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Modeling and Simulation of Fatigue Failure in Bolted Joint under Different Loading Condition

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Abstract - This paper includes the fatigue analysis of bolted joint plates. Experimental analysis of the fatigue test is very costly and time consuming, so there is a need of some CAE testing for the better and time saving results. A CAE environment is created for analysis of fatigue life by using Hyperworks software. Three different cases of shear, bending and axial loading have been modeled for fatigue analysis. Fatigue life and damage in the joint area is prime concern in analysis of the system. The result can be useful to provide details of damage causes for manufacturer. Result shows that the fatigue life is maximum in bending load case when compared with the cases of shear, bending and axial loading. The fatigue damage is localized in bending and axial loading test and in the shear loading test it is throughout the bolt length. Area of maximum bolt damage is just below the head of bolt which is in contact with the plates.

Keywords: Fatigue Failure, CAE, Dynamic Loading, Fatigue Damage, FEM, Life Cycle

I. INTRODUCTION

Bolted joints have the advantage over the other connections such as welding and riveted joints of the ease of dismantling the connected components. This is why they are used in engineering applications such as automobile engineering and aerospace engineering. However, bolted joints may experience self-loosening as an unwanted phenomenon which occurs due to vibration in most of the engineering applications. This self-loosening can cause the fatigue failure of the bolts. Due to frequent loading and unloading and cyclic loading the thread slipping between the bolt and nut may also occur. Following common causes of the relative motion occurs in the threads:

1. Bending or shear of parts which results in forces induced at the friction surface. If slip occurs, the head and threads will slip which can lead to loosening.
2. Thermal effects caused as a result of either differences in temperature or differences in clamped materials.
3. Forces applied on the joint lead to slipping of the threaded joints leading to bolt loosening.[1]

In the bolt connections fatigue crack initiation and growth occurs when cyclic stresses exceed the fatigue strength of local material for a sufficient number of loading cycles. Fatigue performance is affected mainly by mean stress, stress amplitude, assembly parameters, and geometry and fastener materials. Understanding the preloading of the bolts is very critical in fatigue damage of the fasteners because the loading on the bolt is minimal until the working load exceeds preload. If cyclic loads are smaller then the preload no cycle damage or very little damage will occur to the fasteners.[2]

Many of the research in past decade studied the different factors of bolts which influence the fatigue life. Manufacturing method, types of threads and types of loading condition (elastic and plastic limits) are few of them .Many numerical and analytical methods were developed to study the fatigue failure of bolted joints in which it is clearly shown that it is influenced by stress concentration factor at the root of bolt where the intensity of the load is highest. Head fillet, thread run out and first thread engage with nut are three locations where due to high gradient of stress reduction in cross section exhibit which promote the failure[3-7]. K Din Et.al found that structure will have much shorter life which undergoes a low cycle fatigue loading then the one under high fatigue loading cycle.[8]

As fatigue testing is very time and cost consuming method .Finite element analysis can provide an alternative method for the testing analysis. Researchers have carried out analytical studies on the bolted joint using the finite element method. In the past studies numerical methods have been developed through axis symmetrical models to calculate the load and stress distribution along the bolt threads. Pitch difference in nut and bolt is also a deciding factor for fatigue life. Axial fatigue test in term of fatigue limit curve 10^6 cycle shows that a little modification in

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strength of material did not affect the fatigue life much [9-13].

There are research studies where special purpose test rigs developed for the testing of the fasteners. Experimental studies were made on these test rigs to understand the phenomenon of self loosening on fasteners. Special type of test rig have also been developed to understand the phenomenon of self loosening in shear, bending and axial loading condition[15]. But the fatigue life for different loading conditions is not being discussed and investigated. This work attempts to find out the fatigue life of the bolt in shear, bending and axial loading condition employing the finite element method using the Hyperworks software.

