

SIMULATION OF BOLT PRESTRESSING IN STEEL SHEAR CONNECTIONS

Stergios Nasikas, Dimitrios Faidon Kallias Ntroumpis, Apostolis Koukouselis and Euripidis Mistakidis

Laboratory of Structural Analysis and Design, Department of Civil Engineering
University of Thessaly, Volos, GR-38334, Greece

e-mail: emistaki@uth.gr ; web page: <http://lsad.civ.uth.gr/>

Keywords: Prestressed bolts, shear connection, nonlinear analysis, unilateral contact-friction, finite element analysis

Abstract. *The use of prestressed bolts is an effective method for the connection of the highly stressed structural components under alternating loads. The friction forces which develop as a result of the prestressing on the interface of the connected plates contribute to the undertaking of externally applied shear forces. The process of prestressing is a complex phenomenon with many uncertainties and not well understood aspects, addressing capacity of the bolted joints to carry loads, the stress and strain distributions in the connected parts and the methods used for prestressing. The main purpose of this paper is the finite element simulation of a prestressed bolted shear connection during the prestressing process. The model consists of two plates that come into contact when a prescribed rotation between the bolt and the nut is applied. The problem at hand is highly nonlinear as it involves: contact and friction between the surfaces of the connected elements, geometric nonlinearity that stems from the large deformations of the bolt, material nonlinearity due to the elastoplastic behavior of the joints elements.*

1 INTRODUCTION

In cases of steel structures with high demands at alternating loads, the relative displacements in joints between the bolt and the plates should be limited. In such cases prestressed bolts are used in order to ensure that slippage between the connected parts of joint can be prevented. In general, two methods are used for the execution of the prestressing a) the “moment method”, in which the bolt is tightened with a prescribed torque and b) the “rotation method”, in which the bolt is tightened till a prescribed rotation between the bolt and the nut is achieved.

In this paper a high strength M24 bolt was simulated with a 3D finite elements model. For the prestressing process the ‘rotation’ method was chosen. The main purposes of this paper were to compare the results of a tension and a torque test, to estimate the effect of the encircled elements such as plates and/or washers to the results, to estimate the effect of the friction coefficient to the results and to compare the axial force and the reaction moment with the values which the codes suggest. In addition, the results of the analysis provide information for the stresses and strains that develop at any point of the connected bolt during the whole procedure.

The related references either provide us with information according only to experimental results [1],[2] or introduce the bolted joint behavior without simulating the prestressing process [3]. Until now, none full scale 3D finite elements model had been created for the simulation of the actual prestressing process, taking into consideration both the material and the geometric nonlinearity.

2 SIMULATION

The simulation of the prestressing procedure was conducted using 3D Finite Element (FE) models. For the design of the geometry of each part of the connection, the Solidworks [4] software was used. Then the geometry was imported to MSC Patran [5] for meshing, in “parasolid” format. Finally the meshed part was imported to MSC Marc [6] which performed the numerical analysis.

2.1 Geometry

For the understanding and interpretation of the prestressing process, three different models were created (Figure 1):

- Model 1 - Bolt and nut. In this model the “y” displacement of the head of the bolt is restrained. This allows focusing on the interaction between the threaded parts of the bolt and the nut.
- Model 2 - Bolt, nut and two plates.

- Model 3 - Bolt, nut, two plates and two washers.

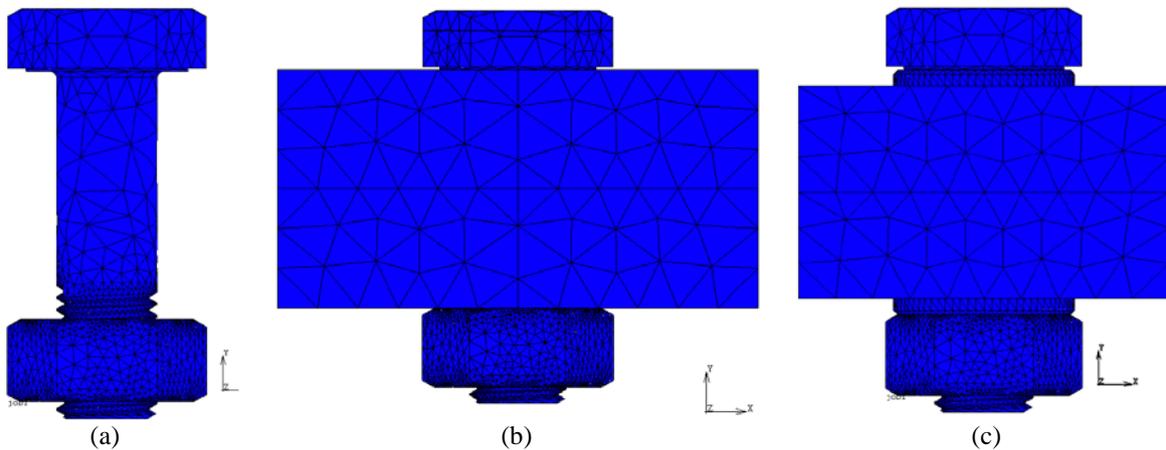


Figure 1. Finite element models of bolt-nut assemblages a) Model 1, b) Model 2 and c) Model 3).

