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Stress Analysis of Flanged Joint Using Finite Element Method

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Abstract: ***In general, most flanging coupling is available in transformation system and automobile industries. "General purpose flange coupling – Basic dimensions" specify the dimensioned for metric threads of nut-bolts. Which type of flange is best for a particular application is a common question. In order to give some answers to that question this paper deals with finite element analysis of bolted joint in different load conditions. This paper presents a theoretical model and a simulation analysis of flange and bolted joints deformation, stresses. The flange and Nut-Bolts force and contact stiffness factor are considered as parameters which are influencing the joint deformation. The flanged joint is modelled and simulation using ANSYS 14 Software. The finite element analysis procedure required in ANSYS simulation is presented as a predefined process to obtain accurate results.***

Keywords: flange joints, nut- bolts, Mesh-model, FEM analysis, results obtained ANSYS

1. Introduction

Bolted flanged joints are generally made up of the bolt group (head, stud, and nut) and the top and bottom flange (Figure 1). Bolted joint is designed to hold two or more parts together to form an assembly in a mechanical structure (Figure 2).Because of different loading conditions, especially high loads, bolted connections can separate. To minimize this effect, a pretension is applied to the bolt. This insures that the connection will not separate, provided the applied load remains less than the pretension. Thus, two primary characteristics in the bolted joint are a pretension and a mating part contact [1, 2]. In order to accurately predict the physical behaviour of the structure with a bolted joint, a detailed three dimensional bolt model is desirable, which fully includes the friction due to the contact on mating parts and pretension effect to tie. This paper deals with finite element modelling of bolted joints with coarse and fine threads in order to reveal their behaviour in different loading conditions.

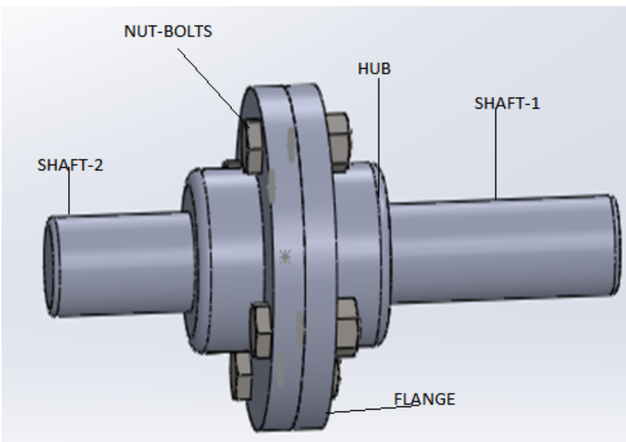

Figure 1: flanged Coupling Assembly with 3-D Model

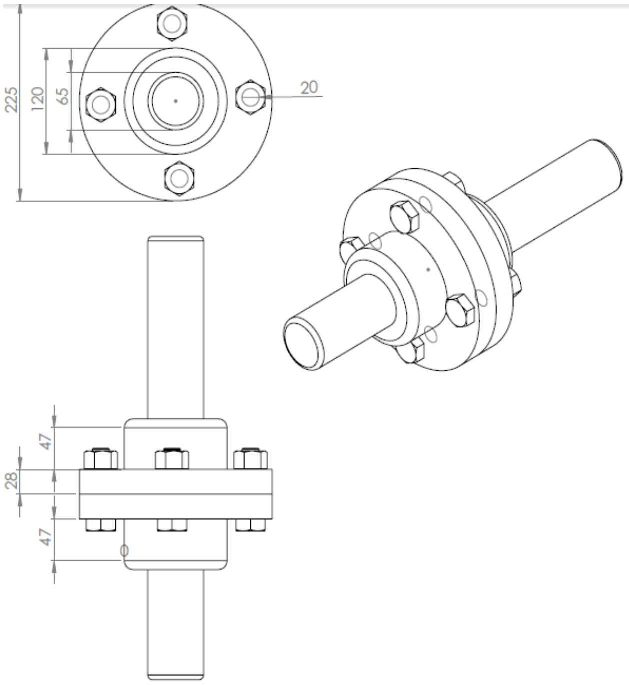

Figure 2: 2-D Flange Coupling Model

2. The FEA Analysis of the Bolted Joint

2.1 The FEA analysis procedure

The detailed finite element analysis for a flanged bolted joint presented in Fig. 4 is exemplified in the following phases:

