

Research Article

An Analytical Model for Rotation Stiffness and Deformation of an Antiloosening Nut under Locking Force

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Screw fasteners are undoubtedly one of the most important machine elements due to their outstanding characteristic to provide a high clamping force just with a simplified design. However, the loosen vibration is their inherent and inevitable fault. The friction locking approach is one of the basic locking fastener categories by enhancing the bearing load on the contact surface of thread by applying a locking force on an antiloosening nut. This locking force may cause more severe deformation in the nut. The contact stress distribution on the nut would be changed and that can cause the variation of the friction torque for the bolt joint. However, there exists no established design calculation procedure that accounts for the rotation deformation and its stiffness of the antiloosening nut under the locking force. The main objective of the work is to develop an analytical solution to the rotation deformation problem encountered in the antiloosening nut. The proposed model is supported by comparison with numerical finite element analysis of different sizes of joint elements and different applied forces.

1. Introduction

Screw fasteners are undoubtedly one of the most important machine elements. Their design simplicity to provide a high clamping force is the outstanding feature of screw fasteners. However, they have an inherent and inevitable fault: they all loosen eventually by vibration [1–3].

Some researches intended to make clear of the phenomenon of self-loosening of bolt based on the vibration theory [4–9]. However, the popularly known antiloosening fasteners do not seem to possess much resistance to loosening by the test results [10, 11] for the effectiveness of screw threads, spring washers, nylon inserted nuts, double nuts, and eccentric nuts of few sizes to resist loosening. Sase et al. [11] in the year 1998 introduced and evaluated the step lock bolt (SLB) with regard to its antiloosening performance using a displacement based loosening device. They found the presence of desirable antiloosening characteristics of SLB.

The friction locking approach is one of three basic locking fastener categories established by major companies [12, 13]. The mechanism of friction locking is to enhance the friction

force by increasing the bearing load or the friction coefficients of contact surfaces of nut or bolt head and thread. One of the methods to enhance the friction force is to increase the bearing load of the contact surface of thread by applying one force on the nut in outside direction perpendicular to the bearing surface of nut. However, the bearing force and the bending deformation of nut will be influenced by this applied force. The bending deformation of the nut is determined by its bending rigidity that may never be discussed by other researches. In the present study, an analytical model was proposed to evaluate the bending rigidity of the nut under preload. The effects of the different forces produced by the small screw and applied on the nut were investigated on the bearing force and the bending deformation of the nut. The theoretical calculation procedure that considers this effect is presented in detail hereafter.

2. Analytical Model

The three basic elements that form an antiloosening nut discussed in present research are the upper nut, the lower

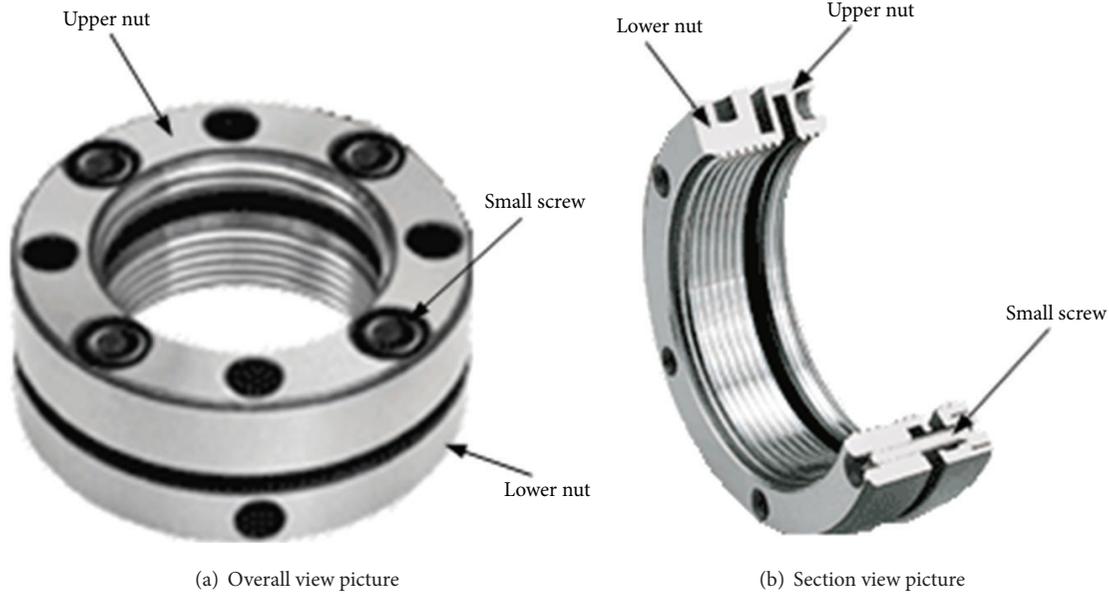


FIGURE 1: Picture of an antiloosening nut.