Each part of the models was designed following the guidelines of DIN (DIN 6914 – bolt [7], DIN 6915 - nut & washers [8] and DIN 13.1 – thread [9]). The simulated models consists of a M24 bolt with a length of 90mm and the corresponding M24 nut. The gripping length is 60mm and the thread above the nut is 12mm. Models also include two plates and two nuts (see Figure 1)

The threaded part of the bolt was modified by converting the curved parts of the profile of the thread into piecewise linear and by excluding thread run-out (Figure 2 a,b).

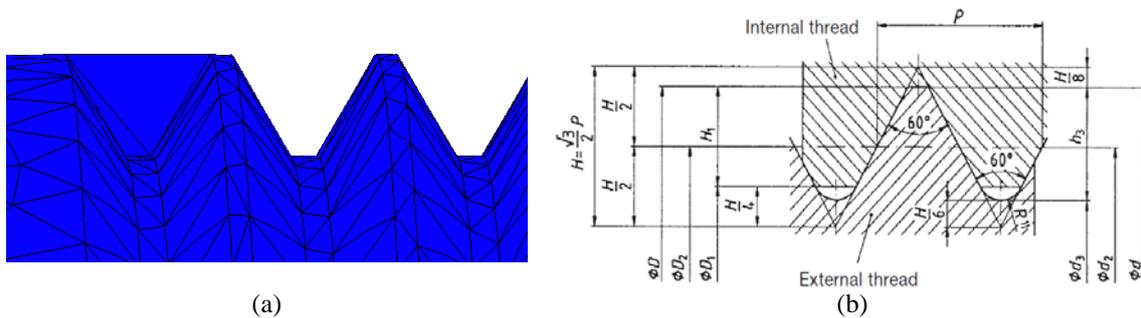


Figure 2. a) Modified thread profile with no thread run-out b) thread profile according to DIN 13.1.

The prestressing procedure was simulated using the “controlled rotation method” by applying a 360° rotation to the head of the bolt. The boundary conditions applied prevent the displacement of the plates on the x-z plane (see Figure 2) and fix lateral faces of the nut.

2.2 Mesh

The meshing procedure of the model was carried out in MSC Patran, using tetrahedral finite elements, of type tetra10. Key aspects of the meshing procedure were the representation of bodies without distortion of their geometry along with a number and size of elements capable of providing reliable results, but without disproportionate increase of the analysis time. Depending on the complexity of the geometry of each part and the required accuracy of the results, different meshing options were used. The numbers of FE and nodes for each part of the assembly are given in Table 1.

	Bolt	Nut	Plate	Washer
Elements	29707	48466	2438	1555
Nodes	44560	79126	4334	3204

Table 1: Number of finite elements and nodes

2.3 Material Properties

For the various parts of the bolt, grade 10.9 steel was used, while for the plates S355 steel was considered. The Young’s Modulus was set equal to 210 GPa and the Poisson’s ratio equal to 0.3. The material laws of the two steel qualities used are given in Figure 3.