- The first phase is modelling the joint using CAD software. The model geometry was generated using CATIA software and then imported as a neutral file in ANSYS. Due to symmetry conditions the model is sectioned. Geometric details, such as chamfers, radii of connection have only a local influence on behaviour of the structure therefore those are neglected. In this analysis we neglect the bolt thread and surface roughness.
- Next, the prepared geometric structure is reproduced by finite elements. The finite elements are connected by nodes that make up the complete finite element mesh.

Each element type contains information on its degree-of-freedom set (e.g. translational, rotational, and thermal), its material properties and its spatial orientation 3D-element type's solid42. The mesh was controlled in order to obtain a fine and good quality mapped mesh. The flange had 31046nodes and nut-bolt node as elements170791and 1455 respectively.

- In order to solve the resulting system equation, boundary and loaded conditions are specified to make the equation solvable. In our model, flange one end fixed and other end has different axial loads were applied. The same the boundary and loaded conditions is bolt were applied. Load variation in range of 20Nto 300N applied, with the Z axes along the bolt and flange length.
- The last phase is interpreting the results. Recorded results plotted forces and deformation ,

2.2 Stiffness analysis of the bolt part

The bolt stiffness *kb* is often determined by simple Calculations, approximating the deformation of the bolt head and of the nut. Assuming uniform tensile stress σ and uniform strain= σ/E for a total contact pressure force P with a uniform bolt cross-sectional area A_b, the authors obtained for the total elastic energy U (sum of the elastic strain energy U and the elastic stress energy where V is the estimated volume obtained with the estimated length L that also account for the stiffness of bolt head and nut. In terms of a reference displacement

$$[K]\{x\}=\{F\} \qquad \text{————————(1)}$$

$$U_b = U_\varepsilon + U_\sigma = \sigma\varepsilon\overline{V} = \frac{\sigma^2 V}{E} = \frac{P^2 A_b L}{A_b{}^2 E} = \frac{P^2 L}{E A_b} \qquad \text{- (2)}$$

$$k_b = \frac{P}{\upsilon_P} = \frac{P^2}{P\upsilon_P} = \frac{P^2}{U_b} = \frac{EA_b}{\overline{L}}, \qquad \text{-------- (3)}$$

$$e\{1\} = \{D_{bp}\} - \{D_{fp}\} = \left([S_{bp}]^{-1} + [S_{fp}]^{-1}\right)\{F_{cp}\} \qquad \text{---- (4)}$$

$$\{F_{cp}\} = \left([S_{bp}]^{-1} + [S_{fp}]^{-1}\right)^{-1} e\{1\}. \qquad \text{----- (5)}$$

$$[K] = \left([S_{bp}]^{-1} + [S_{fp}]^{-1}\right)^{-1} \qquad \text{----- (6)}$$

2.3 Analysis of Load Carrying Capacity

Typically, flanged –bolted jointed are designed for carrying tensile loads and shear loads. In a tensile bolted joint the most critical parameter is the applied pretension load [4]. In such joints, a fastener that can provide the maximum tensile load is preferable. Similarly, if the bolt is subject to shear loads the effective shear area at the shearing planes needs to be maximized. In general, it is assumed that the failure will occur in the bolt (which is preferable and the design criterion for failure). In a first order analysis (assuming total load is evenly distributed over the entire engagement length) on the load carrying capacity of a bolt, it is found to be directly related to the shank diameter. Furthermore, shear failure of the threads should not occur when the bolt is engaged appropriately with the corresponding nut. This means that the threads with larger effective shear areas will be better. The thread profile for ISO metric thread depends on the pitch of the thread, although the general form is the same. Based on the effective tensile stress area of the bolt it could be seen that this area drops as the pitch is increased. For example, a M10x1.25 bolt has an effective tensile stress area of 61.2 mm2 while a M10x1.5 bolt will only have 58 mm2. As such, the failure load of the bolt should be approximately 5.5% larger for the finer pitch bolt. Similarly the thread shear area along the pitch-line per millimetre of the bolt length is 26.57 mm2 for M10x1.25 bolt and 25.62 mm2 for M20x1.5 bolt. This gives approximately 3.7%larger thread shear area for the finer threaded bolt. This effectively reduces the minimum engagement

