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Stress Analysis Of Bolted Joint

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Abstract

This paper presents the simulation and experimental work on the prediction of stress analysis in a single lap bolted joint under shear load. A three-dimensional model of a bolted joint has been developed using modeling software Catia and FEM simulation was carried out by using standard commercial software. In this simulation, stress analysis has been carried out by varying geometrical parameters of bolted joint for optimization. Experimental work was then conducted to measure strains and deformations of the specimens for validation of the developed numerical model. Experimental work was carried out on universal Testing machine and specimen of bolted joint was tested to know ultimate shear strength of bolt. The results from both simulation and experiment were then compared and show good agreement. Several factors that potentially influenced the variation of the results were noted. Finally, critical areas were identified and confirmed with the stress distribution results from simulation.

1. Introduction

Bolted joint is a very popular method of fastening components together. The prime reason for selecting bolts as opposed to welding or rivets are that the connection can be easily released allowing disassembly, maintenance and inspection. It has a various application for mechanical joint like in spacecraft, ship, internal combustion engine, automobile, or oilrig, etc and for civil structure and pipelines.

Based on the service loads there are two types of bolted joints. In tensile joints the bolts are loaded parallel to the bolt axis as shown in Fig.1 while in shear joints the bolts are loaded predominantly perpendicular to the bolt axis as shown in Fig.2. For example the connection of two flanges of a pressure vessel constitutes a tensile joint while the connection of a beam to a column can be considered as a shear joint. [**1**]

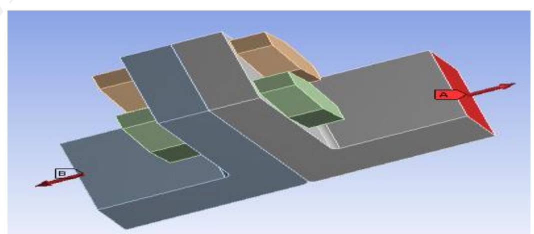

Figure 1 Tensile joint

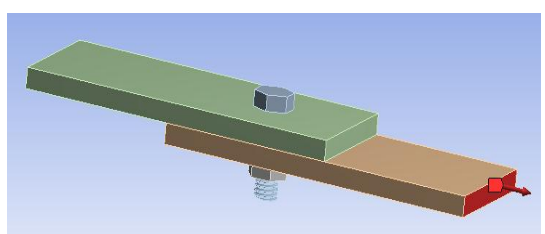

Figure 2 shear joint

Bolted joints are critical structural regions and must be properly designed so that the desired performance from the overall structure is obtained. Because of large stress concentrations, joints can become a source of weakness if proper design practice is not followed. Accordingly, failures typically occurred at connections and interfaces, rather than within the bulk of the system. To provide a safe and cost-effective joint design, it is typical to configure the joint with respect to the geometry and the constituent materials, which affect both strength and failure modes.

Shear joint failure occurs when the joint members are slipped sideways past each other, and eventually cut the fastener. With some shear joints the ultimate joint strength depends only upon the shear strength of the bolts. This type of joint is referred to as a “bearing type” joint. The amount of tension created in the bolts during assembly is relatively unimportant as long as the fastener is retained in the assembly.

Other types of shear joints rely on initial clamp load to resist slip. This type of joint requires a frictional force between the joint members. The shear forces have to overcome the friction developed by the clamp load, which in most cases will be far more than the actual “shear strength” of the fastener itself. This type of joint is common in the structural steel construction industry and may be referred to as a friction-type or slip-critical joint.

Many joints are rarely loaded in pure shear or tension. Some applications subject the joint to a bending force, which results in a combination of tension, and shear load acting simultaneously on the fastener.

1.1 Objectives and Methodology

The chief aim of this project is to predict the stress distribution among region of failures for single lap bolted joint and hence to carry out load carrying capacity of bolt, also to study the effect of various design parameters as mentioned below.

