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A 3D Simulation of Bolted Joint and Fatigue Life Estimation Using Critical Distance Technique

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Abstract

Bolted joints are one of the most common joints in the industry and assemble the most of the machine elements and segments together. Majority of structures are affected by fluctuating forces, therefore there is the risk of fatigue failure that causes countless damages, thus fatigue life estimation of bolted joints have always been important. The value of high stress concentration at the threads root especially first engaged thread causes problems for fatigue life estimation, since by applying stresses lower than yield stress of the bolt material, plastic deformation occurs at zones of thread root that reach to ultimate stress but fracture does not happen and in some cases bolt-nut joints have infinite life, so that maximum stress at thread root is not fatigue life determinant. The modified critical distance technique and expressed stress at this distance were used for determination of fatigue life in joint. In this study, the bolted joint fatigue life prediction using critical distance technique was compared to experimental results. The three-dimensional finite element analysis for bolted joint was performed. Pre-tightening process and tensile axial force were simulated in ABAQUS software after applying two steps of force including rotation displacement to the center of the nut due to clamping joints (applied torque) and tensile force, the stress distribution resultant of different tensile forces by application of the critical distance technique and mechanical properties fatigue life were determined, and S-N curve prediction matched well with experimental data.

Nomenclature

$\Delta\sigma_{w_0}$	The fatigue endurance limit	K_{IC}	The fracture toughness of material
σ_B	The fatigue strength of material	L	The nut length
ΔK_{th}	The coefficient of the crack growth threshold	p	The pitch of the bolt
$r_{p'}$	The critical distance of low cycle fatigue (high stress)	r_p	The critical distance of high cycle fatigue (low stress)

1. Introduction

Since centuries ago, human beings have applied threaded joints in engineering structures. It is so applicable in any types of machinery for its easy installation and separation. The existence of discontinuity on the surface of the screw causes the stress concentration. As long as alternative loading influences most of the structures, it leads to fatigue fracture in connections that results in vital and economical damage. In

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order to prevent the failure of these joints, there are numerous investigations about creating the analysis of the connections. The stress distribution in the regions with high stress concentration, further examination of the techniques reduced stress and improved the fatigue life. In order to estimate the fatigue life of the bolted joints, this subject must be considered that due to high stress concentration in the thread roots during the initial stages of loading, this region enters plastic phase. Among the effective factors on fatigue life (the amount of preload, coarse, and fine threads) are the significant factors. It is noteworthy that maximum concentration factor takes place in the first thread between the bolt and nut, and the researchers concluded that the stress concentration in these zones by changing the shape of the thread reduced. Since the early nineteenth century, the stress distribution in threaded joints has been studied. The theoretical evaluation of maximum stress in the root of the threads depends on the load distribution between the involved threads. Researchers such as Maduschka [1] and Sopwith [2] introduced the theories that verified by photoelectric experiments. This theory was formulated by Stoeckly and Macke [3] in 1952 as a load distribution for bolt and nut joints.

In 1986, Peterson and Kenny [4] performed a frozen stress photo-elasticity test and compared photo-elastic stress distribution to the Stoeckly stress distribution equation, and concluded that before loading of the joint, the level of the nut face reached to the maximum stress, and by increasing the distance from the contact surface of the nut and bolt, stress decreases. Maruyama [5] for the first time obtained the stress distribution for bolted joint by finite element method.

Piscan et al. [6] utilized classical theory for connection analysis and considered bolts and members as linear and parallel springs. Whereas before loading, the bolted joint was tightened so so a nonlinear strain was created therein. Piscan modeled bolted joint using the Ansys software regardless of the thread, and found that strain decreases nonlinearly by increasing the stiffness of the contact point that is non-linear. Williams et al. [7] studied laboratory methods, classical theory, and simulation by Ansys software; simulation was accomplished in two ways: first simple model (without thread), second advanced mode (all of the design details and plastic specifications included); he examined the effect of increasing external load on the bolt applied force.

Lehnhoff et al. [8] calculated member stiffness were by considering stress distribution of the bottom surface of the bolt head uniformly, while Haidar et al. [9] proposed third order function as stress distribution. He also presented a new relationship for the members' stiffness, and results matched well with the experimental data and results from Ansys software. Zhang and Poirier [10] considered the members' stiffness as a function versus external force, because when the external force is added, the surface of the contact changes and the stiffness is a function of the applied force. Dragoni [11] and Waltermire [12] examined the effect of thread pitch and nut shape on stress concentration. One of the remarkable studies on effective factors on fatigue life is Majzoobi's research and et al. [13]. They studied the bolted joint under sinusoidal loading during experimental study using Instron universal apparatus. The results showed that the coarse thread bolts have a longer fatigue life than the fine thread bolts. Among the connections, the bolted joint with the slotted tapered nut and the spring washer has the highest fatigue life than the other joints [14].

Knight et al. [15] examined the effects of factors such as preload, and the area of the spring washer surface on the connection reaction during loading in a structure with two L-shaped sections that were connected by a bolt. Esmaeili et al. [16] showed that in both experimental and numerical methods by increasing the twisting torque of double lap simple bolted and hybrid (bolted/bonded), fatigue life improves due to compressive stress distribution around the hole in the plate. In addition, the compound connection has a longer fatigue life than a simple connection. Biehl [17] simulated tightening torque by ABAQUS software to obtain nut coefficient and the torque simulated as boundary conditions. He also carried out several experiments, and obtained preloads for each torque and finite element results matched well with experiment data.