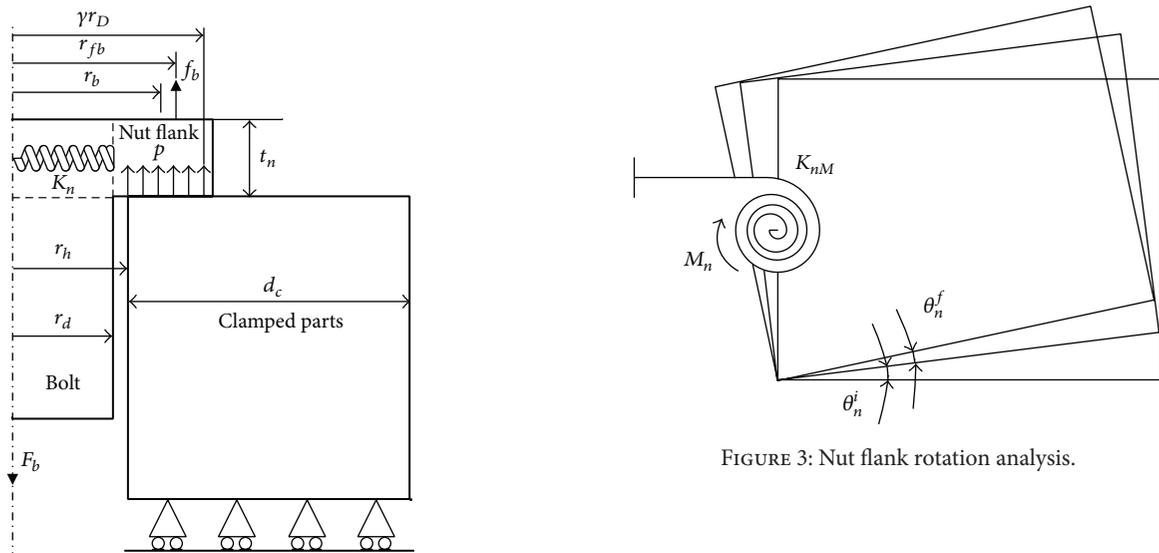


FIGURE 2: Bolted joint model under locking force.

nut, and the small screw, as shown in Figure 1. The lower nut is supported by clamped parts. The small screw is passed through the bolt hole in the upper nut. As the small screw is tightened, the extra large bearing force will be applied on the contact surface of nut threads that can enhance the antiloosening ability of bolted joint. Therefore, the tightening force produced by the small screw can be considered a locking force on the antiloosening nut.

The model used to simulate the joint with the antiloosening nut is shown in Figure 2. For the case involving clamped parts, bolt and nut are included. As with bolt force F_b , when a locking force f_b is applied to the small screw, different

rotation displacements and distortions at the nut flank are generated for the abovementioned joint components.

Before conducting the elastic interaction analyses, it is worth noting that the theoretical reference should be used to treat the different joint elements.

- (1) The nut flank rotation is treated using the theory of beam on an elastic foundation without nonlinear distortion (Figure 3). The rotation angle of the nut flank is considered to be the rotation angle in the point of effective bearing friction radius r_b of the nut bearing surface.
- (2) The bolt part connected to the nut is represented by a compression spring of linear elastic stiffness k_n . The connected point of the spring to the nut is at the center point of the height t_n of the nut.

(3) The connected point of bolt and nut is simplified without the threads.

The elastic spring K_n is given by

$$K_n = \frac{Et_n}{r_d}, \quad (1)$$

where r_d is the radius of the bolt axis, t_n is the height of the nut, and E is Young's modulus.

