Journal of Solid Mechanics Vol. 9, No. 1 (2017) pp. 1-11

Evaluation of Fatigue Life Reduction Factors at Bolt Hole in Double Lap Bolted Joints Using Volumetric Method

F. Esmaeili [1,*], S. Barzegar [2], H. Jafarzadeh [3]

[1] *Department of Mechanical Engineering, University College of Nabi Akram (UCNA),Tabriz, Iran*
[2] *Faculty of Mechanical Engineering, University of Tabriz, Tabriz, Iran*
[3] *Department of Mechanical Engineering, Tabriz Branch, Islamic Azad University ,Tabriz, Iran*

Received 2 September 2016; accepted 15 November 2016

ABSTRACT
In this research, the influence of bolt preload on the fatigue strength of 2024-T3 aluminium alloy double lap bolted joints has been studied experimentally and numerically. To do so, three sets of the specimens were prepared and each of them subjected to tightening torque of 1,2.5 and 5 *N-m* and then fatigue tests were conducted under various cyclic axial load levels. In the numerical method, the influence of bolt preload on the fatigue life of double lap bolted joints were studied using the values of fatigue notch factor obtained by volumetric approach. In order to obtain stress distribution around the notch (hole) which is required for volumetric approach, nonlinear finite element simulations were carried out. To estimate the fatigue life, the S-N curve of plain (un-notched) specimen and the fatigue notch factors obtained from volumetric method were used. The estimated fatigue life was compared with those obtained from the experiments. The investigation reveals that there is a good agreement between the life predicted by the volumetric approach and the experimental results for various specimens with different amounts of bolt preload. The volumetric approach and experimental results showed that the fatigue strength of specimens were improved by increasing the bolt preload as the result of compressive stresses which appeared around the bolt hole. © 2017 IAU, Arak Branch.All rights reserved.

Keywords : Clamping force; Bolted joint; Hybrid joint; Tightening torque.

1 INTRODUCTION

MANY of conventional airplane structures are fabricated by high strength alloys with low density, to provide optimal strength-to-weight ratio material. Aluminum alloys are the most common materials that have been used in aerospace structures due to their advantages such as low weight, relatively high strength and good resistance against corrosion. Among these alloys, aluminum 2024-T3, employed in this study, has appealing properties of high strength and ductility, and is widely used in aerospace structures such as C-130 Hercules aircrafts [1-4].

Mechanical joints are typical tools for assembling the structural components and have been extensively used in the aircraft manufacturing processes for composite or metallic structures. This type of joints offers many advantages over the other joining methods. In the first place, mechanical joining is unique in that it is primarily dependent on the structures which being jointed, and only secondarily dependent on the materials of which these structures are composed. No bonds need to be formed for joining, nor do any need to be broken to disassembly. Hence,

*Corresponding author. Tel.: +98 914 740 8469; Fax: +98 41 34442095.
E-mail address: esmaeili@ucna.ac.ir (F.Esmaeili).

© 2017 IAU, Arak Branch. All rights reserved.

mechanical joining methods allow simple and practical disassembly without damaging the jointed parts. Among the mechanical joints, bolted joints are the vital components in aerospace structures. However, one of the major disadvantages of bolted joints is that, it can cause undesirable stress concentration at the fastener holes [5,6]. Such stress concentration is a particular concern in fatigue critical structures but can also exacerbate or accelerate corrosion. As a result, they are frequent sources of failure in aircraft structures. Conversely, due to ease of assembling and possibility of dismantling, mechanical joining methods have excessive use in aerospace constructions. Therefore, it is necessary to reduce the effect of the stress concentration and reach enhanced fatigue life. According to the results of previous researches, bolted joints have higher tensile and fatigue strengths than welded, riveted and also pinned joints [6-9].

When a nut and bolt are used to joint mechanical members together, the nut is tightened by applying torque and a force is exerted on the nut and then the bolt and nut pulled toward each other's. This force creates tension in the bolt, which clamps the assembled parts of the joint together. Preload is the technical term for the tension caused by tightening the nut that holds the assembled part together [10-12].

As be mentioned, fatigue cracks in structural components usually start at a fillet, groove, hole, rivet, and other discontinuities or notches. It can be found from the theory of elasticity, that the peak stress near the notch root is equal to the average or nominal stress in the surrounding neighborhood multiplied by the elastic stress concentration factor,. Although fatigue strengths are considerably reduced by notches, the reduction is often less than the elastic stress concentration factor, k_t , and a fatigue notch factor, k_f has, therefore, been introduced [13,14].

Notch effects in fatigue are described by the fatigue life for the same stress amplitude being less, for a notched component than for a similar plain (un-notched) specimen. In other words, based on the same fatigue life, the fatigue strength of plain specimen should be higher than notched specimen by a fatigue notch factor [17]. A typical S-N curves for plain (un-notched) and notched specimens were shown in Fig. 1 [18].

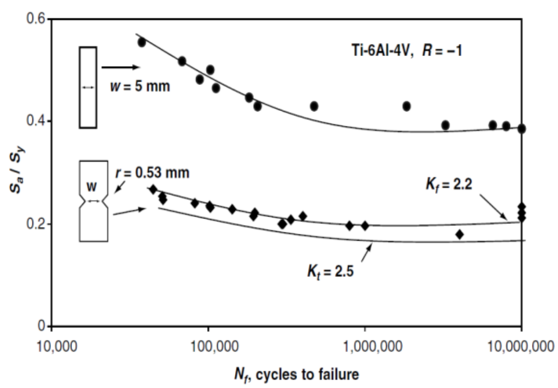


Fig.1
S-N curves for plain and notched specimens [18].

