Structural Engineering and Mechanics, *Vol. 55, No. 3 (2015) 639-654*
DOI: http://dx.doi.org/10.12989/sem.2015.55.3.639

Simplified model to study the dynamic behaviour of a bolted joint and its self loosening

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(Received January 19, 2015, Revised June 27, 2015, Accepted July 7, 2015)

Abstract. Bolted joints are essential elements of mechanical structures and metal constructions. Although their static behaviour is fairly well known, their dynamic behaviour due to shocks and vibrations has been less studied, because of the large size of the finite element models needed for a detailed simulation. This work presents four different simplified models suitable for studying the dynamic behaviour of an elementary bolted joint. Three of them include contact elements to allow sliding of the screw head and the nut on the assembled parts, and the last one allows rotation between screw and nut. A penalty approach based on the Coulomb friction model is used to model contact. The results show that these models effectively represent the dynamic behaviour, with different accuracy depending on the model details. The last model simulates the self loosening of a bolt subjected to transversal vibrations.

Keywords: bolted joint; dynamic study; finite element model; friction; bolt model; self loosening

1. Introduction

Bolted joints are an essential element of mechanical structures. They are widely used in aerospace, railway and automobile construction, and throughout industry. Their role is to assemble isolated elements to create consistent, robust constructions that can be dismantled. The mounting of a bolted joint usually involves applying relative rotation between screw and nut until the desired preload, which will maintain the integrity of the assembly (Guillot 2010, 2011). The tightening method has an influence on the performance of the assembly. In his paper, Abid (2013) presented a finite element method to study the effect of different bolt tightening methods of a bolted flange. In addition, the performance, the reliability required by project led manufacturers to adopt design strategies where experiences play an important role, particularly in the prediction of the behavior of nonlinear mechanical systems. By this context, Katula (2015) established an experience on a bolted end-plate joints of industrial type steel building frames. He provides an experimental background for the design of bolts by studying joints load-bearing capacity, axial load distribution,

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http://www.techno-press.org/?journal=sem&subpage=8 ISSN: 1225-4568 (Print), 1598-6217 (Online)

and end-plate deformations.

Self-loosening of bolted joints is one of the problems encountered in aeronautical structures and can lead to fatigue failure or unintended disassembly. It results from both vibration processes and inadequate control of the contact phenomena. Our ultimate aim is to study the self-loosening of bolted joints when they are subjected to dynamic transverse shear stresses resulting from vibrations within the framework of aeronautical structures.

A frequent cause of self-loosening is the sliding of the bolt head or the nut relative to the assembled parts, which can result in unscrewing of the thread. If there is no sliding under the head or the nut, no unscrewing is possible, even if the joint is subjected to severe vibrations. Most of the studies presented in the literature indicate that the sliding phenomenon appears mainly when bolted joints are excited transversely to the axis of the bolt. Tests based on the so-called "Junker machine" are used to investigate this self-loosening process experimentally.

In the bolt, the geometry of the thread produces a torque on the nut and the screw. This torque depends on the pretension of the screw but, by itself, it is insufficient to cause sliding under the nut or the screw head and to generate unscrewing. If there is an external load resulting from vibration that causes slip under the nut or the screw head, the sliding is deflected by the thread torque. Each relative motion includes a slight rotation of the nut or the screw and thus leads to self loosening of the bolt (Aziz 2003).

Contact problems that affect the self loosening of bolted joints have been studied in the literature. Bouchaala *et al.* (2013) studied micro slip between two flat surfaces, taking a bolted joint as an example.

Some papers have studied the influence of vibration on assembled joints and the contact problems involved, which result in energy dissipation and damping.

Ibrahim and Pettit (2005) focused on bolt relaxation resulting from vibration. Champaney *et al.* (2008) proposed a strategy for analyzing contact problems. They developed an analytical model for elastically deformable structures assemblies by taking into account the nonlinear behaviour resulting from contact and friction.

Jiang (2013) studied the loosening phenomenon of bolts in a curvic® coupling. A three-dimensional finite elements model validates the results.

From an analytical point of view, Nassar (2009), Yang (2011) have proposed a mathematical model to study the loosening of threaded fasteners subjected to a transverse harmonic excitation causing slippage between the contacting surfaces, between the threads and under the bolt head. Integral equations are calculated for torques resulting from the head friction and the nut. The relationship between the shear load and the bending moment of the bolt has been also developed.

In a numerical approach of bolted joints simulation, the accuracy of the results depends on the model chosen, which can range from a very detailed solid model including the real thread geometry, to a very schematic one. Many papers have discussed the solid bolt method. Zhang *et al.* (2007) developed a three-dimensional finite element (FE) model taking the helix angle of the thread into consideration to simulate self-loosening. Other authors (Dinger *et al.* 2011, Koch *et al.* 2012) have proposed a three-dimensional finite element model to reveal the self-loosening process, taking account of friction between the nut and the screw and in all the contact areas.

It's interesting to investigate the self loosening phenomenon but it's more interesting to find a way in order to avoid this phenomenon.

Bhattacharya (2010) has conducted tests to see the ability of different ways to self loosening resistance as Nylock® nut, chemical Lock, flat washer and nylon washer for bolts of different materials, sizes and for different preloads. The loss of the clamping force is an indication of the loosening phenomenon.

All these papers focus on the self-loosening process but most of them are limited to the quasi-static case. Few papers discuss the dynamic behaviour and there is no rigorous description of how the assembled bolt behaves dynamically or how friction affects the dynamic response. This is the aim of the present paper.