The rotational stiffness of the nut flank, which is formed by the nut flank and the connected part of bolt to the nut, is characterized by two parameters k_{nM} and k_{bf} depending on the dimensions of the joint elements. k_{bf} is the bending rigidity of the nut flank due to a bending moment produced by the nut bearing load and can be calculated by the beam theory. k_{nM} is the rotation rigidity due to the bending moment effect produced by the nut bearing load on the connected part of bolt to the nut. The expression of the rigidity K_{bf} is

$$K_{bf} = \frac{2EI_n}{r_b - r_d}, \quad (2)$$

where r_b is the effective bearing friction radius. I_n is moment of inertia. K_{nM} can be equivalently replaced by the compression rigidity of K_n as follows:

$$K_{nM} = K_n \left(\frac{t_n}{2} \right)^2 = \frac{Et_n^3}{4r_d}. \quad (3)$$

Therefore, the rotational stiffness of the nut flank can be written as

$$K = \frac{K_{nM}K_{bf}}{K_{nM} + K_{bf}} = \frac{Et_n^3}{6r_b - 2r_d}, \quad (4)$$

where r_b can be calculated by

$$r_b = \frac{\int_{r_d}^{r_o} pr dr}{\int_{r_d}^{r_o} p dr} \approx \frac{\int_{r_d}^{r_o} pr^2 dr}{\int_{r_d}^{r_o} pr dr}. \quad (5)$$

The equilibrium consideration in the joint is obtained for both the bolt initial preloading state i and final small screw tightening state f , such that

$$P_n^i = F_b, \quad (6)$$

and, from the small screw tightening equilibrium,

$$P_n^f + f_b = F_b, \quad (7)$$

where P_n is the bearing force of nut.

The initial and final rotation deformations of the nut flank are given by the following equations:

$$\begin{aligned} \theta_n^i &= \frac{M_n^i}{K} = \frac{2F_b(3r_b^2 - 4r_b r_d + r_d^2)}{Et_n^3}, \\ \theta_n^f &= \frac{M_n^f}{K} \\ &= (2F_b(3r_b^2 - 4r_b r_d + r_d^2) \\ &\quad + 2f_b(3r_{fb}r_b - r_{fb}r_d - 3r_b^2 + r_b r_d)) \times (Et_n^3)^{-1}, \end{aligned} \quad (8)$$

where

$$\begin{aligned} M_n^i &= P_n^i(r_b - r_d), \\ M_n^f &= P_n^f(r_b - r_d) + f_b(r_{fb} - r_d) \\ &= P_n^i(r_b - r_d) + f_b(r_{fb} - r_b) \end{aligned} \quad (9)$$

in which r_{fb} is the radius of the small screw loading point. It is worth noting that as $f_b = 0$, $\theta_n^i = \theta_n^f$.

The effective bearing friction radius r_b can be obtained from the calculation results from Nassar's theory [14]. In that research, four scenarios of underhead load distribution are discussed (i.e., uniform underhead load, linearly decreasing load, exponentially decreasing load, and sinusoidal underhead load). A uniform pressure distribution simulates the use of a strong steel bolt and nut to clamp onto a much weaker clamping surface, such as plastic surfaces or soft washers. Other pressure distributions considered would simulate contact stress concentration at the edge of the fastener hole or represent the zero pressure conditions at minimum or maximum radii of the contact area. Nassar's theoretical equations [14] can be listed below, where r_{b-t} means the theoretical calculation result of r_b .

Uniform underhead load:

$$r_{b-t} = \frac{0.656(\gamma^{2.9} - 1)r_d}{\gamma^{1.9} - 1}. \quad (10)$$

Linearly decreasing load:

$$r_{b-t} = \frac{0.95r_d(\gamma^2 - 1)}{\gamma^{1.9} - 1} \left(\frac{6\gamma^{3.9} - 23.4\gamma + 17.4}{11.3\gamma^3 - 33.93\gamma + 22.62} \right). \quad (11)$$

Exponentially decreasing load:

$$r_{b-t} = \frac{0.95(\gamma^2 - 1)r_d^{0.1}}{\gamma^{1.9} - 1} \left(\frac{\int_{r_d}^{\gamma r_d} \gamma^{1.9} e^{-a(r-r_d)} dr}{\int_{r_d}^{\gamma r_d} \gamma e^{-a(r-r_d)} dr} \right), \quad (12)$$

where $a = \ln(p_{\max}/p_{\min})/r_d(\gamma - 1)$.