1) To check performance of bolted joint with different design value of preload.
2) To carry out the analysis of bolted joint with different clearance in between the bolt shank and hole on the plate.
3) To check the performance of bolted joint with angle of 45^0 chamfer to vicinity of hole on the plate to be fastened.
4) To carry out the analysis of bolted joint with two different thickness of plate

To meet with above mentioned objectives and for carrying out of analysis, different models have been prepared as mentioned below.

Model 1: Single Lap Bolted Joint which has clearance of 0.2 mm in Between hole on Plate and Bolt Shank. Also value of 32.672 KN was applied.

Model 2: Single Lap Bolted Joint having no clearance in between hole on plate and bolt shank but modeled with value of bolt pretension 32.672 KN

Model 3: Single Lap Bolted Joint having clearance of 0.2 mm in Between hole on Plate and Bolt shank. Also value of 32.672 KN was applied but the thickness of plates is kept 12 mm which is greater than the nominal diameter of bolt.

Model 4: Single Lap Bolted Joint which has clearance of 0.2 mm in between hole on plate and Bolt shank subjected to value of Bolt Pre-tension 32.672 KN. Also Chamfer of angle 45^0 has been given at the vicinity of hole on Plate.

Model 5: Single Lap Bolted Joint having no clearance in between hole on plate and bolt shank but modeled with value of bolt pretension 14.200 KN

For numerical analysis FEM is widely used tool in design, the objective of which is to find the stresses and strain in weaker element of product. So the FEM analysis is carried out by using standard commercial software ANSYS (Workbench) with a version of 14.0 which then compared with analytical results and hence validated with experimental result for one of the case that have been included for analysis.

2. Analytical Calculation

Analytical calculations are done for different model that all are explained in Section1.1. For doing the analytical calculation, material properties and dimensional information should be known and so all the parameters consider for models under analysis are mentioned below.

2.1 Material and Method

A single bolted lap joint consists of a bolt, a nut, washers and two plates. Most often bolts used in machining are made to IS standard 1976 (Reaffirmed 1996). The bolt and nut selected is a hex bolt of ISO Grade 8.8, course series of M10.The mechanical properties and dimensions of the bolts are shown in Table 1. Washers are not used in the design of the specimen because of its influence on the accuracy of torque controlling [5]. The plate material properties used is based on mild steel which is shown in Table 2.

Table 1 Properties and dimension of the bolt and nut

Material type	Medium carbon steel, Quenched and tempered
Modulus of elasticity, E	200 Gpa
Poisson's ratio, Å	0.29
Proof strength	580Mpa
Minimum tensile yield strength	640 Mpa
Minimum tensile ultimate strength	800 Mpa
Nominal length, L	45.74 mm
Nominal diameter, D	10 mm
Height of bolt, H	7.0mm
Width across flat, F	17.9 mm
Width across corners, C	20 mm
Height of nut	8 mm

Table 2 Properties and dimension of the Plates

Material type	Mild steel
Modulus of elasticity, E	200 Gpa
Poisson's ratio, A	0.29
Thickness	10 mm
Width	50 mm

Table 3 Unit and parameter used

Parameter	Unit (SI)
Length	mm
Force	N
Mass	Tonnes
Time	Second
Stress	MPa (N/mm^2)
Density	Tonnes/mm^3

2.2 Design Method

Bolted joints when not properly design may result in structural failure during their life cycle and also due to fatigue loading. In addition a badly design bolted joint may be overweight loading to structural inability .Chief ways in which a bolt can be loaded are in tension, shear and combine shear and tension. A bolt primarily design to withstand tensile forces. While designing a bolted joint in a bolt takes significant amount of tensile and shear loading, proper analysis or calculation must be done to withstand combined stress. Various analysis programe have studied it. [1]

It is also important that in a structural joint, bolt preload or torque applied to secure the component be properly determined. A bolt goes in to a state of tension when torque is applied on the joint some of the factors that affect the bolt tension with the amount of torque applied are nominal bolt diameter, friction coefficient and bolt strength.[5]A rough estimate of the required torque to be applied is given by

$$T=K*D*P$$

When T is the required torque , K is Nut factor, D is bolt diameter and P is total load

2.2.1 Stresses in screwed fastening due to static loading

1) Initial stresses due to screwing up forces
2) Stresses due to external forces
3) Stress due to combination of stresses at 1 and 2.

