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Original scientific paper

Finite Element Analyses of Bolted Joints Using Different Thread Modelling Techniques

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Abstract: Intensive finite element analyses (FEA) were carried out to investigate the effects of thread modelling techniques on torque-force relation in threaded joints. The bolt thread was modelled by solid revolving the ISO thread profile and it was meshed using hexahedral elements. The nut thread was modelled either by solid/cut revolving the ISO thread profile, with/without root fillet or by using hexahedral/tetrahedral elements. Excellent agreement was found between torque-force relationship obtained using the FEA, the experimental result and the theoretical analyses. The FEA results showed the torque-force relationship is independent of both the modelling techniques and the mesh size but highly dependent on coefficient of friction between surfaces in contact. The study outputs support the simplified approach to use the tetrahedral elements to model threaded joints. The value of the equivalent coefficient of friction to be used is approximately the sum of the coefficients of friction at the threads and that at the bearing area.

Keywords: finite element thread modelling; nut-bolt friction; threaded joint; tightening force; tightening torque

1 INTRODUCTION

Threaded joints are non-permanent joints that are widely used in industry. For example, a Boeing 747 jumbo jet contains 2,5 million fasteners. Some of these fasteners cost several dollars [1]. The advantage of using the screwed joints is the ease of assembling and disassembling of the joint in order to conduct maintenance or repair. However, it is crucial to determine the pre-load developed in the joint by applying tightening torque to avoid joint failure [2]. Threaded joints could fail in one of the following modes: looseness that affects their integrity, leakage, blot rupture, etc. [3]. The threaded flanges on the pressure vessels critically affect the vessel sealing performance, safety and reliability [4]. Therefore, it was necessary to introduce a less expensive analysis tool than empirical methods such as the Finite Element Method (FEM) that allows the accurate study of threaded joints. The FEM analyses help in monitoring the developing of pre-tightening load and accompanying stresses in threaded joints. This could lead to minimize testing in vehicle industry [3].

Several researchers conducted FE analyses to evaluate the performance of the threaded joints. For example, Mathurin et al. [5] presented a 3D FE model of thread forming screw during an assembly process, and compared the FE simulations of the radial displacement of thread material with experimental analyses. Moreover, Cardoso et al. [6] performed FE analyses to predict the portion of external load distributed to the bolted joint. Recently, Liu et al. [7] numerically studied, using ABAQUS and MATLAB, the effects of thread fit quality and friction coefficient on the axial load distribution of the bolted joint.

Yu et al. [8] and Yu and Zhou [4] compared the torque-force relationship obtained using the FE analyses with ABAQUS to those obtained using several theoretical models, such as Motosh model and Yamamoto model, and got very good agreement between the FE results and the theoretical results. In addition, they studied the effects of friction between the bolt and the nut on the torque-force relationships. The authors also concluded that the higher the friction the lower the tightening force is. They also studied the effects of other factors such as the pitch, the assembly clearance, and the elastic and plastic materials properties of the joint. In their analyses, they used tetrahedral elements (C3D4in ABAQUS) for the bolt mesh and hexahedral elements (C3D8R in ABAQUS) elements for nut.

Due to the extreme complexity of the helical thread, researchers preferred to use tetrahedral elements to model the threaded joint, e.g. Hwang [3]. Others mix tetrahedral and hexahedral elements to model the joints, e.g. Yu et al. [8]. Fukuoka et al. [9] and Rafatpanah [10] succeeded to model threaded joints using only hexahedral elements. They created the thread by stacking layers of elements one on the top of another where each layer is rotated a calculated angle from the one beneath it. The profile of the first layer is created by a cutting plane which is perpendicular to the bolt axis. This modelling technique creates firstly a 2D layer and meshes it, and then it copies that layer and adds a z-coordinate of each node. The nodes of the element on a specific layer are then connected to the corresponding nodes on the upper layer manually. The more the number of layers the finer the generated mesh of the thread is. The same procedure was used to create the nut. For instance, Rafatpanah [10] created such thread model by manually connecting the nodes on each two successive layers; that is a tedious work.