Sinusoidal underhead load:

$$\begin{aligned} r_{b-t} &= \frac{0.95(\gamma^2 - 1)r_d^{0.1}}{\gamma^{1.9} - 1} \\ &\quad \times \left(\frac{\int_{r_d}^{\gamma r_d} \gamma^{1.9} \sin[\pi(r - r_d)/r_d(\gamma - 1)] dr}{\int_{r_d}^{\gamma r_d} \gamma \sin[\pi(r - r_d)/r_d(\gamma - 1)] dr} \right), \end{aligned} \quad (13)$$

where γ is the value of the maximum to minimum contact radii ratio.

3. Numerical FE Model

To validate the analytical model that estimates the rotation rigidity and deformation of the nut flank, a two-dimensional axisymmetric numerical FE model was constructed. A 2D 4-node element PLANE42 with two degrees of freedom per

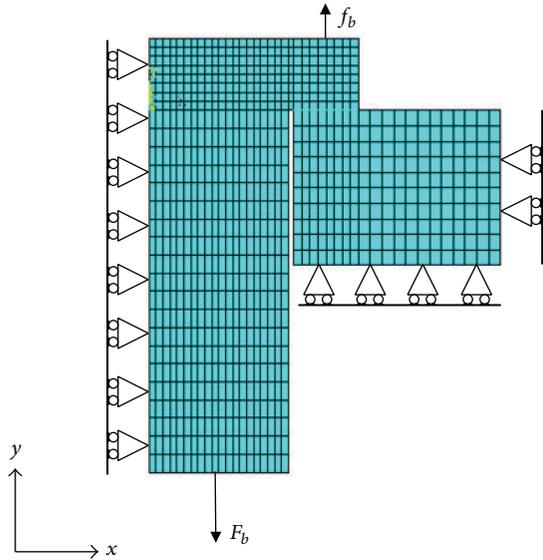


FIGURE 4: 2D finite element model.

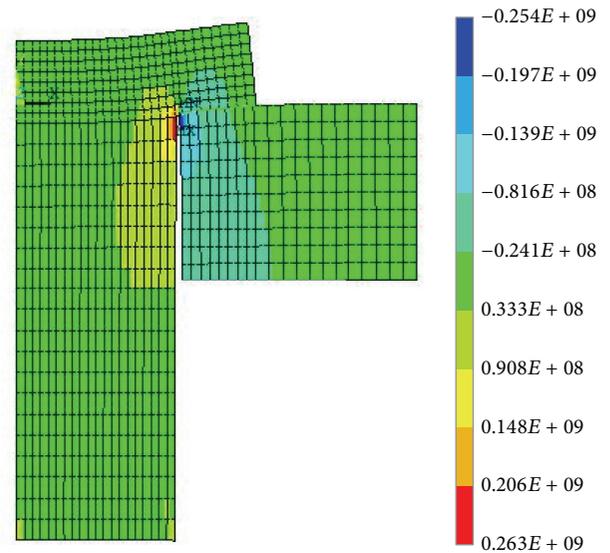
node in ANSYS [15] was used to model the bolt joint, as shown in Figure 4. Young's modulus is 195 GPa and Poisson's ratio is 0.3. In the definition of contact pairs for finite element analysis, the contact surface of clamped part is assigned to be the target element TARGE169, and the contact surface of nut is the contact element CONTA172.

The antisymmetry DOF constraint along x -axis is applied on the bolt axis. The bottom of the clamped part is also restrained in displacement in y direction. The right side of the clamped part is also restrained in displacement in x direction. The loading is applied in two steps. The first step consists of applying an initial nut-up. This is achieved by imposing to the bolt end-plane nodes, an equivalent axial node force to produce the target initial bolt-up stress of 28.8 MPa. A locking force f_b produced by the small screw that depends on the case studied is applied in the second step. Comparisons between the analytical method and finite element method results were conducted on different geometries of joint elements and different applied forces.

4. Results and Discussion

Figure 5 shows the result of the stress distribution along y -axis. The stress concentration can be found at the connecting corner of bolt and nut that produces the stress of about 260 MPa. The rotation of the nut flank can be observed obviously. The results of nut flank rotation after applied loads obtained from the proposed analytical approach are compared to those of FE model for the different applied loads and different sizes of joint elements.