Fukuoka and Nomura [18] proposed an effective mesh generation scheme, which can provide helical thread models with accurate geometry to analyze specific characteristics of the stress concentrations and contact pressure distributions caused by the helical thread geometry; it was shown that how the stress and contact pressure of the thread root vary along the helix and at the nut loaded surface in circumferential direction and why the second peak appears in the distribution of Von-Mises stress at thread root. Hu et al. [19] studied the failure mechanisms under tensile loading for high strength of the bolts with different degrees of tolerance, which is related to coating processes and tolerances degree of the bolt and nut. Chen et al. [20] found that the fatigue life extended by introducing suitable pitch difference. The effect of the bolt-nut fitted clearance on the fatigue failure was investigated. Yu and Zhou [21] in another study, like Biehl research, obtained the relation between preload and torque, except for the applied torque as shear stress to the outer surface of the nut. Yousefzadeh and Torabi [22] created a groove on the thread of the M20 bolt using ABAQUS software to reduce the stress concentration and achieve to the optimum depth of the groove, in which, if the depth is more or less than optimal amount, it has a reverse effect on fatigue life and stress concentration coefficient of the specimens. The other factor that can dramatically decrease the maximum local stress in the root of the threads is increase in the roots radius in the trapezoid thread bolts with large diameters.

In the research of Selah by employing ANSYS Workbench software and three-dimensional finite element simulation, failure analysis of hybrid bonded and bolted single and double lap joints with laminated composite adherents subjected to axial, shear, and bending loads were performed [23]. Cojocaru and Korka [24] using two-dimensional modeling showed that by increase in the root radius of the threads, stress decreases. Recently, Jasztal and Regowski [25] studied fatigue life analysis of the bolted joints using ANSYS software. They presented ways to modify the fatigue life of the bolted joints. Furthermore, they described preload by means of mathematical relations and illustrated the preload effect mechanism on fatigue strength, graphically.

Yu et al. [26] investigated high-temperature low-cycle fatigue life prediction and experimental research of pre-tightened bolts; a new low-cycle fatigue model based on the Von-Mises equivalent stress/strain criterion was proposed, meanwhile, the proposed model was used to predict low-cycle fatigue life of pre-tightened bolts in the high temperature according to the finite element analysis results. In the review paper of Susmel [27], it was clearly shown that the critical distance technique is successful in predicting the high-cycle fatigue strength of real mechanical components weakened by different geometrical features and subjected to both uniaxial and multiaxial fatigue loading. Stress distribution in a bolted joint was obtained in a three-dimensional model with proper accuracy. From the innovations of the article, it is possible to apply the critical distance technique for the first time in order to calculate the fatigue life of the bolt and nut joint.

2. Analysis of Bolted Joint in Software

2.1. The Geometry of Model

A bolted joint similar to study of Refs [13, 14] with all its details including thread pitches, thread profiles based on existing standards (including curvature in the root of the threads), with bushes was modeled. The bushes were connected to a fixture, because of the increase in the volume of the calculations, the fixture in the modeling was not considered and the force directly with a difference percentage due to the exiting friction was applied to the bushes. Schematic of the fixture and bolted joint with bushes are illustrated in Fig. 1. The fixture can clamp various bolts up to 30mm in diameter and 200mm in length. The fixture was made of a material rigid enough to resist fatigue loading (VCN 150) [28]. In Fig. 1, left and right elements at the top of the fixture are not used in modeling.

Since the stress concentration in the threads root, especially the first thread is very high during loading; therefore, these grooved areas determine the fatigue life and strength of the connection. One of the methods for estimating the fatigue life of the grooved specimens is the critical distance technique. For the first time in this study, the modified critical distance technique was used to estimate the fatigue life of the bolted joint. Therefore, the ABAQUS software was used to determine the location and amount of local stresses using finite element method.

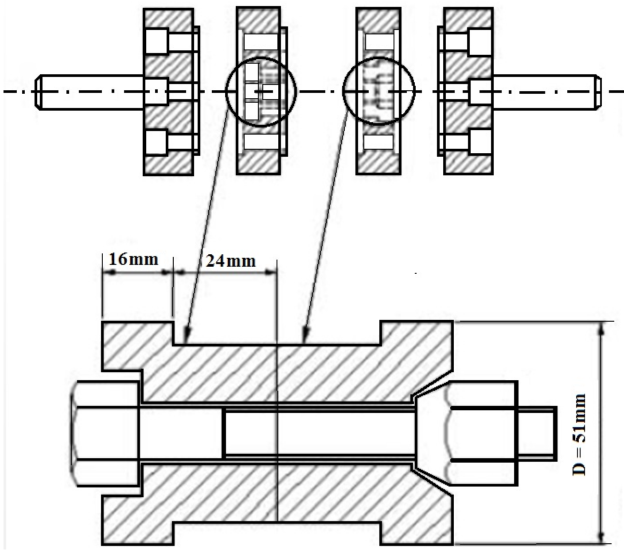

Fig. 1. Schematic of fixture and bolted joint with bush [14].