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2.2.1.1 Initial stresses due to screwing up forces

1) Tensile stress due to stretching of bolt

Initial tension in bolt based on experiment, may be found by relation

$$Pi = 1420d\ N$$

Pi - Initial tension in bolt and (mm)
d – Nominal diameter of bolt (mm)

2) Shear stress across the threads
   a) Average thread shearing stress for screw

$$\tau s = P / (\pi dc \ X\ b\ X\ n)$$

b= width of the thread section at the root.
n- no. of threads in engagement

2.2.1.2 Stresses due to external Forces

1) Tensile Stress
Bolt usually carry a load in the direction of the bolt axis which induces tensile stress in the bolt

$$P = \pi/4(dc^2 * 6t)$$
$$6t = P/ (\pi/4(dc)^2)$$

If the external load is taken up by a number of bolts then

$$P = \pi/4(dc^2 * 6t)*n$$

(If the standard table is not available then for coarse thread, dc = 0.84d, d= Nominal diameter)

2) Shear Stress
When the bolts are subjected to the direct shearing load comes upon a body (i.e. shank) of the bolts & not upon the threaded portion, shearing load carried by the bolts is obtained by using the relation:

$$Ps = \frac{\pi}{4} X\ d^2\ X\ \tau\ X\ n$$

d= Major Diameter
n= No. of bolts.

2.2.1.3 Combined Tension and Shear stresses

When the bolts ore subjected to bolt tension and shear loads in case of coupling bolts or bearing then the obtained from shear loads and that of the threaded part from tensile load . A diameter slightly larger than that required for either shear or tension may be assumed and stresses due to combined load should be checked for the following principle stresses

a) Maximum Principle Shear stresses

$$\tau\ max = 0.5(\ 6t^2 + 4\ \tau^{2})^{1/2}$$

b) Maximum principle tensile stress

$$6t\ (max) = 6t/2+1/2\ (6t^2 + 4\ \tau^{2})^{1/2}$$

These stresses should not exceed the safe permissible values of stresses.

3. Finite Element Analysis of Single Lap Bolted Joint

The three dimensional models were developed by using modeling software Catia and analysis was carried out with use of standard commercial software to predict stress distribution among regions of failure of single lap bolted joints. The FE model results are validated by comparing numerical and analytical results with experimental results for one of the case with design value of pretension, single fastener and clearance in between bolt shank and hole on plate.

3.1 Development and modeling contact surface

Model has three major components on which analysis has performed, they are
1) Flat plates
2) Hexagonal Headed threaded bolt
3) Nut

Overlap two flat Plates are shown in fig. 3 and Hexagonal headed bolt with Nut are shown in fig. 4

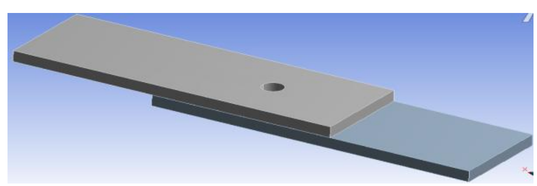

Figure 3 Flat Plates

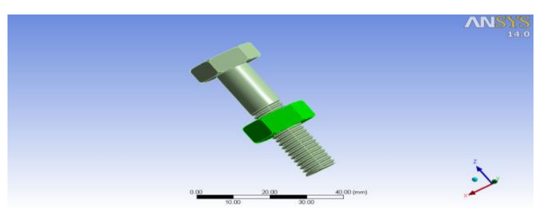

Figure 4 Hexagonal Headed Bolt with Nut

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Modeling of contact in between hole on plate and shank of the bolt, also plate to plate contact has prime importance. Contact in between Hole on plate and shank of Bolt is modeled with No separation contact as bolted joint has to be checked in shear loading so for avoiding penetration of element in to each other No separation contact is best suited. For Plate to plate contact is modeled with frictional contact with coefficient of friction 0.72 .Coefficient of friction (μ) is empirical property of contact materials and contact pressure (p).Product of these two gives the limiting friction shear stress value i.e. μp. The Nut with plate modeled with frictional contact and Nut with threaded shank bolt modeled with bonded contact. All contact modeled for analysis are shown in following fig.5.