Figure 6 shows the variation of the average values of the contact pressure of bolt and nut with the change of the locking force from the small screw. It can be seen that the average value decreases with the increase in the locking force. A difference between the analytical model and FEA that may be owing to the stress concentration at the detached point

FIGURE 5: y -axis stress and deformation of joint under preload force.

of the contact area of the bolt and nut is also shown for two different bolt force cases. Figure 7 depicts the rotation deformation of the nut flank after the locking force is applied. From Figure 7, it can be found that the effective bearing friction radius r_b obtained from the calculation results from Nassar's theory for different underhead load distribution cannot effectively calculate the rotation deformation of the nut flank for different bolt force F_b besides the exponentially decreasing load case. The results given by the uniform distribution and the sinusoidal distribution are very close. On the other hand, the results given by the linear distribution and the exponentially decreasing distribution are also close. However, the exponentially decreasing distribution gives much better results than other distributions as $P_{\max}/P_{\min} = 5$. Hereafter, the proportional coefficient $P_{\max}/P_{\min} = 5$ for the exponentially decreasing load distribution is considered to calculate the rotation stiffness and deformation for the different sizes of joint elements at $F_b = 200$ kN and $f_b = 3500$ N. It should be noted that γ ratio is 1.4 for all cases.

Figure 8 shows the comparison of the results for the different r_D case by the analytical model and FEA.

The analytical and FEA results are in a good agreement with a maximum difference of 0.9% between the two methods. The increase in r_D will generate a smaller rotation stiffness of the nut flank according to (2), which contributes to a larger value of the rotation deformation of the nut flank. From Figure 9, a good agreement between this stiffness and these deformations for different clearance d_n between the bolt and the clamped part is obtained. The decrease in the rotation stiffness k as d_n increase will be enhanced by (4). Therefore, as F_b and f_b are fixed, the bending moment M_n^f can be shown to increase as d_n increases, which leads to the glowing of the rotation deformation of the nut flank. The results of the calculated stiffness and deformations of the nut flank with the analytical model as shown in Figure 10 are in good agreement with those obtained by the FEM for the case

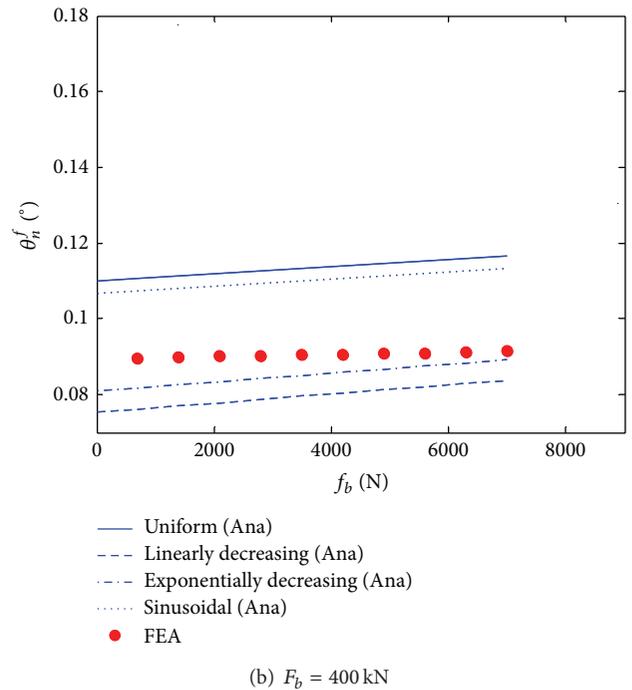
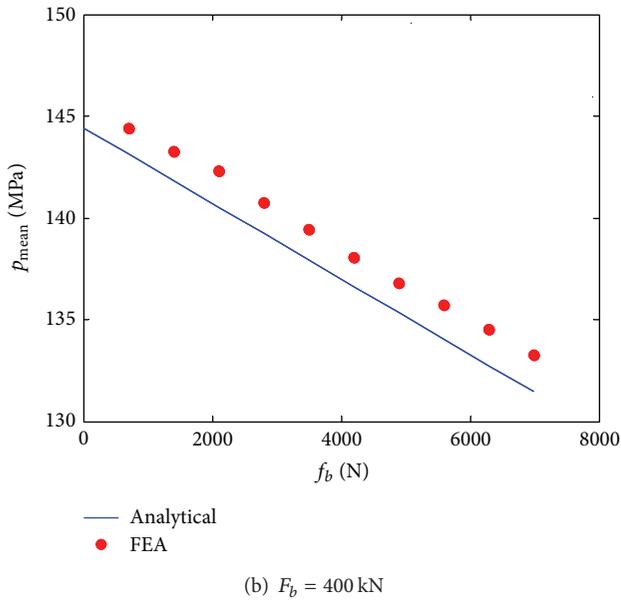
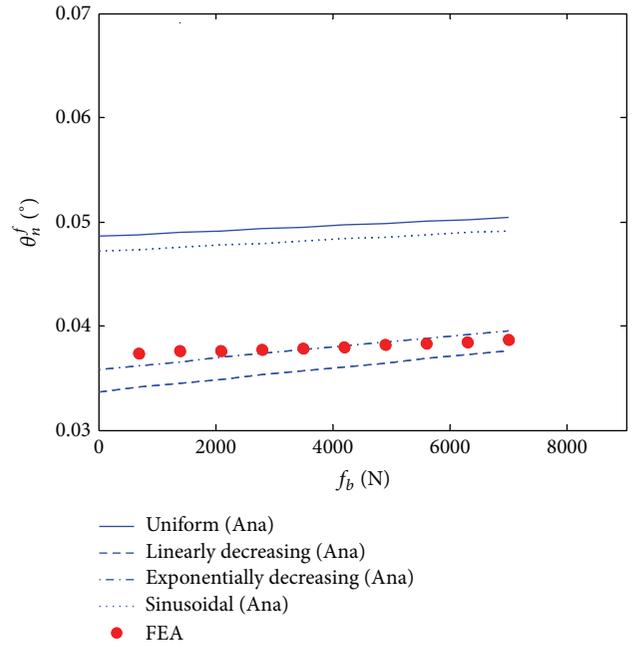
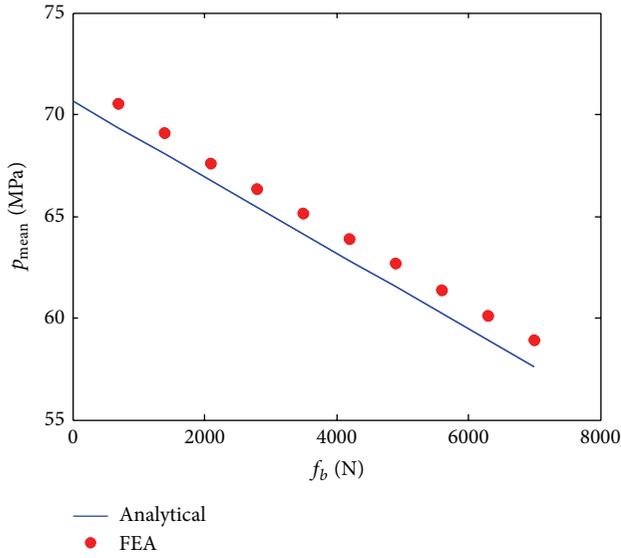