In general, k_f is equal to or less than k_t and is defined by Eq. (1) as follows [18]:

$$k_f = \frac{unnotched\ \ fatigue\ strength}{notched\ \ fatigue\ strength} \tag{1}$$

A relation between the k_f and k_t is given by the notch sensitivity factor, q, which is defined by Eq. (2) as:

$$q = \frac{k_f - 1}{k_t - 1} \tag{2}$$

The value of q lies between 0 and 1. When k_f equals to k_t , $q = 1$ and the material is said to be fully notch sensitive in fatigue. If the presence of a notch does not affect the fatigue strength, $k_f = 1$ and $q = 0$ and the material is notch sensitive. It was found that the values of q is related not only to the material type, but also on the stress distribution around the notch, the size and shape of the specimen, shape of notch, loading type, and etc. Consequently, the values of notch sensitivity cannot be regarded as a material constant.

© 2017 IAU, Arak Branch

Two commonly used equations, proposed for the notch sensitivity factor, are Peterson and Neuber [19,20] equations. Peterson has observed that good approximations for the notch sensitivity factor can also be obtained by Eq. (3):

$$q=\frac{1}{1+\frac{a_p}{r}} \tag{3}$$

where r is the radius at the notch root and a_p is a material constant.

Neuber has developed the following approximate relationship for the notch sensitivity factor:

$$q=\frac{1}{1+\sqrt{\frac{a_N}{r}}} \tag{4}$$

where a_N is the material constant which depends on the grain size. In addition, Heywood [21] represented another empirical equation for relation between k_f and k_t. Heywood's representation of extensive fatigue is somewhat different from Neuber's and Peterson's. It is in the form:

$$k_f=\frac{k_t}{1+2\frac{\sqrt{a'}}{\sqrt{r}}\left(\frac{K_t-1}{K_t}\right)} \tag{5}$$

where a' is related to the length of equivalent material flaws. As mentioned previously, the accuracy of these relations is not known and requires to be assessed by large number of expensive and time consuming experimental fatigue tests.

As mentioned earlier, it has been identified that the use of the maximum peak stress at the notch root is often insufficient for assessing the fatigue life of notched components. The stress distribution in the vicinity of the notch root is important factor in analysis of fatigue process. The stress gradient near the notch root has been used in several early investigations, resulting in the fatigue predictions of Neuber [20] and Peterson [19]. More recently, the method of Neuber was modified by considering both of the elastic and elasto-plastic stress distribution and a different method to calculate the effective distance was proposed in order to estimate the fatigue life of notched specimens. This method is known as the volumetric method. The volumetric method is a critical distance method (TCD) which was developed based on the elastic–plastic stress distribution. TCD can be divided into point method, line method, respectively. In the point method, specific distance from notch tip and corresponding stress along ligament are considered to assess fatigue life of notched components. The critical distance can be summarized as grain size, plastic zone distance, and also, in terms of the parameter which is a function of the material's threshold and fatigue limit. It should be noted that, the critical distance stress is defined as the average stress within the critical distance near the notch root. In the point method, specific distance from notch tip and the corresponding stress from stress distribution are considered to assess fatigue life of notched components. The line methods calculates average stress by numerical integration techniques for effective specific distance from notch roots including or excluding weight effects of stress gradient and material properties. The 2D and 3D methods calculates average stress in stress filed near notches using area and volume integration [18].

In this research, the influence of bolt preload on the fatigue reduction factor at the bolt hole and the fatigue strength of double lap bolted joints will be studied via experimental and numerical analysis. To do so, three sets of specimens will be prepared and each subjected to the torques of 1 *N-m*, 2.5 *N-m* and 5 *N-m* and then fatigue tests will be carried out under various cyclic longitudinal load levels. In the numerical method, the influence of bolt preload on the fatigue strength of double lap bolted joints will be investigated using the values of fatigue notch factors obtained by volumetric approach. To obtain stress distribution around the notch (hole) which is required for volumetric approach, nonlinear finite element simulations were carried out.

© 2017 IAU, Arak Branch

2 FUNDAMENTALS OF VOLUMETRIC METHOD

From a physical point of view, the volumetric method is as an inventive technique in order to predict the fatigue life of notched specimens. This approach is based on two assumptions. Firstly, fatigue damage in notched specimens is dependent on the crack opening stress distribution and stress relative gradients in the vicinity of notch root. The relative stress gradient illustrates the severity of the stress concentration around the notch and crack tips. Secondly, some finite volume of material must be involved for the fatigue damage process to proceed. In other words, the stress that controls the initiation of fatigue damage is not the highest stress at notch root, but rather the somewhat lower value that is the average out to an effective distance [22]. Fig. 2 shows a typical elastic–plastic crack opening stress distribution and stress gradient from a notch root.

