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RESEARCH ARTICLE - MECHANICAL ENGINEERING

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The Effect of Clearance and Pre-Tension on the Performance of a Bolted-Joint Using 3D FEA

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Abstract A bolted joint is a typical connection that is widely used in machine assemblies and structural components. These joints encounter several types of working load. This paper is concerned with the analysis of a bolted joint under shear and tensile loading using a three-dimensional finite element method. Modeling a bolted joint with three-dimensional finite element (FE) applications is not an established technique. We make a full three-dimensional FE model complete with contact surfaces, clearance between bolt and hole, and pre-tension. This model identifies the localized points of high stress concentration, which is not possible with the simplified two-dimensional or axi-symmetric models previously reported. A comprehensive investigation is made on the effect of clearance and pre-tension on the mechanical behavior of a single bolt joint under shear loading. Certain of the numerical results are verified by measuring the performance under load of a bolt shear joint. The study is then extended by analyzing the bolted joint in tension with varying levels of pre-tension.

Keywords Finite element analysis · Three dimensional · Pre-tension · Clearance · Bolted joint · Preload

الخلاصة

تعد براغي التثبيت (مسمار التثبيت اللولبي) من الأشياء الشائعة الاستعمال في ربط وتثبيت قطع الآلات والمنشآت بعضها ببعض، وذلك نظرا لسهولة تبديلها وتثبيتها. إن هذا النوع من التثبيت يتعرض لعدة أنواع من الإجهادات. إن الدراسة المطروحة هذه تتعلق بهذا لنوع من التثبيت تحت تأثير القوى الشدية والقصية وذلك باستخدام طريقة العنصر المحدود بأبعاده الثلاثة . إن الوصول إلى التمثيل الأمثل لهذه الروابط ما يزال محل سؤال. وفي هذا البحث روعي تمثيل هذه الرابطة بأبعادها الثلاثة مع إبراز الأسطح المتلامسة والمسافات البينية بينها وتقدير القوى الشدية الابتدائية. إن هذا الأنموذج ساعد في تحديد مراكز الجهد العالية التي يصعب تحديدها في أنموذج ثنائي الأبعاد أوحتى ذات التناظر الدائري المنشورة في الأبحاث السابقة. وفي هذا البحث كان التركيز منصبا على تأثير المسافات البينية والقوي الشدية الابتدائية على الاجهادات تحت تأثير القوى القصية. كما أمكن التحقق من النتائج الرقمية وذلك بإجراء بعض التجارب المعملية على برغي تثبيت مفرد تحت تأثير القوة القصية. ومن ثم أمكن من دراسة تأثير برغي التثبيت في الشد بمستوى شد سابق و متغير.

1 Introduction

Bolted-joint analysis is a complex procedure involving a number of factors, such as bolt pre-tensioning, contact between the plates, bolt size, number of bolts, bolt layout, bolt deformation, connected member deformation, clearance, loading conditions, supporting conditions, number of plates, and friction flange thickness. Analytical, experimental and numerical techniques have been used to analyze this type of joint. Analytically, the first step towards a bolted joint analysis is to calculate the stiffness of the bolt (k_b) and the connected members (k_c).

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For the stiffness of the bolts, the formula uses the tensile stress area (A_b), major diameter area of the bolt (d), length of threaded portion and unthreaded portion in the grip (l_b). Stiffnesses k_b and k_c are also functions of geometry and the elastic constants of the bolt and connected members. By assuming a one-dimensional condition, the stiffness of the bolt and connected members can be defined by the formula reported by Shigley and Mischke [1].

$$k_b = \left(\frac{A_b E_b}{l_b}\right) \tag{1}$$

$$k_c = \left(\frac{A_c E_c}{l_c}\right) \tag{2}$$

The resultant load on the bolt (F_b) and the resultant load on the member (F_c) can be given by the following formulae:

$$F_b = F_i + \left(\frac{k_b}{k_b + k_c}\right) F \tag{3}$$

$$F_c = F_i - \left(\frac{k_c}{k_b + k_c}\right) F \tag{4}$$

Where F_i is the preload and F is the external tensile load.

Calculating the stiffness for the bolt is relatively easy, but for the connected members the situation is somewhat different. When there are more than two connected members in the grip of a connection, they act like compressive springs in series. If one of the connected members is a soft gasket, its stiffness relative to the other connected members is so small that the total member stiffness is usually taken as being equal to that of the gasket material. However, when there is no gasket, the compression spreads out between the bolt head and the nut. Hence, the area under compression is not uniform and the stiffness of the member is rather difficult to obtain, except by experimentation. Some analytical methods exist for approximating the stiffness. Ito [2] suggested the use of Rotscher's pressure cone method with a variable cone angle. This method is quite complicated, so other methods have been devised, such as Mischke [1] (with cone angle of 30°) and Motosh [3]: both these methods overestimate the clamping stiffness. Once the stiffness is calculated, the resultant bolt load F_b and resultant load on connected members F_c can be calculated using Eqs. 3 and 4. Therefore, analytical methods do have limitations in predicting the stress in the connected members of a bolted joint. Sufficient experimental work requires significant resources and time, and intrinsic result variation makes it difficult to interpret and reproduce. Therefore there are practical advantages to the use of numerical techniques such as the finite element method. For example, models can be altered with ease, and non-linear behavior can be included if necessary.

Considerable research has been carried out to explore the behavior of bolted joints under different types of loading. Al Jefri et al. [4] investigated the characteristics of various geometries of bolted joints with different pre-tension loads. The three-dimensional finite element analysis (FEA) of bolted joints with finite sliding deformable contact has been studied by Chen et al. [5]. The performance of the helical thread and effect of friction on the distribution of load distribution was analyzed. This showed that analytical analysis by Yamamoto's method [6] reaches a lower value of load ratio than that shown by FEA on the first thread. Comparisons of the load distribution on each thread between axi-symmetric model and three-dimensional model were reported. Lehnhoff et al. [7] studied the effect of bolt threads on the stiffness of bolted joints using axi-symmetric FEA. Connected members of different materials were analyzed with different bolt sizes and it was observed that the connected member stiffness decreases with decreasing bolt size. Tanaka et al. [8] used finite-element analysis method to incorporate plasticity theory and the von Mises yield criterion to study the behavior of bolted joints tightened to plastic yielding. Mechanical behavior of bolted joints in various clamping configurations was examined by Fukuoka et al. [9]. In their work, the mechanical behavior of bolted joints in various clamping configurations were analyzed using a 2D finite element model as a multi-body elastic contact problem, and the effects of nominal diameter, friction and pitch error upon stress concentrations were evaluated for through-bolts, studs, and tap bolts. Menzemer et al. [10] carried out experimental work to study the block shear for bolted joints for different arrangements of bolts. Strain distributions around the periphery of the connection were measured and compared to finite element predictions. A similar type of study was done by Tan et al. [11]

