Investigation the effect of tightening torque on the fatigue strength of double lap simple bolted and hybrid (bolted–bonded) joints using volumetric method

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Article history:
Received 19 March 2014
Accepted 10 June 2014
Available online 27 June 2014

Keywords:
Tightening torque
Hybrid joint
Fatigue life
Finite element
Volumetric method

Abstract

In this research, the effect of the tightening torque on the fatigue strength of 2024-T3 double lap simple bolted and hybrid (bolted–bonded) joints have been investigated experimentally by conducting fatigue tests and numerically by implementing finite element analysis. To do so, three sets of specimens were prepared and each of them subjected to tightening torque of 1, 2.5 and 5 Nm and then fatigue tests were carried out under different cyclic longitudinal load levels. In the numerical method, the effect of the tightening torque on the fatigue strength of the considered joints has been studied by means of volumetric method. To obtain stress distribution around the notch (bolt hole) which is required for the volumetric method, nonlinear finite element simulations were carried out. In order to estimate the fatigue life, the available smooth S–N curve of Al2024-T3 and the fatigue notch factors obtained from the volumetric method were used. The estimated fatigue life was compared with the available experimental test results. The investigation shows that there is a good agreement between the life predicted by the volumetric method and the experimental results for different specimens with a various amount of tightening torques. The results obtained from the experimental analysis showed that the hybrid joints have a better fatigue strength compared to the simple bolted joints. In addition, the volumetric method and experimental results revealed that the fatigue life of both kinds of the joints were improved by increasing the clamping force resulting from the torque tightening due to compressive stresses which appeared around the bolt hole.

1. Introduction

Among the mechanical removable joints, including, riveted, pinned or bolted joints, the bolted joint are the most essential components in aerospace structures. However, the existence of geometrical discontinuity in these joints due to essential hole drilling process causes stress concentration and thus increases the tendency of early fatigue crack initiation and growth under cyclic loading [1–3]. Therefore, it is of great importance to reduce the effect of the stress concentration and attain enhanced fatigue life. According to the results of previous studies, bolted joints have higher tensile and fatigue strengths than welded, riveted and pinned joints [4–6].

An alternative method to mechanical fastening is adhesively bonded joint. In order to identify the advantages of adhesive bonding in fatigue, two fundamental differences are important between the two kinds of lap joints. First, in a mechanical joint, the overlapping areas are attached to one another at discrete points only, i.e. by the fasteners. Clearly, severe stress concentrations occur. However, if the connection is made continuously in the full overlapping area by adhesive bonding, these stress concentrations do not occur. Because, the adhesive bonded joints do not need to the fasteners and the fastening holes. Therefore, the stress distributions in the joint are relatively uniform compared to those in the mechanical joint. Secondly, metallic direct contact between the two sheets does not happen in the adhesively bonded joints.

In order to overcome the potential weaknesses of the adhesive bonding, and the mechanical joints, and therefore, to obtain very effective joints, a combination of the mechanical joints (riveted, bolted, etc.) and an adhesive, namely hybrid joints, are used [7–10]. Hybrid joints are used in many engineering application such as aerospace, automotive, and naval industries because of their better performance in comparison with the simple joints such as adhesively bonded, riveted and bolted joints [11,12].

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http://dx.doi.org/10.1016/j.matdes.2014.06.021
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Several researchers [13–17] investigated the hybrid joints. Pirondi and Moroni [18] compared the hybrid weld-bonded, rivet-bonded, clinch-bonded, and simple joints under various conditions, by means of experimental analysis. The effect of the material, geometrical factors, and environment on static strength and energy absorption were evaluated through the analysis of variance. A similar investigation conducted by Schvechkov [19] on the effects of adhesive mechanical properties on the fatigue strength of hybrid (riveted-bonded) joints by means of experimental analysis. In separate investigations, Kelly [14,20] studied the static and fatigue strength of the hybrid (bonded–bolted) single lap joints using different modulus adhesives. The results of studies, revealed that, the hybrid joints with lower modulus adhesives allowed for load sharing between the adhesive and the bolts. In a similar investigation conducted by Fu and Mallick [8], they experimentally showed that the hybrid (bolted–bonded) single lap joints have a higher static and fatigue strength in comparison with only adhesively bonded joints.