Therefore, it can be concluded that the mesh in the threaded joint was created automatically using tetrahedral elements or using mixed tetrahedral-hexahedral elements or manually using only hexahedral elements. Now, the question is: does the modelling technique affect the torque-force of threaded joints? Answering this question is the aim of the current study.

In the current study, FE analyses with ABAQUS were carried out on M8 × 1,25 threaded joint. Several modelling techniques with different element type have been used. The torque-force relationship obtained from the FE analyses was validated with experimental results obtained from the literature and with theoretical results. Good agreement exists between FE, experimental, and theoretical results. Mesh study was carried out on the FE models. In addition, the effects of the coefficient of friction on the torque-force relationship have been addressed.

2 THEORETICAL ANALYSIS

2.1 Thread Geometry, Dimensions, and Materials

The current study was carried out on M8 × 1,25 thread. The ISO standard profile for the metric thread is shown in Fig. 1 for bolts and nuts. The thread angle and the helix angle are shown in Fig. 2. The dimensions, definitions, and values for the M8 × 1,25 thread terms are given in Tab. 1.

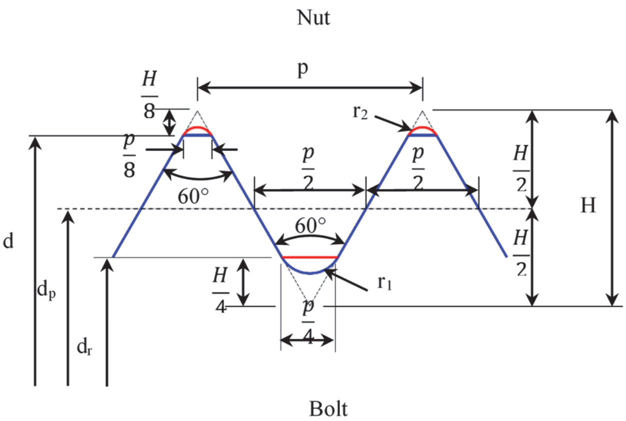


Figure 1 ISO standard profile for metric thread [11]

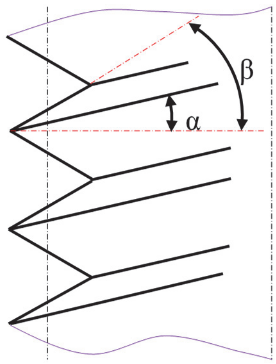


Figure 2 Illustration of the thread angle *β* and the helix angle *α*

Table 1 Definitions and values for an M8 × 1,25 thread terms

Dimension	Definition	Expression	Value for M8 × 1,25
d	Screw nominal diameter	8 mm	8 mm
p	Pitch	1,25 mm	1,25 mm
H	Screw theoretical height	$H=\frac{\sqrt{3}}{2}p$	1,083 mm
d_p	Pitch diameter	$d_p = d - 2\times\frac{3H}{8} = = d - \frac{3\sqrt{3}}{8}p$	7,188 mm
d_r	Bolt root diameter	$d_r = d - 2\times\frac{5H}{8} = = d - \frac{5\sqrt{3}}{8}p$	6,647 mm
r_1	Bolt root fillet	$r_1 = \frac{p}{4\sqrt{3}}$	0,180 mm
r_2	Nut root fillet	$r_2 = \frac{r_1}{2} = \frac{p}{8\sqrt{3}}$	0,09 mm
α	Thread helix angle	$\alpha = \tan^{-1}\left(\frac{p}{\pi d_p}\right)$	3,17° [12]
β	Half of the thread angle	$\beta = \frac{60°}{2}$	30°
r	Radius of bearing contact area between the bolt head in the joint	$r=\frac{1}{2}\left(\frac{3}{2}d\right)=\frac{3}{4}d$	6 mm