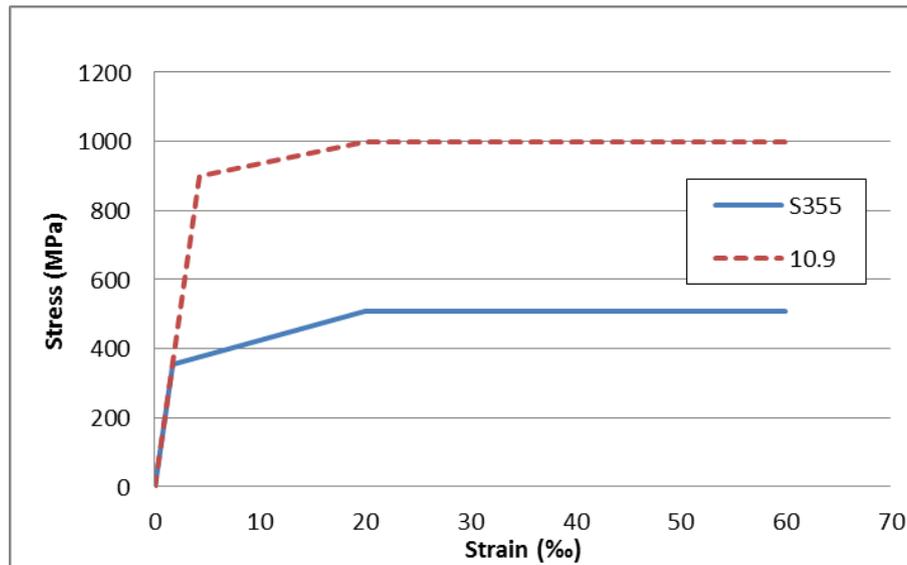


Figure 3. Stress-strain diagram for steel 10.9 and S355

2.4 Contact Properties

The effect of friction on the prestressing procedure was investigated through three cases of different friction coefficients “ μ ”. In the first two cases a uniform friction coefficient was selected for all the contacting bodies ($\mu=0$ and $\mu=0.2$). In the last case, a friction coefficient of $\mu=0.5$ was selected for all the contacting bodies, except of the treaded parts where the value 0.16 was considered. The analysis was carried out using a Coulomb friction model. Double-sided contact was considered between the potential contacting surfaces according to a node to surface algorithm [10].

3 RESULTS

As rotation is applied to the fastener during the prestressing procedure, the portion not resisting to friction will cause elongation of the body of the bolt, while the portion that resists to friction will cause the bolt to twist slightly. As a result of this tightening procedure, a combined tension-torsional stress condition is achieved in the bolt. This fact is lowering the capacity of the bolt to carry axial loads and its effect becomes more evident as the friction between the internal and external threads increases.

In the beginning of the loading the bolt performs elastically and the stresses increase linearly. When the stresses exceed the yield point, plasticity starts spreading throughout the bolt, until a specific cross section has been completely plastified. This means that deformation increases rapidly with a small increase of the applied force. As the deformation increases, stresses at all the points of the plastified cross section tend to reach their tensile strength. At this point the axial force of the bolt reaches its maximum value. The stress distribution is now uniform and equal to the tensile strength of the material and the deformation increases without any force increase. The plastic strains that develop at the bolt during the tightening procedure are presented in Figure 4.