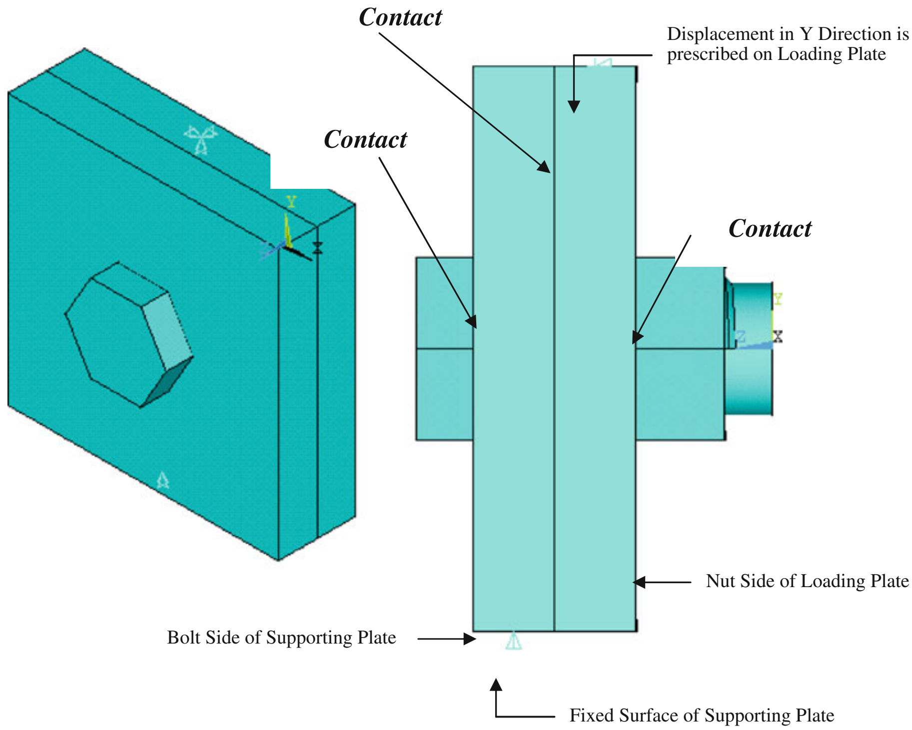

Fig. 1 One-bolt joint showing shear loading and boundary conditions

using an elastic plastic model to study the effect of bolts in rows. Experimental work confirms that there is a reduced effective capacity per bolt with any increase in the bolt number that is placed in a row - known as the row effect. The mechanical behavior of bolted joints during a torque controlled tightening process was analyzed as an elastic-plastic contact problem by Fukuoka et al. [12]. A 3-D analysis using a 2-D finite element mesh with each node having 3 degrees of freedom was conducted. The relationship between axial bolt stress and nut rotation angle was reported. Further study on the mechanical behavior of hybrid (bolted and bonded) joints applied to aeronautical structures was reported by Paroissien et al. [13]. In this study, a fully parametric analytical 2-D model based on the finite element method is presented. A special finite element, the “bonded beam” element is computed to simulate the bonded adherents. Again, Fukuoka et al. in [14] derived a series of closed form algebraic equations which can calculate the true cross sectional areas of internal/ external screw threads with the effects of the helix and root radius taken into account. The equations obtained can be applied to coarse or fine pitch. In [15], Cornwell used the finite element method to accurately estimate the load factor for 4,424 unique combinations over the entire range of the four joint design parameters, namely bolt diameters, joint thickness, individual plate thickness and plate material combinations.

Our survey of the literature shows that there is a scarcity of published data on the effect of factors such as pre-tension, clearance, and layout effect on joint behavior. In addition, researchers have mostly used two-dimensional and axi-symmetric models for numerical investigation. Very little work using three-dimensional FEA has yet been reported. Hassan [16] studied the effect of bolt layout on the mechanical behavior of bolted joint using a 3-D FE model. The problem with 2-D models is that they cannot be used to investigate the effect of pre-tension, clearance, or friction, because of the required simplifying assumptions. In many cases, the stress distribution along the thickness is not uniform and localized points of stress concentration may exist in the bolt and connected members. These critical points cannot be visualized properly with two-dimensional models. Similarly, as the threads are actually a helical form, load distribution on the threads cannot be accurately analyzed with 2D models. All these limitations indicate a need for a full three-dimensional finite element model in which all factors are taken into consideration.

The focus of this paper is to investigate the effect of parameters such as pre-tension, and clearance, on the displacement pattern and stress distribution of the connected members using 3D FEA. The finite element joint model of a one-bolt joint is loaded in shear and tension. Certain of the numerical results are verified by conducting experimental work on a one bolt shear joint.

Table 1 Geometric dimensions of the model

Parts	Dimensions
Loading plate (LP)	70 mm ×70 mm ×10 mm
Supporting plate (SP)	70 mm ×70 mm ×10 mm
Bolt	M16 × 2
Nut	M16 × 2

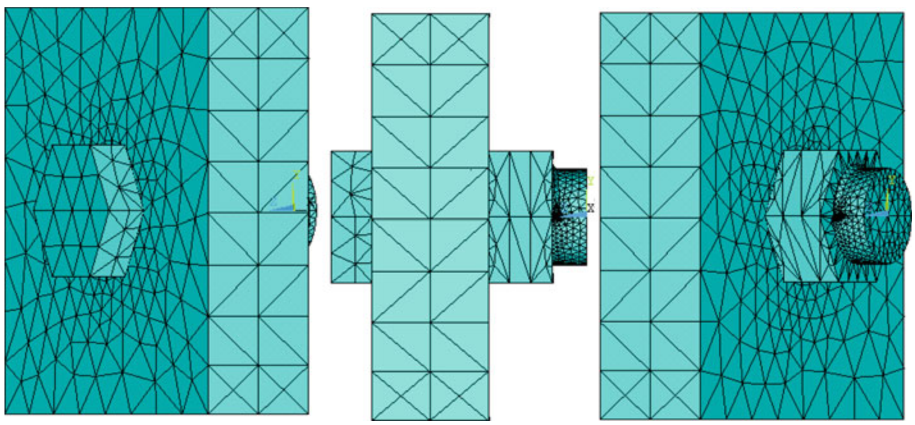

Fig. 2 Typical finite element mesh

2 Computational Model for Shear Loading

Before developing a finite element model, we must determine the bolted joint characteristics to be modeled and understand the capability of the finite element program being used. The bolted joint has many complexities that are impractical to capture in simulation. Two primary characteristics are pre-tension and mating part contact. To investigate the effect of pre-tension and clearance, first a one-bolt joint under shear loading is analyzed. A typical one-bolted shear joint is shown in Fig. 1. For this purpose, a three-dimensional finite element model is developed using the commercial ANSYS FE code [17], assuming material behavior to be linear isotropic. The geometric dimensions used for modeling are given in Table 1. The two plates, and a nut and bolt are modeled using 3D solid elements. Threads are included on the bolt and nut. Figure 2 shows a typical finite element mesh of the complete assembly having 30,000 nodes and 70,000 elements.

When a load is applied to separate a bolted joint, the bolt holds the connected members together. The pre-tension load is used to model a pre-assembly load in a joint fastener, and the pre-tension should be more than the applied load. When the applied load exceeds the pre-tension, the parts will separate. Pre-tension can generally be modeled with temperature, constraint equations, or initial strains. Two ANSYS features, the *PRETS179* pre-tension element and the *PSMESH* pre-tension meshing command, can be used for this type of analysis. The user creates the element and applies the pre-tension load through this element [12]. From a simulation standpoint, the surfaces that are in contact must be able to separate or slide, and this is where the contact elements are used. The other interface surfaces, where separation or slide is not expected, can be fixed together. Due to the non-linear effect of the contact elements, contact problems present two significant difficulties. Depending on the load, material, boundary conditions, and certain other factors, surfaces can come into- and go out of- contact with each other in a largely unpredictable and abrupt manner. The other difficulty is that most contact problems need to account for friction. There are several friction laws and models to choose from, and all are nonlinear. Frictional response can be chaotic, making solution convergence difficult. ANSYS supports five contact models: node-to-node, node-to-surface, surface-to-surface, line-to-line, and line-to-surface. Each type of model uses a different set of ANSYS contact elements. In this type of problem, the surface-to-surface contact elements can be used to model flexible-flexible contact between surfaces. These surface-to-surface elements are well suited to applications such as interference fit assembly, contact or entry contact, forging, and deep-drawing problems.