2.2 Theoretical Models to Relate the Torque *T* to the Clamping Force *F*

Fortunately, several theoretical models are well developed to relate the torque T applied to threaded joints to the axially clamping force F developed in the joint components. Definitions and values of the geometrical parameters used in the following theoretical models are given in Tab. 1. The first theoretical model to consider is presented by Motosh [13] where the relationship between T and F is given by:

$$T = F\left(\frac{p}{2\pi} + \frac{\mu_{\text{th}} d_{\text{p}}}{2\cos(\beta)} + r\mu\right) \tag{1}$$

where μ is the cofficient of friction between the bolt head and the joint bearing area, and μ_{th} is the coefficient of friction between the bolt thread and the nut thread. According to VDI 2230[14], the relationship between T and F is given as:

$$T = F\left(0{,}159p + 0{,}578\mu_{\text{th}} d_{\text{p}} + r\mu\right) \tag{2}$$

According to ISO 16047[15], the relation between T and F is given as:

$$T = F\left(\frac{1}{2}\frac{p + 1{,}154\pi\mu_{\text{th}} d_{\text{p}}}{\pi - 1{,}154\mu_{\text{th}}\frac{p}{d_{\text{p}}}} + r\mu\right) \tag{3}$$

3 EXPERIMENTAL WORK FOR VALIDATION

Numerical analyses which were carried out in the current study were validated by the experimental results presented by Swissi et al. [16]. They related the torque-force for a threaded joint that consists of a low carbon steel (S235 JR) nut and SAE 8,8 bolt. The bolt and nut are of size M8 × 1,25. Swissi et al. [16] designed and manufactured a testing device (see Fig. 3), which enabled them to apply torque to the bolt and to measure the axially clamping force in the joint. They ran their experiments on machined and formed nut samples. The nut (specimen in Fig. 3) was fully fixed and the bolt (screw in Fig. 3) was only allowed to rotate about its axis. As the bolt rotates against the nut, their threads slide over each other and the bolt tends to move axially. This axial movement of the bolt is prevented by a force sensor such that it can read the value of the force. A cylindrical block is located between the end of the bolt and the force sensor to protect the force sensor from damage due to friction with the end of the bolt. Friction between the end of the bolt and the cylindrical block was minimized by reducing the contact area between them.

4 FINITE ELEMENT MODELLING

4.1 Bolt Modelling

Different modelling techniques can be used to model threads in threaded joints. In this article, the first technique is referred to as *solid revolved threading*. In that technique, the thread profile was solidly revolved around a solid cylinder to create the bolt as shown in Fig. 4. By using that threading technique, both the thread and the cylinder can be meshed using hex elements (the brick) or by using hex-dominated elements. In order to simplify the meshing technique, the boundaries between the thread and the cylinder are kept. Other boundaries that were found between the edges of the thread cannot be merged with each other. All the bolts that are used in the current study are modelled using the *solid revolved threading* technique.

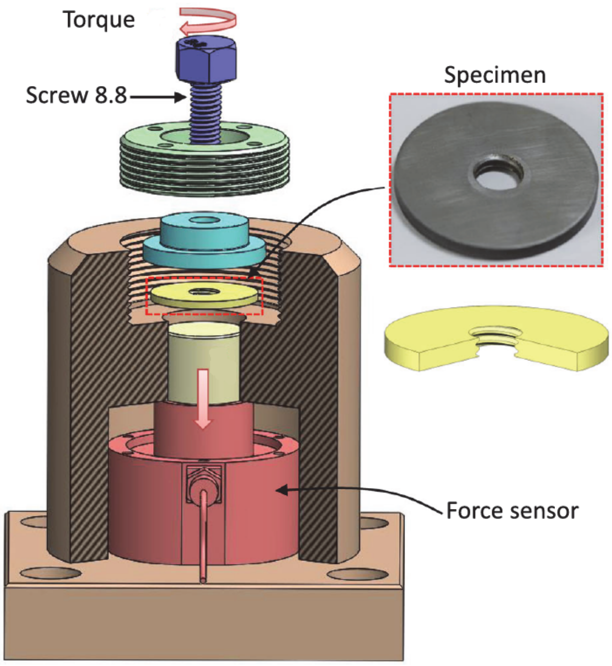


Figure 3 The device used to relate the applied torque to the axially induced force [16]