II. MODELING OF TEST SETUPS

Bolted joint testing machine has been designed and developed using computer aided environment for checking the performance of loosening of threaded fasteners. Unlike a single structure or a component, a whole machine tool structure is composed of many components connected with many joints interfaces. The whole machine structure is an integrated system. Therefore, the structure is a very complex one. Figure 1 shows the complete assembly model. For the fatigue analysis we have considered only the bolt and joining plates of different loading setup to reduce the complexity of analysis[14,15]

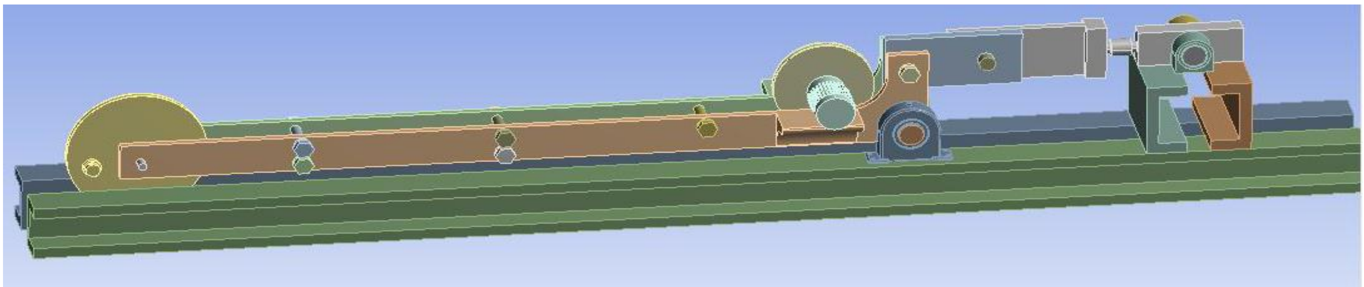

Figure 1 CRE- O model of bolted joint testing rig for Shear Attachment

Figure 2, 3, and 4 show the drawings of plates with actual dimensions for the shear, bending and axial conditions respectively.

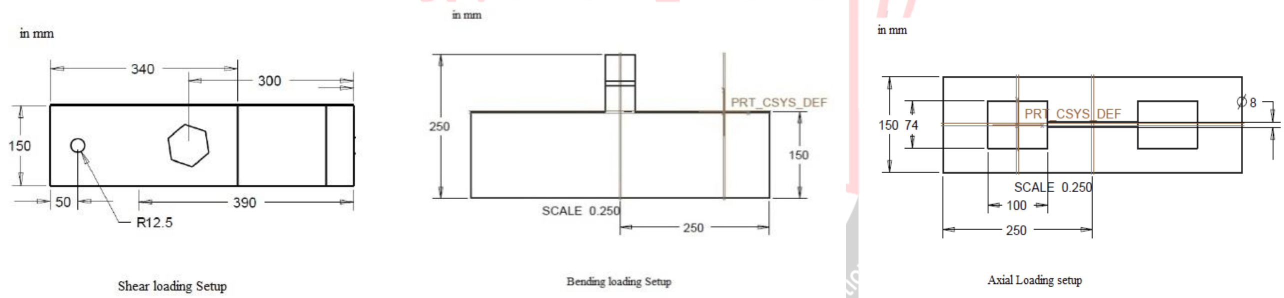

Figure 2,3,4 bolted joint plate dimensions for Shear, axial and bending attachment

Figure 5,6,7 shows the model of plates which are considered for the analysis in shear bending and axial condition.