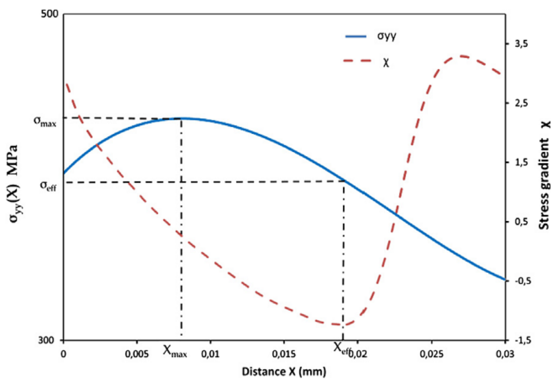


Fig.2
Elastic–plastic stress distribution and stress gradient from a notch root.

According to this figure, three different regions can be recognized from the stress distribution near the notch root. The maximum stress and its distance can be seen in first region. The essential specification of this region is the presence of maximum stress in stress distribution obtained via nonlinear finite element analysis. In second region, the stress value is reducing and its value reaches to the effective stress σ_{eff}. According to the volumetric method, the fatigue notch factor is calculated using the Eq. (6) [16]:

$$k_f = \frac{1}{X_{eff}\sigma_n}\int_0^{X_{eff}} \sigma_{yy}(x)(1-x\chi)dx \tag{6}$$

where X_{eff}, σ_n and σ_{yy} are the effective distance, net stress and crack opening stress, respectively. In addition at above equation χ is the relative stress gradient which is defined by Eq. (7) as [16]:

$$\chi = \frac{1}{\sigma(x)}\frac{d\sigma(x)}{dx} \tag{7}$$

Furthermore, the net stress, σ_N in the vicinity of the bolt hole is given by Eq. (7) as follows:

$$\sigma_N = \frac{F}{(w-d)t} \tag{8}$$

where W is the width of the plate, d the diameter of the bolt hole and t the thickness of the plate.

In the volumetric method, the prediction of fatigue life of notched specimens has been performed by means of the fatigue curve of plain specimen (reference curve), mechanical properties and fatigue notch factor values obtained from Eq. (1). In this method fatigue notch factor is determined using the stress distribution in the vicinity of notch tip by nonlinear elastic-plastic finite element analysis. The fatigue notch factor calculation and the use of reference curve lead to notched specimens fatigue curve.

© 2017 IAU, Arak Branch

3 EXPERMENTAL PROCEDURES

The employed specimens in this study were fabricated from 2024-T3 aluminium alloy with thickness of 2*mm*, whose mechanical properties have been presented in Tables1. In addition, configurations and dimensions of prepared specimens have been illustrated schematically in Fig. 3.

Table 1
Mechanical properties of 2024-T3 aluminium alloy.

Young's modulus (GPa)	Yield stress (MPa)	Tensile strength (MPa)	Poisson's ratio	Elongation (%)
73.4	315	550	0.33	0.18

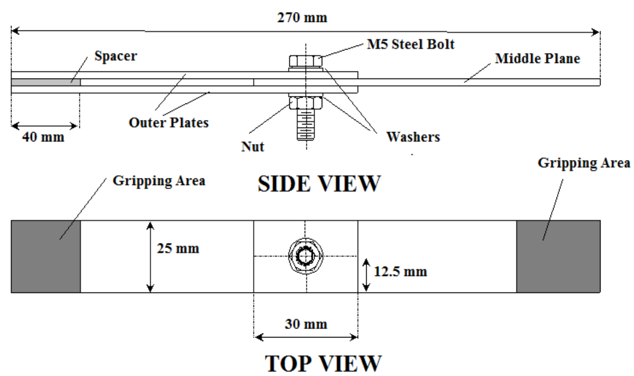
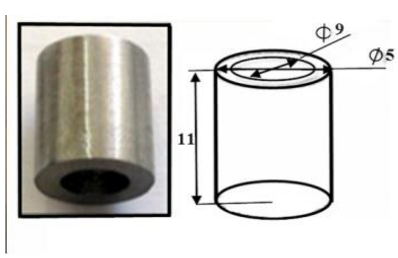


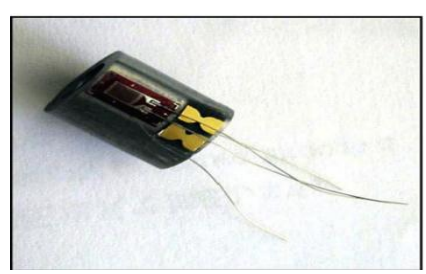
Fig.3
The dimensions of fatigue specimen in *mm*.

To prepare the specimens, fastener holes of diameter of 5 *mm*, were drilled and reamed in the jointed plates. Bolt used in this experiment had hexagonal head and a typical 5 *mm* shank diameter, which was then paired with hexagonal nut as illustrated in Fig. 3. Circular washer was used under both the hexagonal head and nut. The optimum length of the un-threaded part of the shank was selected in order to match the thickness of the plates to be joined, so that the contact between the plates and the bolts is along the un-threaded part of the shank. Finally, the nut is tightened by applying torque using a torque-wrench up to required amounts of torques. To insure that the used bolts and nuts are in the elastic region, a number of preliminary tests were conducted and the obtained results indicated that initial plastic strain started at approximately 8 *Nm* at threads. Therefore, the lower torque level for fatigue testing of the joints was set at 1 *Nm*, (corresponding to finger tightening), in order to create only a small amount of joint compression. The upper torque level for the applied tightening torques was selected to be 5 *Nm* to ensure that only elastic deformation of the thread occurred and also no considerable fretting phenomena happened on the connected plates. Then intermediate torque levels were set at 2.5 *Nm*.

