ABSTRACT

Bolts, rivets or pins are removable joints which are used in Automotive industry for connecting various parts. Therefore, it is of great importance to minimize the effect of the stress concentration and improve the fatigue life. Preload is the technical term for the tension caused by torque tightening the nut that holds the assembled part together. It is evident that, most of machines and structures, such as automotive structures, in their service lives are subjected to multiaxial stresses in which two or three principal stresses vary with time; i.e., the corresponding principal stresses are out-of-phase or the principal directions change during a cycle of loading. Therefore, multiaxial fatigue analysis becomes an important tool for estimating the fatigue strength of these components [1]. In order to estimate fatigue life in the double lap bolted joints, for a reliable design, using multiaxial fatigue criteria is needed due to the complexity of stress distribution as a result of the tightening torque and longitudinal remote load. Therefore, the current investigation has sought to improve the existing body of knowledge about the performance of these multiaxial fatigue criteria in bolted joints in general and the effect of torque tightening on the fatigue life of double shear lap joints in particular. In the present study, the effects of torque tightening on the fatigue life of E34 Steel double lap bolted joints was investigated via experimental and multiaxial fatigue analysis. Cyclic longitudinal loading will be applied to carry out fatigue test.

Keywords – Bolted Joints, Preload, Multiaxial Stresses, Cyclic Fatigue, Tightening Torque

1. INTRODUCTION

In materials science, fatigue is the weakening of a material caused by repeatedly applied loads. It is the progressive and localized structural damage that occurs when a material is subjected to cyclic loading. Fatigue occurs when a material is subjected to repeat loading and unloading. If the loads are above a certain threshold, microscopic cracks will begin to form at the stress concentrators such as the surface, and grain interfaces. Eventually a crack will reach a critical size, the crack will propagate...
suddenly, and the structure will fracture. The shape of the structure will significantly affect the fatigue life; square holes or sharp corners will lead to elevated local stresses where fatigue cracks can initiate. Round holes and smooth transitions or fillets will therefore increase the fatigue strength of the structure. A method for determining the behavior of materials under fluctuating loads. A specified mean load (which may be zero) and an alternating load are applied to a specimen and the number of cycles required to produce failure (fatigue life) is recorded. The bolted joints due to their proven superior performance under static and cyclic loading, have been widely utilized for connecting key components in the structure.

Fatigue failures occur at a stress much below its static strength due to the application of fluctuating stresses. It has been estimated that fatigue contributes to approximately 90% of the mechanical service failures. Fatigue is a problem that can affect any part or component that moves or subjected to high temperatures. Automobile on roads, aircraft wings, and fuel gases, ships at sea, nuclear reactors, jet engines and land-based turbines are all subjected to fatigue failures. There are three basic factors necessary to cause fatigue-

i. A maximum tensile stress of sufficient high value.
ii. A large enough variation or fluctuation in the applied stress.
iii. A sufficiently large number of cycles of the applied stress.

A fluctuating stress is made up of two components- A mean or steady stress and an alternating or variable stress. The stress range is the difference between the maximum and minimum stress in a cycle. And stress ratio and amplitude ratio both are frequently used in presenting fatigue data. These ratios are shown following equations.

\[
\text{Stress Ratio} = \frac{\sigma_{\text{min}}}{\sigma_{\text{max}}} \quad \text{(1)}
\]

\[
\text{Amplitude Ratio} = \frac{\sigma_a}{\sigma_m} = \frac{1-R}{1+R} \quad \text{(2)}
\]
Bolted joints are one of the most common elements in construction and machine design. They consist of fasteners that capture and join other parts, and are secured with the mating of screw threads. There are two main types of bolted joint designs: tension joints and shear joints. In the tension joint, the bolt and clamped components of the joint are designed to transfer an applied tension load through the joint by way of the clamped components by the design of a proper balance of joint and bolt stiffness. The joint should be designed such that the clamp load is never overcome by the external tension forces acting to separate the joint. If the external tension forces overcome the clamp load (bolt preload) the clamped joint components will separate, allowing relative motion of the components.