2.2. Modeling of the Bolted Joint

M12 bolted joint was simulated and analyzed using ABAQUS software. Because of all the details considered, including the bolt pitch, the modeling was three-dimensional. The first step loading, which includes the torque necessary to tighten the connection, was used for the boundary conditions like Biehl research [17]. The geometrical dimensions of bolt are as follows: the external diameter = 12mm, the bolt length equal to 70mm, and bolt pitch = 1.5mm. The modeling of the bolt and nut was done according to DIN 931 and DIN 934 Standards. The hexagonal nut was considered for the bolted joint. The three-dimensional schematic of the bolt and nut, bush and the assembly view of the fixture is illustrated in Fig. 2. The bush was also modeled in two parts, similar to the dimensions of the Ref [14]. It is noteworthy that the analysis was performed in the form of linear elastic, only the characteristic of the Poisson's ratio and the modulus of elasticity of materials are required. The material used to make bolt and nut and Bushes were AISI 1020 and VCN 150, respectively. Mechanical properties of AISI 1020, VCN 150 (V155) are shown in Table 1.

After modeling the components (bolt, nut and bushes), the elements were assembled in such a way that the bolt passes through the inside of the bushes and is tightened using the nut. Of course, in the software, the bolt was displaced so that the threads involved with the bolt and nut with consideration low clearance in accordance with Fig. 3 located in appropriate position. The proper clearance between the threads of bolt and nut in the present model was selected according to Ref [29].

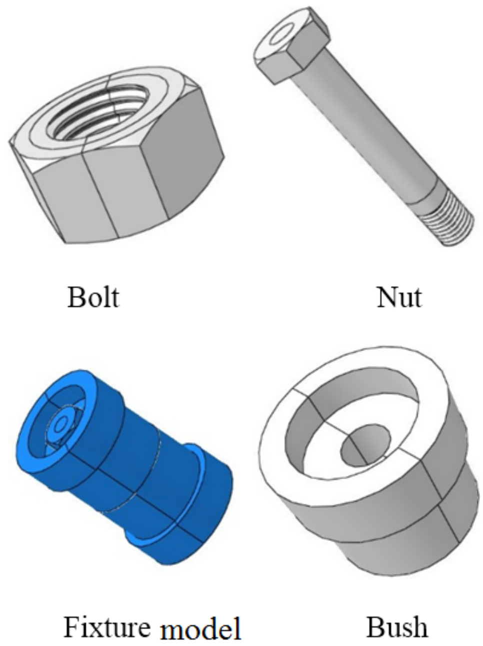

Fig. 2. The three-dimensional schematic of the bolt (M12), nut, bush and fixture in ABAQUS software.

Table 1
Mechanical properties of AISI 1020 and VCN 150 [28].

Property	VCN 150	AISI 1020
Hardness (HB)	111	217
Tensile (Ultimate) strength (MPa)	394.72	744.60
Tensile strength, yield (MPa)	294.74	472.3
Elongation at break (in 50mm)	36.5%	22%
Reduction of area	66%	49.9%
Modulus of elasticity (GPa)	200	210
Bulk modulus (GPa)	140	-
Poisson ratio	0.290	0.30
Shear modulus (GPa)	80	-

In the first step, the loading was applied as a boundary condition for tightening the bolt. Biehl's research [17], similar to the present problem, was used as reference sample that the bolt tightening angle was applied to the point in the center of the nut. In the second step, the axial tension loading was applied, and was used to estimate the stress distribution for evaluating the fatigue life. Moreover, the solving method of model selected static-general, in this analysis, the inertia effects were ignored, and the analysis was considered completely static. Four-contact interaction was considered for all sliding surfaces in the finite element model, that includes the interfaces between mating threads, nut face and lower bush surface, upper and lower flanges surface, surface of the bolt head and the lower bush surface. Bolt shank and hole are not in contact with clearance fit. Contact type was surface to surface. The value of the friction coefficient equal to 0.15 was assigned to all sliding surfaces.

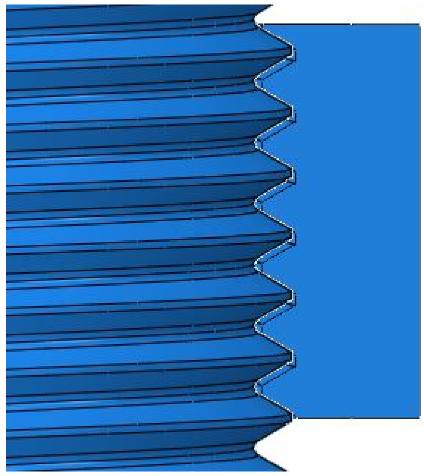

Fig. 3. The schematic of clearance between bolt and nut.

2.3. The Model Meshing

Due to the complexity of the bolt geometry and thread spiral modeling, the tetrahedron element was selected. The maximum concentration of the stress occurs at the threads root, hence finer mesh was assigned to these zones. The bolt and nut were meshed with 4-node linear tetrahedron element (ABAQUS C3D4 element). The meshing of the model (bolt and nut) is illustrated in Fig. 4.