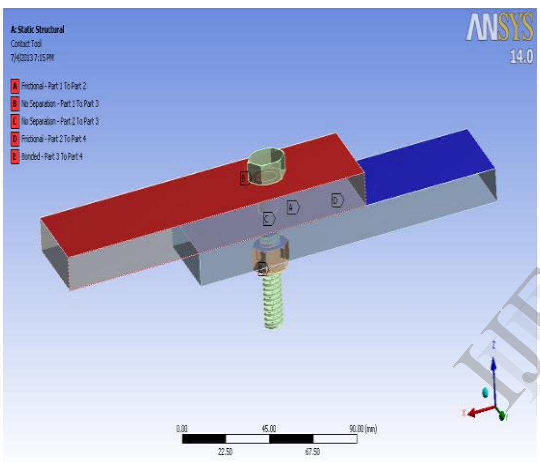

Figure 5 Assembly with preferred contact

3.2 Boundary and Loading Condition

As shown in Fig. 6 boundary condition for different model under consideration have been preferred as follows.

1) Back face of uppermost plate was constrained in X, Y and Z direction.
2) Side faces of both the Plate was constrained in Y and Z direction and kept free in X direction.
3) Bolt Head below face was constrained in X, Y and Z direction.
4) Bolt shank was constrained in Y direction and kept free in X and Z direction.
5) Uniform tangential load was applied at the right end of the bottom plate.

The effect of both non linear material properties and non linear geometry were included automatically in the analysis.

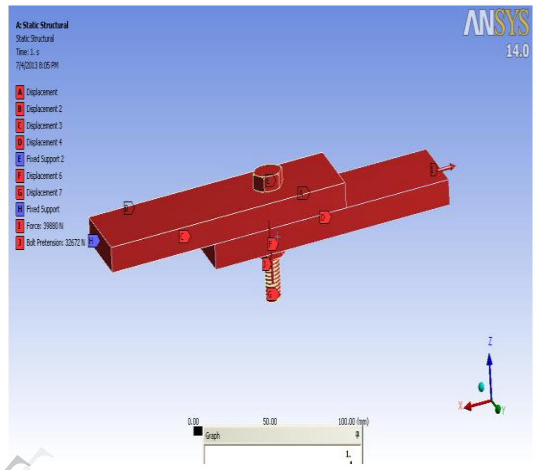

Figure 6 Boundary and Loading Condition

3.3 Pre-tension

In a joint connected using a fastener, it is necessary that the joint is fastened with a particular tension. If the fastener pretension is too tight, it may cause damage to the sructure or the fastener itself might break. On the other hand, if the applied pretension is too less it might result excessive vibration of the structure or unnecessary leaks. So it is necessary that the fastener is tightened with appropriate tension [1]

From analytical and experimental work it has been proved that pretension will provide the best load carrying capacity for the joint. The contribution of the applied load on the bolt load in a pre-loaded tensile joint depends on the stiffness ratio of the bolt and the joint. Also it has been proved that no anti loosening devices are necessary if the bolt is tightened at least 65% of the yield load. Most joint failure are due to insufficient pre-load in the bolt.

For having pre-tension, separate co-ordinate system has been choose where Z axis choose as a Principal axis defined by Global Z axis and four bodies selected for pretension as shown in Fig 7. Model was checked with two different design values of bolt pre-load accordingly 14.2 KN and 32.672 KN to see their effect on load carrying capacity of bolt of single lap bolted joint with shear load.