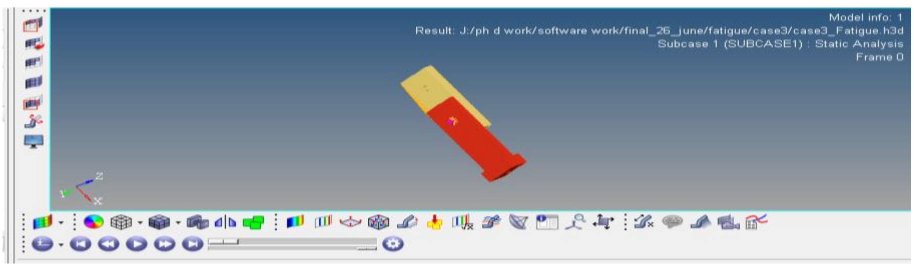

Figure 5 CRE- O model of bolted joint testing plates for Shear Attachment

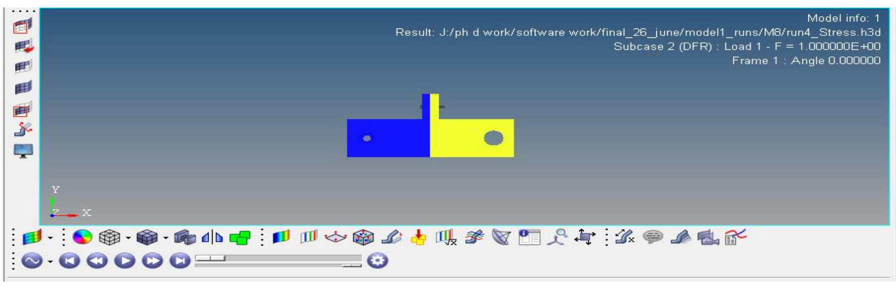

Figure 6CRE- O model of bolted joint testing plates for bending Attachment

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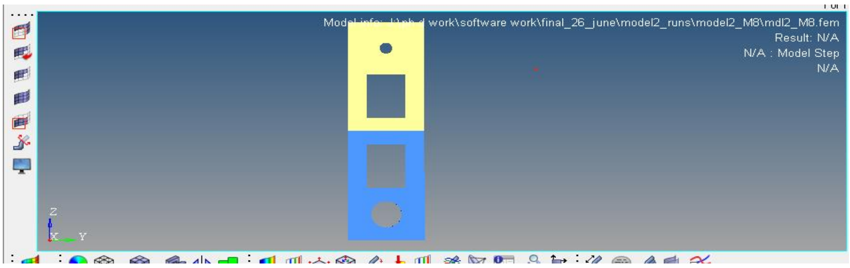

Figure 7 CRE- O model of bolted joint testing plates for Axial Attachment

III. FATIGUE ANALYSIS

To estimate fatigue life of bolts using different multiaxial fatigue criteria, stress and strain distributions in the experimentally tested specimens are required. Due to nonlinearities of the bolt clamped and interference fitted-bolt clamped specimens subjected to cyclic fatigue loading finding analytical solutions around the hole is a very difficult task. Consequently to accomplish this aim, an explanation of the FE simulation of interference fitted-bolt clamped specimens (being the most complex model) is presented here. The FE models were built according to the dimensions of the specimens for all the three setups. To do so, only a joint part of the total test rig was used in the simulation to avoid an unnecessarily large model, as shown in Fig.5, 6 and 7. This simplification does not affect the accuracy of the analysis, because the primary simulations using bigger model revealed that the stress concentration (induced by the hole) diminishes at the selected distance (length) to a uniformly distributed longitudinal stress. Furthermore, as the washer and bolt have almost the same mechanical properties, they were modeled as one part to avoid using extra contact elements which would induce longer processing times. As multiaxial stress states are observed in stress concentration zones, we have implemented Dang Van criteria. The purpose is also to investigate the validity of the so-called critical plane approach versus the integral approach. Dang Van criterion is related to the critical plane approach deals with the integral approach.

It was originally presented through considerations established at the microscopic scale, even if the criterion uses macroscopic stresses which are the alternating part of the shear stress τ_{ha} and the hydrostatic pressure p. The damage fatigue function F of the criterion is a maximization of the damage indicator by plane Eh, defined on the material plane which unit normal vector is h:

$$F_h = Max_t\{\frac{\tau_{ha}(t)+\alpha.P(t)}{\theta}\} \qquad (1)$$

$$\text{Where } P(t) = \frac{I_1(t)}{3} = \frac{\sigma_{11}(t)+\sigma_{22}(t)+\sigma_{33}(t)}{3}$$