Fig. 2 Bolted joint

Fig. 3 Double lap bolted joint

2. LITERATURE REVIEW

F. Esmaeili, T.N. Chakherlou, M. Zehsaz, states “Prediction of fatigue life in aircraft double lap bolted joints using several fatigue criteria” states the effects of torque tightening on the fatigue strength of 2024-T3 aluminum alloy double lap bolted joints have been studied via experimental and multi axial fatigue analysis. A non-linear finite element ANSYS code was used to obtain stress and strain distribution in the joint plates due to torque tightening of bolt and longitudinal applied loads. Fatigue lives of the specimens were estimated with six different multiaxial fatigue criteria by means of local stress and strain distribution obtained from finite element analysis.

T.N. Chakherlou, B. Abazadeh, J. Vogwell, explains, “The effect of bolt clamping force on the fracture strength and the stress intensity factor of a plate containing a fastener hole with edge cracks” states the effect which the clamping force, resulting from torque tightening a nut and bolt, has on the fracture strength and the stress intensity geometry factor of a fastener hole containing a symmetrical pair of edge cracks. The joint fracture strengths were obtained using a tensile testing machine. In the numerical investigation, a finite element package was used to model the three test specimen variants used and thereby establish their stress intensity geometry factors. The results show that the bolt tightening torque, and hence the plate clamping force, has a significant effect on reducing the stress intensity factor, and thus the joint fracture strength compared to bolt-less specimens.

Esmaeili, T. N. Chakherlou, M. Zehsaz and S. Hasanifard, states that, “Investigating the effect of clamping force on the fatigue life of bolted plates using volumetric approach” studies the Effect which the clamping force, on fatigue life have been studied on the value of notch strength reduction factor. Fatigue life was estimated by the available smooth S-N curve of Al7075-T6 and notch strength reduction factor obtained from the volumetric approach was used. Then the estimated Fatigue life was compared with the test experiments. Volumetric approach and experimental result showed that Fatigue life improved due to the compressive stresses generated around the plates.
A. Aragón, J. M. Alegre F. Gutiérrez-Solana explains that, “Effect of clamping force on the fatigue behavior of punched plates subjected to axial loading” states a study of punched plate, with a simple clamping nut-bolt, subjected to cyclic axial loading. The coefficient of friction between the materials of the nut and the plate has been experimentally obtained. After that, a 3D finite element simulation has been performed in order to obtain the contact stresses between the nut, bolt and the plate originated by clamping and axial loading. T.N. Chakherlou, M.J. Razavi, A.B. Aghdam, B. Abazadeh states that, “An experimental investigation of the bolt clamping force and friction effect on the fatigue behavior of aluminum alloy 2024-T3 double shear lap joint” studies the effect which the clamping force, resulting from torque tightening a nut and bolt, has on the fracture strength and the stress intensity geometry factor of a fastener hole containing a symmetrical pair of edge cracks. Fatigue tests were carried out on the bolt clamped double shear lap joint specimens made of aluminum alloy 2024-T3.

Richard G. Budynas, J.Keth Nisbett: “Shigley’s Mechanical Engineering Design” studies the Bolt Clamping Force Formulas and theory of Bolt joint diagram. The textbook details the Clamping force calculation and the types of joints theory.

3. DESIGN DETAILS & METHODOLOGY

In order to estimate fatigue life in the double lap bolted joints, for a reliable design, using multiaxial fatigue criteria is needed due to the complexity of stress distribution as a result of the tightening torque and longitudinal remote load. Therefore, the current investigation has sought to improve the existing body of knowledge about the performance of these multiaxial fatigue criteria in bolted joints in general and the effect of torque tightening on the fatigue life of double shear lap joints in particular. In the present study, the effects of torque tightening on the fatigue life of E34 Steel double lap bolted joints were investigated via experimental and multiaxial fatigue analysis. To do so, three batches of specimens prepared and each subjected to torque of 1.5 N-m, 3.0 N-m and 4.5 N-m and then fatigue tests were carried out at different cyclic longitudinal load levels. E34 Steel material with thickness of 2 mm, was selected for manufacturing of the specimens employed in this investigation. This type of Steel has been extensively used in automotive structures.