Length requirement by approximately the same percentage [5]. In essence, the above discussion shows that fine threads are slightly better than coarse threads in terms of load carrying capacity. If a higher order analysis including the effect of helix angle of the thread and the resulting force resolution is considered, it becomes evident that the higher helix angle inherent to coarse thread will increase the local forces and stresses on the threads [6]. This could lead to bearing failure on the threads and hence galling. In order to confirm conclusions of previous analysis of load carrying capacity and provide deeper insight in behaviour of generic bolted joint with coarse and fine threads it is conducted Commercial software Ansys (Ansys, Canonsburg, Pennsylvania, USA) was used for finite element analysis of bolted joint. First, geometric model of bolted joint was generated with ANSYS APDL. Then ANSYS module Simulation was used to generate the finite element mesh of linear solid 42 elements and hexahedral elements (Figure 4). Material of all elements in bolted joint was, assumed to be homogenous, isotropic and linear elastic with a Poisson's ratio of 0.3 and an elastic modulus of $2x10^2$ MPa. The model is loaded with a tensile load and the axial working loads on head-top flange and nut-bottom flange mating surfaces (Figure 5).

3. FEA Results

The reduced model bolted joint simulations were carried out in order to study its behaviour under the action of external loads taking into account as parameters the normal stiffness, the pretension force and friction coefficient. The influence of these parameters on the deformation of the bolted joint and flange jointed is shown in table 1.To study the influence of force to the Nut-Bold and Flange joint we ranging from a minimum force of 20 N and a maximum force of 300 N. Although the charge of the structure is linear the response of the structure is linear. The influence of pressure on bolted joint deformations, presented in Fig. 6shows a linear behaviour of bolted joint but it hasn't a negligible influence on deformations represented in Fig. 6. The contacts between bolt and nut and between bolt head and flange have the same deformations and, also have the most significant deformations. The contact status is shown in Fig. 6

