To Develop A Methodology For Pretension In The Bolt Using Finite Element Analysis

Nikesh Rokade[1], RajKumar Bhagat[2], U.S. Chavan[3]
[1, 2, 3] Vishwakarma Institute of Technology, Pune, India

Abstract- *Bolted joints are very much useful in joining two parts or structures together. To make the joint leak proof and to avoid separation, pretension plays important role. But before actual practice it is needed analyze it using FEA. Explicit Dynamics does not support bolt pretension tool, so different methods are studied to apply bolt pretension. Due to inability of bolt pretension tool in explicit dynamics, forces are applied in different manners so as to get similar kind of stress pattern as that of bolt pretension tool in ansys18.2. These methods are then compared with it by considering their stress pattern and shear stress. Shear stresses are validated with analytical approach to give best method for pretension using cause and effect matrix.*

Keywords- bolt pretension; VDI guidelines; bolt slicing; stress pattern; Cause and effect matrix.

I. INTRODUCTION

Bolts are the most widely used fastening element. It is required to understand structural behaviour bolted joint, as the failure of bolted joint will lead to serious human and economic consequences. Bolts are having localized connection hence they are subjected to very high stress. Fully tightened bolt can survive much better than untightened or loose bolt. So to ensure the survival of bolt it is needed to preload it with required essential magnitude. If this initial bolt preload is more than that of required then bolt threads may fail due to shearing without application of external load.

To visualize bolted joint loading, joint diagrams are very much useful. A joint diagram is useful for displaying the load deflection characteristics of the bolt and the material that it clamps. In Fig1, joint diagram explains how the bolt extension and clamp compression occurs after application preload to the bolted joint.

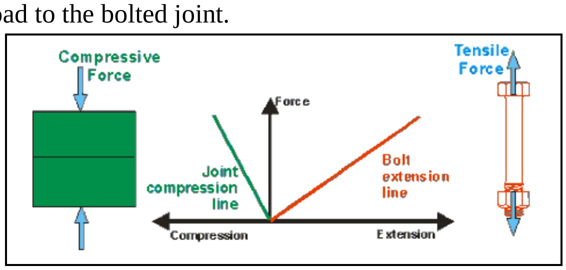


Fig1. Joint diagram for simple bolted joint [1]

With application of external tensile force the joint may reduce some of the clamp force formed due to preload. This is illustrated in the Fig2. The external force acts through the joint material and then subsequently into the bolt. It is clearly visible in Fig2 that bolt force increased and joint clamp force reduced due to application of external force. It must be realized that the load on the bolt cannot be added without decreasing the clamp force acting on the joint. As it can be observed from a study of the diagram, the actual amount of increase in the bolt force is dependent upon the relative stiffness of the bolt to the joint. Force acting on the bolt increases with increase in external force. At the same time the clamp force acting on the joint is decreased. To study the bolted joints, it is required to analyze this bolted joint using FEA prior to that of practical testing of joint. Before developing a finite element model, the analyst must determine the bolted joint characteristics to be modeled and understand the capability of the finite element program being used. This knowledge is useful for the analyst to determine how closely he could simulate the bolted joint. A bolted joint itself is a complex part for simulation, even this can be solved with today's commercial FEA tools with proper element type, step size etc. but still it becomes time consuming, So this bolted joint can further be simplified by omitting geometrical threads and fillets.

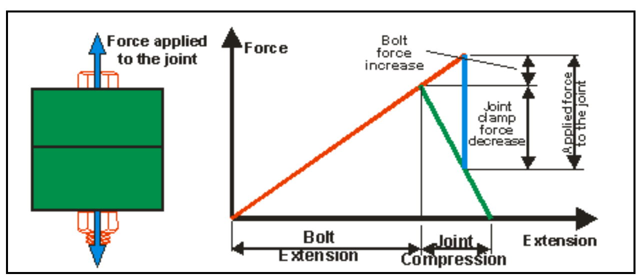


Fig2. Joint diagram for simple bolted joint with external forces [1]

S. H. Ju and C. Y. Fan [2] in there analysis of bolted joint had used bolt pretension with initial displacement scheme and for to make contact between contacting faces they had used node to plane contact so that node can slide across several target faces, but it is difficult to identify amount of displacement that will correlate with the pretension force. N.Tanlak and MA MacCarthy [3] [4] in their paper in order to apply pretension they defined artificial anisotropic coefficient of thermal expansion at one of the washer and using temperature gradient bolt pretension is developed in the assembly. Here as well it will lead to further material investigation for thermal expansion coefficient and to find out temperature gradient.