When a nut and bolt are used to join mechanical members together, the nut or bolt can be pre-tensioned by applying tightening torque using a torque wrench and then the bolt and nut are pulled toward each other. Pre-tension or clamping force is the force which holds the assembled part together [21–25].

Previous works revealed that the clamping force can reduce the stress concentration at the bolted hole region, and therefore improve the strength of the joint considerably [6,21].

As mentioned, fatigue fractures usually occur at notches such as holes and grooves. Since virtually all engineering structures include notches in one form or another, the treatment of this in fatigue life prediction has received considerable attention over the years. The geometry of notch and other notch properties affect the fatigue life. The stress rise will of course be very harmful with respect to fatigue strength, but in fatigue tests of the notched geometries, the effect of $k_t$ appears to have less influence than under static loading, which has led to the definition of a separate fatigue notch factor, $k_f$ [26–28].

The fatigue notch factor relates the unnotched fatigue to the notched, nominal fatigue strength in the same numbers of cycles and the same experimental test conditions. There is not a general experimental method with low cost to determine the $k_f$ values, and this is because of the micro-mechanical complications and uncertainties of the plastic zone at the notch root. In general trend, $k_f$ is equal to or less than $k_t$. A general definition of the fatigue notch factor is the ratio of fatigue strength of the smooth specimen to fatigue strength of the notched specimen in the same numbers of cycles and the same experimental test conditions [29]:

$$k_f = \frac{\text{unnotched fatigue limit}}{\text{notched fatigue limit}}$$

The difference between the fatigue notch factor $k_f$ and the elastic stress concentration factor $k_e$ is expressed by the notch sensitivity factor, $q$, as follows:

$$q = \frac{k_f - 1}{k_e - 1}$$

There are several relations to express the notch sensitivity factor in terms of the notch radius. However, the experimental accuracy of these relations is not very clear, and it is essential to conduct more experimental tests. Notch sensitivity factor for describing the notch effects of the high-cycle fatigue problems have some disadvantages in definition of the relations between $k_f$ and $k_e$, as follows: The empirical relation between the $k_f$ and $k_e$ are established only for the endurance limit. There are some experiment dependent coefficients, which have been obtained for materials used in many years ago. Nowadays, since varieties of materials in industry are ever increasing and new materials and alloys have been produced, using these coefficients is not very appropriate.

Differences between these relations are high and the accuracy of these relations is not clear and the effects of the net stress and stress gradient in these relations have been neglected. Two commonly used equations, among the relations that have been proposed to describe the notch sensitivity factor, are Peterson [30] and Neuber [31] equations.

Peterson assumed that the fatigue failure occurs when the stress at one point which has a critical distance from the notch root ($a_p$) is equal to the fatigue strength of the smooth specimen. Assuming that the stress near the notch root is decreasing linearly, Peterson proposed the following experimental relation:

$$q = \frac{1}{1 + \frac{r}{r_{crit}}}$$

where $r$ is notch radius and $a_p$ is material constant, which depends on the grain size and loading. A widely used empirical expression for $k_f$ for different notch geometries was proposed by Neuber [31]. By considering the stress distribution ahead of a notch with a given notch root radius, he arrived at an expression for $q$ based on the average stress up to some distance ahead of the notch as follows:

$$q = \frac{1}{1 + \sqrt{\frac{a_p}{r}}}$$

where $a_p$ is the material parameter with a unit of length and $r$ is the notch root radius constant which depends on the grain size. In addition, on the basis of a model taking into account how inherent material flaws like cavities, inclusion, defects alter the stress field at the notch root, Heywood [32] suggested a different empirical expression of the notch-fatigue. Heywood’s representation of extensive fatigue is somewhat different from Neuber’s and Peterson’s. Heywood’s proposed expression for the fatigue notch factor is illustrated as follows:

$$k_f = \frac{k_e}{1 + 2 \frac{r}{a’}}$$

where $a’$ corresponds to the length of equivalent material flaws. As mentioned previous, the accuracy of these formulas is not specified and requires to be checked by numerous, costly and time consuming fatigue tests.