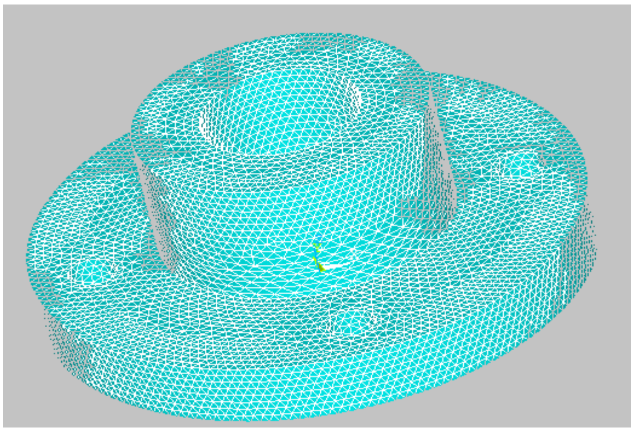

Figure 3: Mesh Flanged

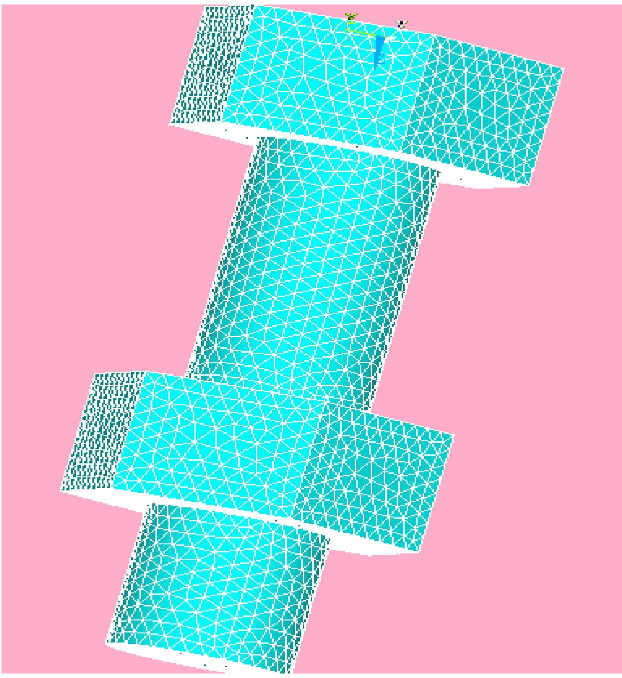

Figure 4: Meshing Nut-Bolts

Represented in Fig. 1, exists in the contact between the bolt head and flange and in the contact between bolt and nut. The value of penetration is small enough. Defining joints is one of the most difficult aspects when simulating the behaviour of machine-tools, because there are many variables that can affect the joint's properties. By using finite element analysis software we can optimize the design process of machine tool components by identifying the parameters that has a influence on the static behaviour of machine tools. By analysing the results obtained in the post-processing phase, the user can evaluate the properties of machine tools still in the design stage, without the need to make prototypes. Based on these preliminary results of bolted joint contact deformation analysis further research will be carried on the model optimization. By knowing the influence of the parameters on the contact deformation we can optimize the stiffness of the structural components of machine tools. In this paper we considered as parameters: Flanged joint force supports a maximum value of 300 N due to the maximum stress that the flange and nut-bolt can be subjected to; we started from the default stiffness factor as generated by the ANSYS 14.0 software and increased to the point that the value of the deformation remains linearly. We found that the deformation and stress has a linear variation; the tensile force between the flanges and nut-bolts variation in 20 to 300N cause a decrease of the bolted joint deformation. Financial Bolted joint is made up of ISO metric M20 bolt and two flanges with thickness of 28 mm. Meshed models of bolted joints with coarse and fine threads was consisted of 31046 finite elements with 172548 nodes and 58733 finite elements with 876652 nodes respectively. Generated finite element models of the bolted joint were used in a few loading simulations in order to demonstrate the capabilities of the models in studding behaviour of the bolted joint. The finite element models of the flanged and bolted joint were subjected to tensile load ranged from 20 N to 300 N and axial working load ranged from 20N to 300 N

Table 1

	Nut and bolt		*Flange*	
Forces	*Stresses*	*Deformation*	*Stresses*	*Deformation*
20	59.2629	0.022283	22.01	0.010653
40	118.526	0.044476	44.021	0.021306
60	177.78	0.066714	66.03	0.031959
80	237.052	0.088953	88.042	0.042612
100	296.315	0.111191	110.05	0.053262
120	355.577	0.1334	132.06	0.06392
140	414.84	0.155667	154.07	0.074571
160	474.103	0.17791	176.08	0.085224
180	533.366	0.200143	198.09	0.095877
200	592.628	0.222381	220.1	0.10653
220	651.892	0.244619	242.11	0.117183
240	711.155	0.266858	264.12	0.127836
260	770.418	0.289096	286.13	0.138489
280	829.681	0.31134	308.141	0.149142
300	888.944	0.333572	330.151	0.159794

As shown in above table 1 are recorded result obtained from ANSYS 14.0 software. All stress distribution over the flange and Nut-Bolt as shown in fig 7 and 8 respectively. it is clear that force is directly proportional to stress in flanged and nut-bolted jointed.

