EFFECT OF BOLT LAYOUT ON THE MECHANICAL BEHAVIOR OF FOUR BOLTED SHEAR JOINT USING THREE – DIMENSIONAL FINITE ELEMENT ANALYSIS

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ABSTRACT

Bolted joint is a typical connection that is widely used in the machine assemblies and construction of structural components etc. Bolted joints when put in use encounter one or more type of working loads. In general, analysis of bolted joint is not simple, it involves many factors such as bolt pretensioning, number of bolts, number of members, clearance between the bolt and hole, flange thickness, loading conditions and friction coefficient, etc. Here, a comprehensive study is conducted to investigate the effect of four-bolted joint layout on the mechanical behavior of the joint. Three-dimensional model is analyzed in shear type of loading. This model helps us to visualize the localized points of high stress concentration that is not possible by using two dimensional or axisymmetric models. Displacement pattern and stress distribution in different arrangements is studied. Critical regions and the critical bolt are being identified.

KEYWORDS: *Finite element analysis; Three dimensional; Layout; Bolted joint; Shear joint*

1.0 INTRODUCTION

Bolted joints are extensively used in most modern machines. The key feature of bolted joints is that they can be dismantled comparatively easily. Purpose of bolted joint analysis is to identify the failure modes like end tear out, bearing, net section fracture and bolt shear. Bolted joint analysis is complex in nature as it involves number of factors to be considered. Factors like bolt pretensioning, contact between plates, bolt deformation, bolt size, clearance, number of bolts, loading conditions, supporting conditions, number of plates, bolt layout, friction flange thickness are important when analyzing a bolted joint structure. Researchers have used different approaches to analyze the bolted

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connections and these are analytical, experimental and numerical techniques. Analytically first step towards a bolted joint analysis is to calculate the stiffness of the bolt and the member. Calculating stiffness for the bolts is easy but for the members situation is somewhat different. Because the compression spreads out between the bolt head and the nut and hence the stressed area is not uniform. But still there are some analytical methods that can be used to calculate the stiffness approximately. Ito, Toyoda and Nagata (1977) suggested the use of Rotscher's pressure cone method with a variable cone angle. This method is quite complicated so there are others such as method of Shigley and Mischke (1989) with cone angle of 30 and method of Motosh (1976). The later two methods overestimate the clamping stiffness and first one is quite complicated and difficult to use. Once the stiffness is calculated the resultant bolt load and resultant load on members can be calculated with the help of formulas. So there is a limitation of the analytical methods to predict the stress in a member in a bolted joint. Carrying out experimental work requires more resources and time and it is difficult to rework in case of any mistake. Time and money are a restriction of doing extensive experimentation. Because of these facts use of numerical methods are useful and time saving. The model can be altered with ease and non-linear behavior can be included if necessary.

Bolted joint analysis has attracted many researchers. Menzemir et al. (1999) studied block shear failures of bolted joints for different arrangements of bolts. Aluminum alloy 6061 was chosen, and an experimental study was performed with the objective of rationalizing block shear failure in connecting elements. Gusset plates were chosen, and samples representing four different bolt patterns mere mechanically deformed. Models to estimate the capacity of the joints are examined and compared with experimental results. Mechanisms governing damage and failure are highlighted in light of the competing influences of load/stress distribution and intrinsic micro structural effects The behavior of truss plate reinforced by single and multiple bolted connections in parallel strand lumber under static tension loading were investigated by Hockey et al. (2000). Sixty single bolt connections were tested and similarly sixty multiple bolt connections were experimented. Their effect on the ultimate tensile strength of the connection was observed. It was also observed that reinforcement significantly improved the ductility in all the connections tested. Design Criteria for Bolted connection elements in Aluminum Alloy 6061 is reported by Menzemir et al. (1999). Plates of relatively thin cross section and extruded shapes held by one or more bolts were tested in tension and shear. Bolt holes along both the tensile and shear planes were elongated. Also those holes located near the edge of the specimen were elongated and noticeably rotated with respect to the far filed load axis. A similar type of study is done by Tan and Smith (1999). They studied the effect of bolts in rows. Experiments confirm that there is a reduced effective capacity per bolt with any increase in the number that is placed in a row. This is called row effect on strength. They actually gave an elasto-plastic model. The mechanical behavior of bolted joints during a torque controlled tightening process was analyzed as an elastic-plastic contact problem by Fukuoka and Takaki (2003). A 3-D analysis using a 2-D finite element mesh with each node having 3 degrees of freedom was conducted. The relationship between axial bolt stress and nut rotation angle was reported. Further study on the mechanical behavior of hybrid (bolted and bonded) joints applied to aeronautical structures was reported by Paroissien and Sartor (2007). In this study, a fully parametric analytical 2-D model based on the finite element method is presented. A special finite element, the "bonded beam" element is computed to simulate the bonded adherents. Again, Fukuoka and Nomura in 2009 derived a series of closed form algebraic equations which can calculate the true cross sectional areas of internal/ external screw threads with the effects of the helix and root radius taken into account. The equations obtained can be applied to coarse or fine pitch. In 2009, Cornwell used the finite element method to accurately estimate the load factor for 4,424 unique combinations over the entire range of the four joint design parameters, namely bolt diameters, joint thickness, individual plate thickness and plate material combinations. Turvey and Wang (2009) described the failure tests on pultruded glass reinforced plastic single bolt tension joints. Four joint layouts were used to determine the effect on joint failure loads. Nonlinear finite element analysis of bearing capacity of joint with combined bolts and welds was done by Wen et al. (2007). They considered the effects on the bearing capacity by different number and different layouts of the bolts. More recently, Hurtuk et al. (2012) investigated the influence of bolt holes, specifically their number and layout on strength and deformation. They determined the maximum load carrying capability and fracture load. All of their work was experimental by deforming the plates under quasi-static loading.