$I_1(t)$ is the first invariant of the stress tensor at time t, $\tau_{ha}(t)$ is the alternating shear stress acting at time t on the material plane; it is obtained by determining the smallest circle surrounding to the load trajectory. The plane where F_h is maximal is called critical plane. It allows the criterion to express the fatigue severity of the multiaxial loading cycle. The fatigue function of the criterion is then written as:

$$F_{DV} = Max_h(F_h) \qquad (2)$$

A fatigue damage model was developed in which fatigue loading was characterized by a constant load equal to the maximum fatigue load. This model was generalized to accommodate the effect of load on all the three setup. The generalized model was employed to predict fatigue response of bolted joints under cyclic loading using the same material damage parameters. In this current paper, this model has been adapted for fatigue loading. Figure 8,9,10 shows the hyperworks software results of fatigue loading testing done for maximum load 2 ton (as per the design of the test rig)[15]. Fracture plane was generally perpendicular to the bolt length, with no significant plastic deformation. When the bolt to be loaded in tension and bending, fracture perpendicular to joint. As indicated, loading on the bolt is minimal until the load exceeds preload. Thus, for a condition where cyclic loads are smaller than preload, little or no cyclic damage occurs to the fastener. The concept of higher preloads resulting in increased fatigue performance is a point of discussion, which suggest that increasing the tightening force of a bolt may increase propensity for failure.

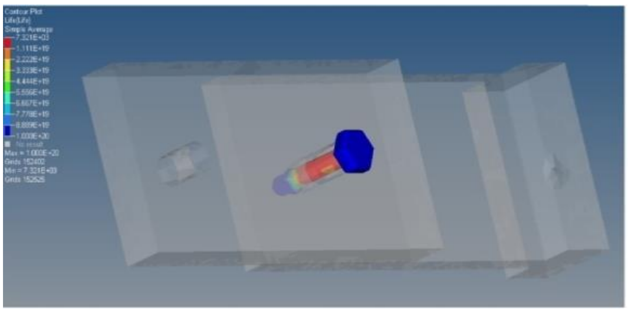

Figure 8 Fatigue life of M8 bolt in shear test rig

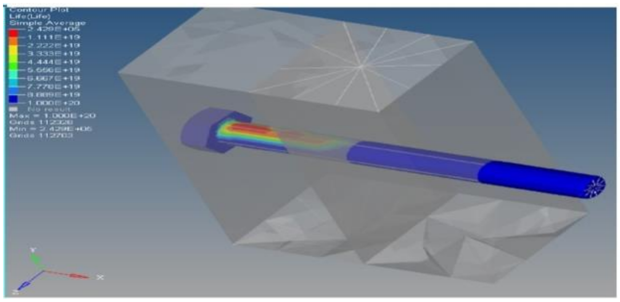

Figure 9 Fatigue life of M8 bolt in bending test rig

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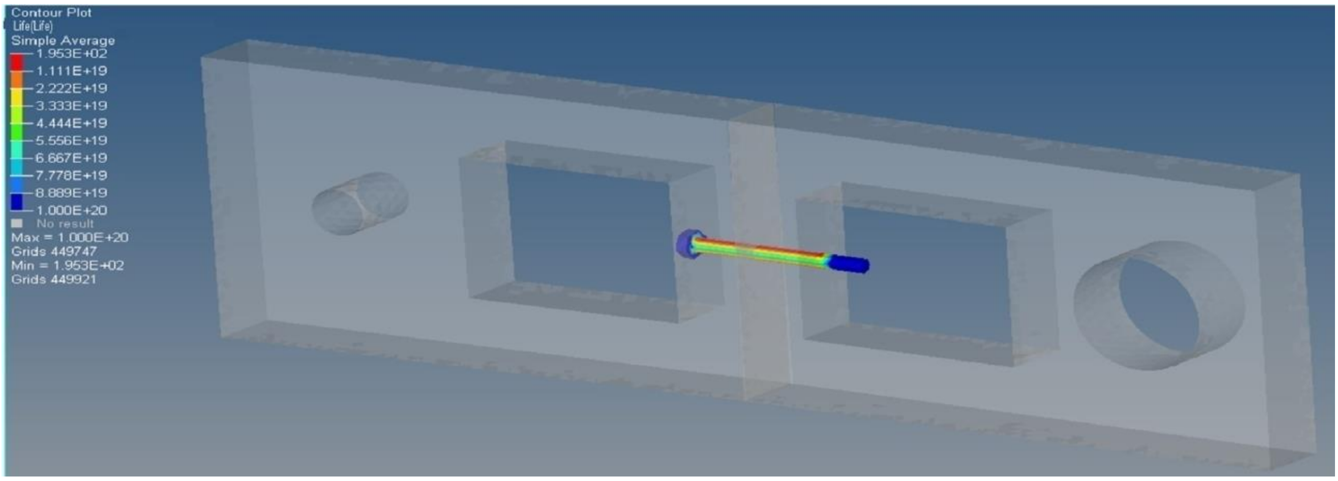