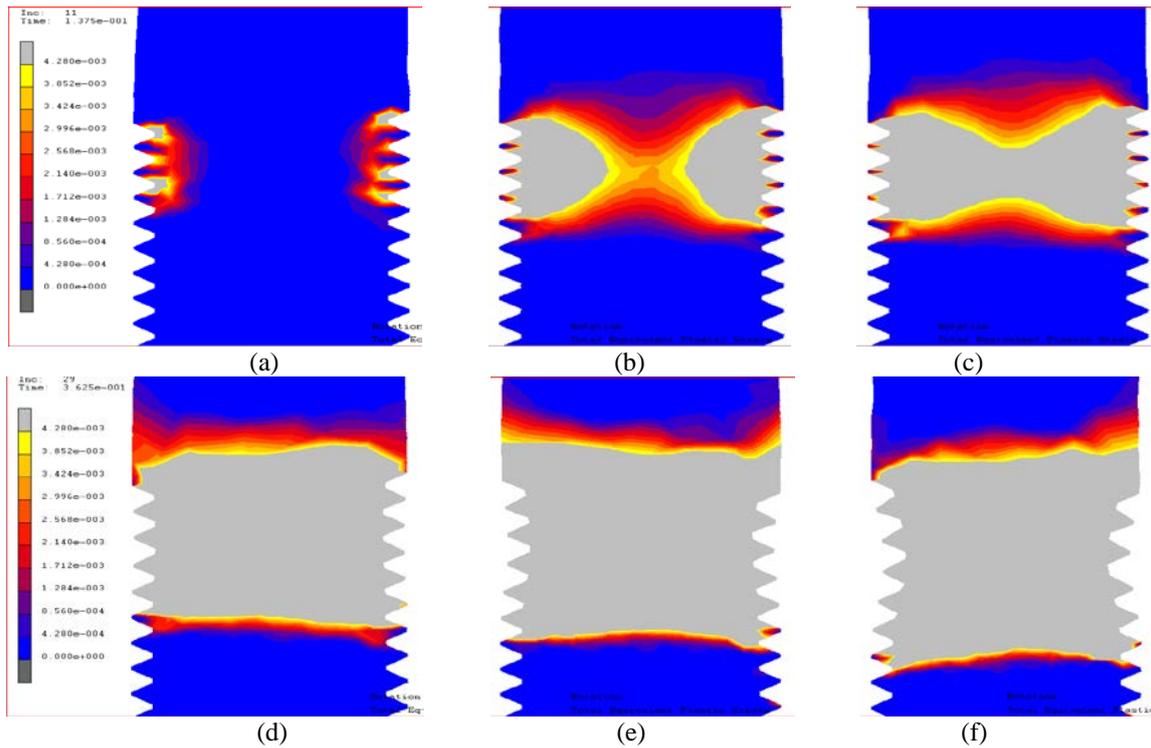


Figure 4. Equivalent plastic strain in the treaded part of the bolt a) first plastic strain develop (49.5°), b) spreading of plasticity (63°), c) first fully plastified cross section (67.5°), d) end of hardening (130.5°), e) design suggested rotation (240°), f) end of analysis (360°).

Then a narrowing of the cross section occurs at the area of the thread where the break will eventually happen (Figure 5). Since the stress distribution is uniform and the cross sectional area is decreasing, the axial force carried by the bolt follows this decrease. This means that an increase in deformation will acquire a lower value of force.

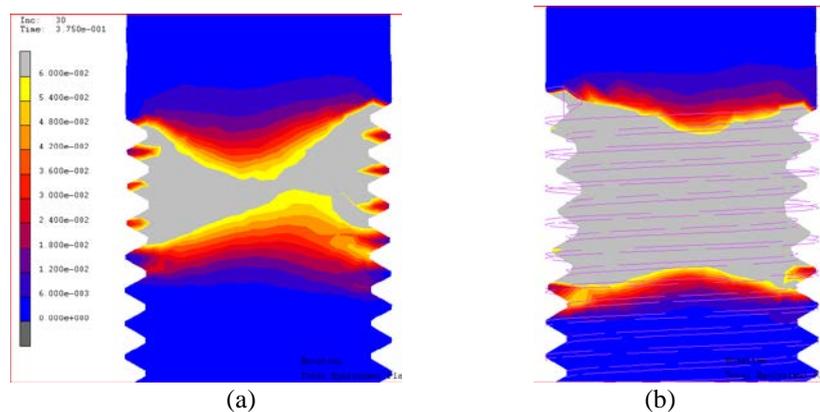


Figure 5. Strain states: a) stresses throughout the cross section have reached the ultimate strength of the material, b) end of analysis (360°)-necking of the cross section

Figure 6 presents the prestressing force versus rotation diagram for the Model 3 assemblage, for a friction coefficient of $\mu=0.2$. The bolt behaves linearly in the beginning of the process until it enters the plastic zone. Then the force increases until the maximum value is reached. After that point the force starts decreasing due to the necking that has occurred at the cross section of the bolt.

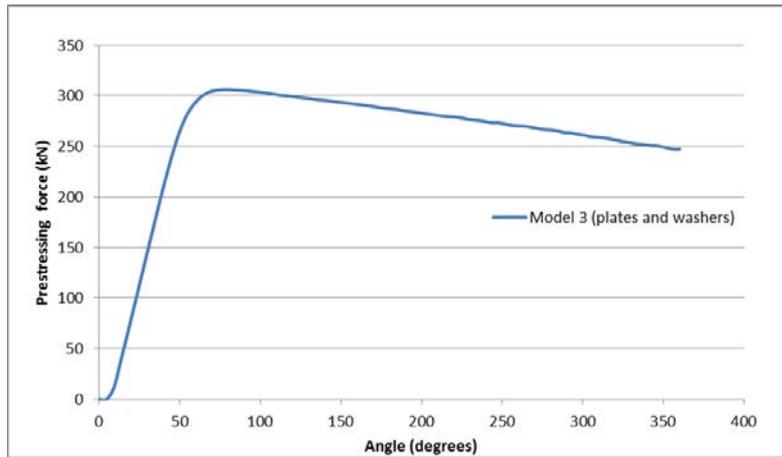


Figure 6. Prestressing force versus angle rotation.

The stresses that develop during the prestressing process are given in Figures 7 and 8.