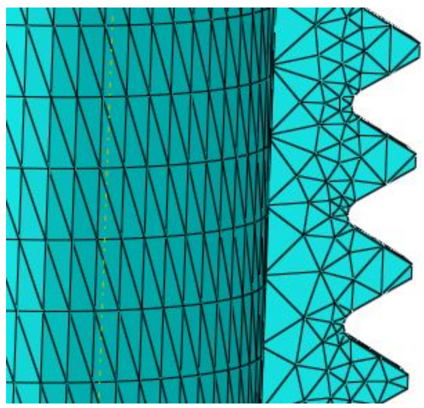

Fig. 4. The meshing of the model (bolt and nut).

2.4. Boundary Conditions and Loading

For the applied boundary conditions, the top surface of the bolt head was fixed; side surfaces of the bushes were constrained in circumferential direction. A multi-point constraint was used to control rotation and prevent lateral translation ($\theta_z = 0.07$, $x = 0$, $y = 0$), the corresponding angle to torque (10Nm) was equal to 0.07 radians. The multi-point constraint as located at the center of the nut, and the nodes around the outer diameter of the nut were connected to each other. A prescribed rotation was applied to the multi-point constraint. In the second step, the tensile load was applied to the bottom surface of the lower bush as pressure in the opposite direction of the loading boundary conditions of the model that is illustrated in Figs. 5 and 6.

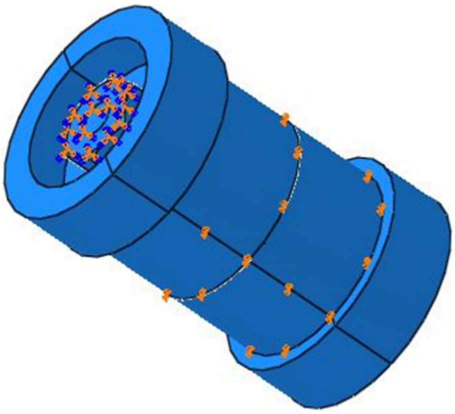

Fig. 5. The boundary conditions of the model in software.

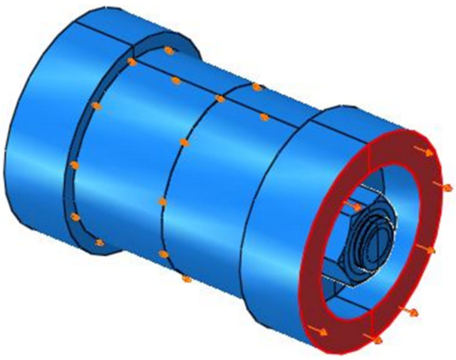

Fig. 6. The stress applied to the bush create tensile in the bolt.

2.5. Validation of Meshing

Mesh density is a significant section that controls accuracy of finite element result. By increasing the density of the mesh (decreasing the dimensions of the elements), the numerical solution of the problem converges to a single solution, hence, mesh congruity should be checked in areas of the model where stresses, strains, contact pressure, or any other parameter that must be accurately calculated.

In order to determine the optimum number of elements, a main parameter, which is the purpose of the analysis of present research, was taken into account, the stress distribution in the threads root was required to calculate the bolted joint fatigue life. The value of the Von-Mises stress obtained from the elements in critical distance was considered.

Due to the fact that the maximum stress concentration creates in the first thread area, the mesh density in this area was increased. The Von-Mises stress diagram was obtained from the maximum stress point to the center of the bolt as a result, and from the intersection of this diagram with the line of the critical distance between the fatigue strength and the critical distance of the fatigue limit, stress at the critical distance was determined. The obtained results for mesh convergence are illustrated in table 2. According to Figs. 13 and 14, it is observed that maximum Von-Mises stress takes place in the threat root and minimum stress belongs to bolt center. In this analysis, the number of elements were equal to 6749.

Table 2
The convergence mesh for Von-Mises stress, axial loading = 47.68kN.

The Von-Mises stress in critical distance (MPa)	The number of elements
238.75	2458
325.21	3542
405.82	4876
487.24	5360
494.73	6849

3. Fatigue Life Estimation Using Critical Distance Technique

The critical distance technique is one of the most effective techniques for estimation of the fatigue life and strength of materials. Neuber [30], after determination of the relation for the characteristic length, showed that the damage due to the real stress was the average stress fluctuation pattern in the fatigue region (Notch surrounding zone); he formulated this relation and called it the linear method. Several years later, Peterson [31] presented a point method, after Tanaka researches [32], Atzori et al. [33] and Taylor [34], finally, the relationship was simplified as follows [35].

$$L = \frac{1}{\pi}\left(\frac{\Delta K_{th}}{\Delta\sigma_0}\right)^2 \qquad (1)$$

If the results obtained from the critical distance technique is less than the endurance limit of the material, then specimen will not fail and fatigue life is infinite. However, if the applied stress is larger than the endurance limit stress, then the specimen has a limited fatigue life. Hattori et al. [36] was able to modify the critical distance technique and estimated fatigue strength and life for low cycles and high stresses, so that the Eqs. (2, 3) were utilized for determination of the critical distance in high- and low-cycle fatigue g, respectively. If the created stress at critical distance is greater than the fatigue strength of the material, the specimen will fail in the first cycle.