Figure 10 Fatigue life of M8 bolt in axial test rig

Tests have shown that having the bolt head face too close to the plate joining region can result in premature failure. The thread run-out region is poorly formed and it is an area of stress concentration.

Table1 Fatigue life of M8 Bolt in different loading conditions

Loading Setup	Fatigue life (no of cycles)
Shear Loading	7.3X10^3
Bending Loading	2.429X10^5
Axial Loading	1.9X10^2

The highest stressed thread is the first one near the bolt head phase, having this region in close proximity which results in a low fatigue strength Table 1 lists the fatigue life of M8 bolt on three different loading setups obtained namely shear loading, bending loading and axial loading. Figure 11,12and 13 show the damage due to fatigue fracture in shear, bending and axial loading. It is very clear from the analysis that damage is maximum at just below the head and contact point of the plate and bolt which correlate with the fatigue test results.

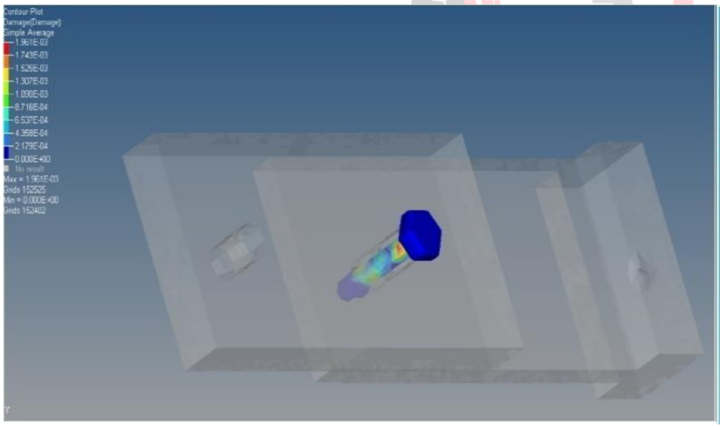

Figure 11 Damage of M8 bolt in shear test rig

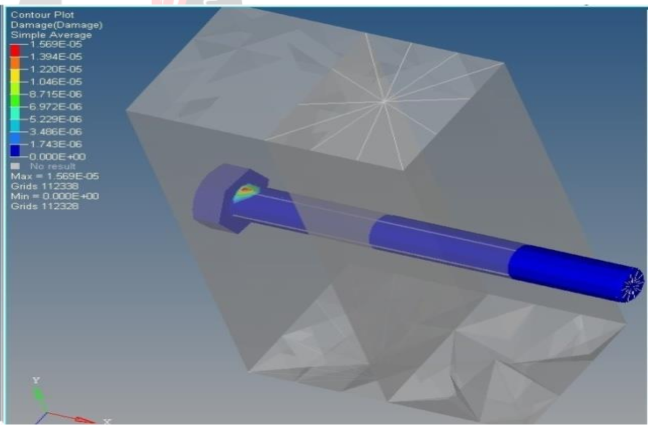

Figure 12 Damage of M8 bolt in bending test

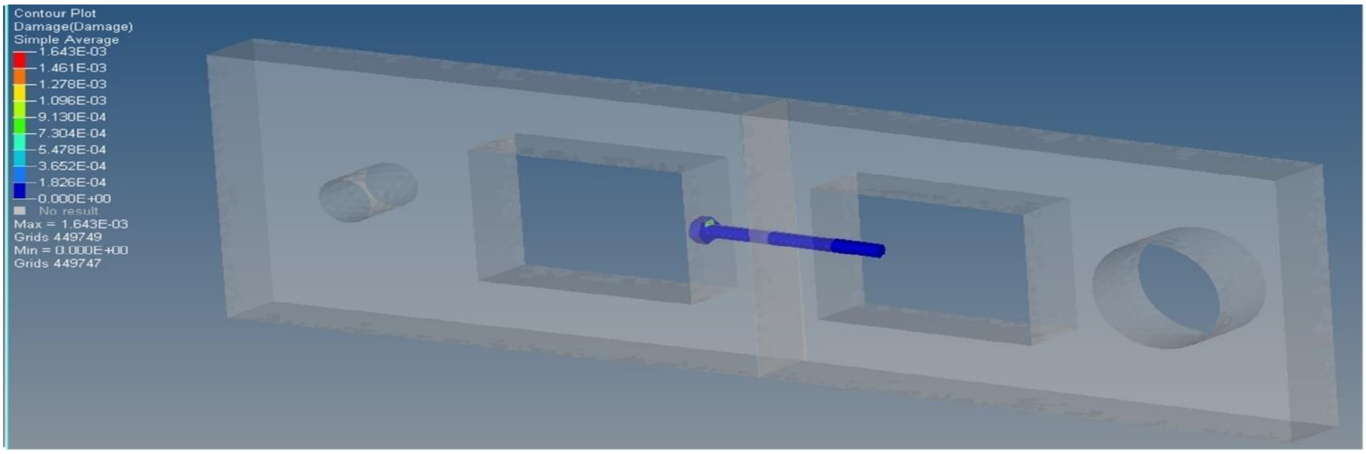

Figure 13 Damage of M8 bolt in Axial test

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In the shear loading the fatigue damage is throughout the bolt length but in the bending and axial loading the damage is only at lower part of bolt edge, which is in contact with plates. These results indicate that the use of washer whether plain or spring washer is really important to increase the fatigue life of the bolts. Result was obtained by CAE Analysis on the crack front in the way already described. With the intention of obtaining enough data points, three specimens were employed to construct. The scatter observed in this graph may be attributed to the FEM Analysis technique, and principally to the use of several specimens, in which there were some variations in the aspect ratio between specimens with similar crack lengths. This results in multiple cracks that initiate on separate planes and eventually grow together to minimize crack surface energy. Fatigue fractures typically exhibit where high stresses cause multiple crack initiation, or in areas of high stress concentration, such as at the threaded bolt head and thread initiation. However, including the fatigue crack growth life can improve the fatigue estimation life and reduce the absolute error of multiaxial fatigue criteria. As a preliminary approximation, with the crack size on the entire range a finite element analysis was performed in order to collaborate with the finding a curve with similar profile. The analysis was performed in very simple way using 3D elements, (i.e. different crack sizes and the same applied load). This point can be improved through more specific studies, either numerical or experimentally.