A.M. Citipitioglu [5] in his paper had used method in which bolt are made shorter and contact is disabled and enabled with plate in two steps to form bolt pretension, but contact death and birth settings are not available in ansys18.2. Jeromy Montgomery and Jeong Kim [6] [7] in their papers of methods for modelling bolts in bolted joint compared various methods of modelling bolts, they compared these methods based on detailing of information required and displacement of outside edge gap on the opposite end of the transverse constraint, from their conclusion in order to obtain good correlation with actual scenario solid bolted model is to be chosen. Jaromy Montgomery [13] in his white paper gave the idea of splitting the bolt in two pieces and pulling these bodies towards each other. He applied forces such a way that lower segment will get pulled towards upper segment and vice-versa. In Design and analysis of metric bolted joints VDI guidelines and FEA by Yung-Li Lee and Hsin-Chung [8] methodology for designing bolted joint is described, which is useful for to determine preload and the shear stress at the connection.
Explicit dynamics does not support bolt pretension tool so it is needed to apply bolt pretension by some other means such application of velocity, displacement or forces instead of it.
In this paper different force application methods are compared with taking FEA bolt pretension tool as a benchmark, best method is selected to be used for analysis considering stress pattern, shear stress. For this method shear stresses are validated by using analytical approach.

II. METHODOLOGY EXPERIMENTAL/ COMPUTATIONAL

- *Model Geometry*

The main objective of this methodology is to get similar stress pattern as that of bolt pretension tool, with application of forces at different locations, So that bolt pretension tool can be replaced by application of forces. To simulate the model with its actual physical condition bolt head and nut are kept hexagonal. The assembly is made up of four separate parts. Frictional surface to surface contact is used in between each pair in contact with coefficient friction of 0.16. In order to reduce the computational time and modelling difficulty, geometrical threads, chamfers, fillets etc. are omitted from the model. Dimension and material properties are described in Table1.

Threads are modelled virtually on nut and bolt using contact settings of Ansys18.2 to simulate its physicality.

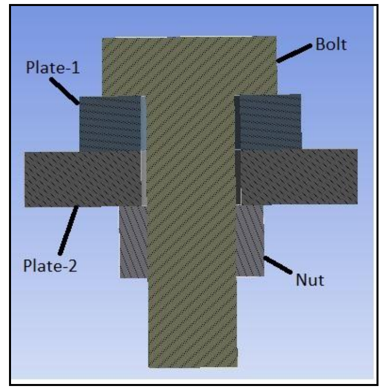


Fig3. Geometry of model

Table1. Geometrical dimensions of assembly

Sr. No.	Description			
	Part Name	*Dimension*	*Units*	*Material Used*
• 1.	• Bolt	M8 x 1.5 x 20	mm	B16
• 2.	• Nut	M8 x 1.5 x 6.6	mm	B16
• 3.	• Plate-1	Φ20 x Φ9 x5	mm	GG15
• 4.	• Plate-2	Φ30 x Φ9 x5	mm	GGG40

In the above table, Dimensions of bolt and nut are standard. Plate dimensions are taken as per convenience and Inner diameter of plates are taken from VDI guidelines [8].

- *Meshing*

Numerous calculations have been conducted to compare the accuracy of all tetrahedral meshes to all hexahedral meshes. First it was shown that the stiffness matrix eigenvalues for linear tetrahedrons were generally larger than that those for linear hexahedrons. This fact demonstrates that linear hexahedrons can generally deform in a lower strain energy state, thus making them more accurate than linear tetrahedrons in numerous situations. Further it produces more error than quadratic linear tetrahedral, linear hexahedral and quadratic hexahedral [9]. Though tetrahedral elements are too much stiff, it is used as a default during meshing to reduce computational time.