$$r_{p'} = \frac{1}{2\pi}\left(\frac{K_{IC}}{\sigma_B}\right)^2 \tag{2}$$

Where the parameters of K_{IC} and σ_B are fracture toughness and fatigue strength of the material, respectively and $r_{p'}$ is critical distance for low-cycle fatigue (high stress).

$$r_p = \frac{1}{2\pi}\left(\frac{\Delta K_{th}}{\Delta\sigma_{w_0}}\right)^2 \tag{3}$$

where ΔK_{th} is the threshold crack growth coefficient, $\Delta\sigma_{w_0}$ is fatigue endurance limit and r_P is critical distance in high-cycle fatigue (low stress). Therefore, interpolation of two points create a line, which intersects with the stress distribution diagram, the critical distance is determined for the specified load by reflecting the stress that is calculated using critical distance on S-N curve, corresponding fatigue life is determined at the obtained stress distribution and for other loadings repeated and for each loading, fatigue life is predicted, therefore by connecting the points, S-N curve is created [37]. Determination of the critical distance using stress distribution with point and line methods is illustrated in Fig. 7.

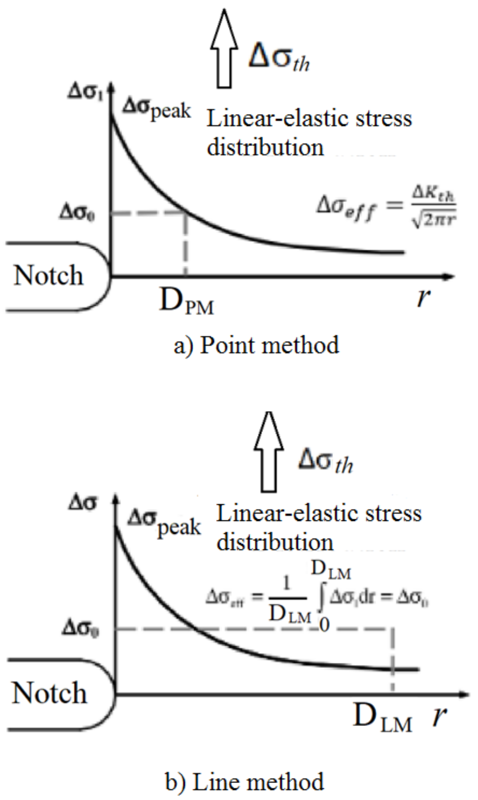

Fig. 7. Determination of the critical distance using stress distribution with point and line methods [27].

A flowchart of the present work procedure is shown in Fig. 8.

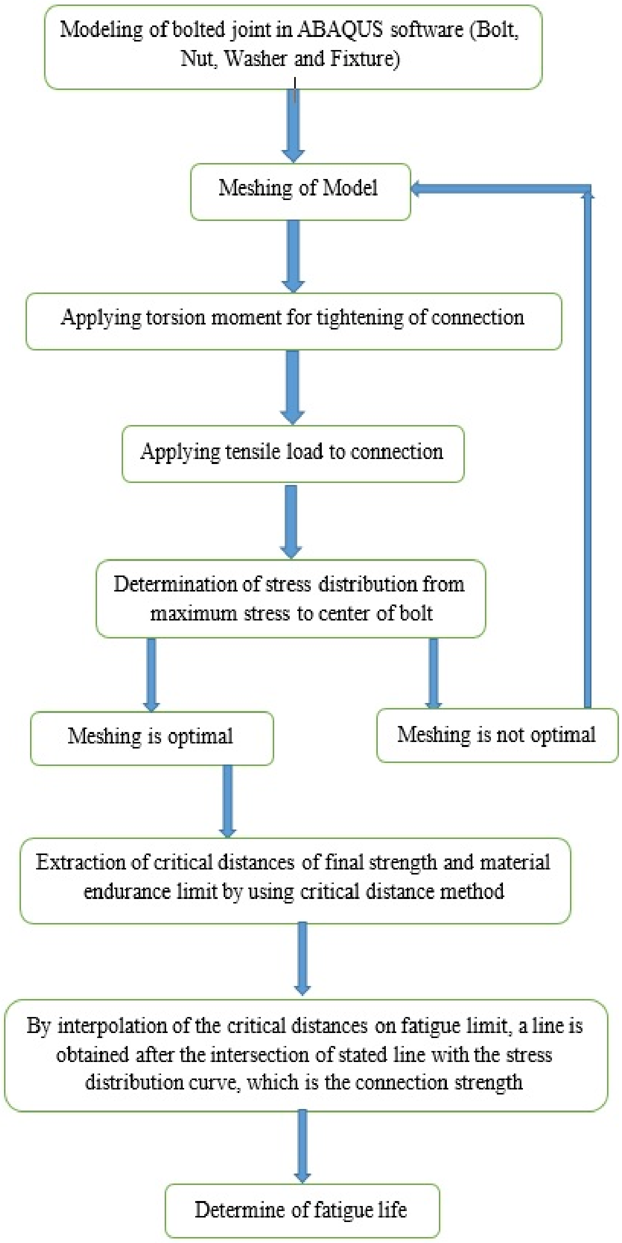

Fig. 8. A flowchart of present work procedure.