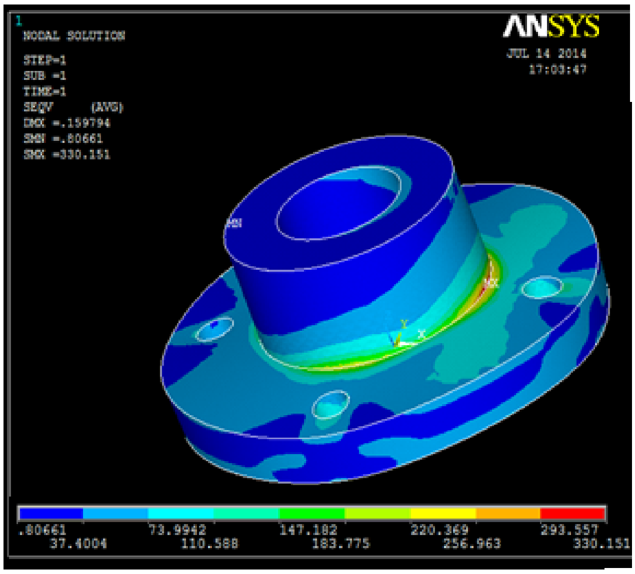

Figure 5: Results of Flanged

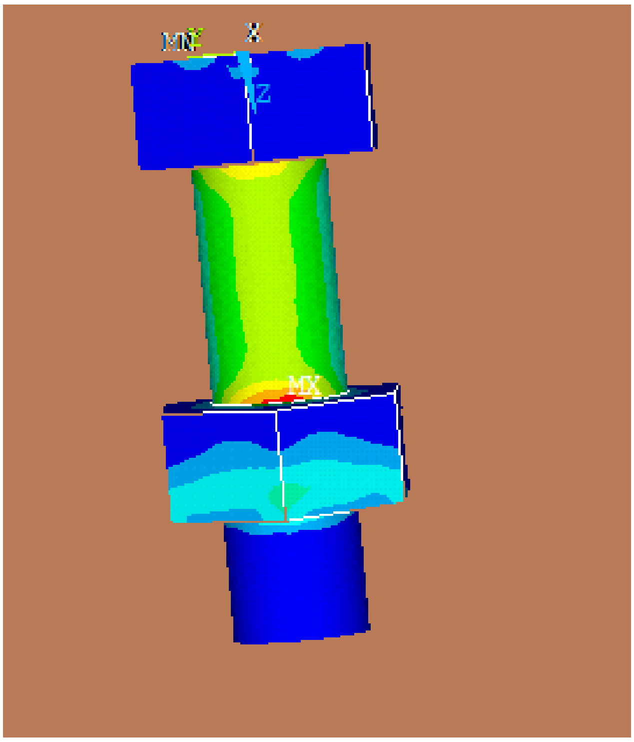

Figure 6: Result of Nut-Bolt

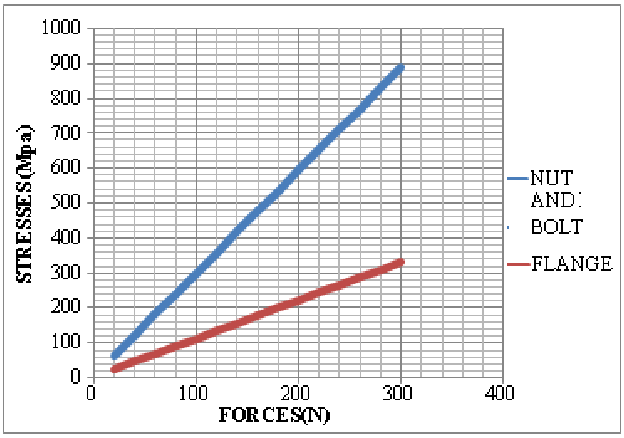

Figure 7: Force V/S Stress Values

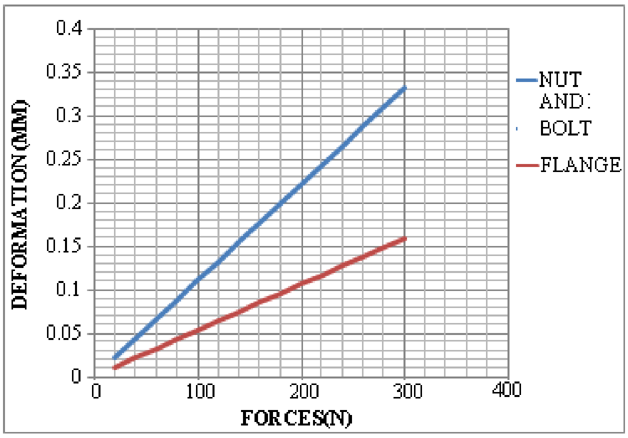

Figure 8: Forces V/S Deformation Values

For the given loads the models were solved for stresses at the nodes as can be expected, the above distribution is affected by the bolt tension and the material properties of flanged the bolt and the nut. In general, considering the stress-force relationship for steel, as the load is increased further the first flanged will reach yield and plastically deform while carrying the maximum possible load. However, a bolted joint should be designed to always force the failure in the bolt shank and not in the thread and therefore, the above findings also suggest that a larger number of engaged threads (fine pitch) will improve the performance of the joint as the stresses are distributed over a larger area reducing the resulting local Stress concentrations.