In order to measure the bolt preload or pretension resulting from the torque tightening, under different applied torques, for both kinds of joints, i.e. simple bolted and hybrid joints, a bolt transducer which located between the plate and nut was used. This bolt transducer consists of a hollow cylinder with two strain gages attached on its outer surface. The suitable strain gauge boxes were used to read the strains values of attached strain gauges as a result of the induced axial strain from the torque tightening and therefore pretension or bolt preload of bolts was calculated by means of Hooke's elasticity law. The proposed approach for assessing the pretension in the bolt and the hollow cylinder dimensions were illustrated in Fig. 4.



(a)



(b)

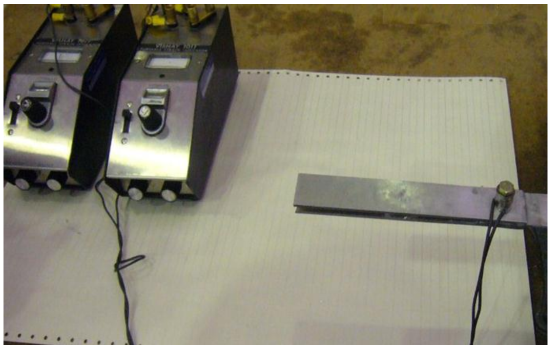

(c)

Fig. 4
Measuring bolt preload with the load cell. (a) Steel bush and its dimensions in mm, (b) The placement of the strain gauges on steel bush, (c) Position of prepared load cell in the joint.

To calibrate the applied torque and bolt preload, torques were applied in 1 *Nm* increments from 1 to 7 *Nm* 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 average amount of compressive strains (ε_m), and determine the corresponding bolt preload using Eq. (9) as follows:

$$F_{preload} = E_C A_C \varepsilon_m = 204188 \times \frac{\pi}{4}(9^2 - 5^2)\varepsilon_m = 89.8 \times 10^5 \varepsilon_m (N) \tag{9}$$

In the above equation A_C is the area of the hollow cylinder cross section. The elastic modulus for the hollow cylinder material (EC) was also experimentally determined in order to obtain the accurate values for the mean axial bolt preload. The relationship among the calculated bolt preloads using Eq. (9) and the applied tightening torques is shown in Fig. 5. As it can be seen from the figure, there is a linear relationship among the calculated bolt preload and the applied tightening torque. This indicates that the hollow cylinder material deforms elastically, for all levels of applied tightening torques.

Then, three sets of specimens were fabricated in which by means of a torque wrench the bolts and nuts were clamped up to required values of tightening torques, i.e. 1 *N-m*, 2.5 *N-m* and 5 *N-m* which created bolt preloads equal to $F_{preload} = 976, 2440$ and 4880 *N* respectively, according to the linear equation obtained from Fig. 5.

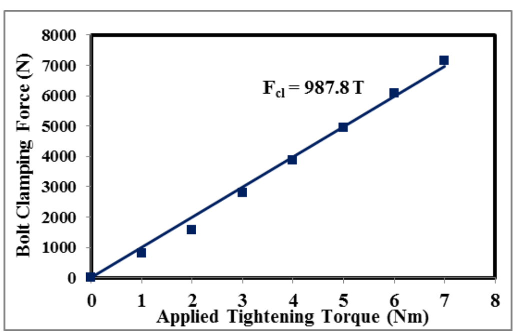


Fig.5
Experimental relation between the applied torques and the bolt preloads at the specimen.

Eighteen specimens were tested by means of a 250 *kN* servo-hydraulic Zwick/Roell test frame. The tests were performed in a laboratory atmosphere with constant amplitude loading mode (for different stress levels of 72, 96, 120, 144, 178 and 192 *MPa*) with a frequency of 10 *Hz*. The load ratio was chosen to be 0.1, in order to avoid probable buckling of the specimens as a result of any compressive loading effects on the test specimens. It should be noted that the fatigue life estimation based on volumetric approach needs plain or reference curve (S-N curve for plain specimens). Therefore, uniaxial tensile fatigue test was performed on plain specimens for the same loading mode (same frequency and load ratio). Configuration and dimensions of fatigue plain specimens were designed and manufactured to conform as nearly as possible to ASTM: E466-07.

The resulting average life from the constant amplitude fatigue tests was plotted in accordance with ASTM 468-90 and is shown in Fig. 6. It can be seen how the bolt preload increases the number of cycles to failure for all the applied stress levels such as anticipated.

© 2017 IAU, Arak Branch

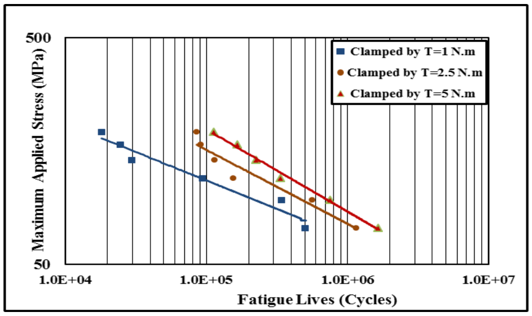


Fig.6
S–N curve attained from experimental fatigue tests.