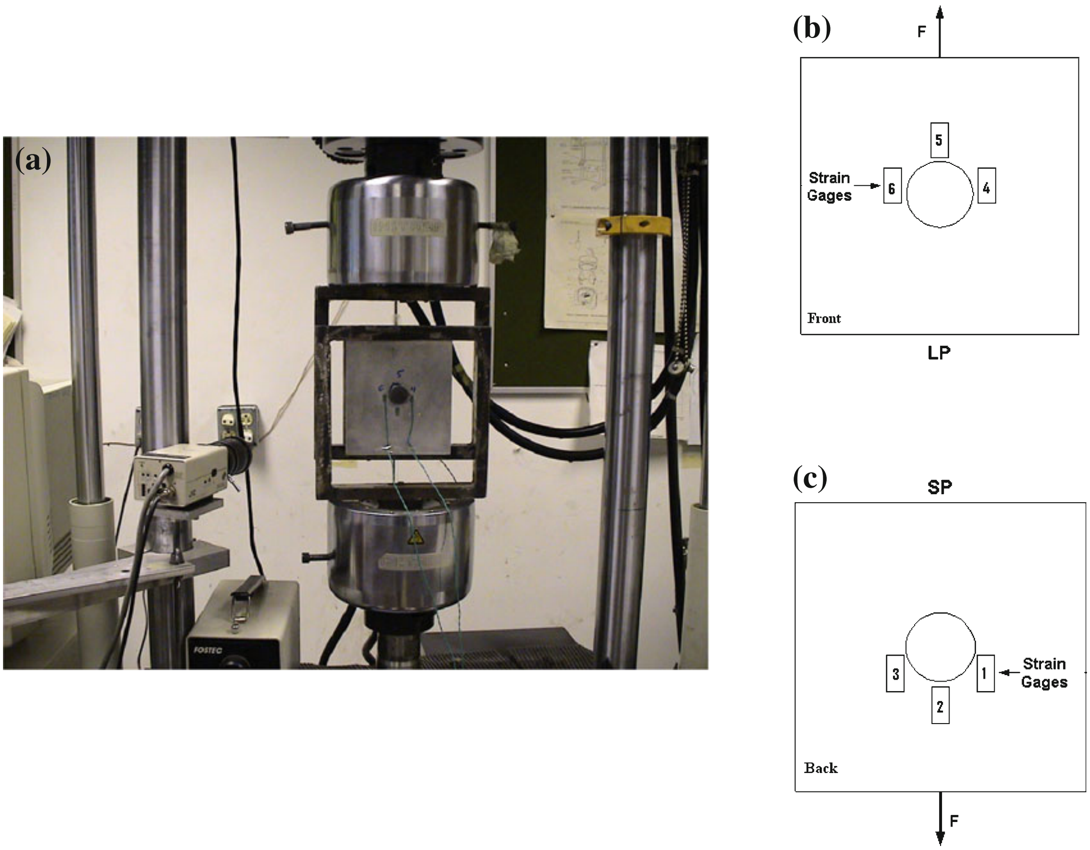

Fig. 3 **a** Experimental set up. **b** Locations and numbering of strain gages. **c** Locations and numbering of strain gages

The present problem involves the relative movement of surfaces when load is applied. Therefore using Coulomb friction, the contact condition is modeled between the interface of two plates, the nut and loading plate, the bolt head and supporting plate, and the bolt shank and the inner hole surfaces of the two plates as shown in Fig. 1. The contact type is provided by the ANSYS surface-to-surface (flexible to flexible) contact through the use of *Targe170* and *Contac174* elements. The interface between the bolt and nut threads is fixed.

Load is imposed on the model through prescribed displacement of the top edge of the loading plate. The lower bottom face of the supporting plate is constrained in the Y direction as illustrated in Fig. 1.

3 Experimental Work

Experimental data was used to validate the numerical model. A 250-kN Instron universal testing machine, fitted with a linear voltage distance transducer (LVDT) extensometer and connected to a computer through a data logger, is used in the tension mode to perform and record the experiments. Figure 3a shows the experimental setup. The steel fixtures are designed in such a way that the upper portion of each connects with either the top or bottom jaw of the machine as appropriate. Each fixture applies a uniform pull on one surface of the plate thus producing a shearing effect. The two plates are made up of aluminum and their nominal dimensions are 140 mm $\times$ 150 mm. An M16$\times$2 nut and bolt is used to clamp the two plates. Three strain gages are placed on each plate. The locations of these strain gages are such that they are positioned around the bolt as closely as possible and towards the loading edge. Figure 3b, c show these locations. Values of strain are recorded from each strain gage using a strain gauge reader.

4 Validation of Numerical Model

To validate the numerical model, the physical set-up was analyzed under the same loading conditions and material properties as the numerical model. Material properties of aluminum (E = 69 MPa, $\nu = 0.3$ and $\mu = 1.3$)

Table 2 Comparison of strains values from the loading and supporting plates for a displacement of 0.07 mm

Strain location	Experimental $\times 10^{-6}$	Numerical $\times 10^{-6}$
Back		
1	45	46
2	65	64
3	22	25
Front		
4	59	51
5	81	83
6	42	41

Table 3 Parameters used in the FEA studies

	Study 1	Study 2
Prescribed displacement (mm)	Constant 0.06	Constant 0.06
Pre-tension (kN)	Variable 0.5, 9.0 and 30	Constant 0.5
Radial clearance (mm)	Constant 0.05	Variable 0.01, 0.05 and 0.1
Plate material	Steel[a]	Steel[a]
Bolt material	Steel[a]	Steel[a]
Coefficient of friction	0.1	0.1

[a] Modulus of elasticity 210 GPa, Poisson's ratio 0.3, yield strength 200 MPa

for the plates and steel (E = 210 GMPa, $\nu = 0.3$ and $\mu = 0.7$) for the bolts are used in the finite element model. Table 2 shows the comparison between the strain values obtained from experimental work and numerical analysis for the six strain gage locations under prescribed loading displacement of 0.07 mm. The values of strain obtained experimentally and numerically are very close. The same trend was observed over a range of displacement loadings.

5 Results and Discussion

Clearly there are many variables that affect the behavior of a bolted joint structure, such as the applied load, pre-tension, and clearance between the bolt and the hole. Table 3 summarizes the two studies that have been carried out on one-bolted shear joint using three-dimensional FEA.

5.1 Effect of Pre-Tension

The purpose of the nut and bolt is to clamp two or more parts together. As a nut is rotated on a bolt thread against a joint, the bolt is stretched. The internal forces within the bolt resist this extension, and a tensile force is generated in the bolt. The reaction to this force is a clamp force that compresses the joint. This clamping force is called the pre-tension or bolt preload. In this study pre-tension force is increased and the clearance (0.05 mm) and the prescribed displacement (0.06 mm) remain constant. Three pre-tension values of 500, 9,000 and 30,000 N are considered. Figure 4 shows the displacement pattern in the Y direction on interface side of supporting plate at a pre-tension (PT) of 30,000 N. By comparing the pattern and the magnitudes with the other obtained using a (PT) of 9,000 N, it observed that the pattern remains the same but there is significant change in the displacement values.

The stress σ_y distribution on the interface side of the loading plate at a pre-tension of 30,000 N is shown in Fig. 5. Examining the region above the bolt hole for different pre-tension values, it is observed that this region is in tension and this tension increases with pre-tension. Figure 6 shows the contour plot of stresses σ_z for pre-tension values of 500, 9,000 and 30,000 N on the interface side of the loading plate. The maximum value of the stress around the bolt hole is written in the center of each hole. The figure shows that with increasing pre-tension the plate is more compressed and the maximum stress value increases. Table 4 summarizes the maximum displacement and stress values for the three levels of pre-tension.