FIGURE 6: Average value of the underhead contact pressure under different locking force.

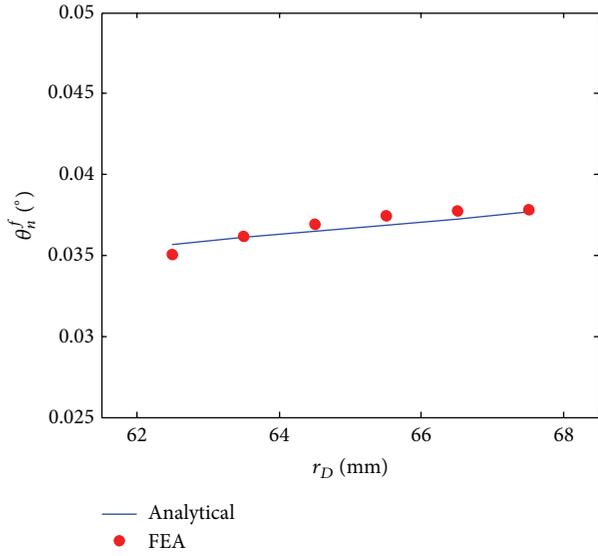
of different height t_n of the nut. The difference between the two methods is less than 20%. The difference between the results is due to the linear simplification made on the nut flank mechanical behavior. According to (4), the increase in the height t_n will raise the rotational stiffness of the nut flank. The rotation deformation will decrease in this case.

In order to analyze the effect of the role position of the locking force f_b , the different r_{fb} case is discussed in this paper. Good agreement results for the different r_{fb} case by the analytical model and by the FEA (Figure 11) are shown. In this case, the rotation stiffness will not be changed because of the same geometry size of the joint elements. The built-up of r_{fb} will increase the bending moment M_h^f , so the rotation deformation is increased with the growth of r_{fb} .