So in this analysis all bodies are meshed with linear tetrahedral (tet4) elements with default element size. Further nut and bolt contact faces are given with finer mesh to get accurate results of shear stresses from analysis. Using these settings mesh count for this model observed as 85979 nodes and 435228 elements.

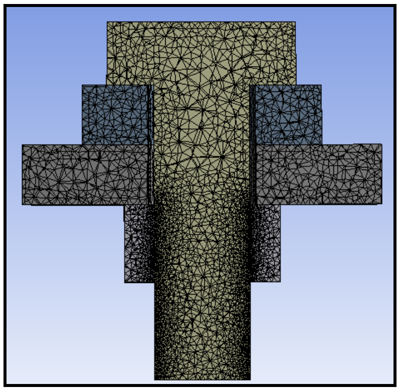


Fig4. Meshing

- *Material*

This bolted joint consists of a bolt, a nut and two plates. Bolt and nut are made up of B16 material. Plate-1is made up of Grey cast iron while plate-2 is made up of ductile iron. The mechanical properties and industrial nomenclature of these parts are shown in Table 2.

Table2. Material Properties [10] [11]

Sr. No.	SiMo ductile iron (EN-GJS-400-15/GGG40)		
	Property	*Magnitude*	*Units*
1.	Density	7300	Kg/m^3
2.	Youngs Modulus	170	GPa
3.	Poissons ratio	0.28	
	Grey cast iron(EN-GJL-150/GG15)		
1.	Density	7100	Kg/m^3
2.	Youngs Modulus	103	GPa
3.	Poissons ratio	0.27	
	B16 (ASTM A193)		
1.	Density	8500	Kg/m^3
2.	Youngs Modulus	97	GPa
3.	Poissons ratio	0.3	

- *Bolt preload*

Instead of using bolt preload formula with nut factor, bolt preload is calculated based on VDI guidelines in which the inputs required are assembly geometry and coefficient of friction. [8]. VDI guidelines are preferred, as in nut factor method exact value of nut factor is difficult to find.

In VDI approach assembly torque is equivalent to the product of the tangential force on the thread contact surface and the thread pitch radius. Tightening torque and coefficient of frictions are selected from VDI approach based on bolt dimensions and material used. [7]

$$M_A = F_M\left[\left(0.16P + 0.58 d_2 \mu_G\right) + 0.5 D_{KM} \mu_K\right] \quad (1)$$

Where,

F_M = Bolt preload due to M_A

μ_G = Coefficient of friction in the thread =0.16

μ_K = Coefficient of friction in the bolt or nut-bearing area=0.16

d_2= Pitch diameter of threads

M_A = Applied assembly torque=29.8 Nm

P = Pitch= 1.25 mm

D_{KM} = Effective diameter of bolt head or nut bearing area $D_{KM} = 0.5(d_w + d_h)$ = 0.5(11.6+9) = 10.3 mm

Hence bolt preload, F_M =17622.2249 N

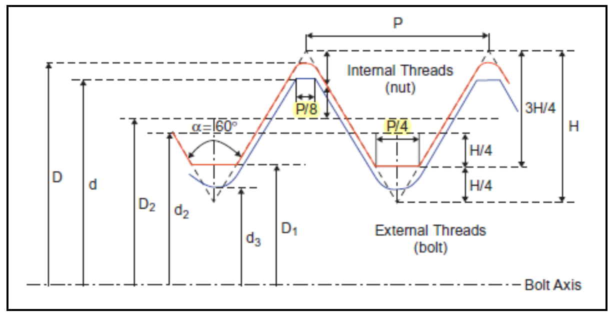


Fig5. Nomenclature of ISO metric threads [8]

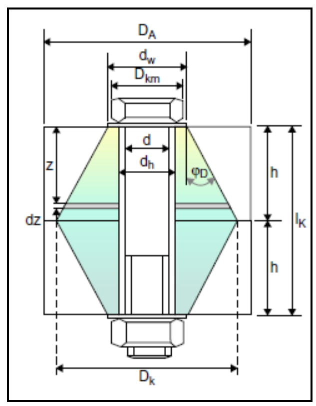


Fig6. Section of throughout bolted joint [8]

- *Analytical shear stress on External and Internal Threads*

For analytical validation of pretension methodology shear stresses at nut and bolt are going to get considered. Stripping of threads will occur when stress in the threads exceeds that of the thread strength. To calculate shear area nut height is taken as m_{eff} .