4 NUMERICAL SIMULATIONS

To obtain the values of effective stress and distance, and also the fatigue notch factors in volumetric method, numerical simulation was used. In current study, to predict the fatigue lives of the bolted plates with various amounts of bolt preload, 3D models of the specimens was simulated by Ansys general multi-purpose finite element code[23]. All of the parts and bolt have been meshed with 20-node SOLID95 structural elements. Furthermore, taking advantages of specimens' symmetry, the analysis was performed only on a quarter-symmetry model. In order to obtain more accurate results, mapped mesh with adequately fine mesh in the vicinity of the hole of the plates has been used in FE analyses. The bottom face of the bolt shank was used to implement the bolt preload.

In this research, in order to apply the bolt preload, a displacement boundary condition (using a trial and error method) was used instead of applying force on the bolt shank. This is obligatory to have an accurate simulation and displacement boundary condition considers the bolt pre-tension relaxation during applying tensile load to the far end of the plate. As the tensile load is applied, a part of initial bolt preload in the bolt is reduced due to contraction of the holed plate thickness due to Poisson's ratio.

The meshed model of the specimens consisted of 17430 elements. In addition, the minimum mesh size in edge of the bolt hole was 0.04 mm. It is to be noted that all the simulations were repeated, when necessary, using different mesh and contact parameters, to ensure the convergence and the mesh independency of the results.

A detailed view of the meshed double lap specimen by symmetric 3D finite element model, applied load and boundary conditions are shown in Fig. 7. The nodes located at the left edge of the FE model were considered to have all their degrees of freedom constrained. Only one quarter of the specimen has been modelled, due to double symmetry (with respect to X-Z and X-Y Cartesian planes) and symmetric displacement boundary condition has been applied to the corresponding planes as shown in the figure.

In order to transfer the pressure between the contacting surfaces of the bolt head (or nut) and the plate, flexible-to-flexible contact state was used. Each contact pair consisted of target element and contact element. TARGET 170 was used as a target element and CONTACT 174 was used as a contact element. The friction effect between the surfaces of the washer (bolt head) and Al-alloy plates was included in the FE model using Elastic Coulomb model with friction coefficient of $\mu = 0.29$ which was obtained from experimental tests based on the sliding of the washer under its own weight on the sloped surface from Al-alloy plate. Furthermore, based on the similar experiments, the friction coefficient for the contact between the Al-alloy plates was considered to be $\mu = 0.4$.

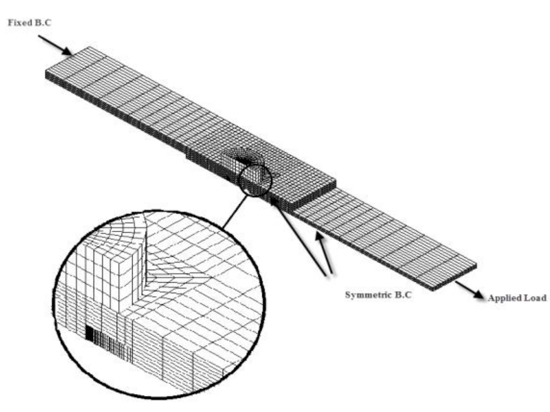


Fig.7
A meshed model of a bolted plate specimen with applied boundary conditions and loading.

© 2017 IAU, Arak Branch

In order to characterize the aluminium alloy 2024-T3 behaviour, a multi-linear kinematic hardening material option with Von Mises criterion within the Ansys software was used to represent stress-strain curve. The stress-strain curve of the material was obtained according to the ASTM 8M from simple static tensile tests and presented in Fig. 8.The elastic modulus and Poisson's ratio were obtained to be $E = 72GPa$ and $v = 0.33$ respectively. Also, for the bolt a linear elastic material relation was considered with elastic modulus of $207GPa$ and Poisson's ratio of 0.30 as it was found that the bolt experiences only elastic deformation while it was subjected to maximum applied tightening torque (8 *N-m*). Additionally, in order to reduce the time of solution process, the washer under the bolt head was included in the thickness of the bolt head model.

Numerical analyses were performed in the following two sequential steps i.e. the application of the bolt preload which was followed by a longitudinal load. In the first step of loading, axial displacement was applied to the bottom face of the bolt shank to simulate the bolt preload. This process was completed for three initial different values of bolt preload ($F_{preload} = 976, 2440, 4880N$) using a trial and error method. After the bolt are preloaded to the same levels as the bolts at the experiments tests, the value of maximum force in each cyclic loading (in accordance with the fatigue test) was applied to the end of the middle plate in the model as a longitudinal remote load.

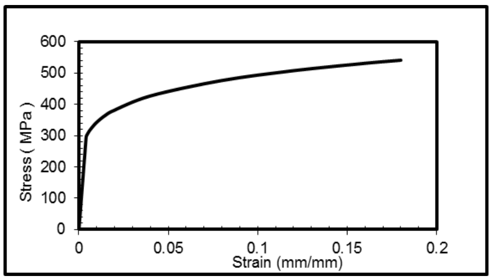


Fig.8
True stress–strain diagram for Al-alloy 2024-T3.