All these previous works try to predict the physical behaviour of a structure with a detailed three-dimensional model since it seems the most efficient approach for these problems. But for a large structure including many screws and for dynamic studies, detailed modelling of the bolt is not appropriate because of the problem size and the huge computational time it would involve. So is it possible for a simple model to highlight the self loosening process? The answer to this question will be presented in this paper. The objective of this study is to analyse the dynamic behaviour of an elementary assembly of two plates connected by a single bolt, without taking the threaded connection between screw and nut into account. Four different models are used: the coupled bolt, the hybrid bolt, the solid bolt and the solid bolt with revolute joint, in order to evaluate the most efficient model for the dynamic study of bolted joints and make a comparative study. A penalty method is used to simulate friction between the different parts. Some results on the dynamic behaviour are then presented, including slip and displacements induced by vibration under the effect of a rapidly varying load.

2. The bolt models

The modelling of the bolt is the key point in this study. Various models are discussed below.

Montgomery *et al.* (2002), Kim *et al.* (2006) have presented a few models, some of which have a large number of elements. A general drawback of dynamic studies is the significant amount of computation required. To solve this problem and reduce the number of elements, many authors have developed simplified models. The first one considered here is called the coupled bolt model (Fig. 1(a)), which consists only of beam elements. One beam element is used to replace the shank, while deformable beam elements are used for the coupling between the shank and the contact surfaces of the assembled parts, acting as the nut and the head of the bolt. This method models the nut and bolt head fairly accurately, allowing transmission of various loads such as tensile and bending loads. With this model, the stiffness of the bolt can be accurately considered and calculated. The approach minimizes the number of elements. Unfortunately, the distribution of forces in the contact zones under the nut and screw head is not very accurate, and sliding in these areas cannot be modelled. Also, the distribution of stress in the cross-section of the bolt shank cannot be determined.

The second model (Fig. 1(b)) is a hybrid one. In this model, the shank of the bolt is modelled by a beam element, while the nut and the bolt head are modelled by solid elements. The bolt load is introduced through the beam element, which is subjected to a tensile load. Deformable coupling elements are used to join the shank to the ends of the bolt, allowing an accurate distribution of the load in the head and the nut and therefore in the assembled parts.

This model can be considered as the closest model to the solid one (Fig. 1(c)), the only difference between the two being the stud part, which is considered as a beam element in the hybrid bolt model. Friction is introduced using a technique similar to that in the solid bolt model: contact elements are placed under the head, under the nut and between the assembled parts. We can conclude that this approach is almost as accurate as the solid bolt model: the shank section is

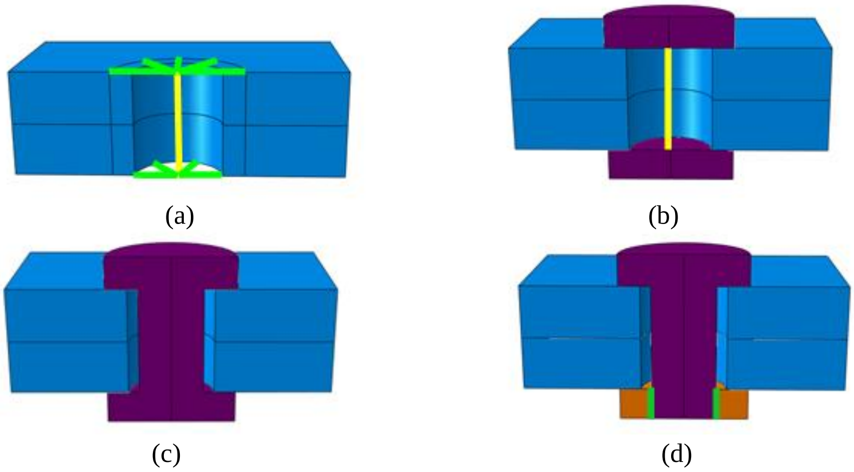


Fig. 1 Various bolt models: (a) coupled, (b) hybrid, (c) solid, (d) solid with revolute joint

modelled using a beam element, the stress distribution in the head and the nut can be calculated, and the tensile, bending and thermal loads are transferred through the beam element. There is no possibility of visualizing the stress distribution in the stud part since it is modelled as a beam element.

The solid bolt model (Fig. 1(c)) is more realistic but more complicated than the two previous models. All the contact details are introduced into this model: under the head, under the nut and between the assembled plates. Actually, three-dimensional brick elements are used for the geometry meshing. Eight nodes define these elements, each node having three degrees of freedom. This approach is close to reality, but not all details of the bolt geometry are introduced, in particular the thread.

These three models are suitable for studying the dynamic behaviour of a bolted joint and identifying shank stresses and contact problems. However, the absence of thread makes them inadequate for studying the self loosening of a bolted joint. The introduction of the thread geometry would greatly complicate the study, especially in the dynamic problem case. In this context, we have tried to find a way to replace the helical connection resulting from thread geometry. This is the aim of our model called the solid bolt model with revolute joint (Fig. 1(d)). This model is similar to the previous one but it includes a revolute joint between the nut and the screw, allowing the rotations of nut and screw to be distinguished, and a torque to be introduced on each of them, which is equivalent to that exerted by the thread.

In order to evaluate the relevance of these models to treat self loosening, we apply them to a simple but representative test structure, which is presented in the next section.

3. Test cases for numerical dynamic analysis

3.1 Geometry

Our test case consisted of two steel plates assembled by a bolt, subjected to a dynamic load transverse to the axis of bolt.