1. Shear stress at internal threads

The shear area (A_{SGM}) for the length of internal thread engagement (m_{eff}) is expressed as follows:

$$A_{SGM} = \frac{\pi d}{P}\left[\frac{P}{2} + (d - D_2)\tan 30°\right] \times m_{eff} \quad (2)$$

Where,

m_{eff} = Length of engagement=6.6 mm
d = Nominal diameter of bolt=8 mm
D_2 = Pitch diameter of internal threads
$=(d - 0.6495P)$ = 7.1881 mm
P = Pitch of thread=1.25 mm
Hence, shear area=A_{SGM}=145.14 mm^2
Shear stress $= \frac{\text{Bolt Preload}}{A_{SGM}}$ =121.41 MPa

2. Shear stress at external threads

The shear area (A_{SGS}) for the length of external thread engagement (m_{eff}) is expressed as follows:

$$A_{SGS} = \frac{\pi D_1}{P}\left[\frac{P}{2} + (d_2 - D_1)\tan 30°\right] \times m_{eff} \qquad (3)$$

Where,
m_{eff} = Length of engagement=6.6 mm
In general calculation pitch diameter of bolt and nut are taken as a same, hence $D_2 = d_2$ [12]
d_2= Pitch diameter of bolt=7.1881 mm
D_1 = Minor diameter of internal threads = $(d - 1.2268P)$ = 6.4665 mm
P = Pitch of threads=1.25 mm
Hence, A_{SGS}=111.7214 mm^2
Shear stress = $\frac{\text{Bolt Preload}}{A_{SGS}}$ =157.72 MPa
These analytical shear stresses are further used to compare with shear stress from FEA.

- *Methods used for Bolt pretensioning*

Method M0: Bolt Pretension tool

This is the regular method for applying bolt pretension. For this method, it is needed to create new local coordinate system. There is bolt pretension tool available in Ansys18.2 by using this tool pretension is applied. Pretension force is loaded from 0 to 17622N and then locked. Fixed support is given at the outer faces of nut.

Method M1: Application of forces as a bolt pretension at outer faces.

In this method equal and opposite force of same magnitude is applied on the bolts upper and lower extreme faces of bolt, whose directions are towards each other. This pretension force is loaded from 0 to 17622N while outer faces of nut are fixed.

Method M2: Application of remote forces

This method is similar to method M1 but instead of regular forces, remote forces are used, because these remote forces gets originated from centre of bolt.

Method M3: Application of forces on sliced bolt away from each other

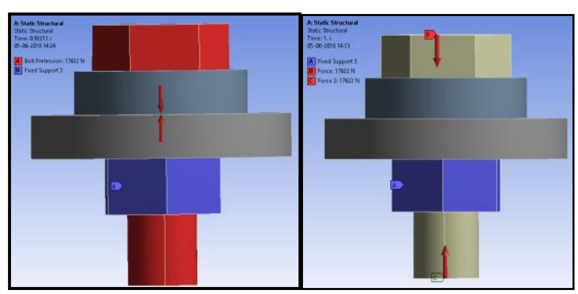
In this method bolt is sliced at the centre of shank above the portion of threads start point. Forces of magnitude of preload force are applied on these sliced faces directing away from each other. In this method displacement boundary conditions are given at the upper and lower extreme faces of so that model will remain in equilibrium. Displacement boundary conditions are also provided at outer faces of bolt to avoid its movement in other directions.

Method M4: Application of forces on sliced bolt towards each other with no contact between sliced parts.