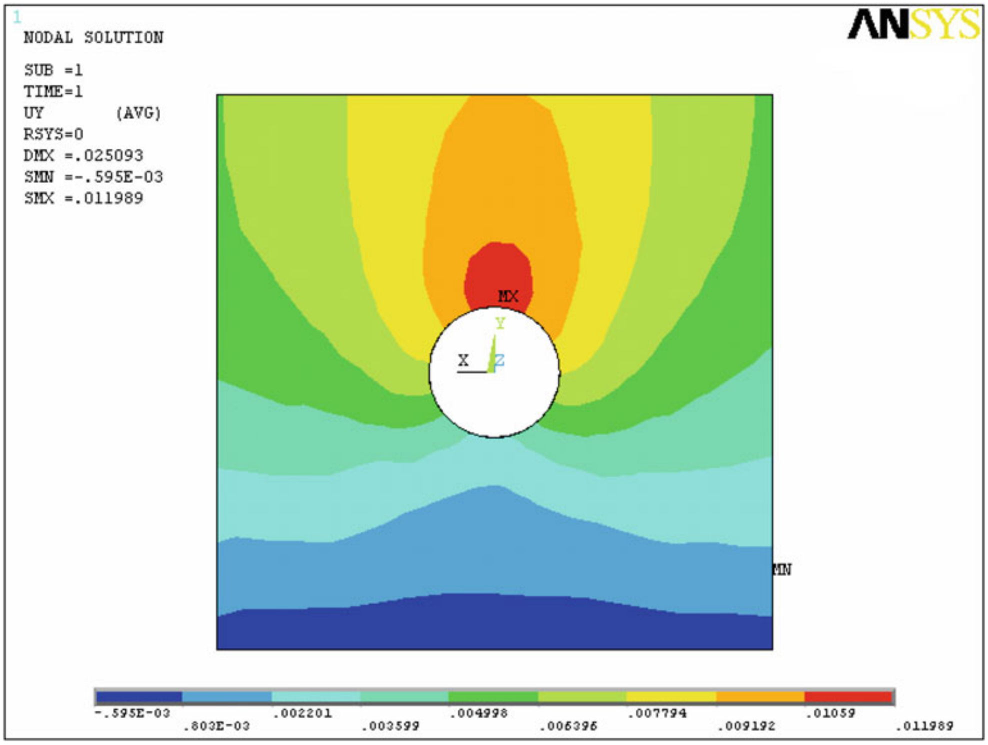

Fig. 4 Distribution of UY on the interface side of the supporting plate at PT = 30,000 N