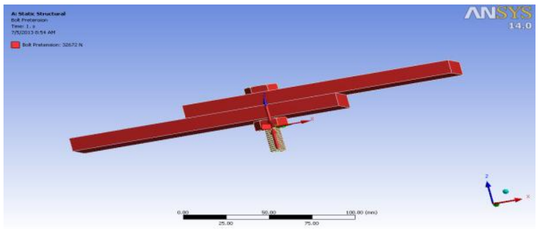

Figure 7 Bolt Pre-tension

3.4 Formulation

For contact analysis and to predict exact stresses induced within the failure regions, the model is meshed with fine meshing of element length 1.7 mm. Flat plates of single lap bolted joint are meshed in square shape and the region around the hole are meshed in circular shape as shown in fig. 8. These plates are meshed using SOLID 186 which is 3D 20- node second order structural solid elements. By combining some of the nodes, the element can degenerate to a triangular base Prism, Quadrilateral based Pyramid, or Tetrahedron. Capability of degeneration is useful since it allows different shapes of element mixed up in a body. Each node has 3 translation degree of freedom i.e. Dx, Dy and Dz. Bolt and nut are meshed using SOLID 187 which is 3D 10 node Tetrahedral second order structural solid elements as shown in fig. 9 and Fig. 10 which have shapes exactly the same as the one degenerate from SOLID 186.

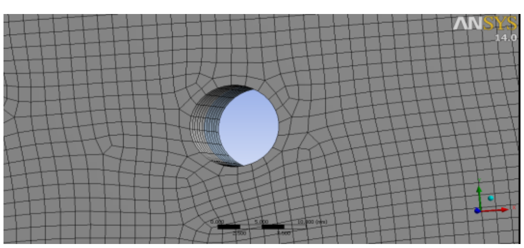

Figure 8 Mesh near the hole

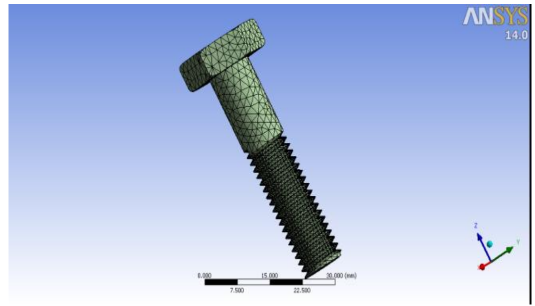

Figure 9 Mesh Model of Nut

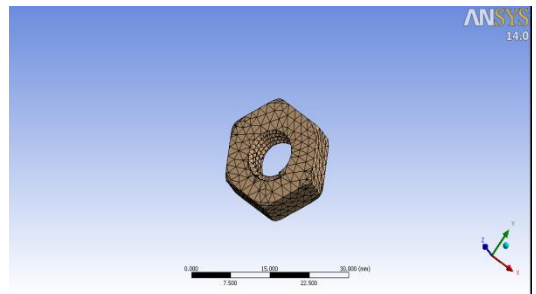

Figure 10 Mesh Model of Nut

4. EXPERIMENTAL PROCEDURE

Experimental work was carried out in strength of material laboratory on digital Universal Testing Machine to carry out corresponding strength values of single lap bolted joint whose physical model for experiment is shown in Fig.11. Design dimension and parameter of model is already listed in table no. 1, 2 and 3.

Experimental model design with clearance of 0.2mm in between hole on plate and shank of bolt with bolt Preload 32.672 KN which is equal to 65% yield value of Bolt. For having appropriate Pre-tension within experimental model, elongation of Bolt with design value of preload is calculated and hence during tightening, elongation of bolt i.e 0.095 mm was measured with the help of Digital Vernier Caliper as shown in Fig. 12. After then the model is mounted within the dies of universal Testing Machine as shown in Fig. 13 and then consecutive load was applied to calculate the Ultimate shear strength of Bolt, reading noted which is equal to 39.880 KN.

Figure 11 Physical Model for Experiment.