Based on a large amount of empirical data, Siebel and Stieler [33] expressed $k_f$ by the relative stress gradient:

$$k_f = \frac{k_e}{1 + \sqrt{\chi}}$$

where, for loading in the $z$ direction using cylindrical coordinates ($z, \theta, r$), the relative stress gradient is defined from the axial stress $\sigma_z$ as:

$$\chi = \frac{1}{\sigma_{max}} \left( \frac{\partial \sigma_z}{\partial r} \right)_{r=0}$$

In this study, the effect of tightening torque on the fatigue strength of 2024-T3 double lap simple bolted and hybrid (bolted/bonded) joints have been investigated both experimentally and numerically. Therefore, two kinds of joints, i.e. double lap simple and hybrid (bolted–bonded) joints were considered. For each kind of the joint, three sets of specimens were prepared and subjected to the tightening torque of 1, 2.5 and 5 Nm and then fatigue tests were carried out on them under different cyclic longitudinal load.
levels. In the numerical section, the influences of tightening torque on the fatigue life of simple and hybrid joints have been studied using the volumetric method. To obtain the stress distribution around the notch (bolt hole) which is required for the volumetric method, nonlinear finite element simulations were carried out.

2. Theoretical aspects of the volumetric method

From the physical point of view, the volumetric method is an innovative technique, in order to estimate the fatigue strength of notched components. In this method, the fatigue failure needs a physical volume to take place, which is called the fatigue processing volume.

The fatigue strength of materials is mainly affected by the specimen size and the relative stress gradient, which are dimensional parameters. This method leads to a two-parameter fatigue initiation criterion, i.e. the effective distance and the effective stress. The effective distance is the diameter of the fatigue processing volume. The shape and size of the volume generally depends on the specimen geometry, loading mode and material properties. This volume is supposed to be the site where the fatigue damage takes place. The effective stress corresponds to an average value over this distance of the stress distribution weighted by the distance and the relative stress gradient [27,29]. It is determined by a nonlinear stress distribution near the notch tip. Therefore, it is a function of the stress field and, depends, as the effective distance, on the part geometry, loading and material. So, for a given geometry, material and loading, the effective stress is, in principal, a function of net stress. A typical elastic–plastic stress distribution near a notch has been shown in Fig. 1.

Three different regions can be recognized from Fig. 1. The maximum stress and its distance can be clarified in zone I. The important characteristic of this zone is the existing of the maximum stress in the elastic–plastic stress distribution, which has been obtained from the finite element analysis. The elastic FE analysis in the notch root for the crack opening stress reveals that stress value is decreasing from the notch root toward the outside [29]. In zone II, the stress value is decreasing and its value reaches to the effective stress $\sigma_{eff}$. According to the volumetric method, the fatigue notch factor is given by the following formula:

$$k_f = \frac{1}{X_{eff}} \int_{0}^{X_{eff}} \frac{\sigma_{eff}(x)}{\sigma_n(x)} |1 - x\chi| dx$$

(8)

where $X_{eff}$ is the effective distance, $\sigma_n$ is the net stress, $\sigma_{eff}$ is the crack opening stress, and $\chi$ is the relative stress gradient which is defined as follows:

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

(9)

In this method, the fatigue notch factor calculated from Eq. (8) and the use of reference curve of smooth specimen lead to notched curve in fatigue according to Eq. (1). In this method fatigue notch factor is calculated by application of the specimen material properties and its geometrical features via finite element analysis.

The volumetric method can also be used in the cases that smooth specimen results are not available but another notched specimen with the same material, specimen geometry and different notch features exist.

3. Materials and experimental methods

The specimens employed in this investigation were made from 2024-T3 aluminium alloy with thickness of 2 mm. Table 1 lists the mechanical properties of the aluminium alloy obtained from tension (static) tests, while Table 2 presents the chemical compositions of the used aluminium alloy.

Two different kinds of joints i.e. double lap simple and hybrid (bolted–bonded) joints were prepared. Test specimens’ configurations and dimensions for both kinds of the joints are illustrated schematically in Fig. 2.

