3-D Finite Element Analysis of Bolted Joint Using Helical Thread Model

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Abstract

Conventional analytical and numerical methods to study the response of a bolted joint is relied on many assumptions and approximations and thus hardly yield satisfied results. 3D finite element model of bolted joints with real helical thread geometry is established and meshed with refined hexahedral elements. This finite element model is used to analyse specific characteristics of stress distribution, stress concentrations, contact pressure distributions, load sharing of the bolted joint components. Sector model of bolted flange joint has been analysed for pretension alone and combination of pretension and axial load. Using the finite element (FE) model with accurate thread geometry with pretension, the thread root stresses, contact pressure along the bolt thread helix, between flanges, on bolt head and at the nut loaded surface have been studied. The peak stress occurs at the first engaged bolt thread root from nut loaded surface and gradually decreases towards the free face of the nut.

Results from analysis with pretension and axial load indicate that the contact separation starts at the inner radius of flange and grows towards outer diameter of flange as the axial load is increased in the bolted flange joint. It is observed from the analyses that most of the load is shared by flanges when the external applied axial load is up to 30% of preload, and beyond this, bolt starts sharing more external load. The maximum stress occurs at the first engaged bolt thread root. Most of the bolt failures are at the first engaged thread.The study suggests that it is necessary to consider threads in FE model to obtain accurate stiffness, bolt load and thread stress predictions. These critical observations provide insight for optimization of bolted flange joint to meet the structural requirements and weight optimisation.

Keywords: Bolted joint, Preload, Joint stiffness, Bolt force , Contact pressure

Introduction

Bolted joints are the most widely used machine elements because they can repeatedly be assembled and disassembled by an easy operation. Structures in aerospace, energy and industrial applications are often connected by bolted flange joints. A basic function of bolted joint is to provide adequate clamping force, joint strength, stiffness and sealing to minimize leakages. Fatigue is one of the critical failure modes in the bolted joint. Bolted joint with preloading minimizes the fatigue damage as the preload reduces the fluctuating stress in the bolt induced by variable external loads. Accurate calculation of stresses, contact pressure , bolt force in a bolted joint under external loads is a fundamental requirement in many industries. Mechanical behaviour parameters of the bolted joints, such as the strength and the stiffness, have been analysed experimentally, theoretically based on elasticity theory, and using numerical method. Finite element method (FEM) is found to be the most powerful numerical method for solving the problems of bolted joints. The development of FEM has made it possible to evaluate joint stiffness, contact pressure distribution, bolt force and the stress concentration at the thread root with high accuracy in bolted joints under external loads.

2.Literature review

Threaded fasteners are the most widely used machine elements because they can repeatedly be assembled and disassembled by an easy operation. Mechanical behaviors of the threaded fasteners, such as the strength and the stiffness of bolted joints, have been analyzed by experiment, theoretical analysis based on elastic theory, and numerical method. Finite element method (FEM) is found to be the most powerful numerical method because of development of numerical techniques for solving the problems of bolted joints.

Nonthreaded FE models [1–2] and 2D axisymmetric thread models [3] have been employed to simulate the mechanical behavior of threaded fasteners (bolt and nut). The nonthreaded models ignore the influence of screw threads on the load transfer in thread connections. The 2D axisymmetric thread models can consider the load transfer and stress concentration in screw threads, but they ignore the helical effect of threads. So it is necessary to build a more effective and accurate model in the case of detailed design. IZUMI, et al [4] investigated the tightening and loosening mechanism of threaded fastener using a 3D FE model with tetrahedral elements, but their model is too rough to accurately obtain the stress distribution in threads. FUKUOKA, et al [5] constructed a 3D FE model with hexahedral elements, which provides an approach for modeling of the helical thread effect of thread connection.