Summarizing the literature survey it is clear that effect of layout has not been reported much. There is scarce information regarding the use of numerical technique. Al Nassar et al. (2012) analyzed the effect of clearance and pre tension on the performance of a single bolted joint using 3D FEA but the layout study is not there. It has been shown by Khurshid (2004) that in many cases, the stress distribution along the thickness is not uniform and localized points of stress concentration may exist in the bolt and connected members. These critical points cannot be visualized properly with two-dimensional models indicating a need of a full three dimensional mode. The focus of the current study is to investigate the effect of layout on the displacement pattern and stress distribution of the members and bolts using three dimensional finite element analysis with software package ANSYS. Four different bolt arrangements loaded in shear are tested. The numerical results are verified by conducting an experiment on four-bolt shear joint. Critical bolt and critical surface in the layouts are identified in terms of maximum stress and individual surfaces of the members are analyzed. Table 1 gives the geometric dimensions of the model used in the testing.

Table 1. Dimensions of the geometric model

Parts	Dimensions
Loading plate	180mm x 180mm x 10mm
Supporting plate	180mm x 180mm x 10mm
Bolt	M16 x 2 (grade 5.6)
Nut	M16 x 2

2.0 ANALYSIS UNDER SHEAR LOADING

2.1 FE Model Description

Four-bolted joint is considered because this combination is used very commonly in the assembly structures. The finite element modeling and analysis has been used to analyze the bolted joint. Bolts, nuts, loading and supporting plate all are made up of solid three-dimensional element. It is defined by eight nodes having three degrees of freedom at each node. These are the translations in the nodal x, y and z directions. Model under study is linear and the properties of steel are used. Young's modulus of 210 GPa, Poisson ratio of 0.29 and yield strength of 200 MPa is used. Smart sizing is used to mesh the whole models. The mesh is refined to such level where after further refinement the displacement and stress level are converging to the previous values. This means that the results we are getting now are numerically correct. The number of elements and nodes in this four-bolt model are 97096 and 19762, respectively.

Elastic analysis with contact elements placed at the contacting surfaces is performed. Contact elements predict the real situation by taking into account the coefficient of friction between the two mating surfaces. The contact type that is used in this case is surface-to-surface contact. The element types, *Targe 170* and *Contac 174* are used to define the contact between surfaces. There are seven contacts areas defined in this model. One contact element pair is between the bolt head and the supporting plate. One is between the interfaces of the two plates. One is between the loading plate and the nut surface. Finally, four are between the bolt shank and the inner surface of the hole of the two plates respectively. Four layouts models have been constructed for analysis. These four are shown in Figure 1. Figure 2 illustrates the typical finite element mesh for the four bolted joint.

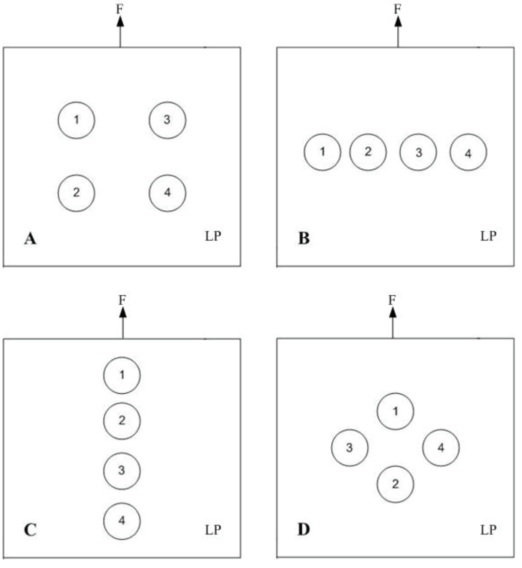

Figure 1. Schematics of four bolted joint layouts (A, B, C and D)