The hybrid joints were manufactured by means of the structural two component epoxy adhesive (Loctite 3421) [34], prepared by mechanical mixing of the resin and hardener in equal amount by weight. In order to obtain the tensile stress–strain curve of the adhesive, several dog-bone specimens were prepared according to ASTM: D638-10. The adhesives were injected into a mold, as shown in Fig. 3, and left to be cured at the room temperature for 24 h. Finally, the prepared specimens were tested on a 100 kN Zwick/2100 static testing machine with a crosshead speed of 5 mm/min. The engineering stress–strain curve of the adhesive is shown in Fig. 4.

In order to eliminate any possible surface scratches, the surface of the plates (specimens) were polished using different grinding (sand) papers with grits of 400, 600 and 1000 at first. To prepare the specimens, fastener holes with diameter of 5 mm, were drilled and reamed in the joint plates. A hex head M5 (class 10.9) steel bolt was used for the mechanical fastening and suitable types of steel washers and nuts were used to prepare the joint as illustrated in Fig. 2. Finally, the nut is tightened by applying torque using a torque-wrench up to required amounts of torques, i.e. 1 Nm, 2.5 Nm and 5 Nm.

![Fig. 1. A typical elastic–plastic stress distribution near a notch and relative stress gradient [31].](image)

Table 1

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
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<tr>
<td>Young’s modulus (GPa)</td>
<td>72</td>
</tr>
<tr>
<td>Yield stress (MPa)</td>
<td>315</td>
</tr>
<tr>
<td>Tensile strength (MPa)</td>
<td>550</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>0.33</td>
</tr>
<tr>
<td>Elongation (%)</td>
<td>18</td>
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</tbody>
</table>

Table 2

<table>
<thead>
<tr>
<th>Element</th>
<th>Cu</th>
<th>Mg</th>
<th>Mn</th>
<th>Fe</th>
<th>Si</th>
<th>Cr</th>
<th>Zn</th>
<th>Ti</th>
<th>Al</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight (%)</td>
<td>4.82</td>
<td>1.67</td>
<td>0.58</td>
<td>0.18</td>
<td>0.07</td>
<td>0.02</td>
<td>0.06</td>
<td>0.15</td>
<td>100</td>
</tr>
</tbody>
</table>
As mentioned, aluminium alloy 2024-T3 sheet was used as an adherend to prepare the hybrid joints in this investigation. The preparation of the hybrid joints has been implemented in two main steps. Firstly, a double lap bonded joint was constructed. In order to obtain high strength joint, the joint plates were cleaned with acetone and then they were let dry, prior to using the adhesive layer. In order to achieve the constant thickness of adhesive layer, 0.5 mm thick sheets were used between adherends. The prepared bonded joints were left in ambient temperature for 72 h, in accordance with the adhesive manufacturer suggestion.

In the second step of preparing the hybrid joints, the double lap bonded specimens with the adhesive were bolt tightened using the same amounts of torques as the simple bolted joints, i.e. 1 Nm, 2.5 Nm and 5 Nm.

3.1. Clamping force measurement

In order to measure the clamping force or bolt pre-tension resulting from the torque tightening, at different applied torques,
for both kinds of the joints, 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 therefore the clamp force has been determined. The used
method and the bush dimensions were shown in Fig. 5.

To calibrate the applied torque and clamping force, 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 mean value of the compressive strains (\(\varepsilon_m\)),
and determine the corresponding clamping forces using Eq. (10).
The relation between the applied tightening torque and the mean
value of compressive strains for both types of joints, are given in
Fig. 6. 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.

\[
F_{cl} = E_{bush}A_{bush}\varepsilon_m = 204,188 \times \frac{\pi}{4} (9^2 - 5^2)\varepsilon_m \\
= 89.8 \times 10^5 \varepsilon_m (N)
\]  

(10)

In the above equation, \(A_{bush}\) is the area of the bush cross section. The
relation between the calculated clamping forces and the applied
torques for the specimen is shown in Fig. 7. As the figure shows,
there is a linear relation between the clamping force and the
applied torque. This confirms that the bush material is still in its
elastic region, even under the maximum applied torque.