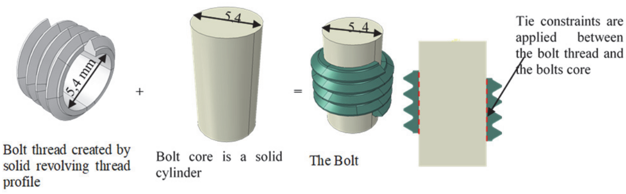


Figure 4 The bolt thread is tied to the bolt core to create the bolt

4.2 Nuts Modelling

For modelling the nut threads, different modelling techniques were used. The *solid revolved threading* technique that is used for the bolts is also used for nuts. However, in the case of nuts, the thread profile was solidly revolved inside a hollow short thick cylinder to create the nut. This short cylinder is then tied with a backing material to act as a support for the nut. Fig. 5 shows the solid revolved nut thread created inside the nut cylinder that is supported by a backing material.

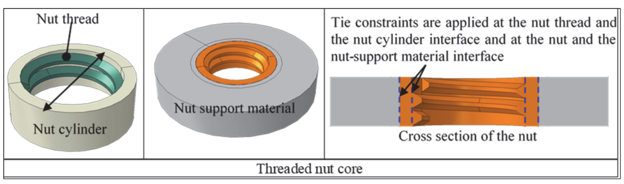


Figure 5 Components of nut model: nut thread, nut cylinder and nut backing

The solid revolved threads were modelled either with fillet at the thread root as shown in Fig. 6a or without fillet at the thread root as shown in Fig. 6b. In addition, the solid revolved thread was modelled with fillet at the thread root and additional support material was created simultaneously with the thread profile as shown in Fig. 6c. Instead of revolving the thread profile, the thread was modelled by revolving the gap between the threads. This technique is referred to as *inverted solid revolved thread*. The inverted solid revolved thread is shown in Fig. 6d.

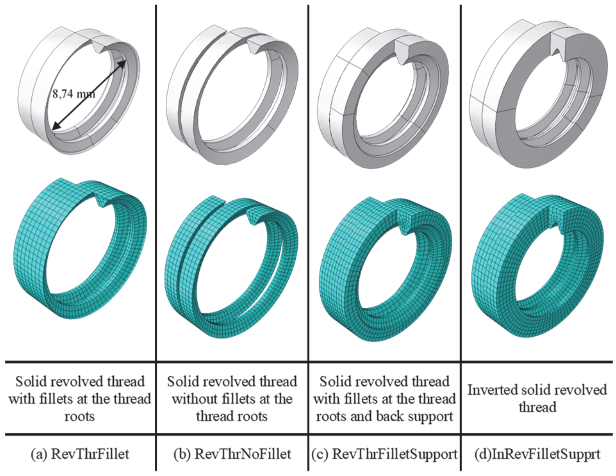


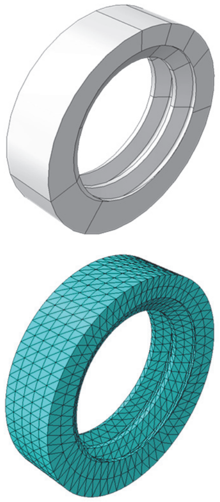
Figure 6 Solid revolving modelling techniques to model nuts

The second modelling technique that is used to model the nut thread is referred to as *cut revolved threading* technique. In that technique, threads are created by cut revolving the thread profile into a hollow short cylinder to create a nut thread. Although this modelling technique resembles the thread manufacturing process, the produced model cannot be meshed using the hex (nor the hex-dominated) elements. It can only be meshed using the Tet elements. Several partitioning techniques were applied to the bolt in order to mesh it using hex (or hex-dominated) elements but they are unsuccessful. The cut revolved threading technique is shown in Fig. 7. Tab. 2 summarizes the thread modelling techniques.