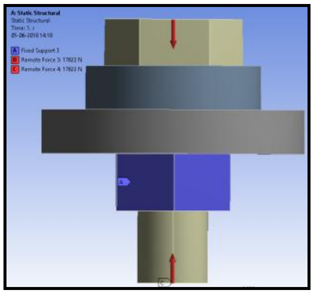
In this method bolt is sliced at the centre of shank above the portion of threads start point. Forces of magnitude of preload force are applied on these sliced faces directing away from each other [13]. Translational joint is used between two split parts of bolt in bolt axis direction to avoid its movement in other directions. No contact is given in between sliced faces. Outer faces of nut are fixed.

Method M5: Application of forces on sliced bolt towards each other with frictional contact between sliced parts.

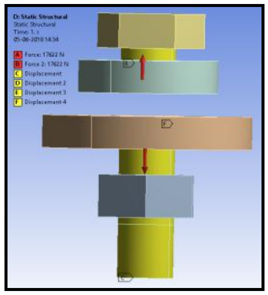
This method is similar to that of M4 except that frictional contact is used in between sliced faces.



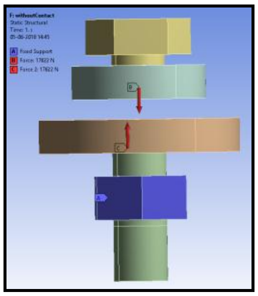
M0 M1



M2



M3



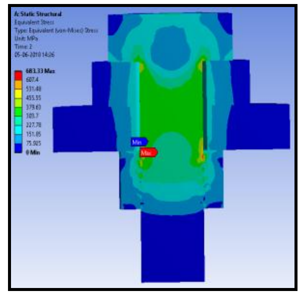
M4, M5

Method M0: Bolt Pretension tool
Method M1: Application of forces as a bolt pretension at outer faces
Method M2: Application of remote forces
Method M3: Application of forces on sliced bolt away from each other
Method M4: Application of forces on sliced bolt towards each other with no contact in between sliced faces, Translational joint in between them.
Method M5: Application of forces on sliced bolt towards each other with frictional contact in between sliced faces

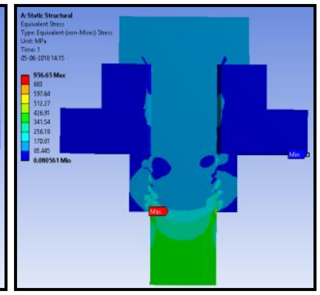
Fig.7 (Color online) Boundary conditions of models

Table3. FEA results

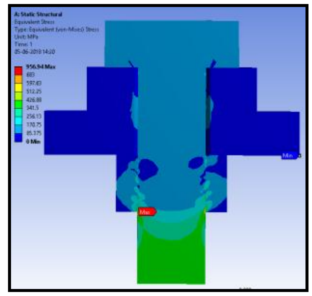
Method	Results			
	Bolt Shear stress (MPa)	*Nut shear stress (MPa)*	*Max Equivalent von mises stress (MPa)*	*Stress Pattern*
M0	155	141	683	At bolt shank
M1	108	120	957	At the bottom of bolt
M2	108	120	957	At the bottom of bolt
M3	100	1.48	336	Throughout the bolt
M4	154	142	841	At the bolt shank
M5	62	81	532	Beneath the nut



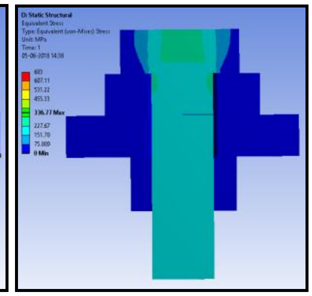
M0



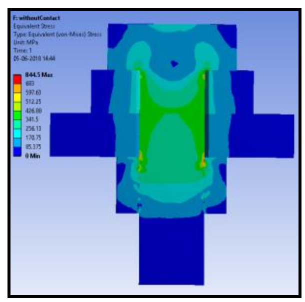
M1



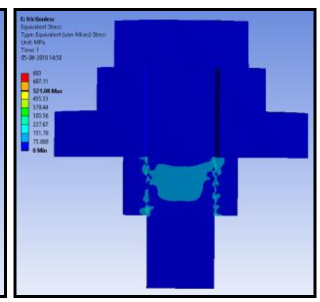
M2



M3



M4



M5

Method M0: Bolt Pretension tool
Method M1: Application of forces as a bolt pretension at outer faces
Method M2: Application of remote forces as a bolt pretension
Method M3: Application of forces on sliced bolt away from each other as a bolt pretension Method M4: Application of forces on sliced bolt towards each other as a bolt pretension, Translational joint in between them.
Method M5: Application of forces on sliced bolt towards each other with frictional contact in between sliced faces
Fig8. (Color online)Stress pattern observed in FEA