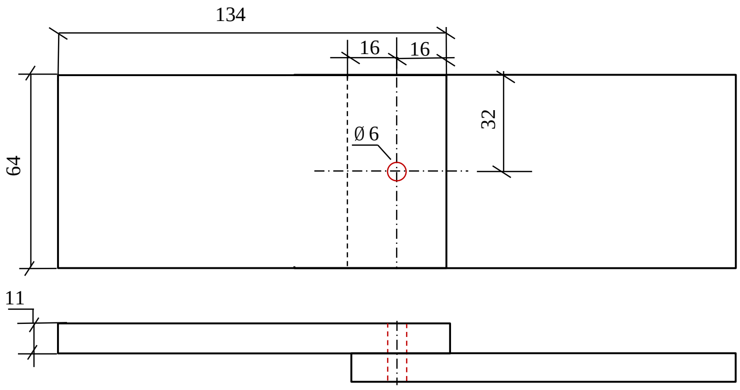


Fig. 2 Test structure

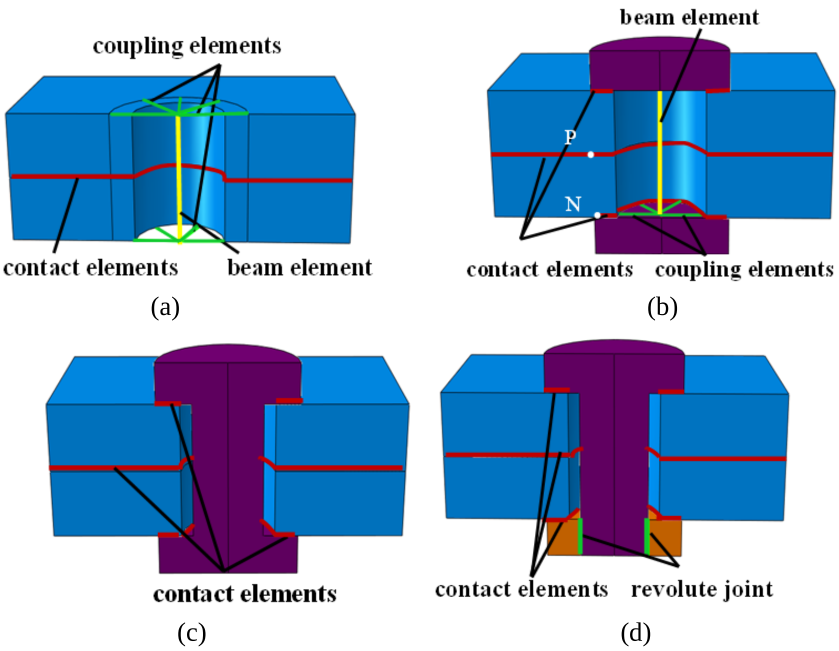


Fig. 3 Different constraints for the bolt assembly modeling

As shown in Fig. 2, the structure was composed of two plates, each 64 mm wide, 134 mm long and 11 mm thick, joined by an M6 bolt of class 8.8.

The finite element analysis software Abaqus\Standard 6.11-2 (2011) was used to model this assembly. Eight-node hexahedral elements with reduced integration (linear bricks C3D8R with an integration point at the centre of the element) were used for the structure meshing. Young's modulus was 210 GPa and Poisson's ratio was 0.3 for the whole structure.

3.2 Constraint elements

The various constraints adopted to model the assembly are described in this section.

Fig. 3 shows the constraint elements used in the different models. In the first model (Fig. 3(a)) the 'uniform coupling constraint' provided by Abaqus software is used for the coupling between the extremities of the bolt and the nodes of the plate facing the nut and the screw head. This constraint allows forces and moments to be transmitted.

For the second model, (Fig. 3(b)) the same coupling elements are used between the extremities of the bolt and the attachment surfaces of the nut and screw head with the shank.

For the third case (Fig. 3(c)), we use the 'tie constraint' between the nut and the screw. It constrains nodes of the slave surface (nut) in such a way that they have the same motion as nodes of the master surface (screw).

The main objective of this study is to simulate the self loosening of a bolted joint, which consists of a rotation of the nut relative to the screw. So in order to allow the screw and the nut to rotate independently, a revolute joint is introduced in the last model (Fig. 3(d)). The torque generated by the helical thread depends on the preload, and we replace it by an external torque applied to the nut and acting in the loosening direction. In reality, sliding can appear under either the nut or the screw head. In this study, we have applied a slightly higher coefficient of friction under the screw head, so that sliding appears under the nut first.

The points P and N in Fig. 3(b) are located at the interfaces between plates and under the nut respectively; they will be used further on to measure the sliding.

3.3 Contact formulation

Modelling the contact is of great importance because of the effects associated with stick-slip during movement, which can lead to divergence problems if not correctly introduced.

The friction coulomb model is used for the modelling of friction. It is a static model and gives the friction forces as a function of the velocity.

When contact extends over an area, the shear and normal loads transmitted by the contact elements are related by the friction coefficient. According to the Coulomb friction model, there is no relative motion when the equivalent shear stress τ_{eq} resulting from the combination of orthogonal shear stress components τ_1 and τ_2 is less than a critical value τ_{crit}, which is proportional to the normal contact pressure P through the friction coefficient μ

$$\tau_{eq} = \sqrt{{\tau_1}^2 + {\tau_2}^2} \tag{1}$$

$$\tau_{crit} = \mu\, P \tag{2}$$

So the limit of the stick and slip regions are shown in Fig. 4.