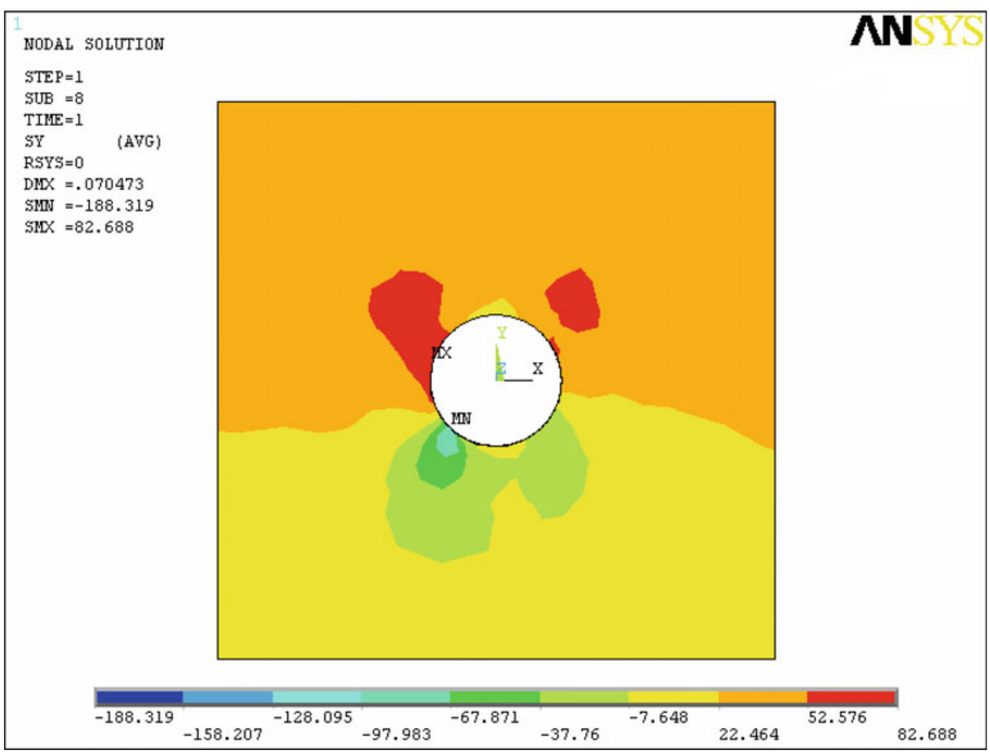

Fig. 5 Stress σ_y on the interface side of loading plate at PT = 30,000 N

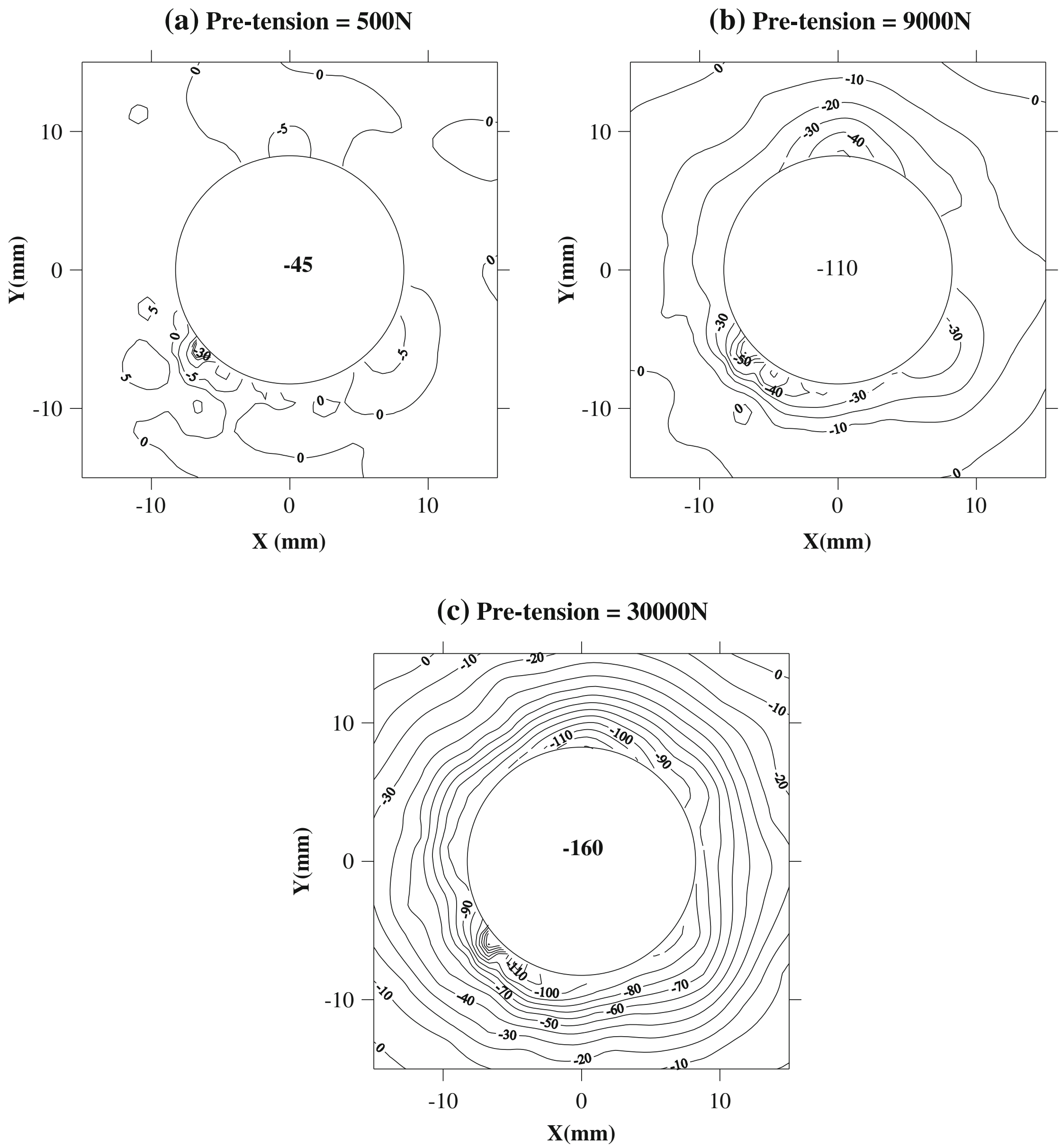

Fig. 6 Stress σ_z contours on the interface side of the loading plate at pre-tensions of **a** 500 N, **b** 9,000 N and **c** 30,000 N

Table 4 Maximum values of displacement and stress for shear joint with increasing pre-tension (PT)

Pre-tension (N)	500 N	9,000 N	30,000 N
Max. y-displacement (UY)			
Bolt (mm)	0.11804	0.045418	0.05405
Nut (mm)	0.099302	0.041542	0.049001
Supporting plate (mm)	0.010855	0.011098	0.01198
Loading plate (mm)	0.06026	0.060978	0.061134
Max. stress σ_z			
Bolt (MPa)	58	225	466
Supporting plate (MPa)	−51	−92	−231
Loading plate (MPa)	−69	−145	−214

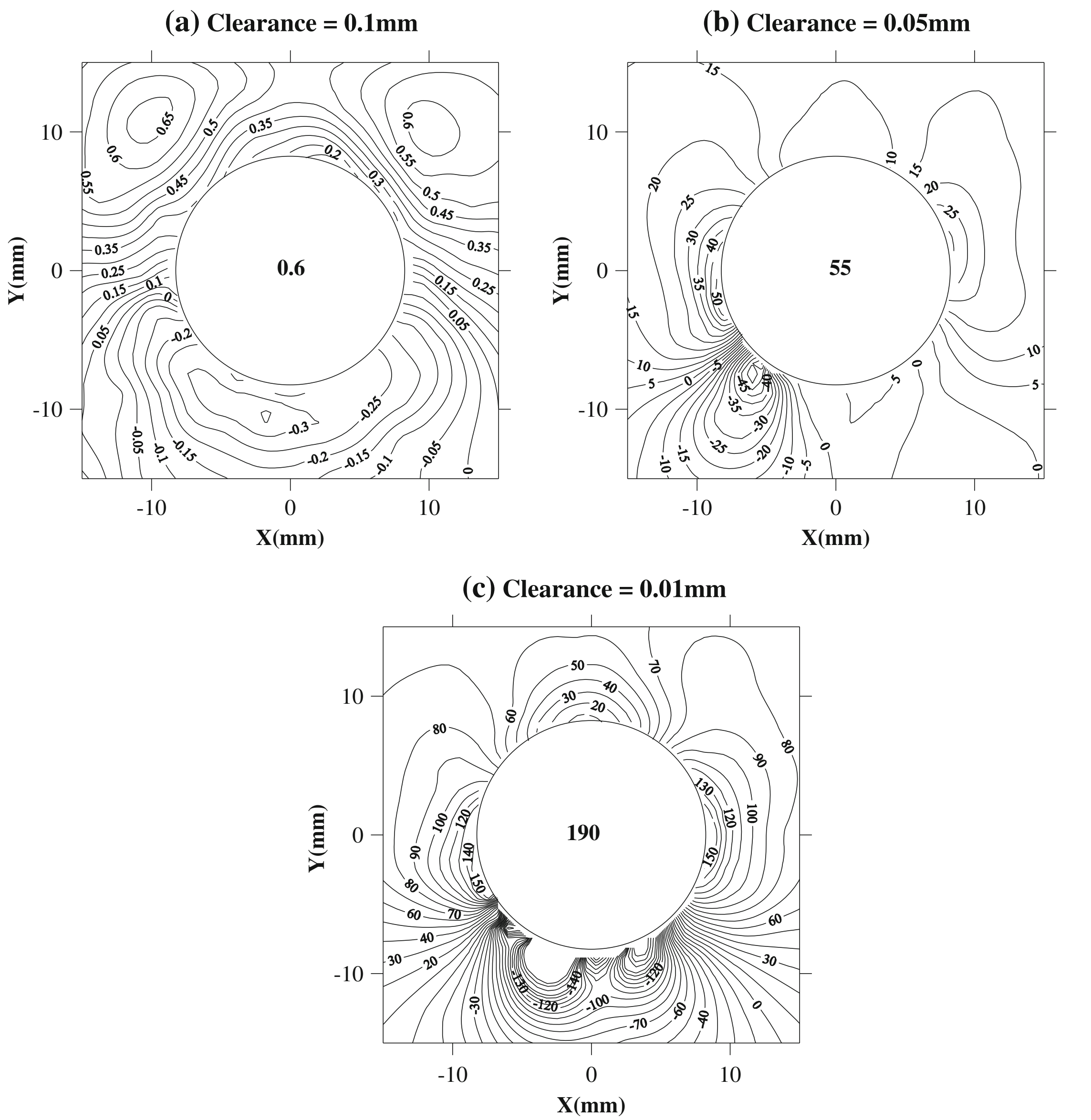

Fig. 7 Contours of stress component σ_y on the interface side of loading plate at radial clearances of **a** 0.1 mm, **b** 0.05 mm and **c** 0.01 mm

5.2 Effect of Radial Clearance

The clearance between the bolt and the hole is an important factor in a bolted joint analysis. Strength and durability of a bolted joint is partially dependent on the closeness of fit between the bolt and hole. The hole should not be so small that the bolt has to be forced in. Equally, it should not be so large that the resulting joint is loose. A loose joint will allow relative motion (wear), moisture ingress (corrosion), work hardening of the bearing surface (leading to cracking), uneven load transmission and eventual cracking under the head of the bolt or thinning of a soft material. The analysis here has examined three clearances −0.01, 0.05 and 0.5 mm. Other parameters such as pre-tension (500 N) and the displacement load (0.06 mm) are kept constant.

Figure 7 shows the contour plot of stress σ_y on the interface side of the loading plate at radial clearance values of 0.01, 0.05 and 0.1 mm. Since the radial clearance (0.1 mm) is greater than the prescribed displacement (0.06 mm), there is no contribution of the stresses on the plate coming from the bolt—as shown in Fig. 7a. Figure 7b shows that as the clearance is decreased to 0.05 mm there is contact of bolt with the hole and the maximum value of stress is increased. In Fig. 7c, the maximum stress around the bolt hole increases, and an

Table 5 Maximum values of displacement and stress for shear joint with increasing radial clearance

Clearance (mm)	0.01 (mm)	0.05 (mm)	0.1 (mm)
Max. y-displacement (UY)			
Bolt (mm)	0.119516	0.11804	0.042238
Nut (mm)	0.102175	0.099302	0.04282
Supporting plate (mm)	0.024466	0.010855	0.00102
Loading plate (mm)	0.063798	0.060268	0.060015
Max. von Mises stress			
Bolt (MPa)	434.865	279.826	5.752
Supporting plate (MPa)	755.038	211.25	3.281
Loading plate (MPa)	837.732	160.467	2.743

increase in stress concentration around the bolt hole is seen over previous cases. Table 5 shows the maximum displacement and stress values at the three radial clearances used. The maximum displacement and stress value decreases with increase in radial clearance.