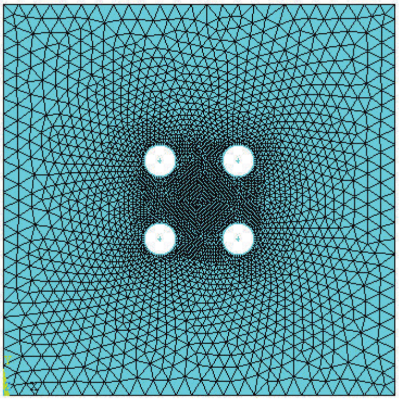

Figure 2. Typical FE mesh of four bolted joint layout for numerical simulation

2.2 Boundary Condition

There are two basic types of boundary conditions to be considered in analyzing the bolted joints. The external constraints and the pre tension in the bolt internally. ANSYS applies the pre tension force by use of special element called *PRETS* 179. Pretension of 30,000 N is applied in the bolt model throughout the analysis. The other type of boundary condition is the constraints and the applied force. Displacement of 0.06 mm is used as a load and all the numerical results are obtained at this value except in the validation phase where two additional loads of 0.07 mm and 0.08 mm are applied to the experimental set up and numerical experiment of Layout A. The lower bottom area of the supporting plate is constrained whereas the upper area of the loading plate is given displacements in the y-direction. The boundary conditions of this problem are shown in Figure 3. The terminologies of loading plate (LP) and supporting plate (SP) are also indicated in this figure. Symmetry is used and half models are tested.

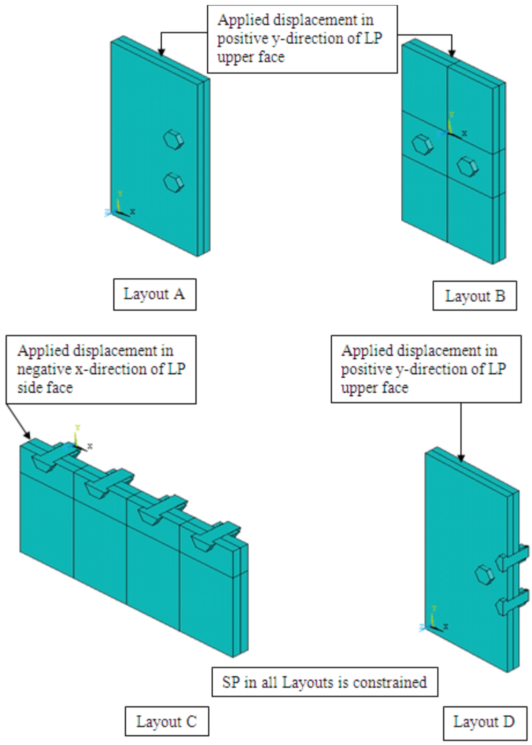

Figure 3. Boundary conditions for layout A, B, C and D

3.0 RESULTS AND DISCUSSION

3.1 Layout A

Figure 4(a) shows the displacement pattern of supporting plate (SP) for layout A. In this figure, isometric view, bolt side view and interface side view are shown respectively. Isometric view shows that the displacement pattern is changing throughout the thickness of the plate. This is very clear by looking at the different pattern on both sides. SP bolt side shows that the sides and bottom region of the plate is not moving with the applied load because of the constraint applied and the displacement is higher around the bolt holes. For SP interface side lower surface is at zero displacement but there is more movement as compared to the bolt side in the upper region. Maximum displacement region is around the bolt 1 near the applied load edge. In this case maximum value of displacement is 0.0314 mm. The upper edge on the interface side is not moving uniformly in the direction of the load. More movement is in the center of the surface. This is because of the arrangement of the bolts. Figure 4(b) gives the isometric view, interface side and nut side view of loading plate (LP) for layout A. Displacement in y-direction is shown. Region close to the loading edge is approaching to the applied load displacement value of 0.06 mm. Displacement decreases as we move away down. Minimum displacement region is around bolt 2. LP nut side shows that the region above the bolt 1 and the side of the surface is approaching the applied load value of 0.06 mm. So there is some upwards movement from the sides while the center being less displaced. Closely inspecting these figures it is clear that the displacement pattern is vice versa the pattern obtained in SP.

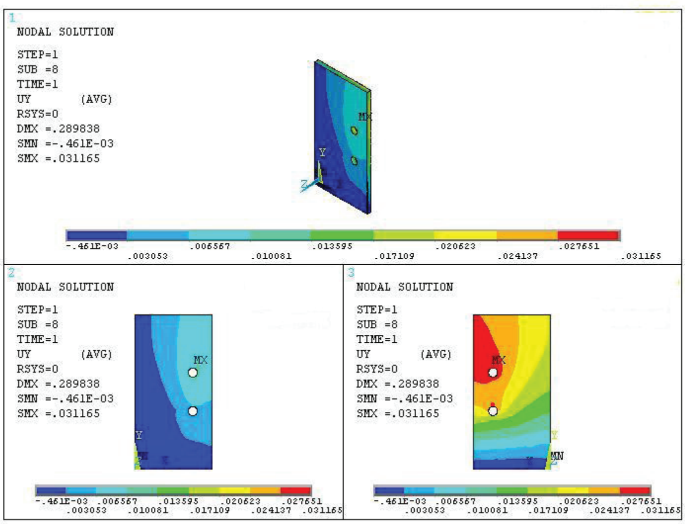

Figure 4(a). y-displacement of SP for layout A
(bolt side and interface side)