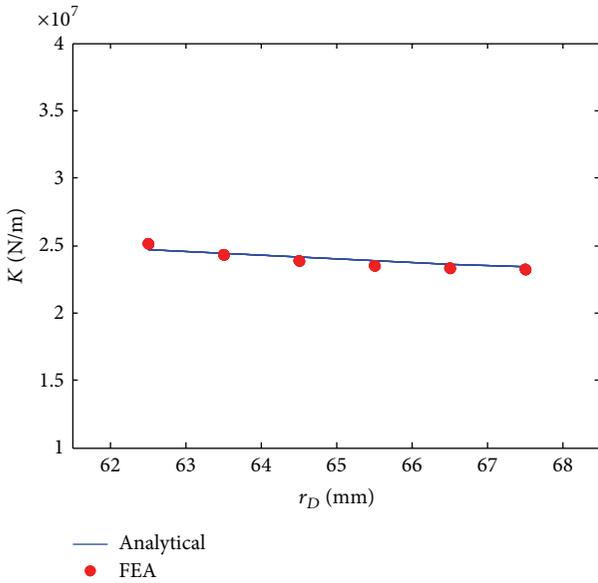
FIGURE 7: Rotation deformation of nut flank.

5. Conclusions

An analytical method has been developed to analyze a bolt joint with an antiloosening nut. The developed model based on the beam theory has shown rationality solution in the rotation stiffness and deformations of the nut flank. The analyses are verified against the accurate 2D FEM on different sizes of joint elements and different applied forces. The analytical results of the rotation stiffness and deformations



(a) Rotation deformation



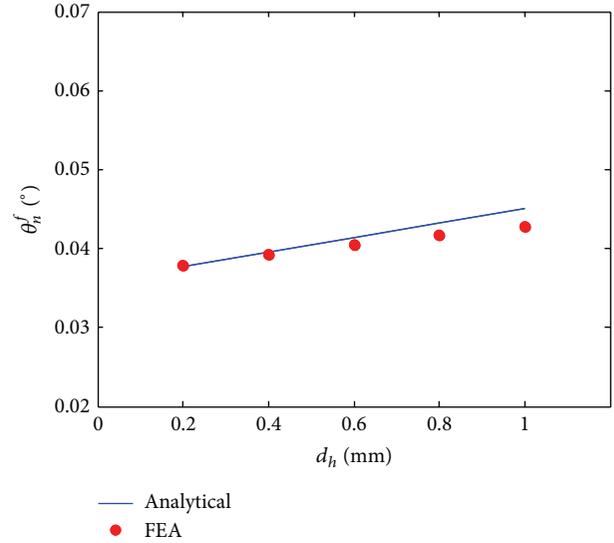
(b) Rotation stiffness

FIGURE 8: Rotation stiffness and deformation of nut flank with variation of r_D .

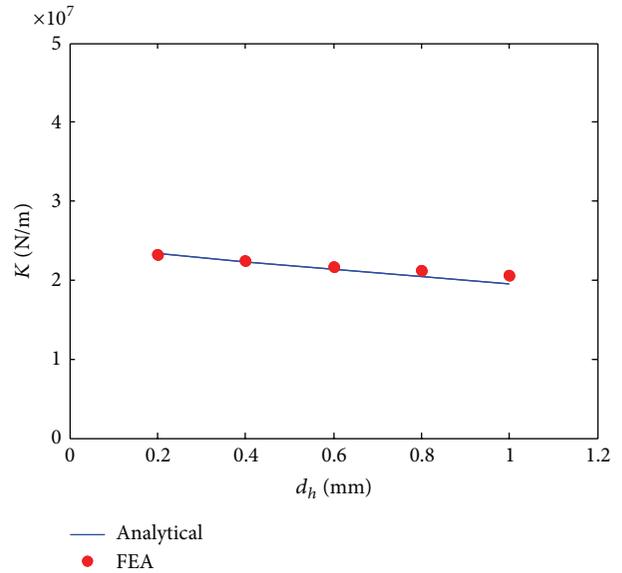
compare quite well with those of FEM. It is confident that the analytical model has the ability to predict the deformation and the contact load of the antiloosening nut.