5.3 Pre-Tension Effect Under Tensile Loading

Preloads in bolts and other structural components often have significant effect on deflection and stress. The ANSYS program provides for the application of pre tension force by use of the element called *PRETS 179*. This element is used to represent a two- or three-dimensional section for a bolted structure using the constraint equations approach. The main assumption and restrictions of this element is that it is not capable of carrying bending or torsion load.

The three-dimensional FE model of the one bolt joint is analyzed to study the effect of pre-tension under tensile loading. The element type and material properties are kept the same. Pressure is applied on the nut side of the loading plate. The three sides of the supporting plate are constrained. Due to symmetry, a half model can be used and symmetric boundary conditions are applied at the lower surfaces in the vertical direction. Figure 8 shows the applied pressure loading and boundary conditions used. Additional contacts between the thread of the bolt and the mating surfaces on the nut are defined. The number of nodes and elements in this half model are reduced to 3,462 and 10,765, respectively. Three different pre-tension values of 2,500, 5,000 and 30,000 N are used. The clearance between the hole and the bolt is taken to be 0.05 mm.

Figure 9 gives the displacement pattern in the axial direction of the bolt at a pre-tension force of 30000 N. Figure 10 shows the stress σ_z distribution of the supporting plate at the same pre-tension force. Due to the pre-tension, highly compressive stresses can be seen around the bolt hole, with the bolt head side being more stressed around the bolt hole. Figure 11 shows the stress σ_z behavior of the bolt when a tension type loading is applied. The two regions (marked by arrows) of highest stress are the regions just below the head of bolt and around the first engaged thread.

Figure 12 shows the graph of stress component σ_z distribution at the root of the engaged threads 1, 2, 3 and 4 at three pre-tension values. The graph shows that the maximum stress is present in the first engaged thread i.e. 1 and it drops significantly in the subsequent threads 2, 3 and 4. In case of pre-tension 30,000 N, the first thread is taking 47 % of the total load. This load share is decreased to 24, 16 and 13% in thread number 2, 3 and 4 respectively. For pre-tension values of 5,000 and 2,500 N the first thread is taking 54 and 49% respectively. Figure 13 shows the variation in stress in thread 1 from root to the tip at three pre-tension values. Table 6 shows that the values of maximum stress in the joint parts and that they have localized plastic deformation at a pre-tension value of 30,000 N.

Calculating the stiffness of the connected members is difficult because the compression spreads out between bolt head and the nut, and hence the area is not uniform. There are, however, some analytical methods that predict an approximate stiffness for the member. In theory, the compression of a member is represented by a frustum of hollow cone. Mischke [1] uses half apex angle $\alpha = 30°$ to calculate the stiffness using formula given below,

$$k = \frac{\pi E d \tan\alpha}{\ln \frac{(1.155t+D-d)(D+d)}{(1.155t+D+d)(D-d)}}$$

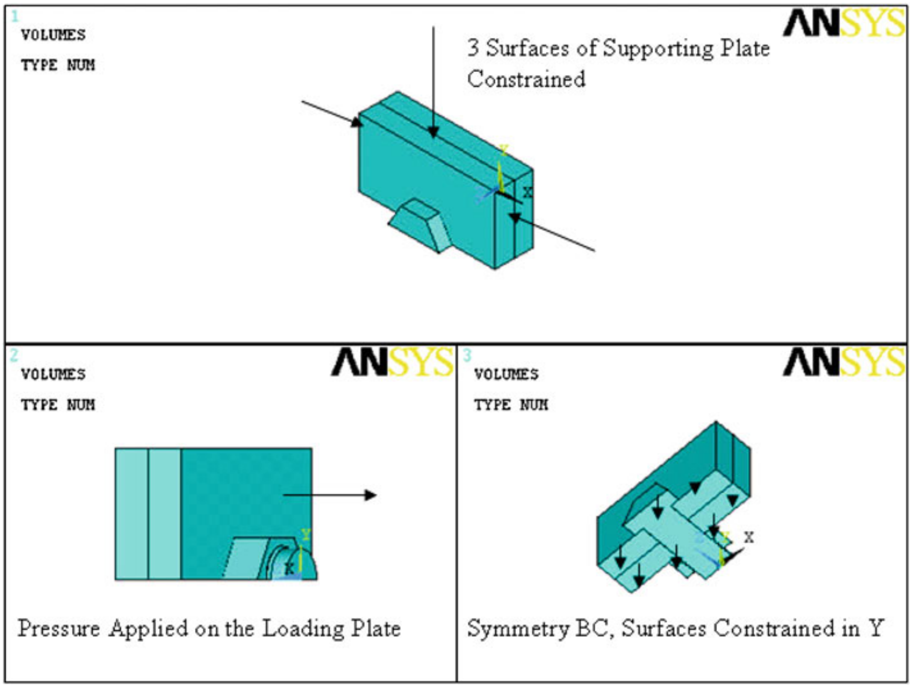

Fig. 8 Applied loading and boundary conditions for a one bolt joint under tension

