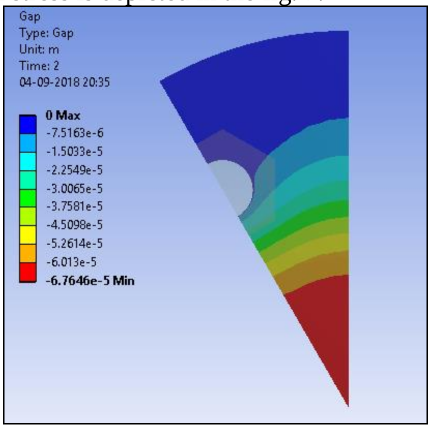


Fig. 6. Gap between the flanges under the deforming force with refined mesh

As we move from bottom of the bolt to the start point of the shank, we see that the equivalent stress along the thread surface varies as shown in fig. 7. Here, initially, the stress induced is on the lower side (varying between 0 MPa to 11.34 MPa as x varies from 0 to 5 mm) as this region is not bonded to any of the remaining part and hence effectively is not under any kind of load.

As we move into the bolt region bonded to nut, the stress suddenly rises owing to contact between the bolt and the nut. Due to contact between the nut and the lower flange, the nut transfers the tensile stress to the bolt.

As we approach the plant of contact between the nut and the lower flange, the stress suddenly rises due to contact stresses between the lower flange and the bolt. This is one of the regions where the contact is firm and as we move towards the plane separating the lower flange and the upper flange, the nodes are loosely in contact. This can be imagined to be like simply supported beam with reaction at the two endpoints.

## VIII. RESULT DISCUSSION

As we approach the plane of separation between the lower flange and the upper flange, the stress gradually rises as a result of gradually increasing moment about the plane of separation, until x = 25 mm. At this point, the stress rises rapidly due to maximum bending moment at this plane.

This is in fact the region of maximum stress in the bolt. The value of stress in this region determines if the bolt will withstand the boundary condition forces and hence is critical from the design point of view.

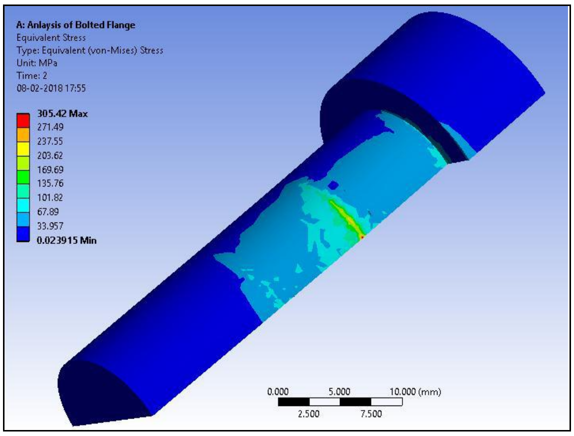


Fig. 7. Equivalent stress induced in the bolt body

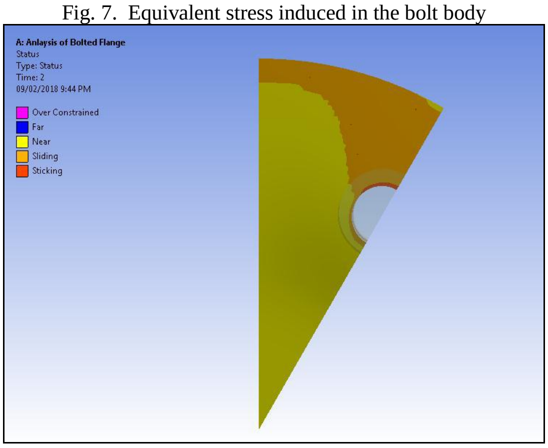


Fig. 8. Contact status for 703.125 N bolt pretension at the end of second load-step

The fig. 8 shows the nature of contact between the two surfaces of the bolted flange. Since the opening takes place on the portion closest to the centreline, the contact surfaces after deformation are close but not exactly in contact. Similarly, this causes the outer surface to press against each other and hence, they slide over each other and are in direct contact with each other. At the same time, the area in the viscinity of the bolt exert high pressure on each other and there is negligible sliding in this region. Thus they are in firm contact and this is evident from the 'sticking' nature of the contact surface.