Table 2 Summary of thread modelling techniques

	Model description	Model Name	Thread elements
Bolt	Solid revolved thread *without* fillet at the thread root	RevThrdNoFillet	Hex or Hex-dominated elements
Nut	Solid revolved thread *without* fillet at the thread root	RevThrdNoFillet	
	Solid revolved thread *with* fillet at the thread root	RevThrdFillet	
	Solid revolved thread *with* fillet at the thread root and support material	RevThrdFilletSupport	
	Solid revolved of the profile inverse *with* fillet and support material	InvRevFilletSupprt	
	Cut revolved thread into a solid cylinder	CutRevTetMesh	Tet elements

In this article, the mechanical response of a threaded joint, which was modelled using different modelling techniques, was investigated. The mechanical response of the joint was evaluated by plotting the relationship between the external moment applied to the joint and the corresponding axial force developed in the joint component.



Cut revolved thread (CutRevTetMesh)

Figure 7 Cut revolved modelling technique to model nuts

4.3 Threaded Joint Assembly

Fig. 8 shows the assembly of the threaded joint used in the current study. It can be seen that the bolt model consists of the bolt thread and the bolt core. The bolt thread is fixed to the bolt core using the *Tie* constraint defined in ABAQUS. The tie constraints constrain all the degrees of freedom of the bolt thread to that of the bolt nut. As mentioned, this bolt model was used in all the analyses that were carried out in the current study. In addition, the nut consists of the nut thread (not shown in Fig. 8) that is tie constrained to a short cylinder (shown with fine mesh in Fig. 8). This cylinder is surrounded by support material.

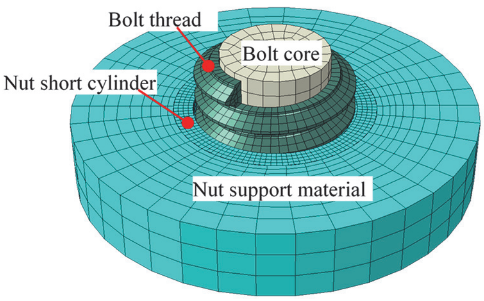


Figure 8 Assembly of the threaded joint model

4.4 Boundary Conditions and Interactions

Fig. 9 shows the boundary conditions applied to the threaded joint assembly. In the interaction module, in ABAQUS, a reference point was created at the centre of the upper face of the bolt core. A kinematic coupling constraint, as described in ABAQUS, is used to connect that reference point to the upper surface of the bolt core. This reference point, and hence the whole bolt, was allowed to only rotate about its axis, i.e. not translational motion was allowed to the bolt. A rotational displacement of π rad imposed on the reference point allowed the bolt to rotate against the fixed nut. The nut thread is tie constrained to the nut cylinder that is tie contained to the nut support material. All the outer surface of the nut body was fully fixed in all directions.

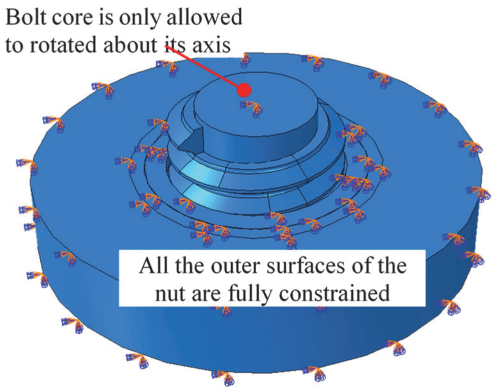


Figure 9 Boundary conditions applied to the threaded joint assembly

The contact surfaces of the bolt thread and the nut thread are shown in Fig. 10. The upper surface of the bolt thread was defined as the master surface while the lower surface of the nut thread was defined as a slave surface. A "surface to surface", as designated in ABAQUS, contact interaction was created between the master and the slave surfaces. The effects of the coefficient between the contact surfaces have been investigated by running analyses at coefficients of friction of 0,05, 0,1, 0,2, and 0,43. The FE coefficient of friction is referred to as μ_{FE}. It is worth mentioning that the FE coefficient of friction μ_{FE} is equivalent to the combined coefficient of frictions at the thread μ_{th} and that at the bolt head μ.