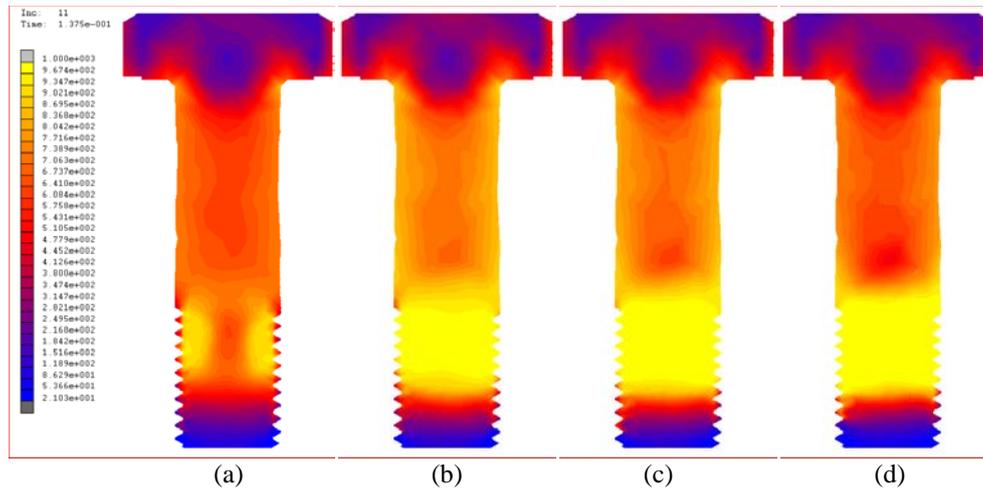


Figure 7. Equivalent Von Mises stress of the bolt at a) end of elastic zone (49.5°), b) first fully plastified cross section (63°), c) first cross section reaching the tensile strength of the material (130.5°), e) design suggested rotation (240°)

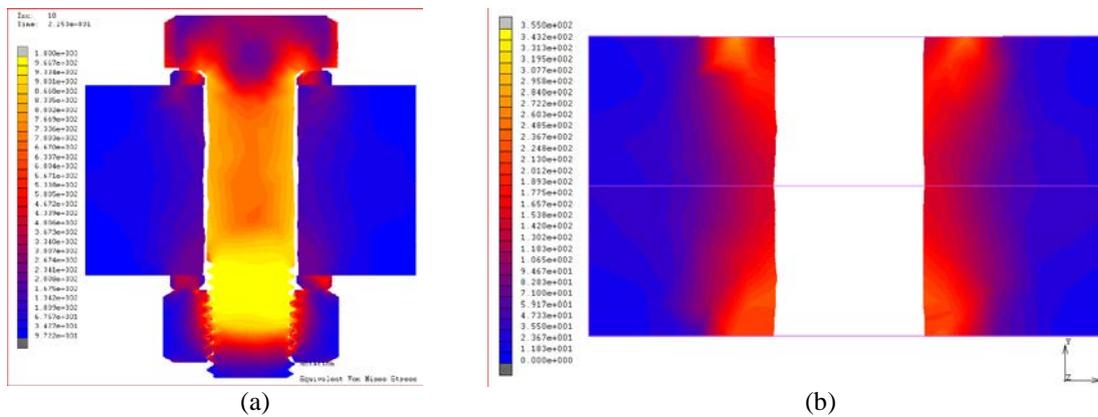


Figure 8. a) Equivalent von Mises stresses at: a) the bolted joint, b) the plates

In this paper, in order to describe the behavior of the bolt during the prestressing process, the load versus elongation relationship is used. As mentioned above a bolt in a direct tension test exhibits different behavior than in a torqued tension one. Figure 9 presents the force versus elongation diagrams for three cases: a) the bolt in direct tension, b) the bolt in torqued tension (Model 1) with no friction ($\mu=0$) and c) the bolt in torqued tension with friction coefficient $\mu=0.2$ between all the contacting surfaces. The numerical tests were performed till an elongation corresponding to half thread pitch was reached (1.5mm). This actually corresponds to rotation of the bolt head by 180° .

In torqued tension tests there is a reduction in both ultimate load and elongation, compared to the values determined from a direct tension test. As expected, the bolt in direct tension reaches the highest axial load and elongation. The effect of friction becomes apparent, as the bolt in torqued tension approaches the behavior of the direct tension test when no friction is applied, whereas there is a large difference in the results as the friction coefficient increases.

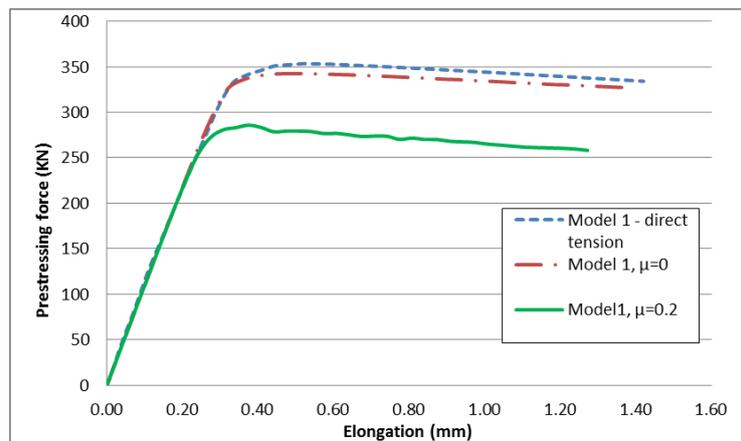


Figure 9. Axial force versus elongation relationship for direct tension and torqued tension ($\mu=0$ & $\mu=0.2$).