4. Results and Discussion

4.1. Numerical Results

The present work has a great similarity with Biehl's research [17], and these two studies have differences in the model, that washer was not considered in present model, also, the results of the first step (Pretightening) should be reassured, washer was added to the model and a rotation of 27.3 degree was applied to the center of the nut as stated in Biehl's research. The stress distribution results and maximum stress are in good agreement with the obtained results of the Biehl study that are illustrated in Figs. 9-11, and a small difference due to different being the shape and number of elements.

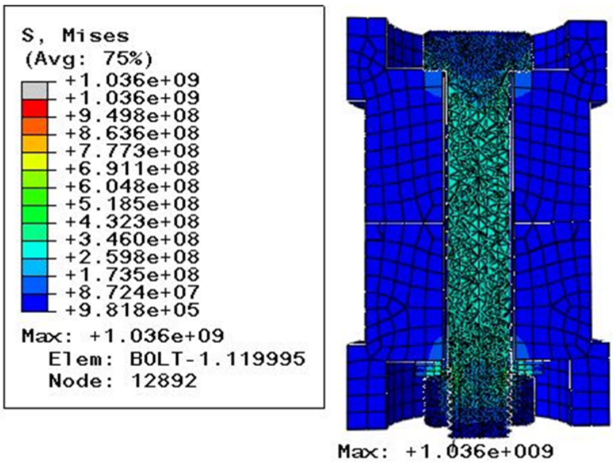

Fig. 9. The Von-Mises stress created in the model.

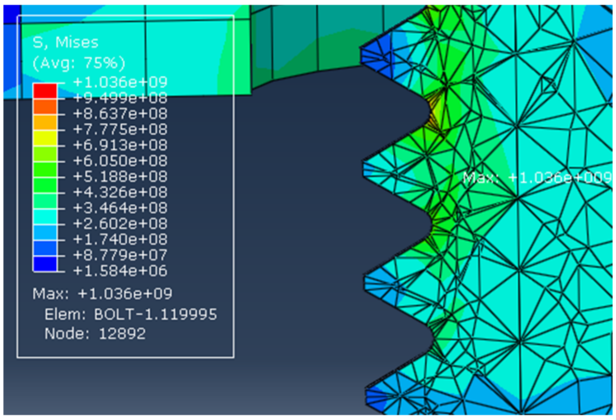

Fig. 10. A view of the concentration of Von-Mises stress in the root of the bolt threads.

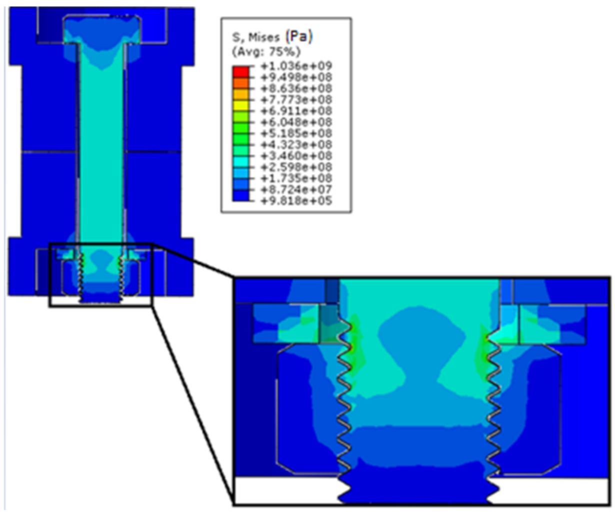

Fig. 11. Distribution of the obtained stress according to Bill research [17].

4.2. The Fatigue Life Estimation

In this section, fatigue life for bolted joint is estimated by the critical distance technique and the results are compared with experimental results. The applied stresses on bolted joint according to the Majzoobi experimental tests [13] are illustrated in Table 3. The results of the stress distribution are used in this section. Due to the repetition of method for different forces, only detailed steps for estimation of the highest stress loading is explained. According to previous studies, the highest concentration of stress occurs in the first root zone of engaged threads, as shown in Fig. 12. Using the nodal coordinates of maximum Von-Mises stress and the center of the bolt cross-section, the path on model was defined. Then, by determining the maximum Von-Mises stress, the nodal coordinates are estimated and path of this node to point in the bolt center on cross-section is determined, this path is illustrated in Fig. 13.

Table 3
The applied tensile loading on model [13].

The number of test	Applied force (kN)	Applied stress (MPa)
1	47.84	567.54
2	43.06	510.78
3	38.73	454.03
4	33.53	397.28
5	28.71	340.52
6	18.18	215.66

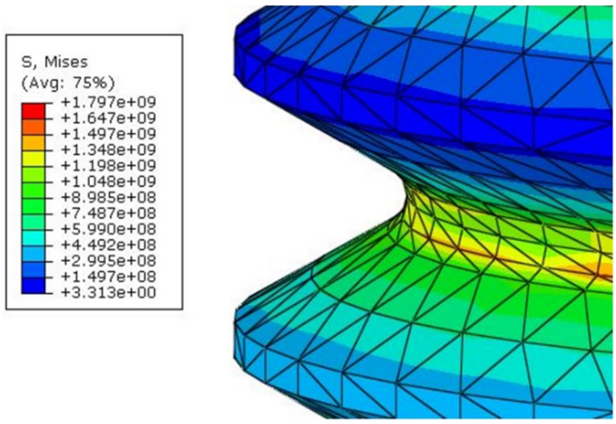

Fig. 12. Stress distribution in the first thread root area, axial load= 47.84kN.