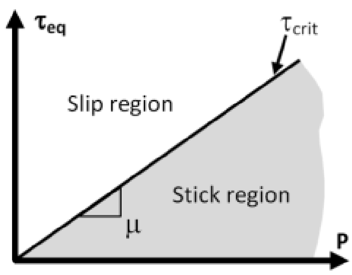


Fig. 4 Stick and slip regions for the Coulomb friction model

Two different discretization methods can be used in Abaqus to model the contact problem: node-to-surface discretization or surface-to-surface discretization. In the first method, each node on one side of the surface interacts with a group of nodes on the other side, and values are interpolated at the projected point. In the second, the contact conditions are enforced on average regions on each side so that the local properties and state of contact present better smoothing. The latter method improves the quality of results and minimizes the penetration of surfaces but is more expensive in computation time. It is best suited for problems involving slightly curved contact surfaces, which was our case. So we chose to use the surface-to-surface discretization method.

Several friction enforcement methods are implemented in Abaqus: a direct method, a penalty method and an augmented Lagrange method. We chose the linear penalty method, which is suitable for stiff surface-to-surface contacts and reduces the number of required iterations. In this method, the normal contact pressure is proportional to the overclosure of the surfaces. It also uses a transversal stiffness that allows some relative motion of the surfaces (elastic slip) when they should be sticking. The condition of no relative motion is approximated by stiff elastic behaviour, so that relative motion before slipping is bounded by a critical value, equal to 0.5% of the average length of all contact elements used in the model.

In our first model, contact is only introduced between the plates. For the following models, additional contact is introduced under the screw head and under the nut. We take a friction coefficient equal to 0.1 between the plates and under the nut, as a representative value for smooth, lubricated surfaces. A slightly higher friction coefficient, equal to 0.11, is chosen under the screw head to force the sliding of the bolt to begin at the nut end rather than the head end, in order to facilitate the measurement of unscrewing.

3.4 Numerical simulation

In this section, the different calculation steps will be presented. The boundary conditions adopted are shown in Fig. 5. The left plate was clamped at its left end whereas the other plate was subjected to a displacement $ux_E(t)$ in the x direction at its right end.

The equation of motion for this problem is expressed by Eq. (3)

$$M\ddot{X} + KX = F \tag{3}$$

The equation of motion of the screw is presented as follows

$$K(X_s - X_N) = F_v \tag{4}$$

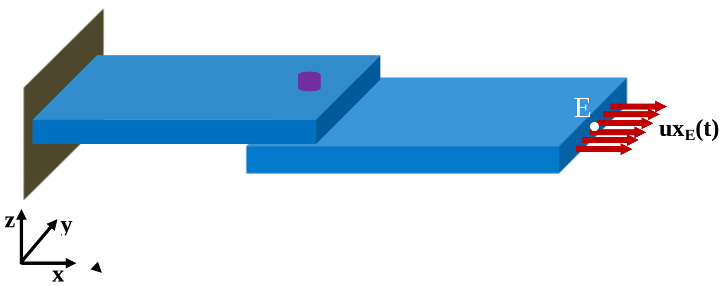


Fig. 5 Boundary conditions

Where:

X_s and X_N are respectively the displacements of the screw and nut

F_v: is the load friction under the screw

The flowchart shown in Fig. 6 presents the various calculation steps.

In the first one, the bolt preload is established in the bolt shaft. To facilitate slipping and loosening of the joint, we limit the preload to 4 kN, corresponding to about 30% of the screw yield strength. So the normal stress in the bolt shaft after preloading is 199 MPa.

This is achieved by means of a special function called ‘bolt load’ in Abaqus, which allows the load to be defined either as a concentrated load or through a change in length of the bolt.

In the second step, the length of the bolt shank is set to reproduce the initial preload and to allow subsequent changes of stresses in the bolt. So the bolt acts as a deformable component, whose length and load can vary when the structure deforms. In addition to that, in case (*d*), a torque of 600 Nmm is introduced on the nut, equivalent uxE to the torque that comes from the thread.

In the third step, an external displacement is suddenly applied. It is made up of a step and a sinusoidal shape, the frequency of which is chosen close to that of the first eigenmode of the structure, in order to maximize the distortion of the plates and the dynamic effects. This displacement $uxE(t)$ has the following expression (in mm)

$$ux_E(t) = 0.08 + 0.12\sin(2*\pi*180*t) \quad (5)$$

The movement and behaviour of the structure are simulated through an implicit dynamic