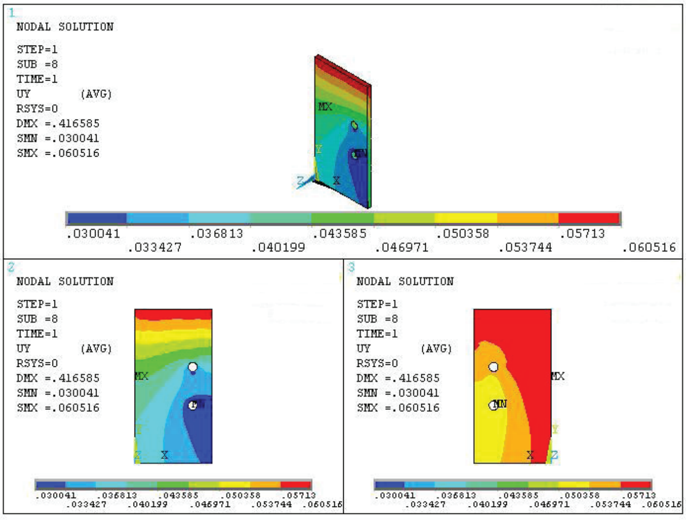

Figure 4(b). *y*-displacement of LP for layout A (interface side and nut side)

Figure 5(a) shows the stress distribution σ_y of SP for layout A. Again isometric view, bolt side and interface side is shown in the figure. Stress distribution is not uniform through the thickness. SP bolt side shows that the region above the bolt 1 is in compression. This is because when load is applied bolt is striking the upper contact surface depending on the clearance level thus compressing it. Region below the bolt comes in tension. SP interface side also shows the similar pattern. The regions above the bolt 1 and bolt 2 are in compression and region below are in tension. In this case maximum value of stress is 111 MPa and it is at the interface side of the plate. Figure 5(b) shows the stress distribution σ_y of LP for layout A. Again the stress distribution is changing throughout the thickness. LP interface side shows the regions of tensile stresses above the bolt 1 and bolt 2. Reason for this, is that now the bolt is striking the lower portion and putting that in compression. The pattern is vice versa as it is seen in SP. The maximum value of stress on this interface side of the plate is given to be 107 MPa. Nut side of LP is mostly in compression. Small regions around the bolt hole are under tensile stress.

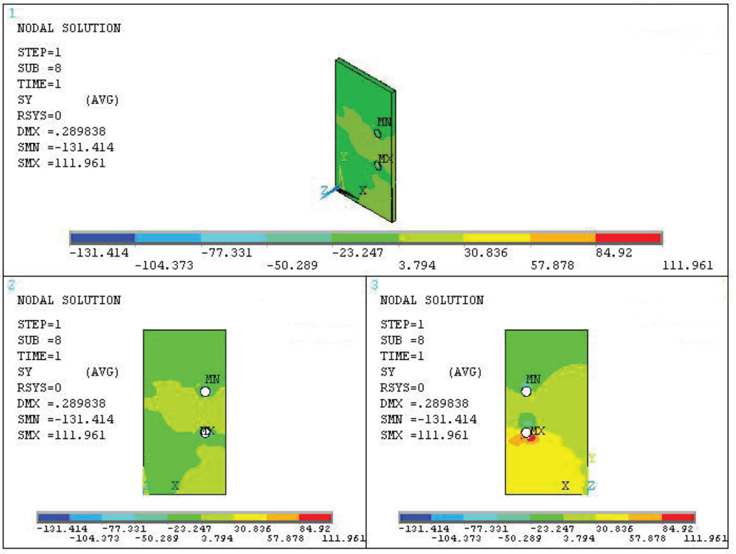

Figure 5(a). Stress σ_y of SP for layout A (bolt side and interface side)