3.2. Fatigue tests

In the next step, the prepared specimens for each kind of the
joints were tested in this investigation to study the effect of the
tightening torque on the fatigue lives of the both kinds of the joining
method.

Fatigue tests have been carried out using constant amplitude
loads in a servo-hydraulic 250 kN Zwick/Roell fatigue testing
machine with a frequency of 10 Hz and stress ratio (load ratio) of
0.1. The fatigue tests have been performed on the double lap sim-
ple bolted specimens with three different amounts of tightening
torques, i.e. 1 Nm, 2.5 Nm and 5 Nm which created clamping forces
equal to \(F_{cl} = 976, 2440 \text{ and } 4880 \text{ N respectively, according to the}
linear equation obtained from Fig. 7. In each case, six fatigue tests
were performed with different maximum remote longitudinal
loads to final failure. The fatigue tests were run until the failure
(fracture) occurred in the main plate. The fatigue test results for
double lap simple bolted specimens have been shown in Fig. 8 in
terms of the number of cycles to failure. In addition, in the case of
double lap hybrid specimens, fatigue tests have been performed
with the same amounts of the tightening torques, as simple bolted
specimens, i.e. 1 Nm, 2.5 Nm and 5 Nm which created clamping
forces equal to \(F_{cl} = 840, 2100 \text{ and } 4200 \text{ N respectively, according to the}
linear equation obtained from Fig. 7. The fatigue test results
for the double lap hybrid specimens have been shown in Fig. 9 in
terms of the number of cycles to failure. It is important to mention
that, failure in all of the specimens occurred in the main plate near the bolt hole edge.

4. Numerical analysis

In order to obtain the effective distance, the effective stress values and the fatigue notch factors in the volumetric method, the finite element analysis has been employed by ANSYS 9.0 general finite element code [35]. In the present investigation, to estimate the fatigue lives of the joint plates for both kinds of the joints with different amounts of the tightening torque, three dimensional models of the specimens have been simulated using Ansys FE code. All of the adherends, adhesive layer and bolt have been meshed with 20-node SOLID95 structural elements. In addition, due to double symmetry condition, only a quarter of the plates, and bolt were needed to be modelled. Symmetric displacement boundary conditions were defined for the nodes on the planes of symmetry. To extract more accurate results, mapped mesh with sufficiently fine mesh around the hole of the plates has been used in FE analyses. The bottom face of the bolt shank was used to implement the bolt clamping force. The finite element mesh of the double lap hybrid joint specimens is presented in Fig. 10, together with its corresponding loading and boundary conditions. The nodes located at the left end of the FE model were considered to have all their degrees of freedom constrained.

In order to transfer the pressure between the contacting surfaces, 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 plate 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 sliding of the washer under its own weight on the sloped surface from Al-alloy plate. Also based on the similar experiments, the friction coefficient was found to be $\mu = 0.4$ for the contact between the plates.

In order to characterize the aluminium alloy 2024-T3 stress–strain behaviour, an elastic–plastic multi-linear kinematic hardening material model with Von Mises criterion was used. This behaviour of the material was obtained from the simple tensile tests and shown in Fig. 11. The elastic modulus and Poisson’s ratio were measured to be $E = 72$ GPa and $\nu = 0.33$ respectively. Also, for adhesive layer the multi-linear isotropic material model was used and the Poisson’s ratio was considered equal to 0.35. Furthermore, for the steel bolt a linear elastic material relation was assumed with Young’s modulus of 207 GPa and Poisson’s ratio of 0.30 as it was observed that the bolt material remained in elastic when it was subjected to maximum applied torque (8 Nm). As the bolt and its washer have approximately the same material properties,

![Fig. 12. Distribution of resultant compressive stress $\sigma_z$ in MPa due to tightening torque of (a) 1 Nm, (b) 2.5 Nm and (c) 5 Nm for simple and hybrid joints.](image-url)
the geometrical model of the washer was added to the bolt head in order to minimize the contact element use (with ignoring contact elements between the bolt head and the washer).