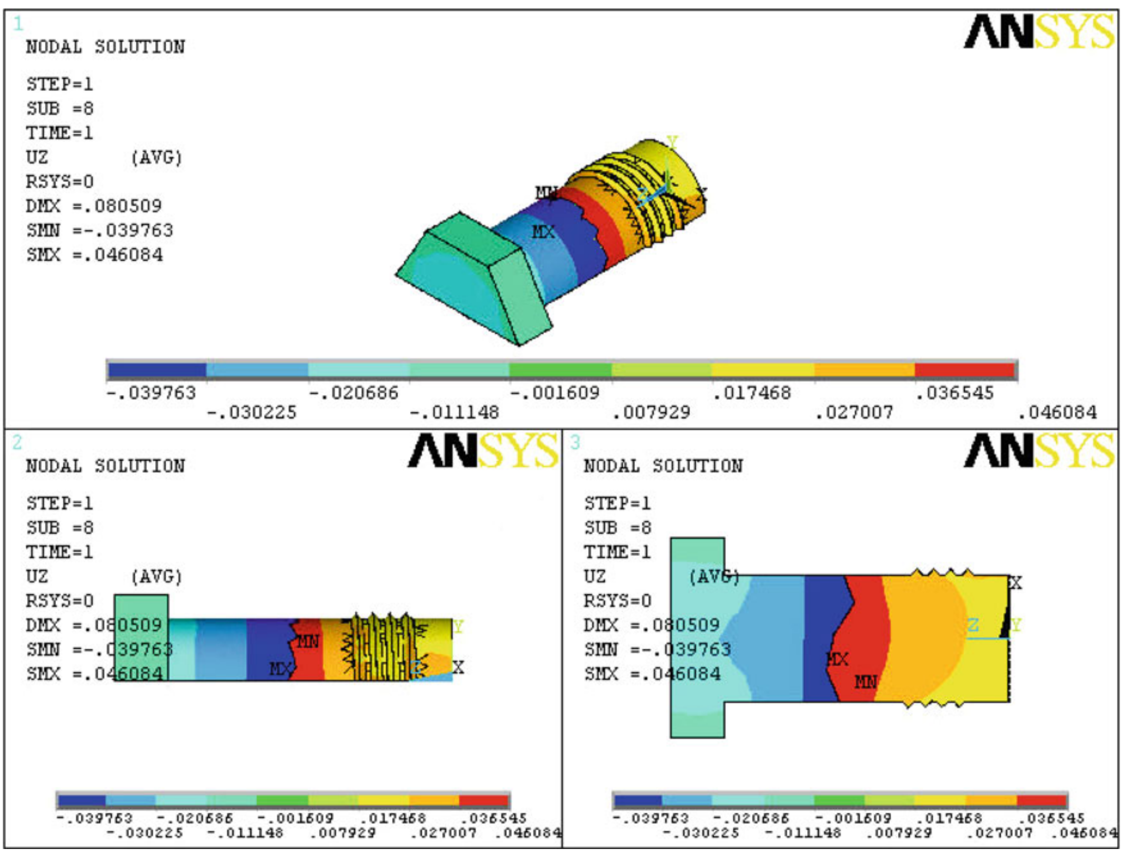

Fig. 9 Displacement in the axial direction of the bolt (UZ) at PT = 30,000 N

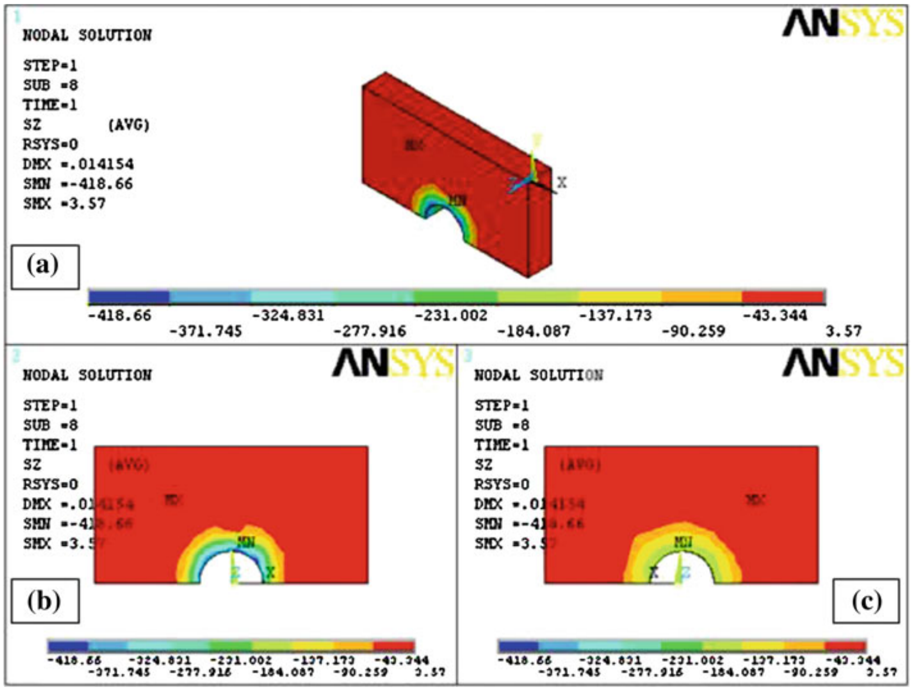

Fig. 10 Distribution of stress component (σ_z) in the supporting plate at 30,000 N. **a** Isometric view, **b** bolt side and **c** interface side

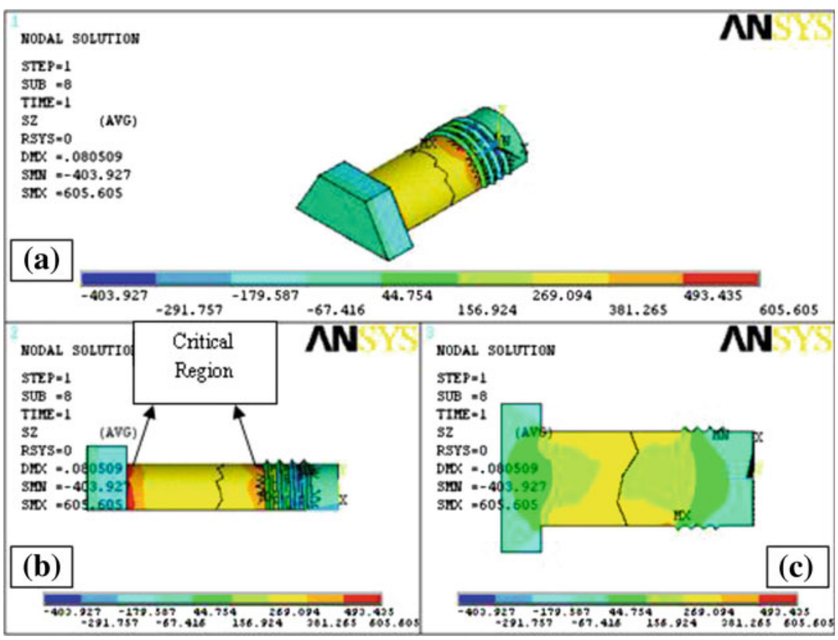

Fig. 11 Distribution of stress component (σ_z) in the bolt at PT = 30,000 N

Wileman et al. [18] conducted a finite element study offering a formula for the ready calculation of the stiffness of the member in this form

$$\frac{k}{Ed} = A \exp\left(Bd/l\right)$$

These formulae contain geometrical parameters such as length l, diameter of washer D, diameter of the bolt d, thickness of the frusta t and angle α, and the modulus of elasticity E.

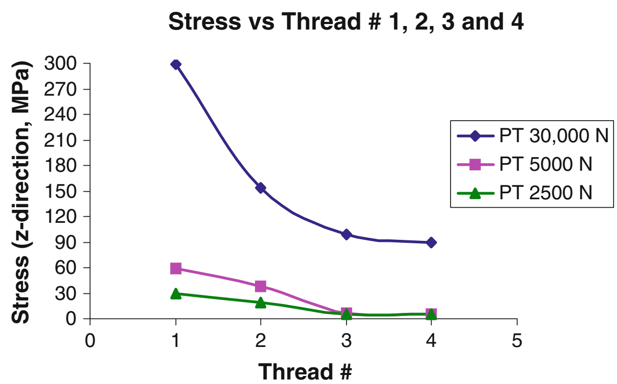

Fig. 12 Stress variation at the root along the threads 1, 2, 3 and 4

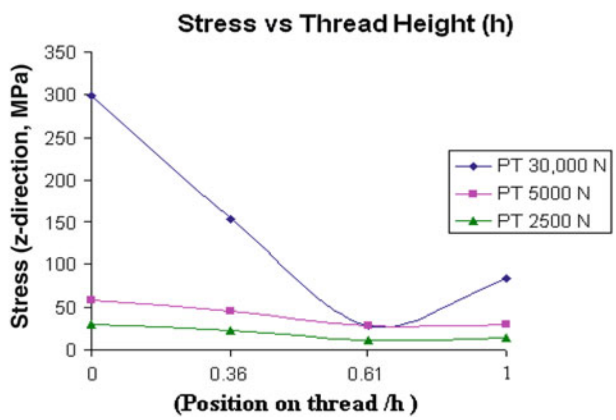

Fig. 13 Stress variation in thread 1 from root to tip

Table 6 Maximum values of stress σ_z for one-bolted joint under tensile loading with increasing pre-tension (PT)

Pre-tension (N)	2,500 N	5,000 N	30,000 N
Bolt (MPa)	80.827	178.85	605.605
Supporting plate (MPa)	−34.351	−69.451	−418.66
Loading plate (MPa)	−22.154	−42.008	−233.499

Figure 14 shows the stress σ_z distribution on the loading plate around the bolt hole at 2,500, 5,000 and 30,000 N in the XZ plane. A cone angle of 30° is used and by inspecting the frusta at angle $\alpha = 30°$ it can be concluded that:

- Stress concentration region outside the frusta is significant in all cases. The concentration increases with pre-tension force.
- The spread of stress concentration (marked by arrows) increases with pre-tension force. The stress lines exceed the frusta when a pre-tension of 30,000 N is used.
- To accurately predict the value of the stiffness of the connected members, the pre-tension force must be incorporated in the above formulae.