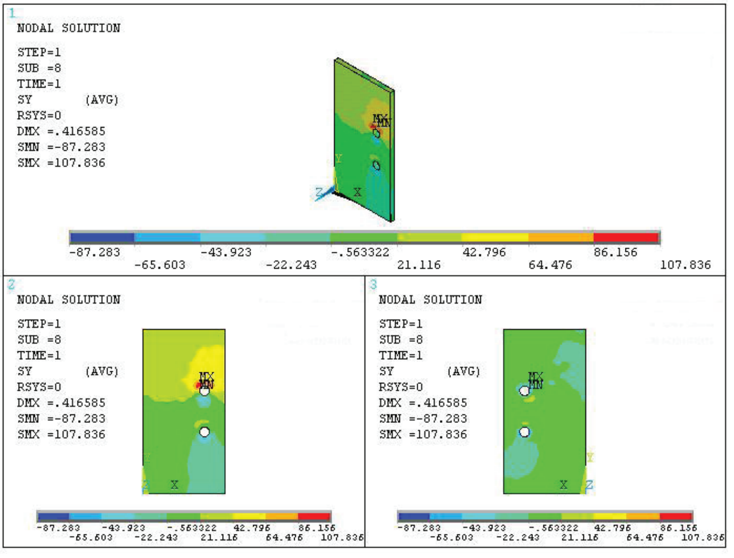

Figure 5(b). Stress σ_y of LP for layout A (interface side and nut side)

3.2 Experimental Validation

To verify the numerical results, an experiment is conducted in which tensile testing is used. A 250 kN Instron universal testing machine, fitted with a linear voltage distance transducer (LVDT) extensometer and connected to a computer through a data logger, is used in tension mode to perform and record the experiment. Experimental set up is shown in the Figure 6(a). Fixtures are used to clamp the bolted joint in the jaws of the machine. The locations of strain gages are shown clearly in Figure 6(b). Strain gages 1, 2, 3 and 4 are placed on LP and 5, 6, 7 and 8 are placed on SP. Three different displacements tests have been performed. The strain gage readings are recorded. For numerical analysis symmetry is employed and half model is used. Strains at location 1 and 3 on loading plate and at location 5 and 7 on the supporting plate are noted from the finite element model at the three displacement values used in the experiment. Table 2(a,b,c) shows the strain values at these locations, at three different load values, that are recorded experimentally and numerically. The first observation by seeing the table is that the values of strains that are obtained experimentally and the values of strains corresponding to these locations obtained by the numerical model are quite close. The important conclusion of this experiment is that the strain produced in the vicinity of bolt, which is closer to the loading edge, is more than the other region. This observation can be seen in both the plates. Location 1 and 7 are closer to the loading edge while 5 and 7 are closer to the supporting edge. The range of numerical strain at location 1 is 83×10^{-6} to 94×10^{-6} and at location 3 the range is 61×10^{-6} to 74×10^{-6}. Same trend is visible in the experimental data. There is a reduction in the strain values around the bolt hole that is away from the loading edge.

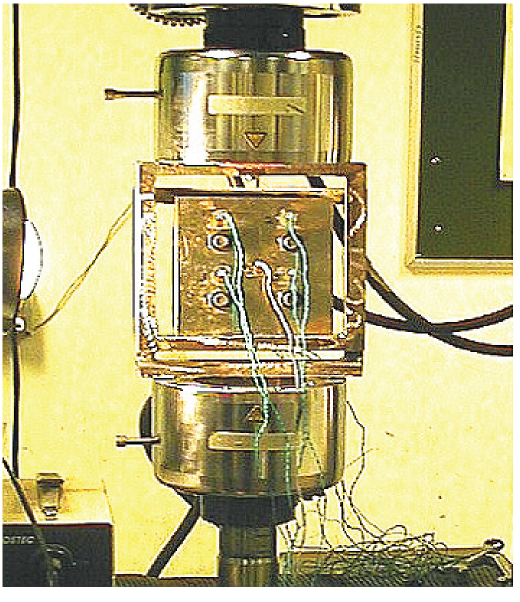

Figure 6(a). Experimental set up of four bolted joint

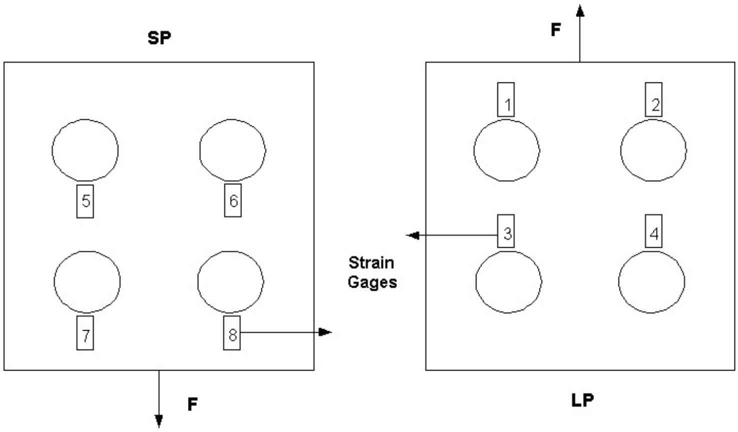

Figure 6(b). Location of strain gages

Table 2(a). Comparison of strain values from the loading and supporting plates for a displacement of 0.06 mm

Strain Location	Experimental (× 10^{-6})	Numerical (× 10^{-6})
LP		
1	81	83
3	59	61
SP		
5	57	60
7	75	76