Numerical analyses were carried out in two main steps including the application of the clamping force which was followed by a longitudinal load to the aluminium alloy. In the first step of loading, axial displacement was applied to the bottom face of the bolt shank to simulate the clamping force. In order to apply clamping force in the bolt, a displacement boundary condition (using a trial and error method) was used instead of applying force on the bolt shank. This is necessary to have an accurate simulation and consider the bolt pre-tension relaxation during applying tensile load to the far end of the plate. As the tensile load is applied to the far end of the plate, a part of initial clamping force in the bolt is reduced due to contraction of the holed plate thickness as a result of Poisson’s ratio. This process was completed for three initial clamping forces resulting from the different amounts of tightening torques for both kinds of the joints using a trial and error method. Then the solution was restarted for each state and in the second step, the value of the maximum force in each cyclic loading (as in the experimental tests) was applied to the end of the plate in the model as a longitudinal static load.

5. Results and discussion

In this section, to investigate the validity and accuracy of the tightening torque on the fatigue strength of 2024-T3 double lap simple bolted and hybrid (bolted–bonded) joints subjected to the longitudinal cyclic loading has been studied by means of the volumetric method. As it has been discussed, the fatigue life estimation based on the volumetric method needs the stress distribution near the notch roots which can be obtained by the nonlinear elastic–plastic finite element analysis and the smooth or reference fatigue curve of the material.

As it was mentioned previously, three different tightening torque values were selected to be applied. To do so, the corresponding clamping force, i.e. \( F_{c1} = 976 \), 2440 and 4880 N for simple bolted joints, and \( F_{c2} = 840 \), 2100 and 4200 N for hybrid joints were to be applied on the plates. Therefore, a displacement boundary condition in the \( Z \) direction was applied on the lower face of the bolt shank to achieve the desired clamping forces equal to experimental test results. The magnitude of the required displacement was found after a few trial and error processes to achieve the desired reaction forces or the clamping forces resulting from tightening torques.

According to the results of the first load step solution of the finite element analysis, some beneficial compressive stresses were observed near the hole of the plates. The compressive stress contours around the bolt hole of the main plate, created due to 1, 2.5 and 5 Nm tightening torques are shown in Fig. 12. As it can be seen, the most compressive stresses are observed near the edge of the hole for simple bolted specimens, which increased from -20 to -101 MPa when the tightening torque increased from 1 to 5 Nm. In addition for the hybrid specimens the compressive stresses increased from -13 to -49 MPa when the tightening torque increased from 1 to 5 Nm.

In the second load step, in accordance with the experimental test process, a tensile remote stress was applied to the FE models to simulate the tensile loading of the specimens. Therefore, a remote tensile stress equal to the applied cyclic load range was applied on the right end of the main plate while the displacement of the left end of the connector plates was constrained. After the application of clamping force which was followed by a longitudinal 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. To realize and recognize the volumetric method concept, the elastic–plastic crack opening stress, and its integration direction should be noticed. In this study, 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. The typical fractured specimens for simple and hybrid joints under fatigue tests with tightening torque of \( T = 1 \) Nm that subjected to maximum remote longitudinal load equal to 7.2 kN is shown in Fig. 13. Figs. 14 and 15 show the typical crack opening stress and the relative stress gradient distributions versus distance from the hole edge toward the plate edge with different amounts of the tightening torque under the applied maximum remote longitudinal load equal to 192 MPa for the simple and hybrid specimens, respectively. As shown in Fig. 14, in the simple bolted specimens the maximum stress values occurred at a distance away from the edge of the hole and their values decreased from 402 to 378 MPa when the tightening torque increased from 1 to 5 Nm. According to Fig. 15 for hybrid specimens the maximum stress values occurred at the edge of the hole and their values decreased from 278 to 252 MPa when the tightening torque increased from 1 to 5 Nm. This is due to elastic stress distribution in main plate for the hybrid specimens. In addition, as it is clear in the figures, increasing the clamping force of the bolts, decreases the magnitudes of maximum and effective stresses, and also decreases the effective distance. Consequently the increase in the clamping force decreases the amounts of the fatigue notch factors, results in improved fatigue lives of the joints.