The objective of this paper is to propose a generally accepted procedure for building a 3D FE model of bolted joint with helical threads, and then to investigate the corresponding mechanical properties. 3D FE model of bolted joint is constructed, and it is meshed with high quality hexahedral elements. The mechanical properties of the helical thread connection are analyzed in detail, including contact pressure distribution at the joint interfaces, stress distribution, stress concentration and load sharing behavior.

3. Problem description

Distinctive mechanical behaviour of bolted joints is caused by the helical shape of thread geometry. Recently, a number of papers have been published to elucidate the strength or loosening phenomena of bolted joints using three-dimensional finite element analysis. In most cases, mesh generations of the bolted joints are implemented with the help of commercial software. The mesh patterns so obtained are, therefore, not necessarily adequate for analyzing the stress concentration and contact pressure distributions, which are the primary concerns when designing bolted joints. In this work, an effective mesh generation scheme is considered, which can provide helical thread models with accurate geometry to analyze specific characteristics of stress concentrations, contact pressure distributions and load distribution caused by the helical thread geometry. A steel bolt-nut set joining two flanged shells is analysed for pretension and operating axial load to study joint behaviour.

4. Finite element modeling and analysis

4.1Modelling

The design considered here is a steel bolt-nut set joining two flanged shells in a circular pattern. Because of symmetry sector model is considered for modelling and analysis. Since 90 bolts are used to connect two shell structures, 4° sector model is considered for study. 3/8-inch steel bolt (UNF threads) is used to clamp two 0.5-in-thick Aluminium and Titanium flanges. The length of the shell is considered such that, it will not have any effect on joint behavior. The dimensions of bolted flange joint is shown below fig.1.

Dimensions : Thickness of flange and shell (t_f)= 0.5inch, Flange outer radius = 16.7inch, Inner radius of shell = 15 inch, bolt head diameter = 0.75 inch, shell length = 4 inch, Bolt specification : 3/8-inch bolt , 24 threads per inch.

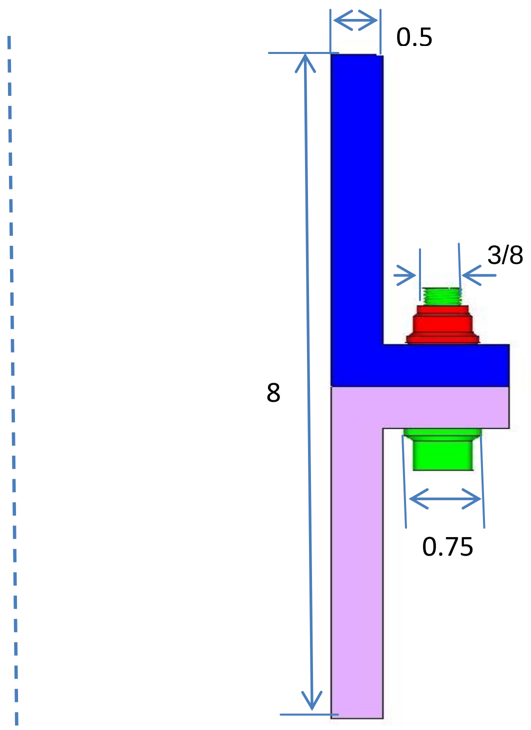


Fig.1. Geometry of bolted flange joint

4.2Meshing

FE modelling of the entire joint is usually neither practical nor necessary, and would detract from the basic insights. It is sufficient to model a single fastener location, with boundary conditions carefully chosen to replicate the constraint existing in the actual joint, and to load the fastener with a force representing the local per-fastener force. This section describes the meshing of model, type of meshing used and reasons for the same. When analysing the mechanical behaviour of bolted joints with three- dimensional analysis, it has been a common practice that the threaded portion of the FE models has axisymmetric geometry, where the effects of lead angle are neglected because of its small value. That is, external and internal threads are modelled by stacking an appropriate number of threads with axisymmetric geometry. Recently, some researchers start to use helical thread models because of a growing recognition of the importance of helical effects, e.g., loosening phenomena of bolted joints. The present approach used to create helical threads is more accurate and procedure is given below. The finite element software ANSYS is used for meshing and analysis