In the prestressing process, the applied rotation causes the bolt to elongate while the encircled elements of the connection are in compression. The different stiffness between the parts of the assembly leads to larger deformations in the body of the bolt. The threaded part of the bolt enters the plastic zone, while both plates and washers (if any) remain in the elastic zone. When plasticity occurs, the elongation of the bolt increases rapidly leaving the plates to expand and return to a lower stress condition. The shortening between the gripping length of the bolt versus rotation is given in Figure 10. The highest of the shortening of the gripping length is reached, when the bolt carries the ultimate prestressing load. As noticed in Figure 10, Model 2 with no washers, has a smaller deformation than Model 3 which includes washers. However in both models the deformation of the encircled elements is negligible compared to the elongation of the bolt.

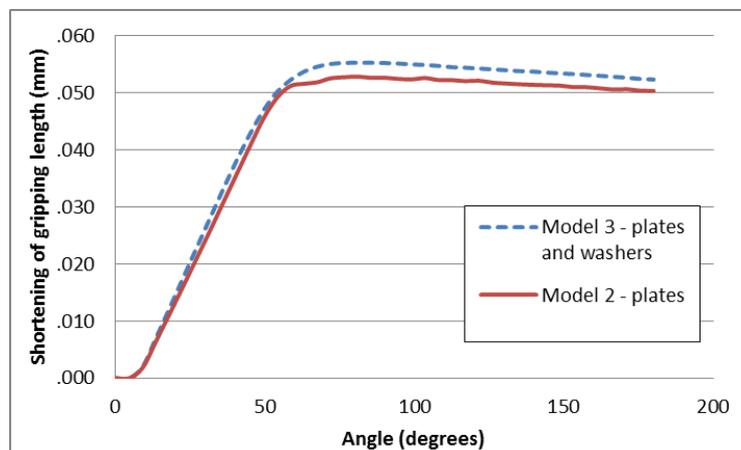


Figure 10. Shortening of gripping length versus angle of rotation diagram

As a result, the prestressing force is not affected by the behavior of the plates and the washers. It is apparent that in Figure 11, both Model 2 and Model 3 exhibit the same behavior in axial load as Model 1. This means that the simplified Model 1, with the assumption of zero vertical displacement of the bolts head, is adequate for the simulation of the bolts behavior.

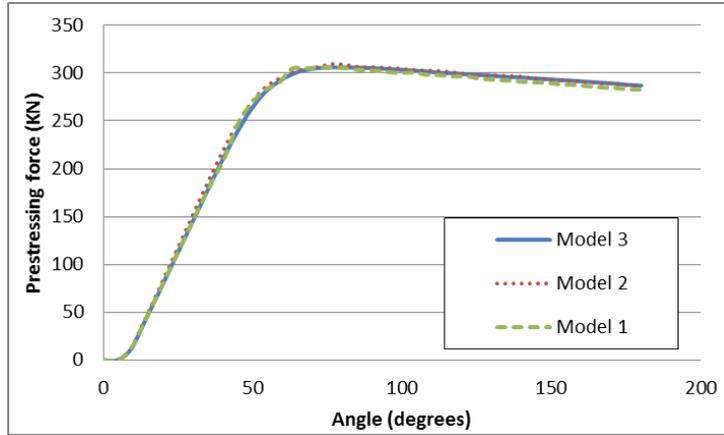


Figure 11. Axial force - angle diagram for the three different models

Comparison of axial force results

Table 2 presents the values of axial force in the bolt. In the second column are given the maximum values obtained from the analysis, while in the third and fourth column give the values suggested from the codes. The calculations below assume a circular cross section, of area A_s , diameter $d_m = \frac{d_3 + d}{2}$ and ultimate tensile strength $f_{ub} = 1000 \text{ MPa}$.

Assuming a circular cross section of d_m in tension at the ultimate load, the resultant force is $F_u = 352.3 \text{ kN}$. This value is very close to the value calculated analytically in the direct tension test (353.4 kN), which means that d_m is the effective diameter.

It is obvious that there is a significant deviation between the values, which is increasing as the friction decreases. However there is still a significant difference as a result of the material overstrength, that cannot be ignored. The axial force during the prestressing reaches higher values than the ones proposed by the regulations [11], which seem to be rather conservative.