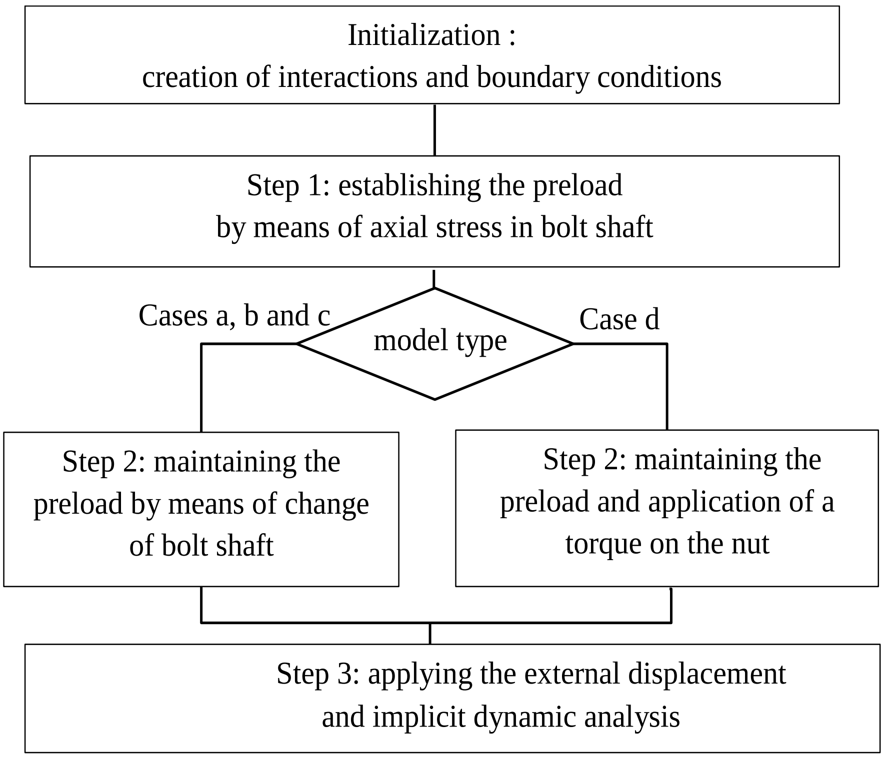


Fig. 6 Flowchart of successive steps

analysis. The resolution method used in Abaqus is presented by Sun *et al.* (2000) after Hilber and Hughes (1978). It is based on the full Newton iterative method with an increment strategy as shown in Eq. (6)

$$\Delta u^{i+1} = \Delta u^{i} + \mathrm{K}_t^{-1}\,(F^{i} - I^{i}) \tag{6}$$

where Δu is the displacement increment, K_t is the tangent stiffness matrix, and F and I are the external and internal forces.

The implicit dynamic procedure is described by

$$M\ddot{u}^{i+1} + (1+\alpha)Ku^{\mathrm{i}+1} - \alpha K u^{i} = F^{i+1} \tag{7}$$

$$u^{i+1} = u^{i} + \Delta t\dot{u}^{i} + \Delta t^{2}\left(\left(\frac{1}{2} - \beta\right)\ddot{u}^{\mathrm{i}} + \beta\ddot{u}^{i+1}\right) \tag{8}$$

$$\dot{u}^{i+1} = \dot{u}^{i} + \Delta t\,\left((1-\gamma)\ddot{u}^{i} + \gamma\ddot{u}^{i+1}\right) \tag{9}$$

where M is the mass matrix, u the displacement, K the stiffness matrix and F the applied force. The coefficients β and γ depend on a value α according to

$$\beta = \frac{1}{4}\,(1 - \alpha^{2}) \tag{10}$$

$$\gamma = \frac{1}{2}\,(1 - \alpha) \tag{11}$$

We use the default α value of −0.4144 to obtain numeric damping that smoothes and stabilizes the result.

4. Results and discussion

The results obtained from the dynamic analysis are given in this section. The variation in contact areas and the extension of sliding zones under the nut are presented and the behaviour of the plates, nut and screw under the effect of the external displacement ux_E are highlighted. The best approach for modelling self loosening is selected.

4.1 Modal behavior

First we want to verify whether our different models of the bolt affect the linear dynamic behaviour of the structure. Fig. 7 shows the mass, the first three eigenfrequencies and the shapes of the eigenmodes for the different cases studied.

We consider only the modes whose displacements remain in the *XZ* plane and are excited by the ux_E displacement we impose. Their shapes are similar to that of a beam with a constant section which bends in the *XZ* plane. The additional modes, corresponding to bending out of the *XZ* plane and twisting, are not excited by the ux_E displacement we impose and they do not affect the behaviour considered here.

The mass and modes are very close for the four models studied. We can conclude that subsequent differences in behaviour are only due to different ways of modelling the bolt.

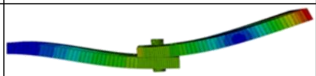
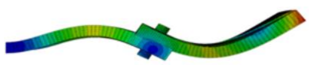
	Case a	**Case b**	**Case c**	**Case d**	**Mode shape**
Total mass model (kg)	1.471	1.479	1.479	1.479	
First frequency (Hz)	162.1	162.2	162.3	162.4	
Second frequency (Hz)	933.5	931.5	931.9	931.9	
Third frequency (Hz)	2574.9	2571.4	2571.7	2571.9	

Fig. 7 First three eigenmodes. Colors of the shapes indicate magnitude of displacement

4.2 Dynamic behavior under excitation

Now we consider the behaviour of the structure when it is excited by the displacement $ux_E(t)$ defined in Eq. (5). We first focus on the slipping that occurs between plates, then on the slipping under the nut. Fig. 8 shows the displacements and the state of contact at node *P*, located at the interface between plates and belonging to the lower, mobile plate as shown in Fig. 3(b) The first