4.2.1Mathematical expressions of thread cross sectionprofile

The cross sectional profile along the bolt axis including the thread root radius is shown below fig.2. Assuming that the rounded portion of the thread root is a part of a single circle with diameter ρ , the surface of external thread can be divided into three parts such as A-B (root radius), B-C (thread flank), and C-D (crest). The thread profile perpendicular to the bolt axis can be obtained by expanding those three parts into the plane, as shown in Fig.3. Its shape is naturally identical at any cross section along the bolt axis. The helical thread models are constructed by utilizing the characteristics explained here. The given below equations are used to generate external(1) and internal (2)thread profiles respectively.

$$r=\begin{cases} \frac{d}{2}-\frac{7}{8}H+2\rho-\sqrt{\rho^2-\frac{P^2}{4\pi^2}\theta^2} & (0\leqslant\theta\leqslant\theta_1) \\ \frac{H}{\pi}\theta+\frac{d}{2}-\frac{7}{8}H & (\theta_1\leqslant\theta\leqslant\theta_2) \\ \frac{d}{2} & (\theta_2\leqslant\theta\leqslant\pi) \end{cases} \quad (1)$$

θ=√3π⁄p θ=7/8π p≤√3/12p H=√3/2p

Where d and h are nominal diameter and thread overlap.

$$r=\begin{cases} \frac{d_1}{2} & (0\leqslant\theta\leqslant\theta_1) \\ \frac{H}{\pi}\theta+\frac{d}{2}-\frac{7}{8}H & (\theta_1\leqslant\theta\leqslant\theta_2) \\ \frac{d}{2}+\frac{H}{8}-2\rho_n+\sqrt{\rho_n^2-\frac{P^2}{4\pi^2}(\pi-\theta)^2} & (\theta_2\leqslant\theta\leqslant\pi) \end{cases} \quad (2)$$

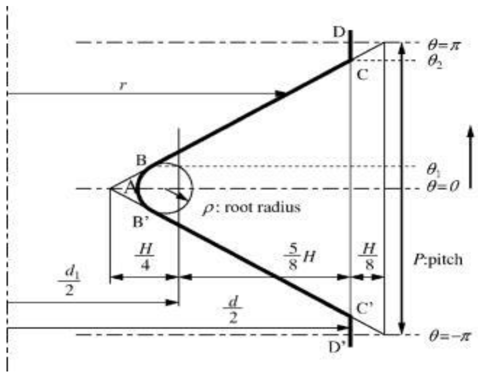


Fig .2. Thread cross section along bolt axis

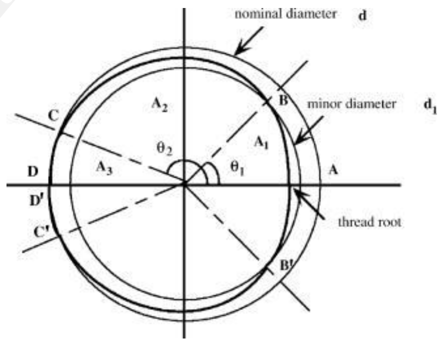
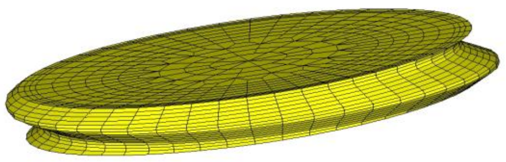


Fig .3. Profile of the cross section of external thread perpendicular to the bolt axis