Figure 12 Bolt tightened up to desired elongation to develop design pretension.

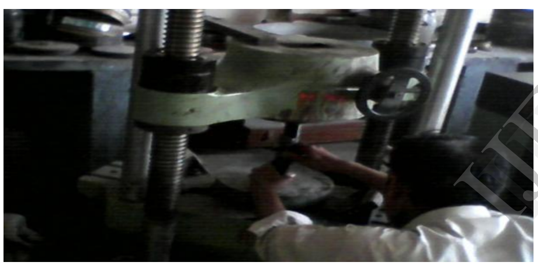

Figure 13 Experimental model position in between dies of UTM.

5. Result and Discussion

From the overview of above analytical and numerical results, it is observed that when there is clearance in between hole on plate and shank of bolt with maximum value of bolt pre-tension i.e. 32672 N as specified in model-1, all the stresses including maximum shear stresses, maximum principal stresses and equivalent stresses are close to analytical results but greater than stresses concenerned with the model-2 where there is no clearance and with same maximum value of Pre-tension, means it is very obvious that when we are not keeping the clearance in between hole on plate and shank of bolt as specified for model-2 magnitude of stresses are reduced.

It is also observed that stresses incurred for Model-5 where there is no clearance in between hole on plate and shank of bolt with minimum design value of bolt pretension i.e. 14200 N are less compared to Model-1 and Model-2, means it is also proved that with increase value of clearance and bolt pretension stresses that induced are increased.

For Model-3 where there is clearance in between hole on plate and bolt shank with Bolt pretension 32.672 KN but the thickness of plates are kept 12 mm which is greater than the nominal diameter of bolt, stresses induced are found maximum as compared other Model, means it is also proved that thickness of the plate greater than nominal diameter of shank is undesirable.

At last for model-4, where there is clearance in between hole on plate and shank of bolt with chamfer angle of 45^0 given at sharp corners of hole on plates, it is observed that stresses induced are very close to analytical results same with Model-1 but the stresses that localized around sharp corners of hole on plates for all other models and which can cause failure of joint due to distortion of corners are moved to safe zones.

Experiment was carried out to calculate the ultimate shear strength of bolt and hence to carry out analytical and numerical analysis with ultimate shear load so as to visualized and validate the failures regions by comparison.

Shear of Bolt along bolt shank is shown in following fig. 14 and fig. 15.

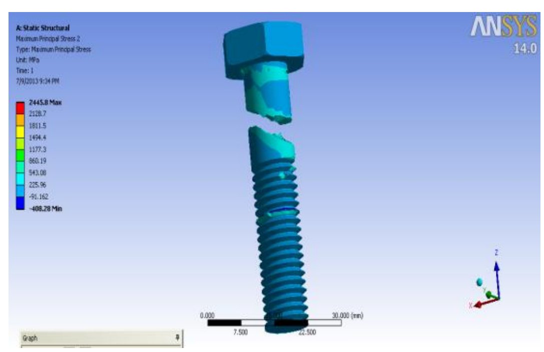

Figure 14 Bolt Sheared along shank region

Figure 15 Bolt Sheared along shank region

6. Conclusion

The results obtained from numerical and analytical analysis are compared for different models of single Lap Bolted joints under shear load and hence following specific conclusion are made.

1) Because of tolerances in the positioning of the holes and the tolerances of bolt diameter clearance is necessary so minimum clearance need to be provided with bolted joints which reduces magnitude of stresses in region of failures.
2) Under dynamic loading condition to avoid self loosening of bolt, it is better to tight the bolt up to 65% of yield load but under static loading, it is better to tight the bolt with design value of preload i.e. Pi = 1420d N
3) Greater thickness of Plate than nominal diameter of bolt increases the stresses within the regions of failure of joint so it is suitable to keep the thickness of plate less than the nominal diameter of bolt.
4) With chamfer angle of 45^0 at the vicinity of hole, stresses which are localized at the corner edge of hole can be shift to safer zones.

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