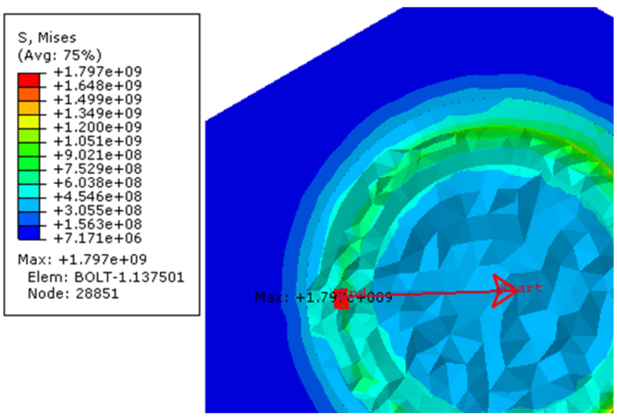

Fig. 13. Determined path from maximum Von-Mises to bolt center.

As a result, the stress distribution diagram based on the distance from the node (the location of the highest concentration of stress) for applied loading (force= 47.84kN) to the bottom of the nut is illustrated in Fig. 14. The AISI 1020 crack growth curve is available for the load ratio (stress) equal to 0.1 and 0.7 in Ferreira's research et al. [38], since the stress ratio (R) in Majzoobi study was equal to zero, the initial torque applied on the connection, causes a small elongation in the bolt, in which $R = 0.1$ is considered, and the crack growth pattern (Fig. 15) was used. The values of the parameters used to estimate the fatigue life are as below [38]:

$$\Delta K_{th} = 11.6\text{MPa}\sqrt{\text{m}},\ K_{IC} = 40\text{MPa}\sqrt{\text{m}},$$

$$\sigma_B = 520\text{MPa},\ \Delta\sigma_{w_0} = 0.504S_{ut},\ \Delta\sigma_{w_0} = 262\text{MPa}$$

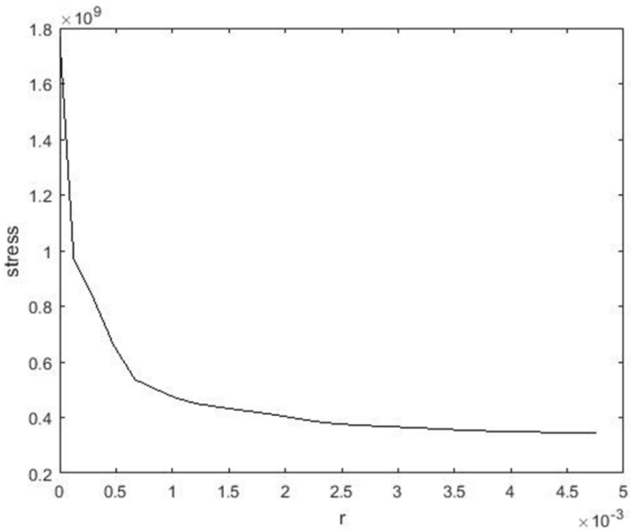

Fig. 14. Stress distribution curve, applied load equal to 47.84kN.

The values of endurance limit and critical fatigue strength distance are equal to 0.3mm and 0.94mm, respectively. By interpolation of the critical distances on fatigue limit, a line is obtained after the intersection of stated line with the stress distribution diagram (Fig. 16), the obtained point is fatigue limit that determines fatigue life by reflecting this point on the S-N curve, the amount of fatigue life for specified loading was estimated equal to 59280 cycles, as illustrated in Fig. 17, fatigue limit stress is equal to 494.737MPa, when tensile stress is equal to 563.758MPa applied.

The mentioned steps repeated for different applied stresses, and the fatigue S-N curve critical distance technique was compared with experimental results (Fig. 18). In addition, the comparison of the data obtained from the present study and the experimental results are illustrated in Table 4.

Table 4
The comparison of the data obtained from the present study and the experimental results.

Applied stress (MPa)	Fatigue life (experiment) [Ref. 13]	Fatigue life (critical distance Technique)
216.107	515011	475059
338.255	156193	154150
389.933	101343	114624
446.309	69386	90909
507.383	56607	67190
563.758	43585	59280