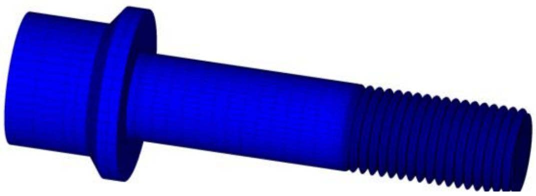
θ=√3π⁄p θ=7/8π p≤√3/12p H=√3/2p

4.2.2 Helical thread modeling

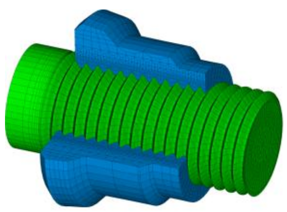
In order to mesh the helical thread model with high quality hexahedral elements, the threaded zones of the bolt and nut are meshed separately and joined to the respective remaining zones. The threaded zone is constructed using the same cross section shape as shown in fig.3. but the adjacent sections have an angle difference of 22.5°(360°/16) along the bolt axis. Each pitch of the external thread is divided into 16 layers of solid elements. Similarly, each pitch of the internal thread is divided into 16 layers of solid elements .The mesh of one pitch of the bolt thread, bolt and the nut are shown in Fig.4.The proposed geometry modeling of the bolt and the nut enable the model to be meshed mostly with 8node solid hexahedral elements so as to improve the solution efficiency and accuracy. The modeling method of threads can not only guarantee the discrete accuracy of FE elements, but also well control the element number.



One pitch external thread mesh



Bolt mesh



Nut and bolt threaded zone

Fig. 4. Finite element model of bolt and nut

4.2.3 Meshing of flanges with shell

Sector model of flange and shell portion has been modelled using 8 noded brick elements. Very fine mesh is maintained around bolt hole of the flange to capture results accurately. The elements across thickness of the flange and shell maintained as 6 and 4 to capture bending accurately.

4.2.4 Contact modelling :

Surface to surface contact elements are used between flanges, bolt head& flange, Nut &flange, bolt & nut threads . The coefficient of friction between mating components is 0.15. Pretension element is used for applying preload.

The Finite element model of bolted joint is shown below fig.5.

No.of elements : 230885
No.of nodes : 227870

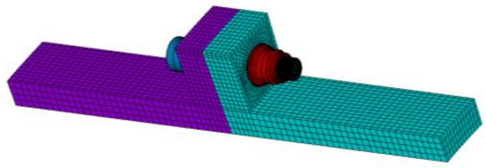


Fig.5. Finite element model of bolted flange joint

4.3 Material properties

The two casings (shells) are assumed to be made of Aluminium and Titanium .The bolt is made of high-strength, SAE Grade 8, 120 ksi proof bolt, with yield at 130 ksi. The ultimate strength is 150 ksi at 12% total elongation. The material properties are shown in below table1.

Table1:Material properties of the joint parts

S.NO	Part name	Material	Young's modulus E (psi)	Poison's ratio v
1	flange1	Aluminium	10.0×10^6	0.33
2	flange2	Titanium	16.9×10^6	0.31
3	Bolt & nut	High-strength steel	30.0×10^6	0.30

4.4 Boundary conditions

One end of the shell is constrained in all directions. The other end is subjected to axial load of 7000 lbf. The sector face nodes have been constrained in a tangential direction. The pretension value of 7000 lbf is applied at the middle of the bolt using pretension element. The boundary conditions plot is shown below fig.6.

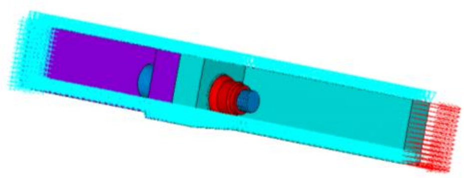


Fig.6. Boundary conditions plot

4.5Analysis

Nonlinear contact analysis has been carried out for two different load cases using ANSYSver13.0. In load case1, only pretension has been considered to study joint response. In loads case2, pretension and axial load of 7000 lbf are considered for analysis to study joint behavior. The total load is divided in to small increments and each increment is applied individually. Nonlinearity is treated as piecewise linear. Checks are made in the form of iterations to ensure that the equilibrium of forces maintained. Equilibrium is satisfied for each load increment to ensure that the solution is acceptable. This is done by estimating residual or out of balance forces within the structure and reducing to negligible value. The total deformation is calculated as the sum of the deformations of each increment. The deformation and stresses of the contacting bodies are not linearly dependent on the applied loads