Table 2(b). Comparison of strain values from the loading and supporting plates for a displacement of 0.07 mm

Strain Location	Experimental (× 10^{-6})	Numerical (× 10^{-6})
LP		
1	84	88
3	65	69
SP		
5	64	67
7	78	80

Table 2(c). Comparison of strain values from the loading and supporting plates for a displacement of 0.08

Strain Location	Experimental (× 10^{-6})	Numerical (× 10^{-6})
LP		
1	**89**	**94**
3	**70**	**74**
SP		
5	**68**	**71**
7	**83**	**86**

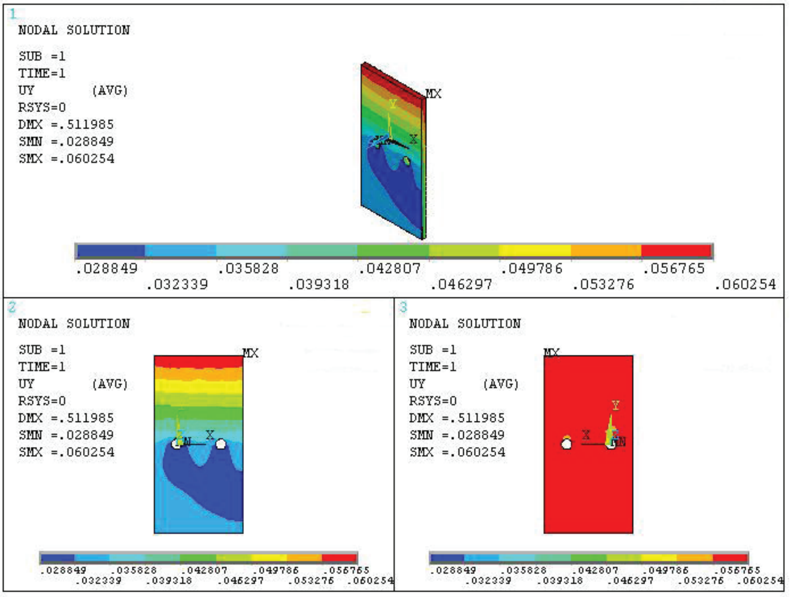

Figure 7(a). y-displacement of LP for layout B (interface side and nut side)

Figure 7(b) shows the stress distribution σ_y of LP for layout B. Hence upper half region of LP interface side is under tensile stress with a maximum stress value of 97 MPa located near bolt 1 on the interface side. Nut side of LP is again not much stressed.

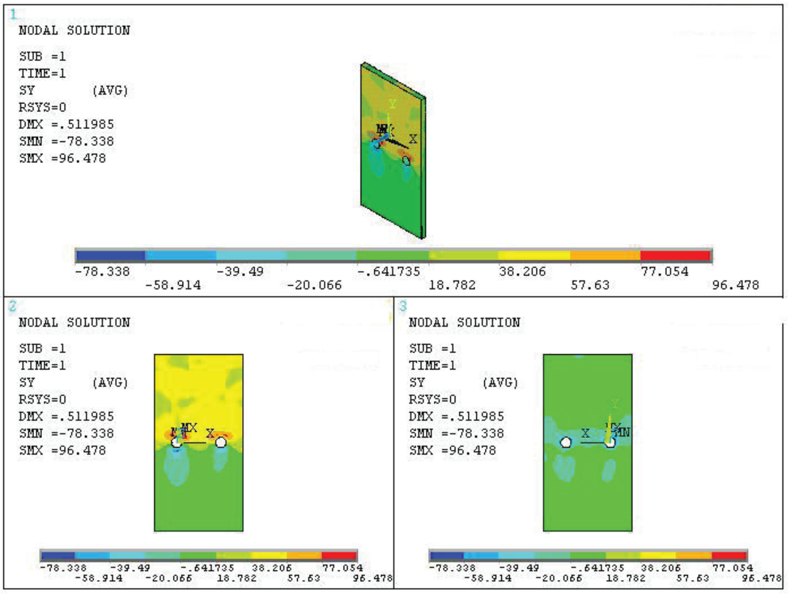

Figure 7(b). Stress σ_y of LP for layout B (interface side and nut side)

3.4 Layout C

Figure 8(a) shows x-displacement pattern of LP for layout C. Bolt side of LP shows that the surface is moving from the sides more. The effect is that there is movement in direction of applied force from the sides while the movement decreases as we move to the center. This is because of the vertical positioning of the bolts. SP interface side shows that region around bolt 4 is moving with the least displacement which is along the loading direction.