Model	Numerically obtained ultimate strength (kN)	Code calculated values (kN) $F_p = 0.7f_{ub}A_s$	Differences between numerically obtained and code calculated values
Direct tension	353.40		
Plates & washers, $\mu=0$	342.50	246.62	38.9%
Plates & washers, $\mu=0.16$	330.30	246.62	33.9%
Plates & washers, $\mu=0.20$	306.00	246.62	24.1%

Table 2: Axial forces obtained by different FEA models, suggested values from Eurocode and differences between them.

Moment

In order to reach the desired prestressing force, the applied torques value has to overcome the resistance of:

- The torque that is caused during the interaction of the internal and external threads that come into contact during the prestressing. The magnitude of this moment is independent of the friction coefficient and is characteristic only of the geometry of the threads and of the mechanical properties of the fastener.
- The moment that results from the friction on the surfaces of the various contacting bodies. Apparently, the calculation of this value is more complex, as it depends on the total area in contact, the contact coefficient " μ " and the normal reaction force on these surfaces.

Two formulas are provided in [12] in order to calculate the required prestressing torque in a joint model of a bolt, a nut and two plates (see also Figure 1). The first formula concerns a theoretically obtained prestressing torque value:

$$M_{theoretical} = P_i \frac{d_m \mu_0 \sec\theta + t g \alpha}{2 (1 - \mu \sec\theta \tan\alpha)} + \mu P_i \frac{d_{m\pi}}{2} \quad (Nm) \quad (1)$$

with:

P_i is the required prestressing force

$$d_m = \frac{d_3 + d}{2}$$

θ is the thread profile angle

$$\tan\alpha = \frac{p}{\pi d},$$

p is the thread pitch

$$d_{m\pi} = \frac{d_1 + d}{2}$$

The second formula gives an empirical value for the prestressing torque:

$$M_{empirical} = K d P_i \quad (Nm) \quad (2)$$

where: $K = 0.2$, a rotational constant.

In Table 3 are given the torque values derived from the analysis and the calculated ones from the formulas.

Model	Numerically obtained		Formulas	
	Axial Force (kN)	Torque (Nm)	Analytical Torque (Nm)	Empirical Torque (Nm)
Model 1, $\mu=0.00$	342.53	164.36	75.54	1644.14
Model 3, $\mu=0.5$ & $\mu_0=0.16$	330.30	3656.32	2599.43	1585.44
Model 3, $\mu=0.20$	306.00	1965.50	1539.10	1468.82
Model 2, $\mu=0.00$	341.97	165.00	75.42	1641.46
Model 2, $\mu=0.20$	309.53	1432.68	1556.85	1485.74

Table 3: Torque values calculated from different FEA models and suggested values from formulas (1) and (2).

From the data provided in Table 3, becomes apparent the effect of friction on the required prestressing torque. For the models with no friction the formulas lead to irrational values that do not correspond to reality. The same happens when washers are added to the model, where the additional friction surfaces lead to a 24% increase in the required torque value. In the model with the plates, the values calculated approach better the ones of the formulas. However there is still a gap between them, as the values proposed by the formulas lead to a higher design torque (8% and 3.6% respectively).

The design prestressing force for a M24 bolt, encircling two plates is $F_p=246.62$ kN. This force leads to required torque values of $M_{theoretical} = 1240.43$ Nm (1) and $M_{empirical} = 1183.78$ Nm (2). These values are significantly lower than the ones calculated analytically, partly due to the higher prestressing force of the bolt, which in turn leads to an average 15.4% increase in the required torque value.

The torque versus - rotation diagram mirrors the axial force - rotation diagram. This is something expected, because the required torque value depends on the friction forces, which in turn depend on the normal reaction force between the contacting bodies. As a result, the moment diagram follows the same upward trend in the elastic zone, reaches its maximum value for the maximum prestressing force and then starts decreasing in a slow rate, as the force is decreasing.