- **Chemical properties of E34 Steel**-

<table>
<thead>
<tr>
<th>Specification</th>
<th>Grade</th>
<th>C% Max</th>
<th>Mn % Max</th>
<th>P % Max</th>
<th>S % Max</th>
<th>Si % Max</th>
</tr>
</thead>
<tbody>
<tr>
<td>E34</td>
<td>SAIL FORM 34</td>
<td>0.10</td>
<td>0.70</td>
<td>0.030</td>
<td>0.030</td>
<td>0.20</td>
</tr>
</tbody>
</table>

- **Material properties of E34 Steel**-

<table>
<thead>
<tr>
<th>Young’s modulus (GPa)</th>
<th>Yield stress (MPa)</th>
<th>Tensile stress (MPa)</th>
<th>Poisson’s ratio</th>
<th>Elongation (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.85</td>
<td>340</td>
<td>440</td>
<td>0.29</td>
<td>22</td>
</tr>
</tbody>
</table>

The mechanical properties of this material have been illustrated in Tables 3.1 which obtained from tension (static) tests in accordance with ASTM: E8/E8M-13a [11]. Fastener holes of 5 mm diameter were drilled and reamed. Hex head M5 bolts along with suitable types of steel washers and nuts were used to prepare the joint. At the bush outer surface, two strain...
gauges were stuck to measure the compressive axial strain and so the stress in the bush using Hooke’s stress–strain law. Having the bush cross-sectional area at hand and the axial stress, the axial force in the bush and then the clamp force has been determined. To calculate the Clamping force Theoretical formula as given in Eq.2 is used where F_cl is ‘Clamping Force’, K = ‘Coefficient of friction’ and D is ‘Dia of Bolt’. Generally, the Friction between Steel is considered in range of 0.15-0.2. To calibrate the applied torque and clamping force, torques were applied in 1 N m increments from 1 to 7 N m to the nut using a torque wrench, and then the axial strains were recorded for each value of the torques. This test was repeated three times for each case to obtain the mean value of compressive strains (ɛ_m), and determine the corresponding clamping forces using Eq. (2). The elastic modulus for the bush material (E_bush) was also experimentally determined in order to obtain the accurate values for the mean axial clamping force.

- Theoretical Calculation-
  $F_{cl} = \frac{T}{KD} = \frac{1500}{(0.2 \times 5)} = 1000\text{N}$  \hspace{1cm} (3)

- Experimental Calculation-
  $F_d = \frac{E_{bush}A_{bush} \varepsilon_m}{4} = 204.188 \times \frac{\pi}{4} \left(92^2 - 5^2\right) \varepsilon_m = 89.8 \times 10^5 \varepsilon_m (N)$  \hspace{1cm} (4)

Fig. 4 Test specimen configuration and dimensions

Fig. 5 Isometric View
Any assembly will have many components to fulfill the functional requirements. Globally, the OEM practice is to validate any component design after the drawing is created. The created design should meet the performance standards. To test the validity of the designed component is in accordance to the desired performance standard; the designer opts for physical or virtual validation. All the components in the assembly will have their own structural properties like natural frequency, stiffness, mass etc. Hence, the process of validation starts with static test.

![Fig. 6 CAE methodology in design loop](image)

### i. Non-Linear Analysis

When structure response (deformation, stress and strain) is linearly proportional to the magnitude of the load (force, pressure, moment, torque etc.) then the analysis of such a structure is known as linear analysis. When the load to response relationship is not linearly proportional, then the analysis falls under non-linear analysis. e.g., force (stress) vs displacement (strain) curve is non-liner (polynomial). Stiffness \([K]\) is the function of displacement \([d]\) and deals with true stress & strain.

\[
[M] \ddot{x} (t) + [C] \dot{x} (t) + [K] x (t) = F(t) \quad \text{-------- (5)}
\]

Where,

\(M \ddot{x} (t) = \frac{d^2x}{dt^2}\) = are the inertia forces, which are functions of the mass matrix \(M\) and the nodal acceleration vector \(\ddot{x} (t)\),

\(C \dot{x} (t) = \frac{dx}{dt}\) = the viscous damping forces, which are functions of the damping matrix \(C\) and the nodal velocities \(\dot{x} (t)\),

\(K x (t) = \) the elastic forces, which are functions of the stiffness matrix \(K\) and the nodal displacement vector \(x (t)\) and \(F (t) = \) the applied loads.