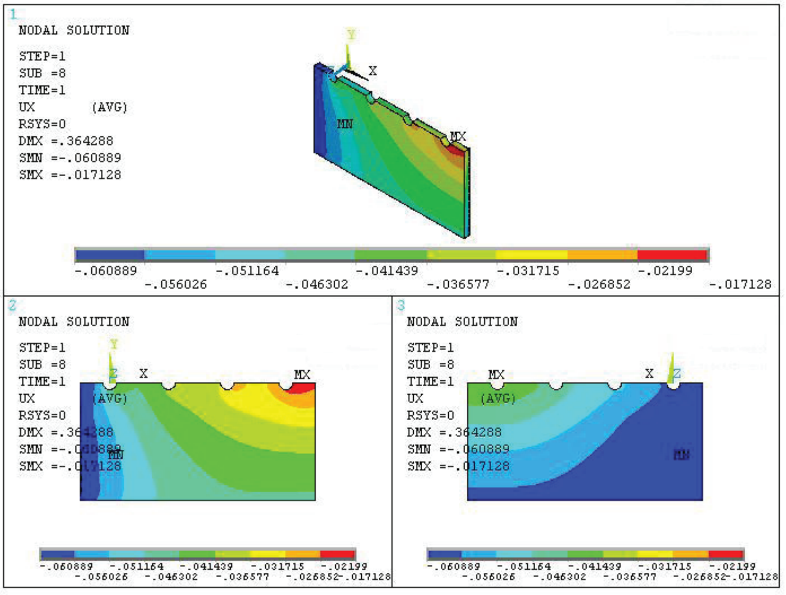

Figure 8(a). *y*-displacement of LP for layout C
(interface side and nut side)

Figure 8(b) shows the stress distribution on LP for layout C. LP interface side shows more stressed surface. Maximum value of stress is around bolt 1. Upper half region up till bolt 2 is in tension and the other half is in compression. On LP nut side tensile stresses are around bolt holes. Hence the maximum value of stress reaches a value of 222 MPa.

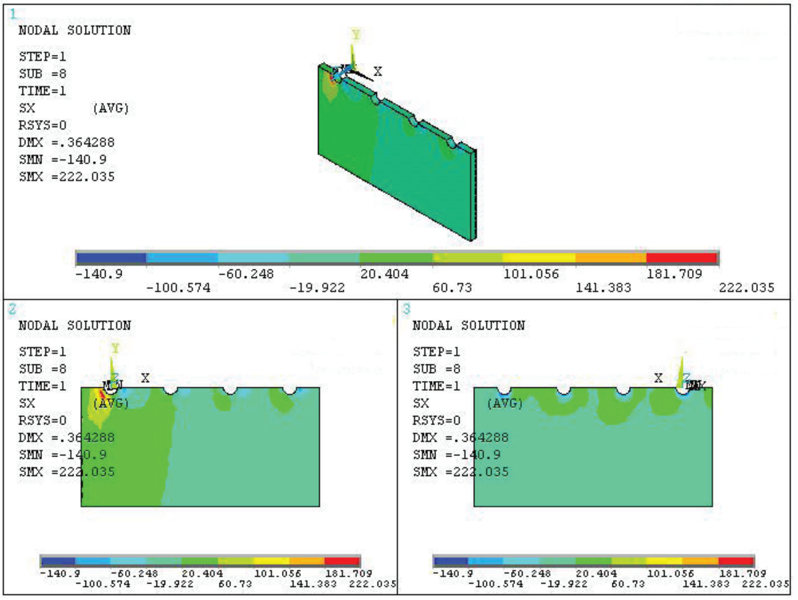

Figure 8(b). Stress σ_y of LP for layout C (interface side and nut side)

3.5 Layout D

Figure 9(a) gives the displacement pattern of LP for layout D. Isometric view shows that displacement is not uniform along the thickness. LP interface side shows more relative movement regions as compared to the nut side due to the slipping phenomena. Minimum value of displacement is in the region around the bolt 2. LP nut side region near the loading edge is moving with displacement value equal to the applied load. Sides of this surface also show this movement thus it can be said that due to the positioning of the bolts plate is moving in upward direction from the sides.

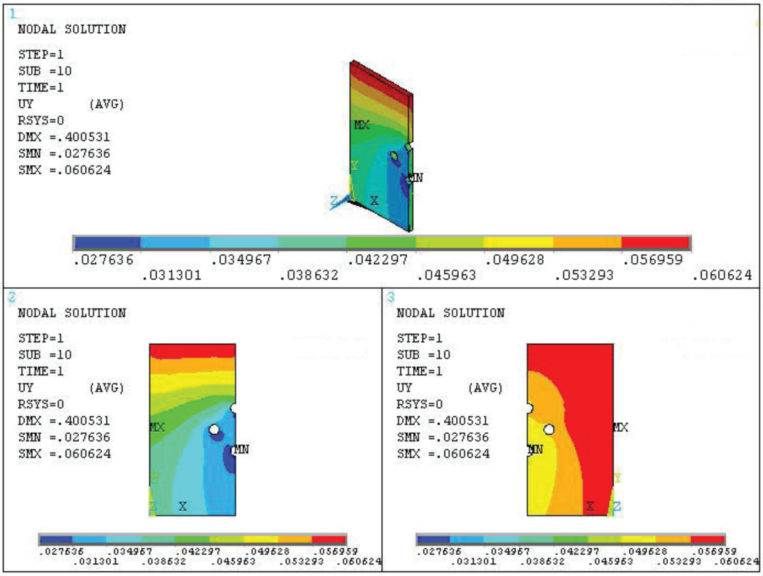

Figure 9(a). y-displacement of LP for layout D (interface side and nut side)