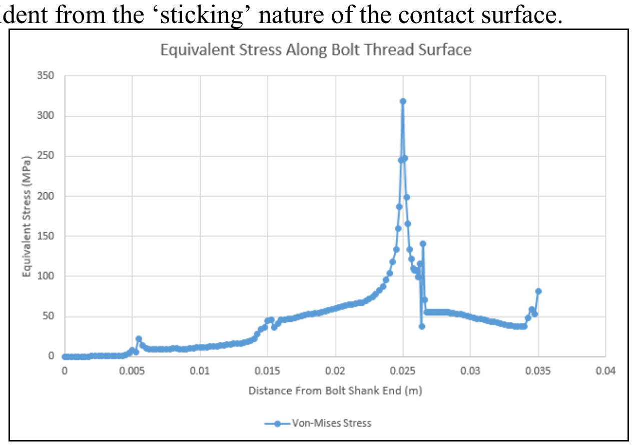


Fig. 9. Equivalent stress along bolt surface as measured from the shank end

The stresses induced from x=26 to x=34 mm are almost mirror values image of stress from x=16 to 24mm. This is due to symmetrical nature of loading. The stress though rises suddenly as x approaches x=35mm. At this point due to stress concentration resulting from change in cross section from shank to bolt head, the stress is slightly amplified and this can be seen from the stress – position curve shown in fig. 9.

The table shows the results of the case studies performed on the above mentioned bolted flange assembly models. The values of the maximum gap and the corresponding equivalent stress induced are enlisted in table . The values are recorded upto three significant figures.

TABLE 4. CASE STUDY OF GAP INDUCED AT THE CONTACT SURFACES

| Case No. | Bolt Pretension (N) | Max. Gap ($\times 10^{-2}$mm) | Linear Interpolation Values | % Difference in max. gap |
|---|---|---|---|---|
| Case 1 | 351.562 | 6.020 | 6.020 | 0.000 |
| Case 2 | 703.125 | 5.823 | 5.834 | -0.193 |
| Case 3 | 1054.688 | 5.653 | 5.649 | 0.066 |
| Case 4 | 1406.252 | 5.453 | 5.464 | -0.195 |
| Case 5 | 1757.812 | 5.264 | 5.279 | -0.278 |
| Case 6 | 2109.375 | 5.093 | 5.093 | 0.000 |

## IX. CONCLUSION

The above analysis thus proves that the gap induced between the contact surface of a bolted flange is proportional to the first order of the magnitude of the bolt pretension applied on the flange geometry. Graphically the results are shown in fig. 10.

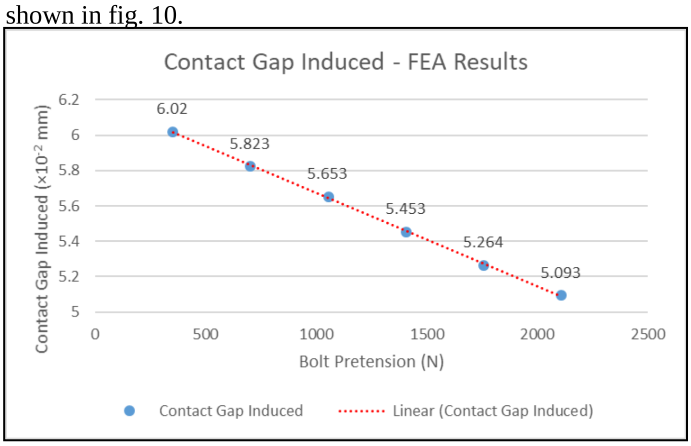


Fig. 10. Comparison of FEA and linear interpolation curve

As is evident, the linear interpolation values agree closely with the FEA calculated values.

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[2] Practical Aspects of Finite Element Simulation, A Study Guide, 3rd edition, Altair University
[3] Design of Machine Elements, VB Bhandari, 5th Edition
[4] Image courtesy ANSYS® Mechanical Help, Version 14.0