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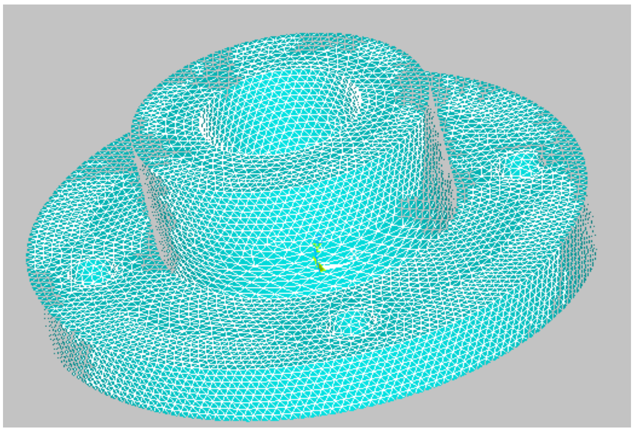

Figure 3: Mesh Flanged

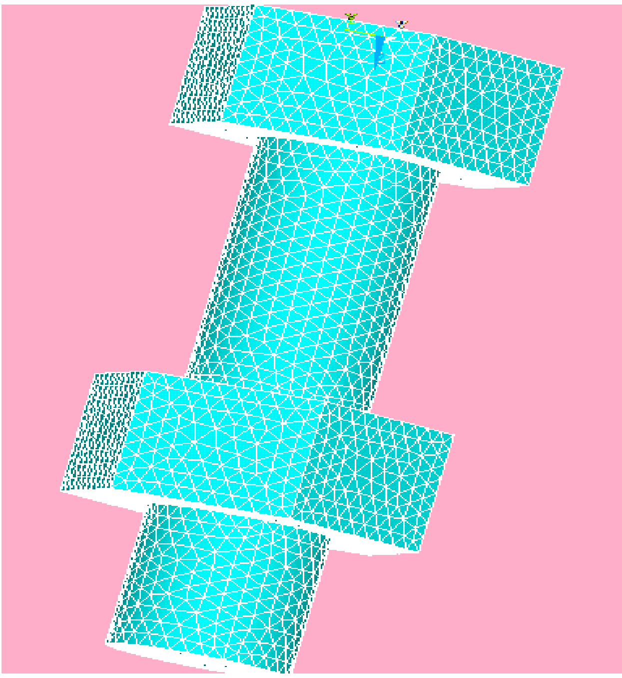

Figure 4: Meshing Nut-Bolts

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.

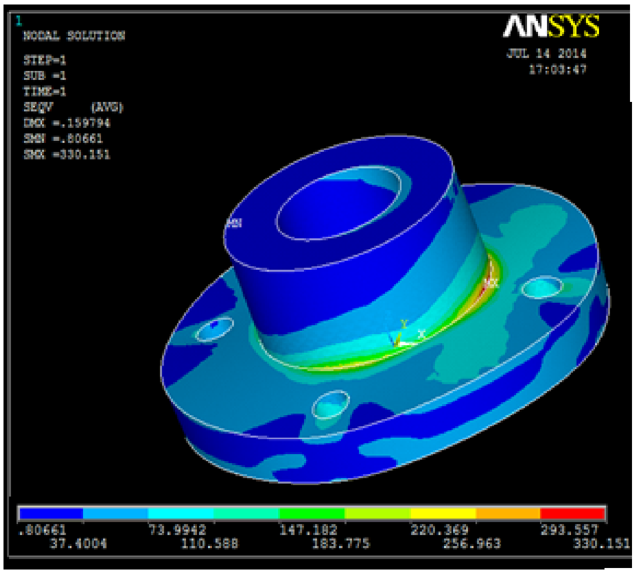

Figure 5: Results of Flanged

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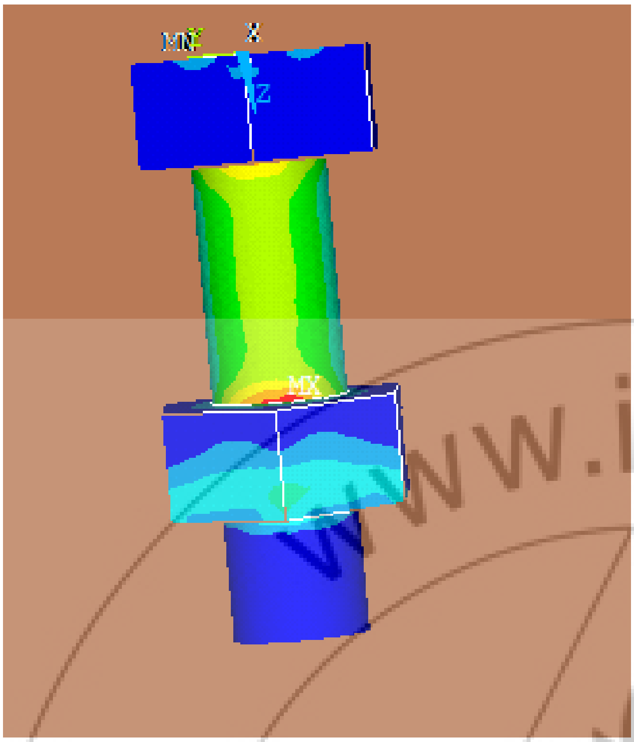

Figure 6: Result of Nut-Bolt

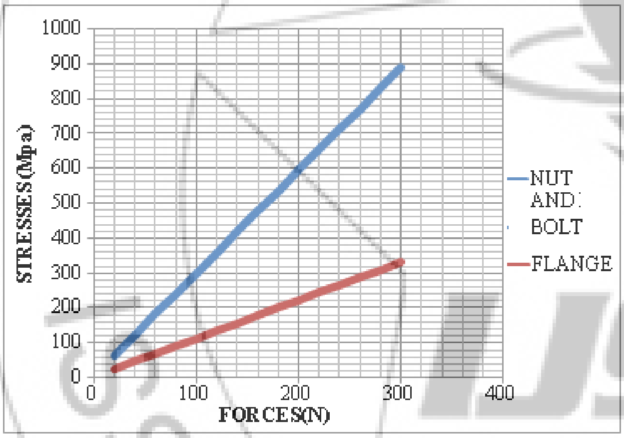

Figure 7: Force V/S Stress Values

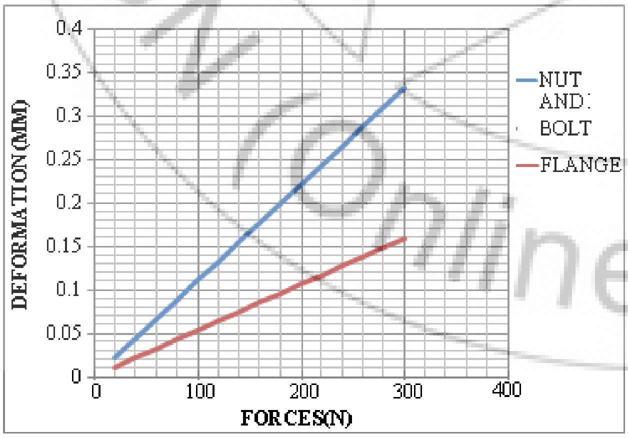

Figure 8: Forces V/S Deformation Values

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.

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