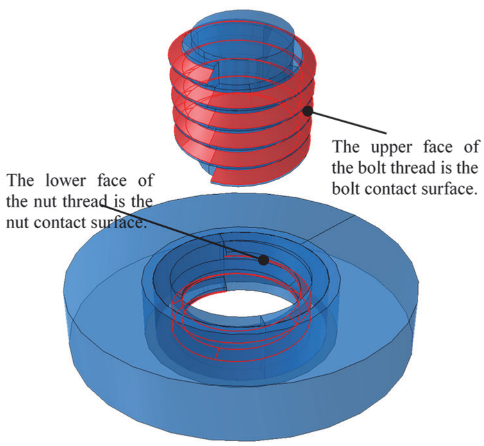


Figure 10 Contact surfaces of the nut and the bolt

5 RESULTS AND DISCUSSION

A set of FE analyses was carried out to evaluate the performance of threaded joints when modelled using different techniques. In all of those analyses, the bolt model was kept the same, which consists of a solid revolved thread that is tied to a cylinder. Each nut has only two threads. The nut was modelled using different techniques that are summarized in Tab. 2.

5.1 Mesh Study

Firstly, a mesh study was carried out in order to find the effects of the nut element size on the torque-clamping force relationship. The solid revolved without fillet thread modelling technique (*RevThrdNoFillet*) was used in the mesh study. Linear hexahedral elements of type C3D8R and linear wedge elements of type C3D6, as designated in ABAQUS, were used to mesh the nut. The bolt was meshed using the linear hexahedral elements of type C3D8R. The global element sizes of 0,2 mm, 0,3 mm, 0,4 mm, and 0,5 mm were used to mesh the nut. The element size of 0,38 mm was used to mesh the bolt in all of the analyses. The coefficient of friction of 0,43 was used in the mesh study; however, other values of it were used later in investigating the effects of friction on the torque-clamping force relationship. It was found that the relationships between the applied torque and the clamping force for the four meshes are identical (see Fig. 11). Moreover, since the analyses were elastic analyses, the torque-force relationships were found to be linear. However, the effects of mesh on the central processing unit (CPU) time is quite significant. Tab. 3 compares the number of elements used in each mesh along with its corresponding CPU time.

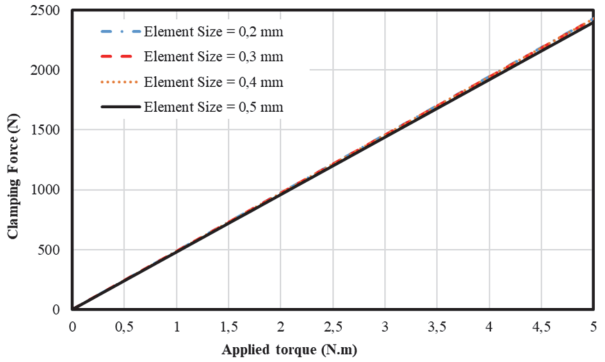


Figure 11 Torque-force relationship for the solidly revolved thread without root fillet (*RevThrdNoFillet*) using different element sizes for the nut

Table 3 Effects of mesh element size on the CPU time

Element Size / mm	Number of element of the nut	Total number of elements of the model	CPU Time / s
0,2	12793	20479	3510
0,3	4260	11946	1712
0,4	2296	9982	1331
0,5	1148	8834	953