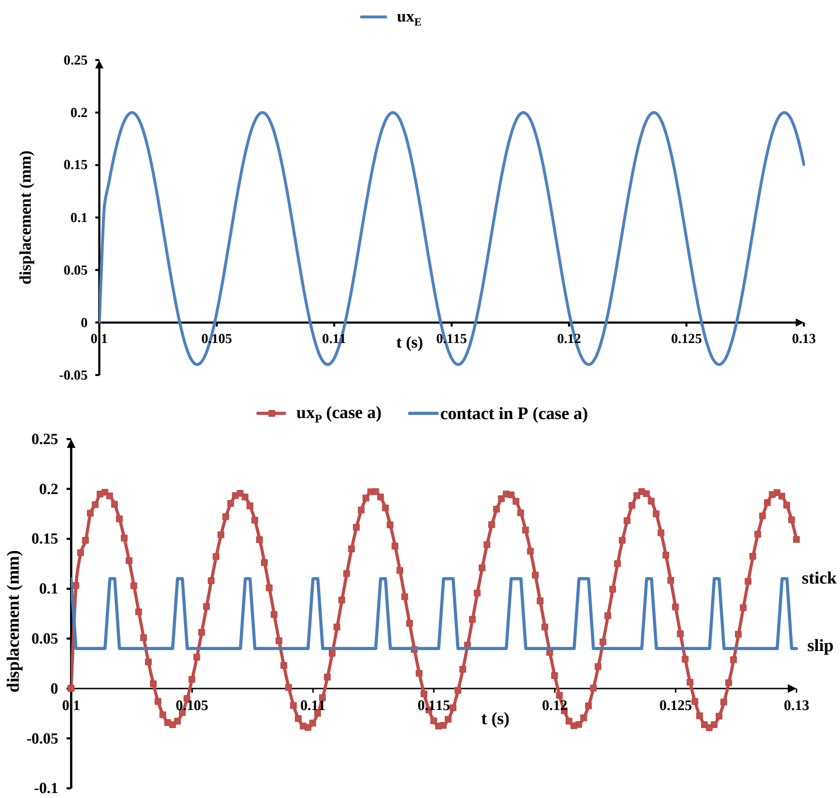


Fig. 8 Displacement imposed and response at point *P* located between plates

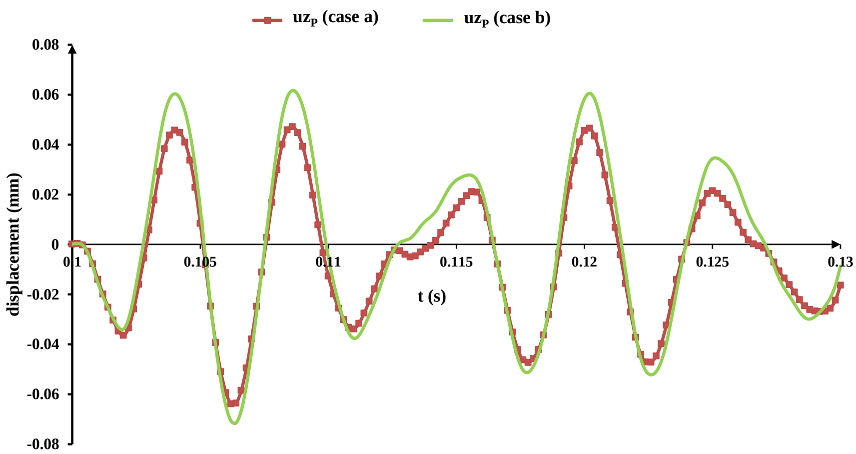


Fig. 8 Continued

part of this figure shows the excitation displacement ux_E, while the second shows the response displacement ux_P in the x direction, and the third shows the response displacements uz_P in the z direction for cases *a* and *b*. In the second part, we can see that the plates follow the displacement ux_E imposed in the x direction closely, with a few irregularities at the very beginning coming from high frequency response to the initial step. Slipping occurs immediately after the initial step and, later, after each change of direction, as soon as the relative displacement ux_P exceeds about 0.01 mm. This response is nearly identical for the 4 cases studied.

The third part of Fig. 8 shows the displacement ux_Z in the z direction, transverse to the one imposed at the end of the plate. It comes from the bending that results from this excitation. It shows a beat phenomenon due to the summing of two sinusoidal movements with close frequencies:

- the linear response of the structure to the sinusoidal excitation force, at 180 Hz, induces a sinusoidal movement at the same frequency;
- the transient response of the first mode to the initial excitation step induces an approximately sinusoidal bending movement at the first eigenfrequency, 162 Hz.

The weak differences between responses in z for cases *a* and *b* come from the different distribution of the load and stiffness in these models, and from the slipping that appears under the nut in case *b*. Cases *c* and *d* have responses almost identical to case *b*. The beat phenomenon does not appear on the displacements in the x direction.

In cases *b*, *c* and *d*, slipping can occur under the nut. No slipping appears under the screw head because the friction coefficient is greater here than under the nut, so tangential force at these interfaces is limited by slipping of the nut.

Fig. 9 shows displacements in the x direction and the state of contact at node N, located at the interface between the nut and the plate as shown in Fig. 3(b), for case *b*.

The plate undergoes the excitation displacement ux_E, while the nut is held by the screw. The initial step is sufficient to induce slipping from the beginning. When the movement reverses at t=1.1013 s, the nut sticks on the plate, until the bending of the screw shank is sufficient to overcome the sticking force. This occurs at t=0.1033 s, and then the nut slips on the plate. This sequence repeats in the opposite direction at the next phase of movement, after t=0.1043s.