For non-linear analysis,

\([M] \ddot{x} = 0, [C] \dot{x} = 0, [K] \) is function of \([d]\), \(F(t) = \) constant

Abhilash D. Bhosale, Ashish R. Pawar
ii. Fatigue Analysis

The FEMFAT BASIC program uses the load spectrum as the external stress, similar to the concept of nominal stress. The treatment of the various stress relationships ($R_i$, $a_i$) in the individual load spectrum stages is performed in FEMFAT by determining the local component S/N curve corresponding to the respective stress relationship ($R_i$, $a_i$).

![Schematic representation of a component S/N curve](image)

A local component S/N curve, such as that shown in figure 4.3, is described in a log-log plot by the following three parameters:

- The endurance stress limit $\sigma_{ef}$
- The endurance cycle limit $N_{ef}$
- The slope $k_c$.

Miner’s rule is probably the simplest cumulative damage model. It states that if there are k different stress levels and the average number of cycles to failure at the ith stress, $S_i$, is $N_i$, then the damage fraction, C, is:

$$
C = \sum_{i=1}^{k} \frac{N_i}{N_i^{ef}}
$$

Where,
n_i is the number of cycles accumulated at stress S_i.
C is the fraction of life consumed by exposure to the cycles at the different stress levels.
In general, when the damage fraction reaches 1, failure occurs. The above equation can be thought of as assessing the proportion of life consumed at each stress level and then adding the proportions for all the levels together. Often an index for quantifying the damage is defined as the product of stress and the number of cycles operated under this stress, which is
\[ W_i = n_i \times S_i \] \hspace{1cm} (7)
Assuming that the critical damage is the same across all the stress levels, then:
\[ W_{\text{failure}} = N_i \times S_i \] \hspace{1cm} (8)
The stress amplitude \( \sigma_a \) and the mean stress \( \sigma_m \) can be defined as the nominal or local stress. Accordingly, these concepts are known as the nominal stress concept or principal of local strains. In FEMFAT, these are the local elastic stresses, and can be determined from the external stress by means of a structural elastic FEM analysis of each component.

Using the elementary Miner rule, the creep strength domain S/N curve with slope ‘k’ is extended until stress amplitude \( \sigma_a = 0 \).
Below the endurance stress limit \( \sigma_e \), the modified Miner rule employs a fictitious extension to the creep strength domain with a slope 2k-1.
In automotive industries, fatigue life is estimated by using a numerical technique including both a discretized model of the structure and an associated load-time history that represents a specified event in the life of the structure. To facilitate this technique, FEM can be used. FEM is certainly a great advantage when the fatigue life of complex geometries is being estimated. In this chapter, the advanced fatigue mechanisms were discussed. Whether it is the S-N or E-N approach, high-cycles or low-cycles fatigue, they all require obtaining stresses and strains [8].

4. FINITE ELEMENT ANALYSIS

4.1 Element Type
In analysis C3D10M element are used for analysis. For a 3D stress analysis, ABAQUS offers 4 different classifications of quadratic tetrahedral elements namely C3D10, C3D10M, C3D10H and...
C3D10I. Although it is very clear that C3D10H are hybrid elements and primarily intended for simulating incompressible materials (e.g., Hyperplastic behavior modeling with rubber, human muscle tissue etc.), many users often tend to confuse between using the remaining elements. Let us take a look at using these elements with respect to contact analysis in ABAQUS/Standard.C3D10 are textbook formulation second-order tetrahedral elements. Upon applying a uniform pressure load to its face these elements generate zero nodal forces at corner node rendering them. C3D10 elements are recommended for any contact involving finite sliding with node to surface or surface to surface formulation when used with penalty enforcement method [9].