5.Results &discussions

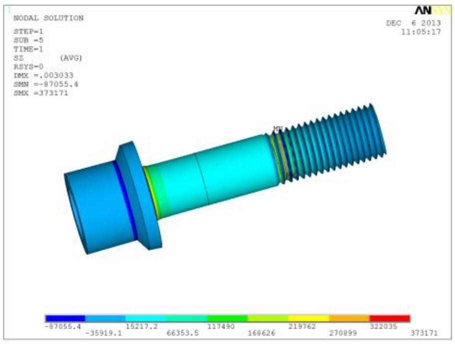
Nonlinear contact analysis has been carried out for two different load cases.

Case1 : When the joint is subjected to only preload of 7000lbf, bolt undergoes elongation and flanges are subjected to compression. Peak stresses are induced at threads of bolt , flange bolt holes .

5. 1 Stress distribution pattern

5.1.1Bolt stresses

The von Mises equivalent stress(σe), and the axial stress (σz) of the joint are reported The peak stresses are induced at thread root, on shank near to bolt head and are highly localised. These zones prone to local plastic deformation and stress concentration. The nominal axial stress on the shank is 63ksi and is exactly matching with hand calculation. The peak axial stress is 373 ksi and highly localised. This max. stress is induced at first engaged thread at nut loaded surface and gradually reduces towards the nut free surface. The peak von Mises stress is 304 ksi and is highly localised. This max. stress is induced at first engaged thread at nut loaded surface and gradually reduces towards the nut free surface. The streses induced in the bolt is tensile in nature. The stress contour plots are shown in fig.7 to 8. The variation of von Mises stress in radial direction is shown below Fig.9. The stress concentrations is distributed over very small length of around 0.010 inch at thread root.



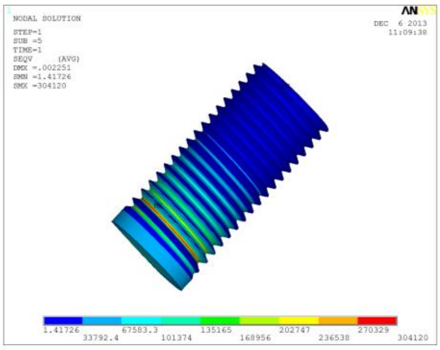


Fig.7.von Mises stress contour plot of bolt

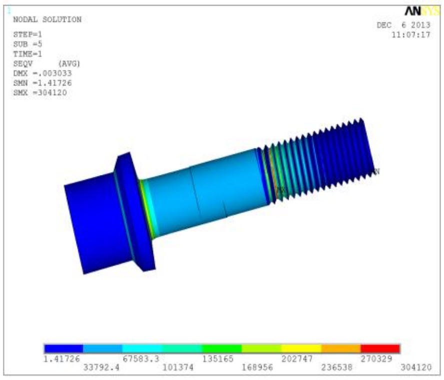


Fig. 8. Axial stress contour plot of bolt

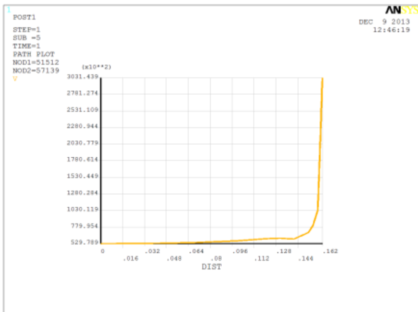


Fig.9.Variation of von Mises stress in radial direction at thread root

5.1.2 Flange stresses

The flanges are subjected to compression due to preload and compressive stresses are induced. The max. compressive stress induced at flange hole is 30.6 ksi . This compressive stress is useful to improve the life of the component at operating condition.The von Mises equivalent stress is 47 ksi This max. stress is induced in the bolt hole edge where it comes in to contact with bolt head. The compressive stress of the members indicates a hollow conical shape. The stress contour plots are shown in fig.10 to 11.