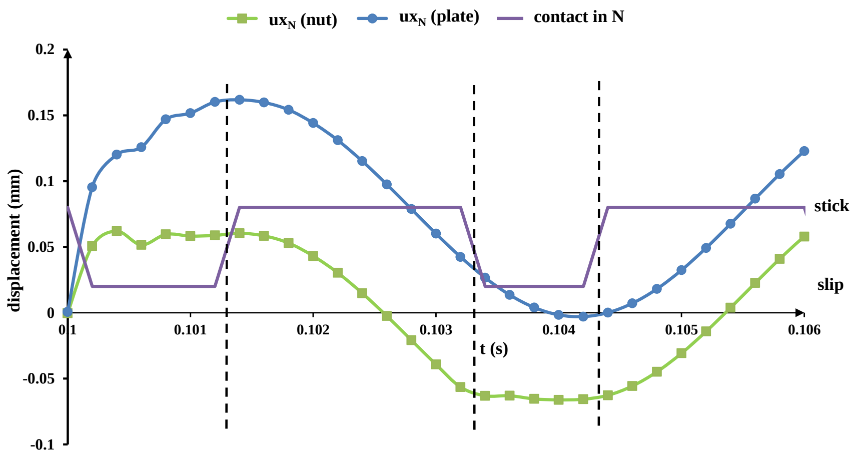


Fig. 9 Displacements in direction *x* and state of contact at node *N* for case *b*

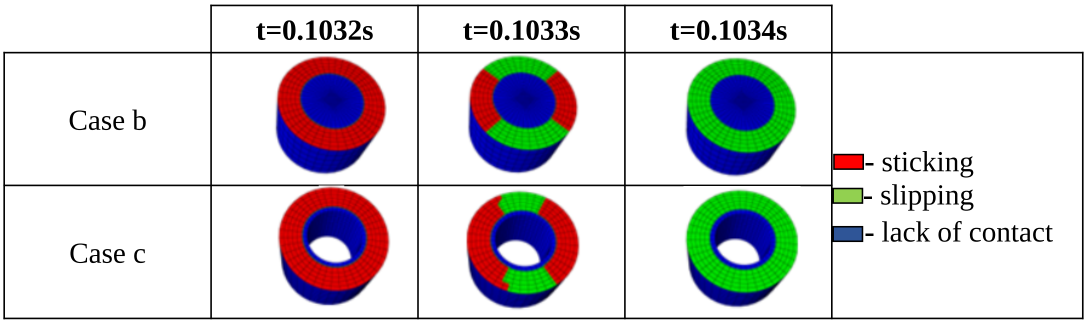


Fig. 10 Contact status under the nut (red=sticking, green=slipping, blue=lack of contact)

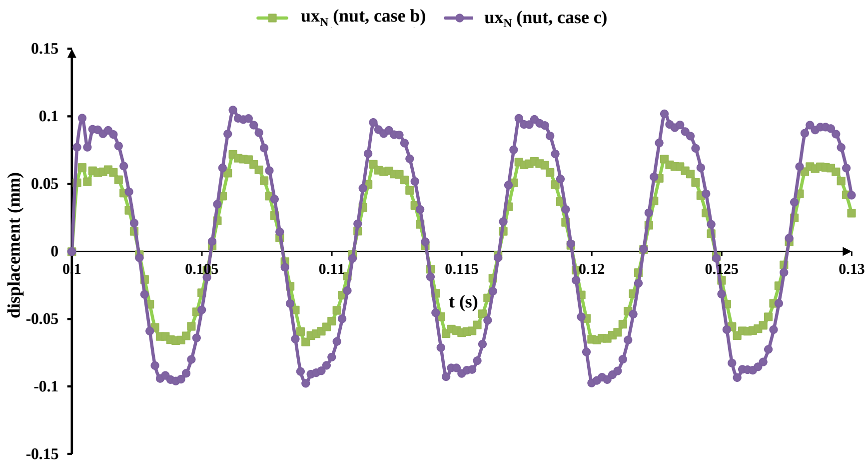


Fig. 11 Displacements under the nut in direction *x* for cases *b* and *c*

Transition between sticking and slipping is almost instantaneous on the whole contact surface between nut and plate. This is shown in Fig. 10 for cases *b* and *c*. The differences between cases are small and result from the different distributions of the normal load on the contact array.

Fig. 11 shows the response in the *x* direction, at node *N*, for cases *b* and *c*. Slipping occurs between the nut and the plate, twice for each movement period. The amplitude of movement is smaller for case *b*, which demonstrates that slipping is higher. This comes from the stiffer coupling introduced between the screw and the nut in case *b*. Although sliding appears at the same time in every case, displacement ux_N is different. So it is important to model the stiffness of the screw ends accurately and, for that reason, model *c* is preferred to model *b*.

4.3 Simulation of self loosening

We now consider the behaviour of the model with the revolute joint (case *d*), subjected to the same displacement $ux_E(t)$, and to torque equivalent to that coming from the thread.

The first part of Fig. 12 shows, for point *N*, the displacement ux_N of the nut in the *x* direction, the contact status, and the slipping displacement sx_N, defined by the difference between the displacements of the plate and the nut at the same node, *N*