Nomenclature

d_n : Clearance between the bolt and the clamped part
 F_b : Bolt force
 f_b : Locking force
 E : Young's modulus
 I_n : Moment of inertia
 k_{bf} : Bending rigidity of the nut flank



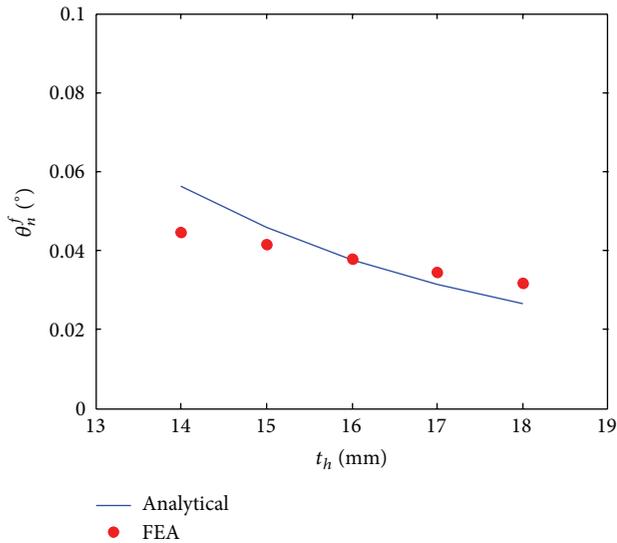
(a) Rotation deformation



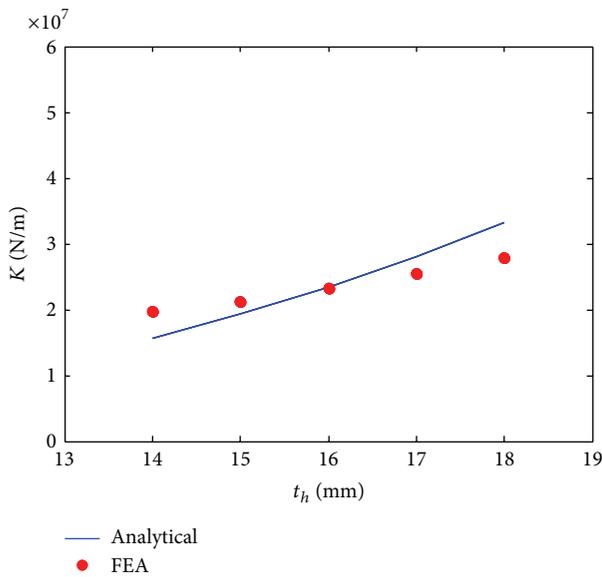
(b) Rotation stiffness

FIGURE 9: Rotation stiffness and deformation of nut flank with variation of d_n .

k_n : Linear elastic stiffness of compression spring
 k_{nM} : Rotation rigidity due to the bending moment effect
 P_n : Bearing force of nut
 r_b : Effective bearing friction radius
 r_{b-t} : The theoretical calculation result of r_b
 r_d : Radius of the bolt
 r_{fb} : Radius of the small screw loading point
 t_n : Height of the nut
 θ_n^i : Initial and final rotation deformations
 θ_n^f : Final rotation deformations.



(a) Rotation deformation



(b) Rotation stiffness

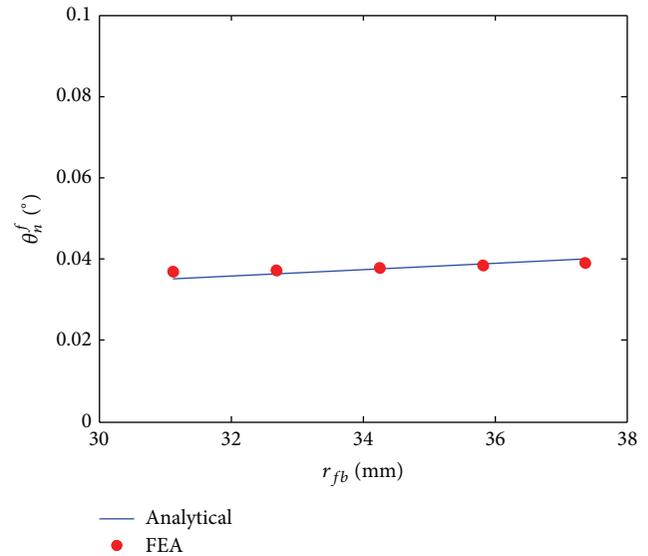
FIGURE 10: Rotation stiffness and deformation of nut flank under different locking force with variation of t_h .

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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FIGURE 11: Rotation deformation of nut flank under different locking force with variation of r_{fb} .

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