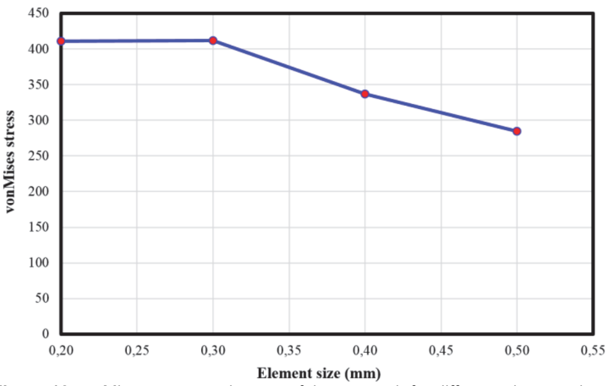


Figure 12 vonMises stress at the root of the nut teeth for different element sizes

Although the torque-clamping force relationship is independent of the element size, see Fig. 11, the stresses at the root of the nut teeth are dependent of the element size, see Fig. 12. It can be seen that the larger the element size the smaller value of the vonMises stress at the teeth root. However, for element sizes of 0.2 mm and 0.3 mm the values of vonMises stress are almost equal. This indicates that it is appropriate to use element size of 0.3 mm or less. The finer element size could result in higher CPU time, see Tab. 3.

5.2 Theoretical Results against Experimental Ones

Eqs. (1), (2) and (3) were used to find the theoretical torque-force relationships. The value of the coefficient of friction at the bearing surface between the bolt and the joint (μ) was assumed to vary between 0,1 to 0,3 and the corresponding values of the thread coefficient of friction μ_{th} was obtained. Many researchers adopted this value of μ, e.g. Hwang [3] and Swissi et al. [16]. The Excel solver technique was used to compare the theoretically calculated torque-force relationship to the experimental results. The objective function of the solver was to minimize the mean root square of the error between the average theoretically calculated and the experimentally obtained values of the force at specific value of the applied torque. Fig. 13 compares the torque-force relationships obtained from the three theoretical models, at $\mu = 0,2$ and $\mu_{th} = 0,18$, with the experimental results. It can be seen that all the theoretical models resulted in identical curves.

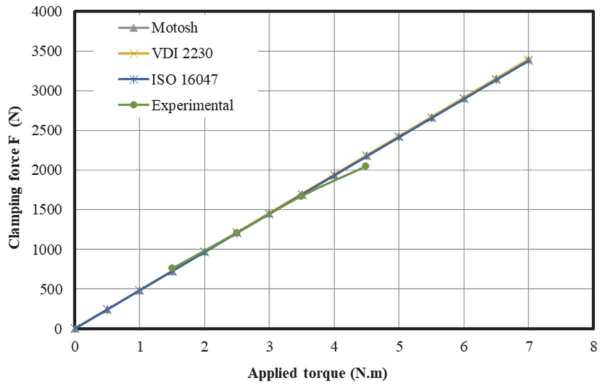


Figure 13 Theoretical torque-force relationship

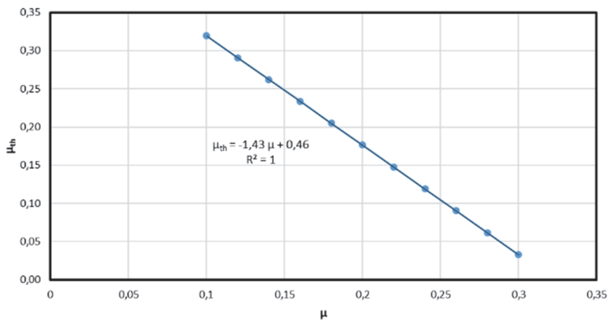


Figure 14 Relationship between the coefficient of friction at bearing area μ and the coefficient of friction at thread μ_{th}

Fig. 14 compares the values of μ to those of μ_{th} such that the resulted torque-force relationships agree well with the experiment. It can be seen that as the μ_{th} increases, the value of μ should be decreased and vice versa. It is worth noting that, if μ was set to zero, the value of μ_{th} would be

0,46. Setting the value of μ to zero is only available in the FE modelling as will be seen in the next sections.