5 RESULTS AND DISCUSSIONS

In the present research, the influence of preload of the bolt on the fatigue life of bolted joints, subjected to longitudinal cyclic loading, has been studied by means of volumetric method. As it has been discussed, the fatigue life estimation using the volumetric method requires stress distribution at the edge of the bolt hole which can be obtained by nonlinear elastic-plastic finite element analysis and plain or reference fatigue curve of the material.

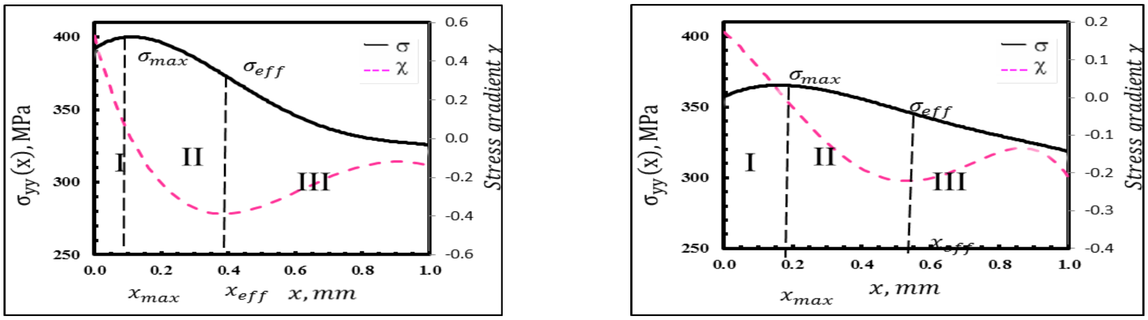
The process of finite element method solution has been illustrated in previous section. After the applying the bolt preload which was followed by a longitudinal remote load, the solution of the numerical analysis has been implemented. Then the crack opening stress and the relative stress gradient distributions versus distance from the hole edge which are important to calculate the fatigue notch factor, can be obtained. Application of the volumetric method requires the elastic-plastic crack opening stress, and its integration direction. In this research, the cracks propagate at root of the plate hole and the crack opening stress is perpendicular to crack face and parallel to longitudinal applied load at the free end of the plate. As mentioned earlier, to calculate the fatigue reduction factor (by Eq.(6)), the stress distribution around the notch (bolt hole) which is required for volumetric approach, was obtained using the nonlinear finite element simulations. The steps consist in obtaining the stress distribution as a function and subsequently establishing the equation of the opening stress. To do so, a curve fitting is performed to assess a polynomial representation by the mathematical software such as MATLAB. In next step, the value of fatigue reduction factor has been calculated by integration of Eq.(6) over the critical distance using the mathematical software such as MATLAB and Maple.

Figs. 9 and 10, show the crack opening stress (bulk stress) and relative stress gradient distributions against the distance from the hole edge toward the plate edge for specimens with two different values of bolt preload.

As it is clear in the figures, increasing the preload of the bolts, decreases the magnitudes of maximum and effective stresses, and also decreases the effective distance. Consequently the increase in the bolt preload, decreases the amounts of the notch reduction factors, and in result improves the fatigue lives of the joints.

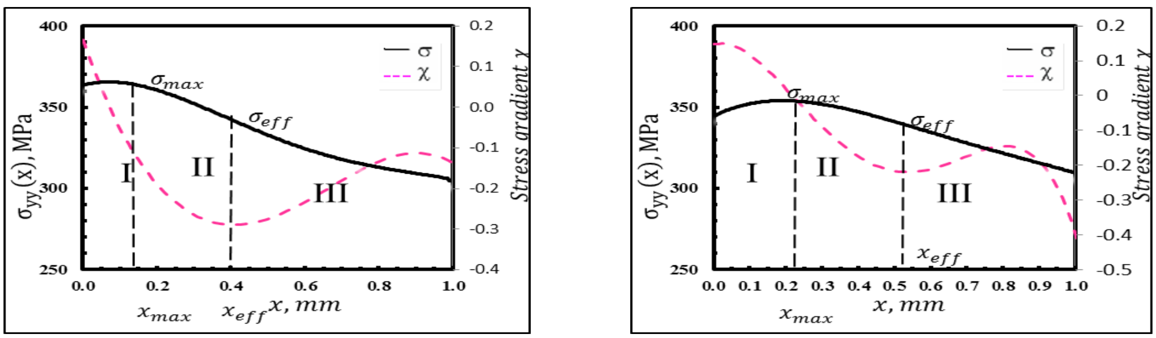
This research revealed that, the fatigue strength enhancement of double lap bolted joints is related to the compressive pre-stress near the bolt hole caused by the compression of the plates by means of bolt preload. As the tightening torque of the bolt increases, the bolt is stretched more and the compression in plates increases. Therefore, this compressive pre-stress leads to delay the fatigue crack initiation and propagation at the edge of the bolt hole and consequently improves the fatigue life of the joints.

© 2017 IAU, Arak Branch



(a) (b)

Fig.9
Crack opening stress and relative stress gradient distributions for the specimen with preload of $F_{preload}=976N$ at different levels of applied load; a) $S_{\max}=192MPa$ b) $S_{\max}=144MPa$.



(a) (b)

Fig.10
Crack opening stress and relative stress gradient distributions for the specimen with preload of $F_{preload}=4880N$ at different levels of applied load; a) $S_{\max}=192MPa$ b) $S_{\max}=144MPa$.