4. Conclusions

The following conclusions are drawn from the present work:

1) The stresses of flange obtained in static analysis are within the allowable stress limit888.944Mpa.
2) The maximum deflection for the flanged is0.333572mm and the maximum deflection for the Nut-bolt is 0.275 mm.
3) The maximum normal stress for the nut-bolt is 9.011 N/mm^2 and the maximum normal stress for the flange is330.151N/mm^2.
4) The first, second and third principle stresses for flange are288.74 N/mm^2, 99.679N/mm^2,71.883 N/mm^2.
5) The first, second and third principle stresses for Nut-Bolt are 435.43 N/mm^2, 78.229 N/mm^2 and 32.041 N/mm^2.

References

[1] J.F.Tsai, H.J.L.a., Analysis of Underwater Free Vibrations of a Composite Propeller Blade. Journal of REINFORCED PLASTICS AND COMPOSITES, 2008. Vol. 27, No. 5: p. 447-458.
[2] H.J.Lin, W.M.L.a.Y.M.K., effect of stacking sequence of nonlinear hydro elastic behaviour composite propeller Journal of Mechanics, 2010. Vol. 26, No. 3: p. 293-298.
[3] Toshio Yamatogi, H.M., Kiyoshi Uzawa, Kazuro Kageya, Study on Cavitation Erosion of Composite Materials for Marine Propeller.2010.
[4] Young, Y.L., Fluid–structure interaction analysis of flexible composite marine propellers. Journal of Fluids and Structures, 2008. 24(6): p.799-818.
[5] Taylor,D.W, "The speed and power of ships", Washington,1933.
[6] J.E.Conolly, "Strength of Propellers", reads in London at a meeting of the royal intuition of naval architects on dec 1, 1960, pp 139- 160.
[7] Chang-sup lee, yong-jikkim,gun-do kim and in-siknho."Case Study on the Structural Failure of Marine Propeller Blades" Aeronautical Journal, Jan 1972, pp87-98
[8] G.H.M.Beek, visitor, Lips B.V., Drunen. **"**Hub-Blade Interaction In Propeller Strength**",** the society of naval architects and marine engineers, May 24-25, 1978, pp 19-1-19-14.
[9] W.J.Colclough and J.G.Russel. "The Development of a Composite Propeller Blade with a CFRP Spar" Aeronautical Journal, Jan 1972, pp53-57.
[10] Gau-Feng Lin "Three Dimensional Stress Analysis of a Fiber Reinforced Composite Thruster Blade", the society of naval architects and marine engineers, 1991.
[11] J.P Ghosh and R.P Gokaran." Basic ship propulsion", 2004.

[12] Looram M., Merrick E..: Bolted joint maintenance & applications guide, Aptech engineering services, Report TR-104213, USA, 1995
[13] Bolt and screw compendum, Kamax, USA, 2006.
[14] ISO 724: General purpose metric screw threads – Basic dimensiones, 1993.
[15] Saman F.: An engineering insight to the fundamental behaviour of tensile bolted joints, Steel constructions, Vol.35 No.1, 2001.
[16] Saman F.: Minimum Thread Engagement: What is the optimum engagement lingth? Technical note AFI/01/002, Ajax Fasteners Innovations, 2001.
[17] Saman F.: An engineering insight to the fundamentals of screwed sheet fastening to resist cyclic loading, 9th Wind Engineering WorkShop, Townsville, 2001.
[18] Saman F.: Fine thread versus coarse thread: Which one is better thread? Technical note AFI/01/001, Ajax Fasteners Innovations, 2001.