5.3 Effects of Friction

The FE models given above consider only the friction between the threads of the nut and the bolt. Therefore, an equivalent coefficient of friction should be used in the modelling. This coefficient of friction is referred to as μ_{FE}.

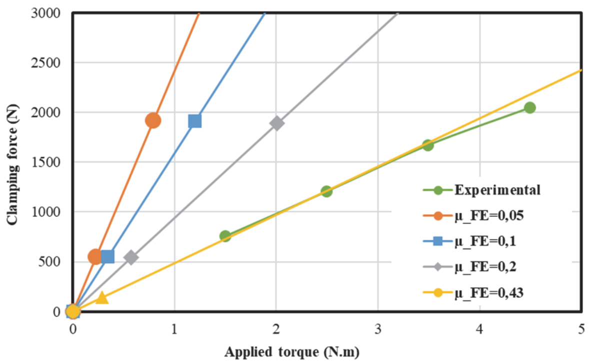


Figure 15 Effects of friction on the torque-force relationship

Fig. 15 compares the FE torque-force relationship for different values of μ_{FE} with the experimental results. The *RevThrdNoFillet* threading technique was used with element size of 0,3 mm. It can be seen that as the value of μ_{FE} coefficient of friction increases, the axial force induced in the joint decreases for the same applied torque. This can be attributed to the higher torque (energy) needed to overcome the frictional resistance during tightening the bolt. This result is supported by the finding obtained by Yu et al. [8]. Moreover, it can be seen that the experimental results agree well with coefficient of friction of 0,43. This value of friction seems too high. However, it is worth noting that this is an equivalent value of both μ and μ_{th}. This result highlights the importance of studying the FE models to check whether the contact at the bearing area was considered in the analysis along with contact at the threads.

5.4 FE Torque-Force Relationship Versus Experimental

Fig. 16 compares FE torque-force relationships to the experimental torque-force relationship that is given in [16].

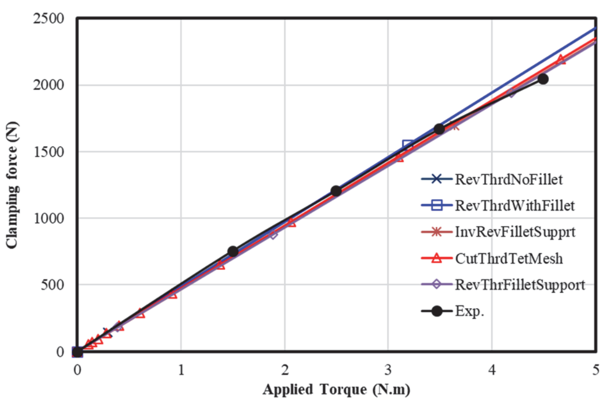


Figure 16 FE torque-force relationship compared to the experimental results obtained from [16]

The FE results are obtained from different thread modelling techniques. The thread of the nut in all FE models is meshed using elements of global size of 0,3 mm. The FE coefficient of friction μ_{FE} of 0,43 was used in these analyses. It can be seen that all the FE results are consistent and in good agreement with the experimental results. The consistency of the FE results allows users to apply any of the thread modelling techniques given in this article to model threaded joints. However, for the sake of simplicity and ability to model complex threaded joints, the *CutRevTetMesh* threading technique is recommended.

6 CONCLUSION

Different modelling techniques can be used in modelling the threaded joints. This article focused on thread generating techniques in the nut of an M8 × 1,25 steel joint. The relationship between the applied torque and the resulting clamping force was obtained for different modelling techniques. The following conclusions can be drawn:

- The torque-force relationship was found to be independent of both the modelling technique and the mesh size.
- The mesh study revealed that an element size of 0,3 mm is satisfactory in such analyses.
- When modelling the thread only, an equivalent coefficient of friction μ_{FE} should be used. The value of this coefficient can be approximated by the sum of μ and μ_{th}.
- Similar to the theoretical analyses and the experimental results, the FEM resulted in a linear torque-force relationship, in good agreement with the experimental one.

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