![Tetrahedral Element](image)

**Fig. 10 Tetrahedral Element**

### 4.2 Material

Most materials that exhibit ductile behavior (large inelastic strains) yield at stress levels that are orders of magnitude less than the elastic modulus of the material, which implies that the relevant stress and strain measures are “true” stress (Cauchy stress) and logarithmic strain. Material data for all of these models should, therefore, be given in these measures. If you have nominal stress-strain data for a uniaxial test and the material is isotropic, a simple conversion to true stress and logarithmic plastic strain is

\[
\sigma_{\text{true}} = \sigma_{\text{nom}} (1 + \varepsilon_{\text{nom}}), \quad \text{----------------- (9)}
\]

\[
\varepsilon_{\text{pl}}^{\ln} = \ln (1 + \varepsilon_{\text{nom}}) - \frac{\sigma_{\text{true}}}{E}, \quad \text{----------------- (10)}
\]

The example below illustrates the input of material data for the classical metal plasticity model with isotropic hardening. Stress-strain data representing the material hardening behavior are necessary to define the model. An experimental hardening curve might appear as that shown below.
First yield occurs at 200 MPa. The material then hardens to 300 MPa at one percent strain, after which it is perfectly plastic. Assuming that the Young's modulus is 200000 MPa, the plastic strain at the one percent strain point is .01 – 300/200000 = .0085. When the units are newton’s and millimeters, the input is

\[
\text{Table 1 Stress-strain data}
\]

<table>
<thead>
<tr>
<th>Yield Stress (N/mm²)</th>
<th>Plastic Strain</th>
</tr>
</thead>
<tbody>
<tr>
<td>200</td>
<td>0.0</td>
</tr>
<tr>
<td>300</td>
<td>0.0085</td>
</tr>
</tbody>
</table>

4.3 Contact Non-Linearity

Abaqus provides more than one approach for defining contact. Abaqus/Standard includes the following approaches for defining contact:

1. General contact
2. Contact pairs
3. Contact elements

4.5 Modelling Details

![Fig.12 Schematic Fig of Double Lap Shear Joint](image)

![Fig.13 Plate orientation for Lap Joint](image)
Table 2 Material Properties

<table>
<thead>
<tr>
<th>Material</th>
<th>Young’s Modulus (MPa)</th>
<th>Poisson’s ratio</th>
<th>Density (ton/mm^3)</th>
<th>Yield Strength (MPa)</th>
<th>Ultimate Strength (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel (E34)</td>
<td>2.1e5</td>
<td>0.29</td>
<td>7.8e-9</td>
<td>340</td>
<td>440</td>
</tr>
</tbody>
</table>

4.6 Boundary Conditions

![Boundary Conditions of Plate for Lap Joint](image)

4.7 Finite Element Analysis Results

1. Case1: Clamping force = 1500 N, Pretension value = 1500N

![Finite Element Analysis Results](image)

Fig. 15 Fig. shows the values of Longitudinal load = 300N, Frequency = 10 Hz.
Compressive stress generated in plate because of clamping = 176 MPa.
Fatigue life at 1500 N clamping force
EXPERIMENTAL & NUMERICAL INVESTIGATION OF PRETENTION EFFECT ON FATIGUE LIFE OF DOUBLE LAP BOLTED JOINT UNDER DYNAMIC SHEAR LOADING

Fig. 16 Fig. shows the values of Fatigue life (min) = 14200 cycles & Fatigue life (max) = 1e6 cycles

2. Case 2: Clamping force = 3000 N, Pretension value = 3000N

Fig. 17 Fig. shows the values of Longitudinal load = 300N, Frequency = 10 Hz
Compressive stress generated in plate because of clamping = 301 MPa
Fatigue life at 3000 N clamping force

Fig. 18 Fig. shows the values of Fatigue life (min) = 64000 cycles & Fatigue life (max) = 1e6 cycles

3. Case 3: Clamping force = 4500 N, Pretension value = 4500N
Fig. 19 Fig. shows the values of Longitudinal load = 300N, Frequency = 10 Hz
Compressive stress generated in plate because of clamping = 426 MPa
Fatigue life at 4500 N clamping force

Fig. 20 Fig. shows the values of Fatigue life (min) = 147000 cycles
Fatigue life (max) = 1e6 cycles

4.8 Effect of Clamping Force on S-N Curve

Case1: Clamping force = 1500 N,
Pretension value = 1500N

Case2: Clamping force = 3000 N, Pretension value = 3000N
EXPERIMENTAL & NUMERICAL INVESTIGATION OF PRETENTION EFFECT ON FATIGUE LIFE OF DOUBLE LAP BOLTED JOINT UNDER DYNAMIC SHEAR LOADING

Case 3: Clamping force = 4500 N, Pretension value = 4500 N

From above 3 case studies of Clamping force on the S-N curve of the material it is quite clear that as the compressive stresses around the plate increase the Fatigue life of the Double Lap shear joint increases.