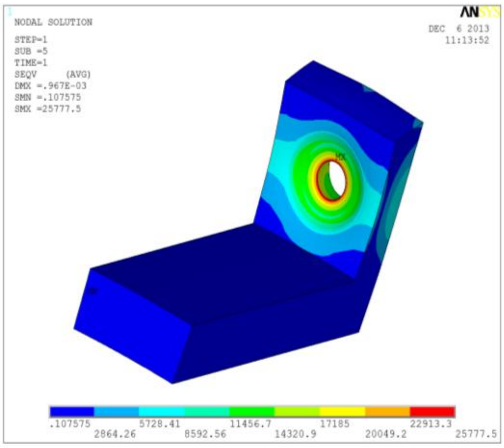


Fig.10. Von Mises stress contour plot

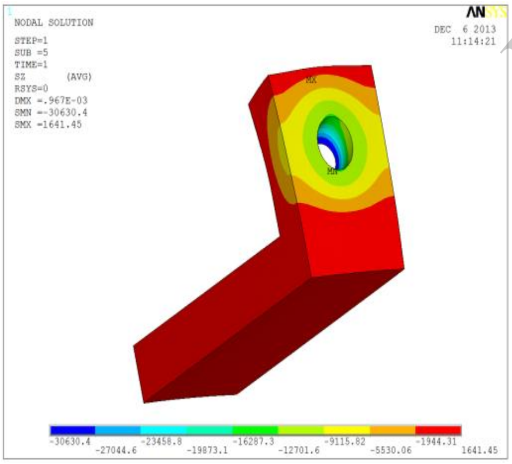


Fig.11. Axial compressive stress contour plot

5.1.3 Contact pressure distribution

The max.contact pressure at bolt head surface is 34 ksi and occurring at inner radius. The contact pressure in the bolt head bearing surfaces shows concave nonlinearity because of the stress concentration at the interface edges. The contact pressure distribution plot is shown in Fig.12. The max.contact pressure at nut loaded surface is 71 ksi . The contact pressure at the nut loaded surface varies in the circumferential direction due to the effect of the helical thread geometry. The contact pressure distribution plot is shown in Fig.13. The maximum contact pressure near the first engaged thread is 274.6 ksi ,and decreases with the distance from the nut bearing surface. The contact pressure distribution plot is shown fig .14.
The maximum contact pressure between flange contact surface is 12.6ksi and the distribution pattern is shown in figure15.

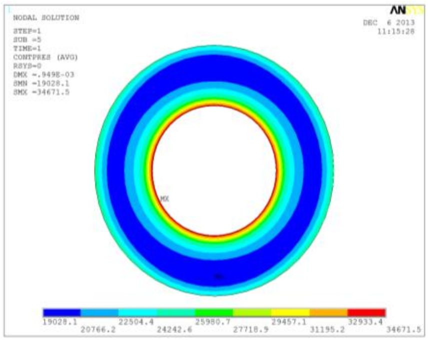


Fig.12. Contact pressure contour plot at bolt head and flange contact area.

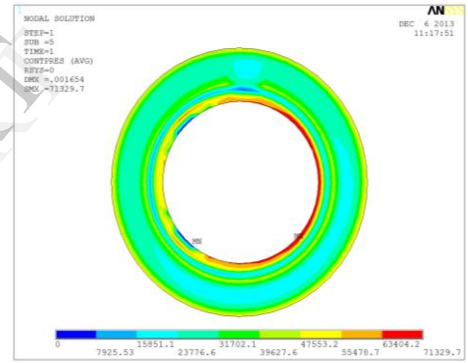


Fig.13.Contact pressure contour plot at nut and flange contact area.