The improvement in fatigue life in the bolted specimens can also be attributed to the method that the joint transmit the applied load. As the tightening torque is increased, a large part of the load is transmitted by friction (at the plate faces). The remaining part of the load is transmitted by the reaction of the bolt against the bolt hole surface. The transmitted load between the connection plates (as a result of friction) versus the applied longitudinal load for three different applied tightening torques has been shown in Fig. 11.

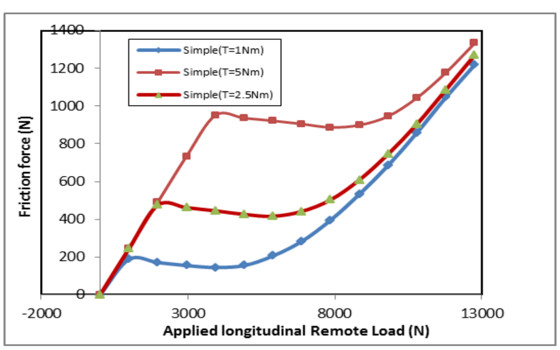


Fig.11
The load transmitted by friction between the plates versus the applied longitudinal load for three different applied tightening torques.

As it can be seen from the figure, the transmitted load by means of friction initially increases as the longitudinal load is increased. This is because of an initial gap between the bolt shank and the bolt hole. At the beginning of the loading, the total of applied longitudinal load is transmitted only by means of friction between the plates. However,

© 2017 IAU, Arak Branch

the friction force stops to increase when the bolt touches the hole after which point the applied load is divided between the bolt and the plates. The bolt preload in joints creates friction on the contact surface. This friction changes the load transfer mechanism in joint. Therefore, friction force reduces a portion of load transferred through the bolt as shown in Fig. 11. As a result, the local stress at the edge of the hole is lower, since part of the total load is absorbed by friction depending on the bolt preload and the friction coefficient of the contact surface.

In this section, to investigate the validity and accuracy of the volumetric approach, the fatigue lives predicted with this method have been compared with those which obtained from experimental analysis. The fatigue strengths of specimens with different amounts of bolt preload against the number of cycles to fracture using the volumetric method and experimental test results were illustrated in Fig. 12. According to the Fig. 12, the volumetric method results were found to agree very well with the obtained results from experimental analysis and presents very good and reasonable results.

It must be mentioned that, among the technical literature and surveys which are available in the field of the fatigue life estimating, a few researches can be found regarding to the application of the volumetric approach, in order to obtain the fatigue life of the different types of specimens [15-17]. According to the reference [16], the volumetric approach has been successfully employed for fatigue life assessment of the multi resistance spot welded joints. As mentioned previously, although the essential parameters of the volumetric approach are finite element analysis and the values of fatigue strength reduction factor and fatigue life of plain specimens, the important prerequisite of this procedure is to recognize the direction of the fatigue crack path of the notched specimens.

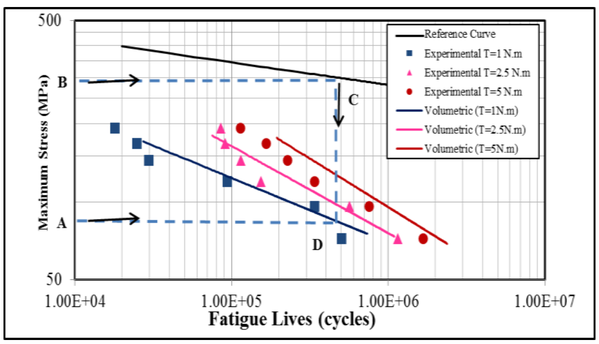


Fig.12
Fatigue strength versus the number of cycles to failure for different specimens based on the volumetric approach and comparison with experimental results.

As discussed in manuscript, the fatigue life estimation using the volumetric method requires stress distribution at the edge of the bolt hole which can be obtained by nonlinear elastic-plastic finite element analyses and plain or the reference fatigue curve of the material. The first step to predict the fatigue notch effect is to obtain a fatigue reference curve for the same loading mode. Therefore, uniaxial tensile fatigue test was performed on plain specimens for the same loading mode (same frequency and load ratio). The reference fatigue curve is presented in Fig. 12. In the next step, nonlinear finite element analyses were performed in order to obtain the stress distribution near the notch (bolt hole) root. Based on the obtained stress distribution, the fatigue life reduction factor was calculated by Eq.(6) (as mentioned in manuscript). Finally, the fatigue strength of plain specimen (point A in Fig. 12), σ_{smooth} ,was calculated by multiplying the applied remote stress or fatigue strength of notched specimen (point B) with obtained fatigue strength reduction factor from the volumetric approach. The fatigue strength of plain specimen obtained by the volumetric approach reported on the fatigue reference curve, gives directly the fatigue life of the notched specimen (Fig. 12). In next step, the fatigue strength of plain specimen was projected on the reference curve (point C). The intersection between the projection of (point C) on the fatigue life axis and the value of the applied remote stress, $\sigma_{notched}$ gives a (point D). By repeating this operation for several loads, the entire curve, can be determined.

The volumetric fatigue life prediction is not similar to the other recognized methods such as Peterson and Neuber [21,22]. The volumetric approach uses the fatigue curve of plain specimen, and the stress distribution near notch tip which obtained by means of the finite element analysis. Using the finite element analysis, loading mode, relative stress gradient and geometrical effects are considered and it is clear that this method provides better results. The predicted fatigue life using the volumetric approach, have very good agreement with the experimental results such as different notched specimens, spot welded joints and simple bolted plates [15-17].The results of this work and other researches [15-17] indicates that this approach gives a relative good description of notch effect.