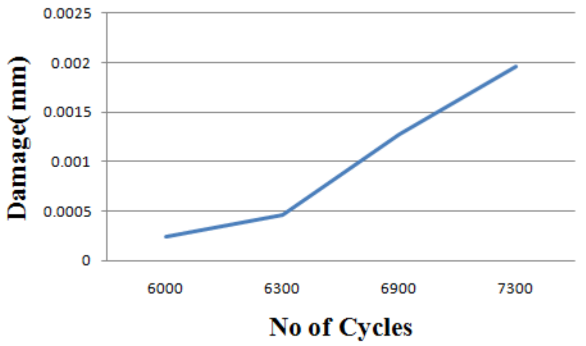


Figure 14 Damage Vs No of Cycle in Shear loading

Figure 15 Damage Vs No of Cycle in Bending loading

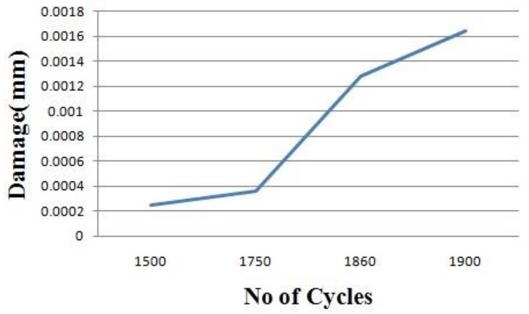
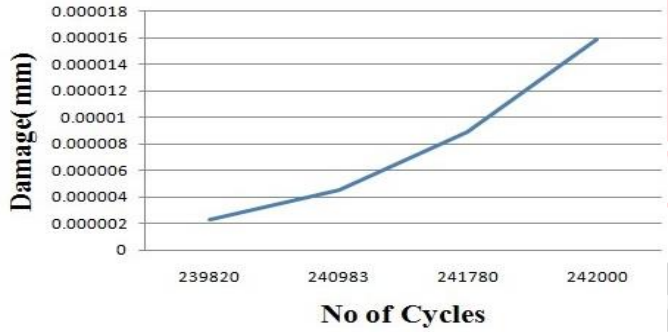


Figure 16 Damage Vs No of Cycle in Axial loading



It can be seen from the figure 14, 15, 16 that crack growths curves that loading stage with the lower fatigue load level had a longer crack length before final failure. That was because a smaller ligament of undamaged material was required to sustain this lower fatigue load level. Furthermore, crack length before final failure for axial and bending loading was noticeably shorter than the crack length of shear loading before final failure. Although the maximum fatigue loading cycle of the bending loading is maximum and fatigue load of the axial loading is more severe than the shear and bending loading. This was somewhat surprising that more loading cycles were applied before the final failure and most of these were at the lower level of loading which produced damage that was more uniformly spread although slower in rate in case of shear loading. As shear loading damage was more widespread, it is required to prevent static failure at the higher maximum fatigue load. Fatigue cracks usually initiate in the thread roots but can also initiate under the bolt head. Often failure in this location is the result of an inadequate under head radius (resulting in a high stress concentration) or the bolt being mounted on an inclined surface. A small joint angle (such as two degrees) can have a dire effect on fatigue strength. This effect has caused many a service problem in the past; a comparatively common problem when bolting welded structures together. The preload improves the fatigue behavior of the bolted joints by making decrease the stress amplitude even if the stress mean value increases. Additionally the increase of the increase of the bolt location has a detrimental influence. Taking into account the fillet radius under the bolt head greatly improves the fatigue resistance of the bolt by reducing the stress concentration factor. In this area stress states induced by the loading of the tee-stub are multi-axial and require thus to be analyzed by multi-axial fatigue criteria.

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IV. CONCLUSION

The models presented in the study are based on the actual experiment setup which used for studding loosening behavior of the bolted joint under dynamic condition. In this research study an attempt was made to determine fatigue life of the M8 bolt under the maximum loading condition of the design test rig.

The fatigue behavior of the bolted joints under shear bending and axial cyclic loading was predicted using a fatigue damage model. Model is reduced to avoid complexity of the analysis because there are so many parts in the assembly. Analysis result shows that the fatigue life (No of Cycles) is very less in the axial loading test compare to other two tests of bending and shear loading. Also bending loading has the maximum fatigue life then other two. The damage is local area concentrated in axial and bending loading condition and in shear loading it is throughout the bolt length. The FE analysis has shown that for a given applied stress, the intensity factor at the tip of a crack at the edge of a bolt of a bolted joint is lower edge cracks. This is due to the normal stress and friction force between contacting surface of bolt head and plate which acts as a resistant force against external applied load and reduces the stress (caused by external load) around the hole and crack tip. the fatigue life of the bolt-nut connection can be improved by introducing the suitable pitch difference and thread bottom radius because the initiation and the propagation of cracks may be changed. It can be concluded that the ratio between the number of cycles to crack initiation 25 and the number of cycles used in the crack propagation process is affected by the relative slip amplitude The intensity factor reduction is more pronounced at the crack tip at the plate surface compared to the plate plane. But in all the cases the maximum damage is near the bolt head just at joining face of the plate and bolt, which indicate towards the need of the washers (plain or spring) to prevent the fatigue damage so early in the fatigue life.

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