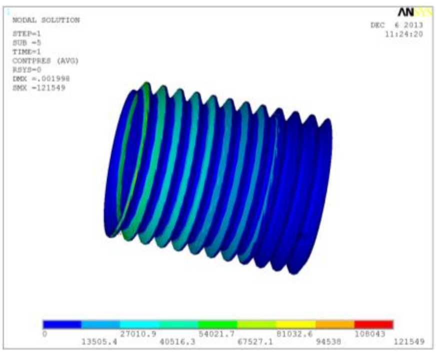


Fig.14. Contact pressure contour plot between bolt and nut threads.

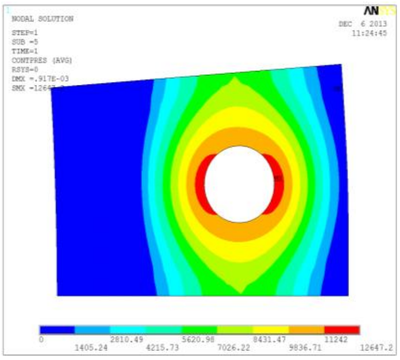


Fig.14. Contact pressure contour plot between two flanges

5.2 Case2 : Pretension and axial load of 7000 lbf

Nonlinear contact analysis was carried out by considering pretension and axial load. The results indicate that the contact separation starts at the inner radius of flange and grows towards outer diameter of flange as the axial load is increased in the bolted flange joint. The resultant displacement contour plot is shown in Fig.16. It indicates about opening of the flanges due to external load.

It is observed from the analyses that most of the load is shared by flanges when the external applied axial load is up to 30% of preload, and beyond this, bolt starts sharing more external load. The load sharing behaviour of bolt and nut is shown in Fig.17 .

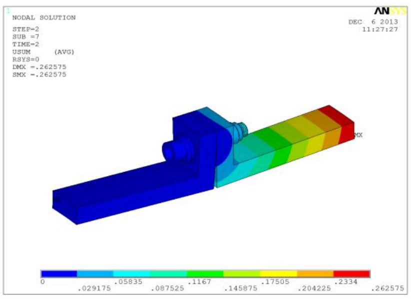


Fig. 16. Resultant displacement contour plot

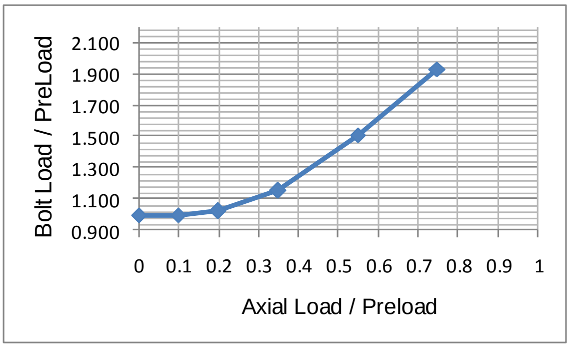


Fig. 16. The load sharing behaviour of bolt and flange

6.Conclusion

An effective three-dimensional thread modeling scheme, which can accurately take account of its helical geometry, is proposed using the equations defining the real configuration of the thread cross section perpendicular to the bolt axis. It is shown how the thread root stress varies along the helix and that the maximum stress occurs at the first engaged thread. The locations of stress concentrations are shown in detail. The stress distribution in the flanges and the stress concentration at flange bolt holes are analysed using 3D FE model. The contact pressure distribution at joint interfaces ie between flanges, bolt head and flange, Nut and flange, Bolt and Nut are analysed based on analysis. It is shown how the contact pressure at the nut loaded surface varies in the circumferential direction due to the effect of the helical thread geometry

It is observed from the analyses that most of the load is shared by flanges when the external applied axial load is up to 30% of preload, and beyond this, bolt starts sharing more external load. The maximum stress occurs at the first engaged bolt thread root. Most of the bolt failures are at the first engaged thread. Experimental results also substantiate the behaviour observed in the analyses

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