© 2017 IAU, Arak Branch

6 CONCLUSIONS

In this research, the influence of the bolt preload on the fatigue life of 2024-T3 aluminium alloy double lap bolted specimens has been investigated using experimental and numerical analysis. In the numerical method, to estimate the fatigue life, the S-N curve of plain (un-notched) specimen and the fatigue reduction factor obtained from volumetric method were used. The estimated fatigue life was compared with the results which obtained from experimental analysis. The investigation shows that there is a good and reasonable agreement among the life predicted by the volumetric approach and the experimental results for various specimens with a different amounts of the bolt preload. Volumetric approach and experimental results revealed that the fatigue life of double lap bolted joints were improved by increasing the bolt preload due to compressive stresses which appeared around the hole.

REFERENCES

[1] Huda Z., Zaharinie T., Min G., 2010, Temperature effects on material behavior of aerospace aluminum alloys for subsonic and supersonic aircraft, *Journal of Aerospace Engineering* **23**(2): 124-128.

[2] Mahadevan S., Shi P., 2001, Corrosion fatigue reliability of aging aircraft structures, *Progress in Structural Engineering and Materials* **3**(2): 188-197.

[3] Buyuk M., Kan S., Loikkanen M., 2009, Explicit finite-element analysis of 2024-t3/t351 aluminum material under impact loading for airplane engine containment and fragment shielding, *Journal of Aerospace Engineering* **22**(3): 287-295.

[4] Mao J., Kang S.B., Park J.O., 2005, Grain refinement, thermal stability and tensile properties of 2024 aluminum alloy after equal-channel angular pressing, *Journal of Materials Processing Technology* **159**(3): 314-320.

[5] Pratt J., Pardoen G., 2002, Numerical modeling of bolted lap joint behavior, *Journal of Aerospace Engineering* **15**(1): 20-31.

[6] Ireman T., 1998, Three-dimensional stress analysis of bolted single-lap composite joints, *Composite Structures* **43**(3): 195-216.

[7] Pratt J., Pardoen G., 2002, Comparative behavior of single-bolted and dual-bolted lap joints, *Journal of Aerospace Engineering* **15**(2): 55-63.

[8] Esmaeili F., Chakherlou T.N., Zehsaz M., Hasanifard S., 2013, Investigating the effect of clamping force on the fatigue life of bolted plates using volumetric approach, *Journal of Mechanical Science and Technology* **27**(12): 3657-3664.

[9] Valtinat G., Hadrych I., Huhn H., 2000, Strengthening of riveted and bolted steel constructions under fatigue loading by preloaded fasteners experimental and theoretical investigations, *Connections in Steel Structures IV*.

[10] Chakherlou T.N., Abazadeh B., Vogwell J., 2009, 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, *Engineering Failure Analysis* **16**(1): 242-253.

[11] Chakherlou T.N., Alvandi-Tabrizi Y., Kiani A., 2011, On the fatigue behavior of cold expanded fastener holes subjected to bolt tightening, *International Journal of Fatigue* **33**(6): 800-810.

[12] Budynas R.G., Nisbett J.K., 2011, *Shigley's Mechanical Engineering Design,* 9rd ed, McGraw-Hill.

[13] Dowling N., 2012, *Mechanical Behavior of Materials: Engineering Methods for Deformation, Fracture and Fatigue*, Prentice Hall Inc, New Jersey.

[14] Esmaeili F., Hassanifard S., Zehsaz M., 2011, Fatigue life prediction of notched specimens using the volumetric approach, *Journal of Solid Mechanics and Materials Engineering* **5**(9): 508-518.

[15] Hassanifard S., Zehsaz M., Esmaeili F., 2011, Spot weld arrangement effects on the fatigue behavior of multi-spot welded joints, *Journal of Mechanical Science and Technology* **25**(3): 647-653.

[16] Adib H., Pluvinage G., 2003, Theoretical and numerical aspects of the volumetric approach for fatigue life prediction in notched components, *International Journal of Fatigue* **25**(1): 67-76.

[17] Tryon R. G., Dey A., 2003, Reliability-based model for fatigue notch effect, *SAE International* **1**: 01-0462.

[18] Pluvinage G., 2004, *Fracture and Fatigue Emanating From Stress Concentrators*, Kluwer Academic Publishers, Dordrecht.

[19] Peterson R. E., 1974, *Stress Concentration Factors*, John Wiley & Sons, New York.

[20] Neuber H., 1961, *Theory of Notch Stresses: Principles for Exact Calculation of Strength with Reference to Structural Form and Material*, USAEC Office of Technical Information, Oak Ridge, Tenn.

[21] Heywood R.B., 1962, *Design Against Fatigue*, London, Chapman and Hall.

[22] Pluvinage G., 1997, Application of notch fracture mechanics to fracture emanating from stress concentrators. *Advances in Computational Engineering Congress of Computational Engineering Sciences* **97**: 213-218.

[23] Swanson Analysis Systems Inc, 2004, ANSYS, Release 9.

© 2017 IAU, Arak Branch