Figure 9(b) shows the stress distribution on LP for layout D. Three views are shown. Upper half portion of the interface side is in tension. The High stress regions are around the bolt holes. The maximum stress value is 104 MPa and is on the interface side of the LP. Small regions of tensile stresses are there on LP nut side.

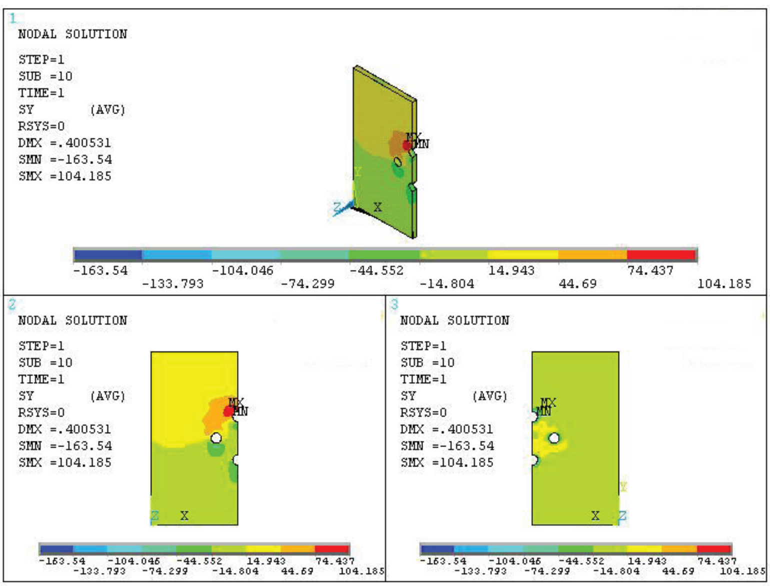

Figure 9(b). Stress σ_y of LP for layout D (interface side and nut side)

4.0 COMPARISON

After analyzing the layouts individually Table 3 lists the maximum von Mises stress values on SP and LP. It is clear from this table that layout C and D are showing the highest stress value for loading and supporting plate, respectively. While the layout B has the minimum stress values. Stress values for layout A and layout B are comparable so layouts A and B are better than layouts C and D. Table 4 lists the maximum stress values in the direction of applied load in the bolts. It is clear that the highest stress is in the critical bolt of layout C. The minimum stress is again in the layout B. It is clear from these two tables that there is a relationship between the high stress regions of loading plate with the critical bolt experiencing high stress in a specific layout.

Table 3. Maximum von Mises, σv Stress on SP and LP

	σ_v (MPa) at SP	σ_v (MPa) at LP
Layout A	173	192
Layout B	157	153
Layout C	329	331
Layout D	348	308

Table 4. Maximum stress, σ_y on bolts

	σ_y (MPa) at bolt 1	σ_y (MPa) at bolt 2	σ_y (MPa) at bolt 3	σ_y (MPa) at bolt 4
Layout A	105.725	93.745	105.725	93.745
Layout B	94.767	94.151	94.151	94.151
Layout C	197.93	185.32	182.12	156.758
Layout D	166.662	102.943	159.374	159.374

5.0 CONCLUSIONS

A detailed study is conducted to investigate the effect of layout for four bolted joint in shear. On the basis of the study some conclusions can be drawn as follows:

- The values of maximum von Mises stress in layout C and D is higher in both SP and LP than the values for lay out A and B. It is concluded that the last mentioned layouts are better.
- Reason for this is that more the bolts are spread and away from the centerline of the plate in a bolted arrangement, better is their stress distribution pattern with less von Misses stress as compared to the layouts where the arrangement of bolt is concentrated in the center of the plate.
- Looking at the layouts individually LP interface side is more critical as compared to the LP nut side.
- For lay out A, B, C and D, on LP, stress value is more in the region around the bolt hole 1. The bolt hole near the loading

edge is more stressed in all the cases.

- For lay out A, B and C critical bolt is bolt 1 as it has higher stress value than the other bolts in the layout. For layout B, bolt 1 and 2 are both critical because of the horizontal arrangement of the bolts.
- It can be concluded that the distribution of stress is not similar around every bolt hole in the member as usually assumed in design procedure calculations. The stress distribution changes with the change of arrangement of bolts.
- It is also observed that the critical region in the LP is the same where the bolt is critical too.

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