In this section, to investigate the validity and accuracy of the volumetric method, the fatigue lives predicted with this method have been compared with the experimental test results. Fig. 16 shows the fatigue strength versus the number of cycles to fracture for different specimens on the volumetric method and experimen-
tal test results. As it can be seen in this figure, the volumetric method has a good agreement with the experimental test results 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 [6,26–28]. According to the Ref. [28], 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 smooth specimens, the important prerequisite of this procedure is to recognize the direction of the fatigue crack path of the notched specimens.

The volumetric fatigue life prediction is not similar to the other recognized methods such as Neuber and Peterson [30,31]. The volumetric approach uses the fatigue curve of smooth 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 [6,26–28]. The results of this work and other researches [6,26–28] indicates that this approach gives a relative good description of notch effect.

Fig. 14. Crack opening stress and relative stress gradient distributions for simple bolted specimen due to tightening torque (a) \( T = 1 \text{ Nm} \), (b) \( T = 2.5 \text{ Nm} \) and (c) \( T = 5 \text{ Nm} \) and applied remote longitudinal tensile stress of 192 MPa.

Fig. 15. Crack opening stress and relative stress gradient distributions for hybrid specimen due to tightening torque (a) \( T = 1 \text{ Nm} \), (b) \( T = 2.5 \text{ Nm} \) and (c) \( T = 5 \text{ Nm} \) and applied remote longitudinal tensile stress of 192 MPa.
The advantages of the volumetric method include the possibility of predicting fatigue life for many loading cases using notched geometry, the absence of the empirical and ambiguous coefficients used in traditional methods and the opportunity to obtain rapid and cost-effective results using a finite element method.

Based on this investigation, in the lightly fastened joints, there are only very small compressive stresses around the hole due to the small clamping force resulting from torque tightening. Consequently, in such cases, bolts and nuts play only a small role in delaying the fatigue crack initiation and propagation.

The results obtained from the experimental tests, revealed that the fatigue life of the joints were improved by increasing the tightening torque due to the compressive stresses which appeared around the hole. The improvement in fatigue life can be attributed to the method that the joint transmit the applied load. As the tightening torque is increased, a large portion of the load is transmitted by friction. The remaining portion of the load is transmitted by the bearing (through the bolt hole surface). The improvement in fatigue life can also be related to compression around the hole caused by the compression of the plates by the bolt pre-tension. Such negative stresses can reduce the total amount of resultant stresses that cause fatigue crack initiation and propagation in the plate due to the applied tensile external loads. This is in complete agreement with available results of fatigue life in simple bolted plate and double shear lap bolted joints clamped with tightening torques [2,3,6,11].

In addition, in the hybrid joints, the stress concentration around the hole is reduced significantly. Therefore, the local stress at the edge of the hole is lower, since some portion of the total load is transmitted by adhesive layer.

Finally, the comparison of the obtained results from the experimental tests, confirms that the hybrid joints have better fatigue performance than simple joints for all levels of the tightening torque. In addition, it should be noted that, in the hybrid joints, direct metallic contact between the plates does not occur (due to the adhesive layer presence); therefore, the possible fretting fatigue that causes an early fatigue crack initiation is eliminated. This is in appropriate agreement with some of the previous results reported in the literature, which discussed in introduction such as [8,18,20].

6. Conclusions

In this research, the effects of the torque tightening on the fatigue strength of the double lap simple bolted and hybrid joints have been investigated via experimental and numerical analysis.

Obtained results from the experimental analysis revealed that, increasing the tightening torque or clamping force on the joint, leads to improve the fatigue life. It was also observed that, the hybrid joints have longer fatigue lives in comparison with the simple bolted joints. In the numerical method, in order to estimate the fatigue life, a simple process based on the volumetric method by means of the available smooth S–N curve of Al2024-T3 and the fatigue notch factor was used. The estimated fatigue life was compared with the available experimental test results. The investigation revealed that the fatigue life estimated by the volumetric method has a good agreement with the experimental results for various specimens under the different amounts of the tightening torque. The obtained results showed that the fatigue lives of specimens were improved by increasing the tightening torque due to compressive stresses which appeared around the hole. The improvement in fatigue life can be attributed to the method that the joint transmits the applied load. Furthermore, in the hybrid joints, direct metallic contact between the plates does not occur; therefore, the possible fretting fatigue is eliminated.

References