$$sx_N = ux_N\,(\text{plate}) - ux_N\,(\text{nut}) \tag{12}$$

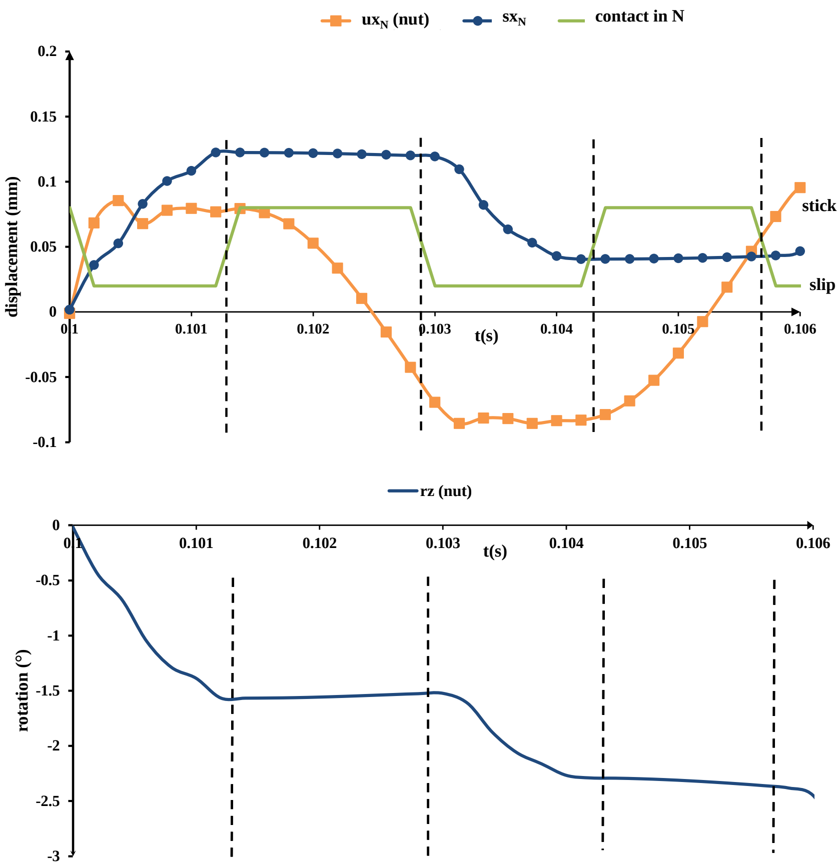


Fig. 12 Displacements under the nut in direction *x* and nut rotation *rz* for case *d*

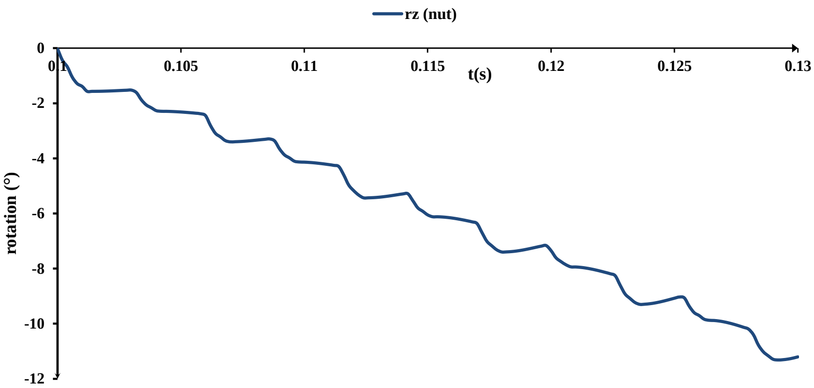


Fig. 13 Nut rotation resulting in self loosening for case *d*

We can see the variation in slipping displacement sx_N during the slipping time.

The second part of Fig. 12 shows the rotation *rz* of the nut under the action of the torque coming from the thread. Rotation in the loosening direction appears each time the nut slips on the plate, whatever the direction of the movement. We can see that the shape of the rotation *rz* is very similar to that of the slipping displacement sx_N, but the direction is different: the slipping displacement increases and decreases alternately, while rotation always goes in the same direction.

Fig. 13 shows the evolution of rotation during successive periods of movement.

The nut turns in the loosening direction at each sliding stage, and the angular displacement *rz* increases rapidly, so we can see the development of self loosening. This simulation is not quite exact because we assume that the unscrewing torque coming from the thread is constant whereas, actually, it decreases when the nut unscrews. But this shows that the model is appropriate for simulating phenomena including relative rotation of screw and nut.

5. Conclusions

Bolts are widely used in mechanical systems, and they must be modelled to simulate the static and dynamic behaviour of many structures. Several models are presented in the literature, varying from very elementary to very detailed ones. For solving dynamic problems, such as the self loosening phenomenon, a simplified model, including few elements, is essential.

We presented 4 simplified models: (a) a coupled beam, (b) a beam with solid ends and contact areas, (c) a solid bolt with contact areas, and (d) a solid screw and nut with contact areas. In each case, a penalty approach based on the Coulomb friction model was used to model contact between the bolted plates and in the contact areas under the bolt ends. Model (a), or one of its variants, is currently used for the static and dynamic calculation of structures including many bolts.

We showed that the dynamic behaviours of these 4 models are very close, as long as no slipping occurs under the bolt ends. Models (b), (c) and (d) can take such slipping into account but models (c) and (d) represent it more accurately. Finally, model (d) can simulate the whole dynamic behaviour, including the modal behaviour of the structure, slipping between bolted parts, slipping under the bolt, and even self loosening.

We can conclude that model (a), the simplest one, is sufficient for modelling structures when no slipping occurs under the bolts. Model (c) is slightly more complicated than model (b) but it is more accurate. We recommend model (d) for phenomena including rotation of the screw or nut, such as occur in self loosening.

The next step will be to confront our models with a very detailed one, which includes the thread shape and contact in the thread. Finally, an experiment will be carried out, which will impose dynamic excitation on the structure, leading to the self loosening of a bolt. All these results will be compared to validate the quality and efficiency of our models.

Acknowledgments

This work was granted access to the HPC resources of CALMIP under allocation 2014-P1121.

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