It is observed that stress pattern for method M4 in which sliced bolt and forces towards each other without contact in between sliced parts is showing good correlation with stress pattern by bolt pretension tool.

In order to select best method out of tried methods, cause and effect matrix is formed as shown in table4. By keeping bolt pretension as benchmark. Weightages are given to each criteria according to its importance. Ratings are given as for good correlation 9, poor correlation 3, and for very poor 1 or 0 based on observations and results.

After rating, these methods are ranked based on their weighted sum.

Table4. Cause and effect matrix

Criteria	Ratings					
	Weightage	*M1*	*M2*	*M3*	*M4*	*M5*
• Stress Pattern	• 10	• 1	• 1	3	9	1
• Bolt Shear stress	• 10	• 3	• 3	3	9	1
• Nut shear stress	• 8	• 3	• 3	0	9	3
• Max Equivalnet von mises stress	• 6	• 9	• 9	1	3	3
• Applicability with Explicit dynamics	• 10	• 3	• 3	9	9	9
• Total	•	• 148	• 148	96	360	152
• Rank	•	• 3	• 4	5	1	2

From table4 it is observed that method M4 is having highest weighted sum. So that this method can be used for pretension in explicit dynamics. To validate the results Shear stresses of method M4 are compared with the benchmarked pretension method. Also benchmarked pretension method is validated with shear stresses from analytical approach.

Table5. Comparison of percent variation

Comparison	Percent variation	
	Shear stress on Bolt	*Shear stress on Nut*
• Pretension tool and analytical approach	1.89%	16.52%
• Method M4 with Pretension tool	0.64%	0.7%

III. CONCLUSION

After ranking all these methods it is concluded that Method M4 in which bolt is sliced at the center of shank is best suitable to pretension. This method can be used for bolt pretension instead of pretension tool.

REFERENCES

[1] "A tutorial on basics of bolted joint" http://www.boltscience.com/pages/basics1.htm

[2] S. H. Ju, C. Y. Fan, "Three dimensional finite element of steel bolted connection," Elsevier, 2003.

[3] N. Tanlak, FO Sonmez, "Detailed and simplified model of bolted joints under impact loading," in J. strain analysis Vol 46, 2010, pp. 213-225.

[4] M.A. McCarthy, C.T. McCarthy, "Three-dimensional finite element analysis of single-bolt, single-lap composite bolted joints: part I—model development and validation," Elsevier, 2004

[5] A.M. Citipitioglu, R.M. Haj-Ali, "Refined 3D finite element modeling of partiallyrestrained connections including," Elsevier, 2001

[6] Jeromy Mongomery, Methods of modelling Bolts in the bolted joint, Siemens Westinghouse Power Corp, Orlando, FL.

[7] Jeong Kim, Joo-Cheol Yoon, "Finite element analysis and modeling of structure with bolted joint," Elsevier, 2006.

[8] "A tutorial on basics of bolted joint" http://www.boltscience.com/pages/basics1.htm

[9] Steven E. Benzley, Ernest Perry, "A Comparison of All Hexagonal and All Tetrahedral Finite Element Meshes for Elastic and Elasto-plastic Analysis".

[10] https://www.matbase.com,"the free and independent online materials properties resource, Sandia National Laboratories"

[11] http://www.matweb.com, "Material data exporterts for Ansys, Solidworks."

[12] WWW.F-M-S.DK, "Finding nominal pitch diameter",2.01.2011/1 1-2

[13] Jerome Montgomery, "Boundary Condition Influences on Shank Stress in 3D Solid Bolt Simulation", Siemens Westinghouse Power Corp, Orlando, FL