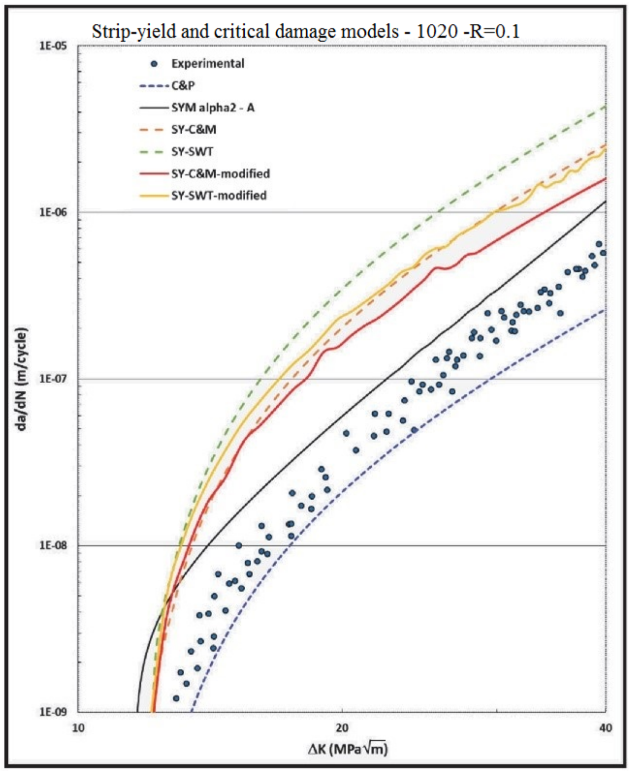

Fig. 15. The AISI 1020 crack growth curve [38].

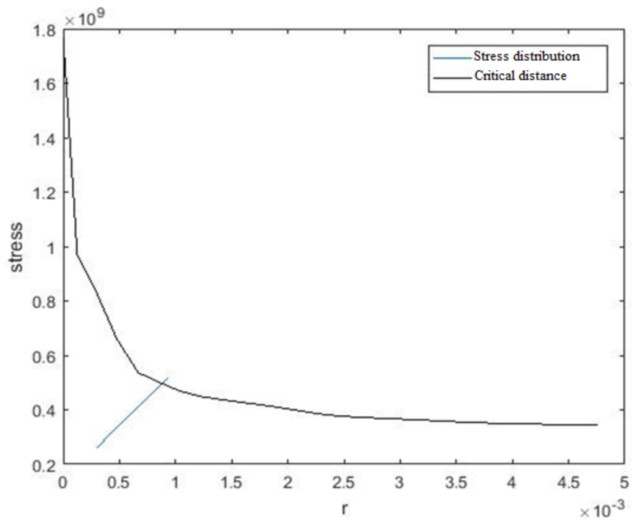

Fig. 16. Intersection of stress distribution diagram and critical distance line.

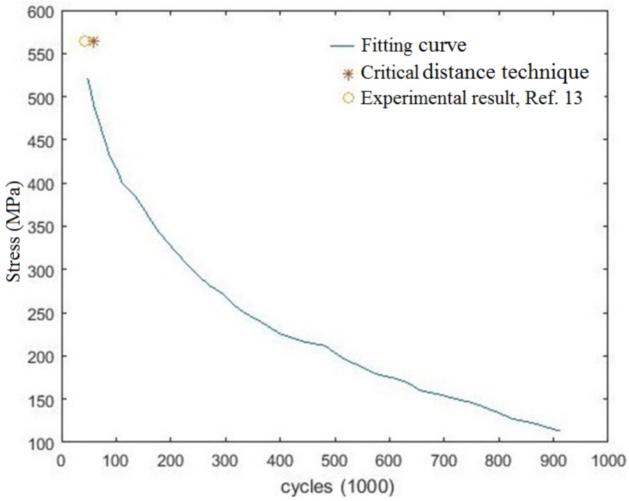

Fig. 17. Experimental S-N curve [13].

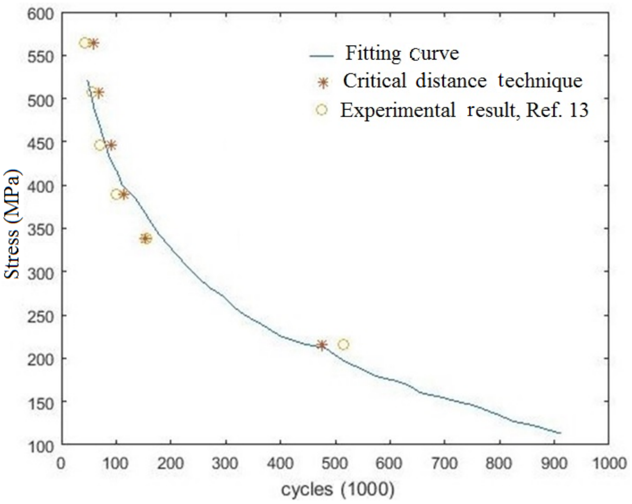

Fig. 18. The comparison of experimental [Ref. 13] and critical distance technique fatigue life.

5. Conclusions

In this study, a three-dimensional finite element model of the bolted joint considering helix angle for threads was analyzed to simulate two steps, including pre-tightening process and axial tensile loading. Due to the applied linear elastic analysis method, stress in the threads root exceeds yield stress of the material, and singularities occur in these regions. Therefore, the appropriate method for fatigue life estimation of notched specimens must be utilized. In this research, the modified Critical Distance Technique was applied for fatigue life estimation, in addition to estimation of the high-cycle fatigue life, which also covers low-cycle fatigue life.

According to the comparison of the data obtained from the present study and the experimental results according table 4, the fatigue life in low-cycle fatigue has better correspondence than high-cycle fatigue. There is a good agreement between the results and it can be concluded that the present technique is suitable for fatigue life prediction of the bolted Joint. It is noteworthy that the conventional critical technique was only used for calculation of fatigue strength, while the modified method is used to determine the fatigue life. For the first time, the critical distance technique was used to calculate the fatigue life of the bolt and nut joint.

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