Fig. 21. Stress Histories

Fig. 22. Mean stress effects on the S-N curve
From above figures it is clear for material of same mechanical properties, compressive stress increases the number of cycles to failure (yellow curve), while tensile stress decreases the number of cycles to failure (magenta). A zero mean stress curve falls in between (green) [10].

4.9 Result Table

<table>
<thead>
<tr>
<th>Clamping Force (N)</th>
<th>Compressive stress generated in plate (MPa)</th>
<th>Fatigue Life cycles (Min.)</th>
<th>Fatigue Life cycles (Max.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1</td>
<td>1500</td>
<td>176</td>
<td>14200</td>
</tr>
<tr>
<td>Case 2</td>
<td>3000</td>
<td>301</td>
<td>64000</td>
</tr>
<tr>
<td>Case 3</td>
<td>4500</td>
<td>426</td>
<td>1.47E5</td>
</tr>
</tbody>
</table>

5. EXPERIMENTAL INVESTIGATION

A. Experimental measurement of Clamping Force using Load Cell

To measure the clamp force (bolt axial tension) at different applied torques, a special experimental method was designed using a steel bush that was placed between the nut and the plate. At the bush outer surface, two strain gauges were stuck to measure the compressive axial strain and so the stress in the bush using Hooke’s stress–strain law. Having the bush cross-sectional area at hand and the axial stress, the axial force in the bush and then the clamp force has been determined. The used method and the bush dimensions were shown in Fig. 23.

To calibrate the applied torque and clamping force, torques were applied in 1 N m increments from 1 to 7 N m to the nut using a torque wrench, and then the axial strains were recorded for each value of the torques. This test was repeated three times for each case to obtain the mean value of compressive strains ($\varepsilon_m$), and determine the corresponding clamping forces using below equation

$$F_d = E_{bush}A_{bush}\varepsilon_m = 204.188 \times \frac{\pi}{4} \left(9^2 - 5^2\right)\varepsilon_m = 89.8 \times 10^2 \varepsilon_m (N)$$

The elastic modulus for the bush material ($E_{bush}$) was also experimentally determined in order to obtain the accurate values for the mean axial clamping force. Where ‘A’ bush is the area of the bush cross section. As the figure shows, the relationship between the clamping force and the applied torque is linear in the range of applied torques. This indicates that the bush material is still in its elastic region, even under the maximum applied torque.

In the next step, three batches of specimens were prepared in which with using a torque-wrench the bolts were tightened up to required amounts of torques, i.e., 1.5 N m, 3.0 N m and 4.5 N m and the
observed $\varepsilon_m = 0.01086$ mm, 0.031626 mm and 0.04654 mm respectively which created clamping forces equal to $F_{cl} = 1120$, 2840 and 4180 N respectively, according to the linear equation obtained from Fig. 24.

Fig. 23 Steel bush and its dimensions in mm and (b) position of prepared load cell in the joint.

Fig. 24 Linear relation between applied tightening torque and clamping force

**B. Experimental measurement of Fatigue Life**

Fatigue tests have been carried out for stress ratio of 0.1 and frequency of 10 Hz using servo-hydraulic 250 KN Zwick/ Roell fatigue testing machine.
Fig. 25 Servo-hydraulic 250 KN Zwick/Roell fatigue testing machine and fatigue test specimen

Fig. 26 S–N curve attained from experimental fatigue tests

Fig. 27 Fractured fatigue specimen with clamping force

$F_{cl} = 1280$ N that subjected to remote longitudinal load 7.0 KN

Fig. 26 shows that total 6 specimen were tested for each of the Pretention values and a curve was fit for all three cases. It was quite clear that increasing the clamping force increased the fatigue life of the specimen.
6. CONCLUSIONS

In this investigation, the effects of clamping force on fatigue strength of double lap bolted joints have been studied via experimental & fatigue analysis based on obtained results following conclusions can be drawn,

- As the tightening torque or the Clamping force is increase it leads to the improvement in the fatigue life of Double Lap Shear joint.
- By increasing the Clamping force, the Double lap shear joint induces compressive stresses around the hole of the plate and this helps in enhancement of the fatigue life of the joint.

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