6 Conclusions

A full three-dimensional model of a bolted joint is developed. This includes the thread geometry, contact between the mating surfaces and modeling of the plates, bolts and nuts as solid elements. The finite element method is used to analyze the bolted joint in shear and tension loading. From the numerical results, the following conclusions are drawn:

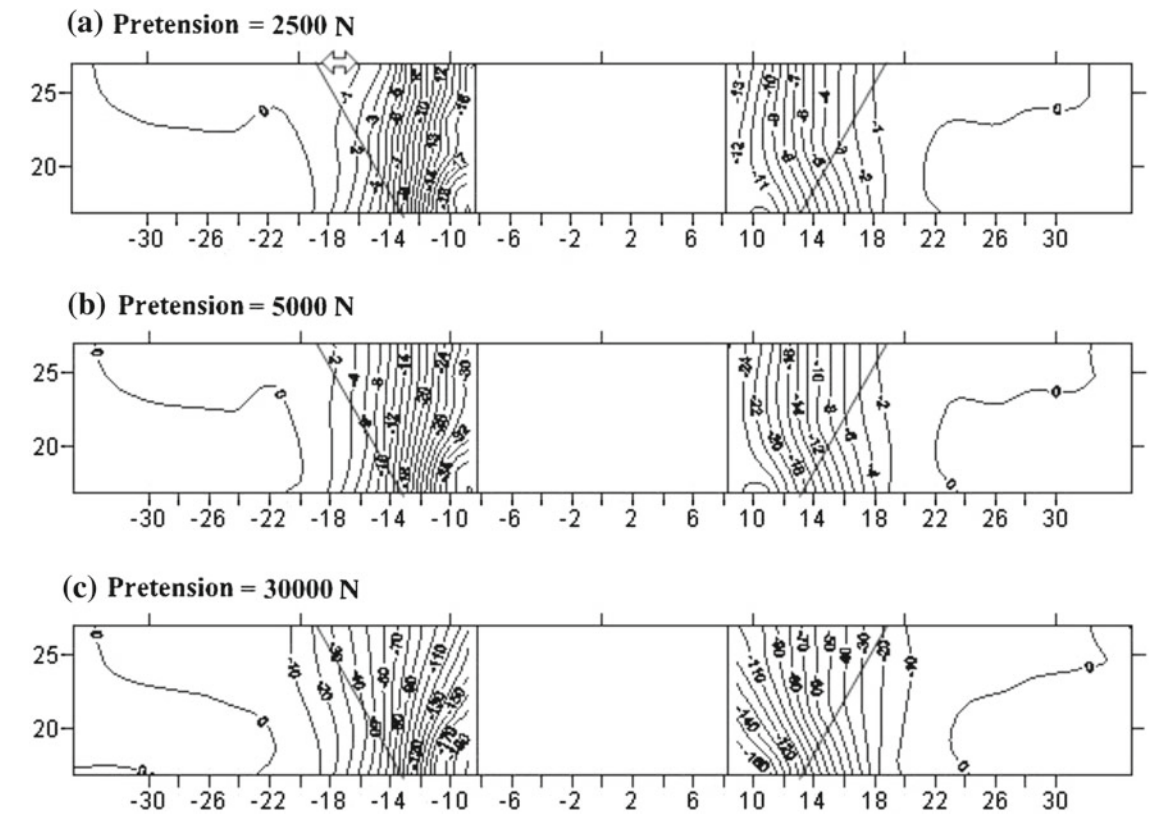

Fig. 14 Stress distribution on loading plate in the xz-plane at pre-tensions of **a** 2,500 N, **b** 5,000 N and **c** 30,000 N

1. Pre-tension and clearance affect the displacement pattern and stress distribution of bolted joint in shear loading.
2. In shear loading, the maximum displacement and stress value decreases as the clearance is increased.
3. The value of maximum compressive stress increases with pre-tension. This is valid for both tension and shear type of loading.
4. Pre-tension has dominant effect on the stress in z direction when the joint is loaded in tension.
5. The critical regions in the bolt are the regions just below the head of the bolt and around the first engaged thread.
6. The threads have different load shares when loaded in tension. The first thread bears the highest load.
7. The numerical results are validated by experimental measurements.

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References

1. Shigley, J.E.; Mischke, C.R.: Mechanical Engineering Design, 5th edn. McGraw Hill, New York (1989)
2. Ito, Y.; Toyoda, J.; Nagata, S.: Interface pressure distribution in a bolt-flange assembly. ASME Paper no. 77-WA/DE-11 (1977)
3. Motosh, N.: Determination of joint stiffness in bolted connections. ASME J. Eng. Ind. **198**(3), 858–861 (1976)
4. Al Jefri, A.M.; Abdel Latif, A.K.: Characteristic of bolted joint under elastic loading conditions. Eng. Syst. Design Anal. Conf. ASME **PD-80**, 173–185 (1996)
5. Chen, J.J.; Shih, Y.S.: A study of the helical effects on the thread connection by three dimensional finite element analysis. Nucl. Eng. Design, **191**, 109–116 (1999)
6. Yamamoto, A.: The Theory and Computation of Thread Connection, pp. 39–54. Youkendo, Tokyo (1980, in Japanese)
7. Lehnhoff, T.F.; Bunyard, B.A.: Effects of bolt threads on stiffness of bolted joints. ASME J. Press. Vessel Technol. **123**, 161–165 (2001)

8. Tanaka, M.; Aoike, T.: Behavior of bolted joints tightened in plastic region. Anal. Bolted Jt. ASME, **PVP- 367**, 89–95 (1998)
9. Fukuoka, T.; Takaki, T.: Mechanical behavior of bolted join in various clamping configurations. J. Struct. Eng. **120**, 226–231 (1998)
10. Menzemer, C.C.; Fei, L.; Srivatsan, T.S.: Failure of bolted connections in an aluminium alloy. J. Mater. Eng. perform. **8**, 197–204 (1999)
11. Tan, D.; Smith, I.: Failure in the row model for bolted timber connections. J. Struct. Eng. **125**, 713–718 (1999)
12. Fukuoka, T.; Takaki, T.: Elastic plastic finite element analysis of bolted joint during tightening process. J. Mech. Design **125**(4), 823–853 (2003)
13. Paroissien, E.; Sartor, M.: Analytical two-dimensional model of a hybrid (bolted/bonded) single-lap joint. J. Aircraft **44**(2), 573–655 (2007)
14. Fukuoka, T.; Nomura, M.: True cross sectional area of screw threads with helix root radius geometries taken into consideration. J. Press. Vessel Technol. **131**(2), 024501 (2009)
15. Cornwell, R.E.: Computation of load factors in bolted connections, proceeding of the institution of mechanical engineers. Part C. J. Mech. Eng. Sci. **223**(C4), 795–808 (2009)
16. Khurshid, H.: Effect of bolt layout on the mechanical behavior of bolted joint. MS Thesis, KFUPM (2004)
17. ANSYS User's Reference Manuals, Release 7.0, SAS IP. Inc., Cheyenne (2000)
18. Wileman, J.; Choudhury, M.; Green, I.: Computation of member stiffness in bolted connections. J. Mech. Des. **113**(4), 432–437 (1991)