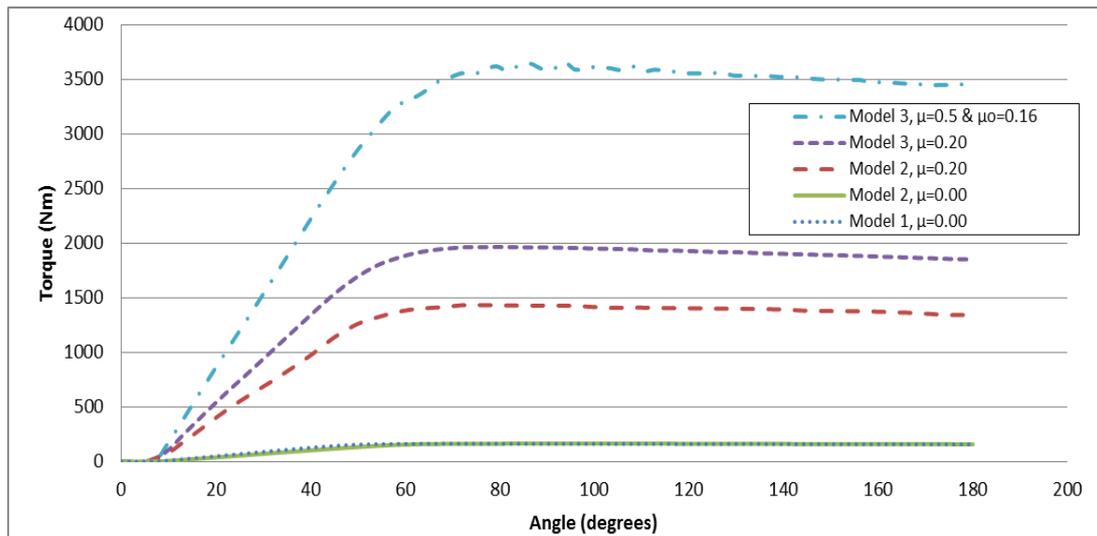


Figure 12. Torque-angle diagram for five different models

4 CONCLUSIONS

The analysis carried out suggests that in a prestressed bolted connection, most of the deformation occurs in the threaded part of the bolt, between the underside of the nut and the unthreaded part of the bolt. Therefore this threaded part determines the behavior of the bolt during the prestressing process.

Friction is another crucial factor for the prestressing process. It affects the capacity of the bolt to carry axial loads, the maximum elongation of the bolt, the required prestressing torque and the stress distribution in the connection. A low friction coefficient leads to high axial forces and elongations in the bolt, while the required torque is decreasing.

The values of design prestressing force and torque provided by the codes differ slightly from the values calculated analytically. More specifically the codes suggest for a M24 bolt with 60mm gripping length the application of a rotation of 240 [13], which causes a design prestressing force equal to 246.6 kN. On the other hand the value provided by the model after that rotation was 273.3 kN, while the maximum value of the prestressing force reached 309 kN. Regarding the design prestressing torque, the codes suggest a value of about 1200 Nm, which is much lower than the values calculated in the models by 15.4%.

5 REFERENCES

- [1] Kulak, G. L., Fisher, J. W., and Struik, J. H. A. (1987), "Guide to Design Criteria for Bolted and Riveted Joints, Chapter 4: Bolts", Second Edition, *Wiley-Interscience*, New York.
- [2] Papanikolas, P. (2005), "Prestressed bolts used at Rio-Antirrio bridge", *Metal Structures*, issue 4. (In Greek).
- [3] Molnár, L., Váradi, K., Liktó, B. (2014), "Stress Analysis of Bolted Joints Part I: Numerical Dimensioning Method", *Modern Mechanical Engineering*, Vol. 4 No. 1, pp. 35-45.
- [4] MSC.Marc 2011, User's Guide, "Volume A: Theory and User Information", MSC Software Corporation
- [5] DS SolidWorks 2013, What's New, "21, SolidWorks Simulation", Dassault Systèmes SOLIDWORKS Corporation
- [6] MSC.Patran 2011, Patran Reference Manual, "Part 3: Finite Element Modeling", chapter 1: "Introduction to Finite Element Modeling", MSC Software Corporation.
- [7] DIN 6914, High-strength hexagon head bolts with large widths across flats for structural steel bolting, 1989.
- [8] DIN 6915, Steel hexagon nuts with large width across flats for high-strength structural bolting, 1999.

[9] DIN 13-1, General purpose ISO metric screw threads, Part 1, “Nominal sizes for 1 mm to 68 mm diameter coarse pitch threads”, 1999.

[10] McCarthy, C.T., McCarthy, M.A., Padhi, G.S. (2002), “Automated Three-Dimensional Finite Element Modelling of Composite Aircraft Bolted Joints - Modelling Issues”, *Worldwide Aerospace Conference and Technology Showcase*, Toulouse, 8-10 April.

[11] Eurocode 3, Design of steel structures - Part 1-1, “General rules and rules for buildings”, 1993.

[12] Kermanidis, A. ,educational material, “Bolts”, University of Thessaly, Department of mechanical engineering, Machine Design Laboratory.

[13] DIN 18800, Part 1, “Structural steelwork – Design and construction”, 1990.