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NUMERICAL INVESTIGATION OF STATIC BEHAVIOR OF BOLTED JOINTS

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This paper investigates the static behavior of bolted joints and the extent of its control in the design of assembled structures. An analysis method is thus first developed highlighting stresses distribution in the junction to dimension the functional performance area. A description of the joint characteristics is presented. Numerical simulations, comparing the complete and simplified finite element models relevance, are then carried out. The integration of results of this analysis in the design of multiple bolted joints structures in finally presented. Experiments on a testbed, where a viscoelastic material is introduced in joints interfaces to enhance global damping, validate the approach developed.

Keywords: bolted joint, finite element, interface, stress, viscoelastic

1. Introduction

The bolted joint is a non-destructive device to join structures. It is used in most mechanical structures regarding its low cost, easiness of assembling and a wide range of forms and dimensions. It can be used to couple structures with standard materials and composite ones. Scientists have shown much interest in it over decades for its interesting advantages, and they tried to develop models to enhance its integration in various types of mechanical studies.

Understanding the static of the bolted joints is crucial to support other advanced studies. In this context, many works were developed. They focused on the clamping performance examination. This was done by the study of pressure distribution in the joint. Gould and Mikic (1972) used an analytical approach based on the finite element method to determine the pressure distribution in the contact area. The bolt however was not modeled. It was replaced by a load distribution that described pressure defined under the bolt head. The joint stiffness calculation is thus neglected. Naik and Crews (1986) developed an analysis method to determine stresses around the joint hole. Nevertheless, the bolt was considered rigid and frictionless. The study of the clearance influence was the goal of the work. Ireman (1998) developed a three-dimensional finite element model to take into consideration all important factors that had influence on the stress distribution. Marshall *et al.* (2006) studied the contact pressure distribution by experimental tests. The authors used an ultrasonic waves technique to detect contact pressure at the clamped interface. Mantelli *et al.* (2010) proposed a statistical method to determine the shape of the contact pressure distribution. Żyliński and Buczkowski (2010) studied the clamped bolt joint subjected to working pressure through a three-dimensional finite element model while considering a progressive opening of the joint.

Several finite element models were developed over the time to support these previous works and others. The choice of the model depends on the level of accuracy and simplicity of integration in the numerical simulation process. In real applications, the bolted joint presents nonlinear behavior due to friction between components and pieces to be assembled. This non-linearity can be taken into consideration or not, depending of the importance of its impact on the global dynamic response of the structure under investigation. Bolt joints models vary from a simple spring--damping element to a three-dimensional element. Few works were done to compare different models in the literature (Guzas *et al.*, 2015; Ibrahim, 2020; Li *et al.*, 2020; Giannella *et al.*, 2021) from the relevance and the computational efficiency point of view.

Bolted joints have been studied for decades for the mechanical engineering improvement goal. Recently, industrials and researchers have shown interest in these junctions from the design point of view, especially in aeronautics and space sectors. Structures complexity with several junctions have to be appropiately treated (Wang *et al.*, 2005; Crocombe *et al.*, 2006).

In this paper, an analysis method of the static behavior of bolted joints is presented to be implemented in a next step in the design of mechanical assemblies with multiple bolted joints. These assemblies have more complex geometries than what was treated in the literature regarding the studies on bolted joints. They represent an important problem of size that should be token in consideration when choosing bolts finite elements models. Two types of these finite elements models are studied to detect the influence of the model on the static behavior. To achieve this goal, the static preload and the contact surface are the first to be investigated. These characteristics are important since they condition the coupling functionality of the junction. Numerical simulation on a lap-joint structure is carried out to analyze the contact area. The normal stresses distribution, in the contact interface and through the assembly thickness, is presented. Finally, an experimental testbed created to study assemblies with multiple bolted joints is used to validate the numerical results and discuss its extent on more advanced studies such as damping design in global structures.

2. Contact surface and static preload

The bolted joint assures connection between components of mechanical structures. The goal is to attain cohesive behavior. Such a joint is composed of the bolt, nut and structural components stuck between the bolt head and the nut (Fig. 1). Contact interfaces take place between all these components.

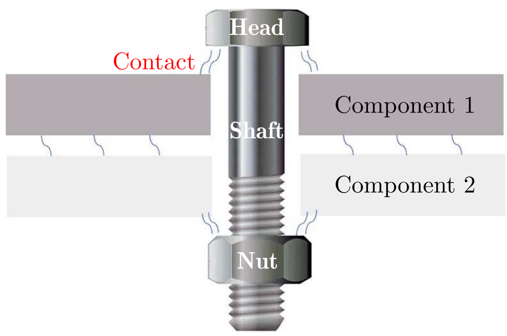

Fig. 1. Bolted joint model

To maintain connection, an area of the assembled parts is always under compression under the bolt head/nut after clamping (Fig. 2). This area does not represent any sliding motion unless desired (Reid and Hiser 2005). Elsewhere, it can occur. The compressed area prevents from bending and/or shear of the bolt. It should only work on tension/compression when connected. This is due to shear strength that is 70% lower than in compression (Guillot, 2007). Several experimental and theoretical works were carried out to determine the stress distribution in this area. Rotscher (1927) proposed first to equate the compressed area with a cone of 45° angle for each component. The stress was equal in each section of that cone. Lehnhoff *et al.* (1994) chose to work with an angle of 30° to have more conservative values. Fixing the angle method has limits. Some works developed formulas depending on geometric characteristics of the junction (Rasmussen, 1978). The conic contact area can be assimilated to a cylinder of the equivalent section in order to calculate its stiffness (Fig. 2).

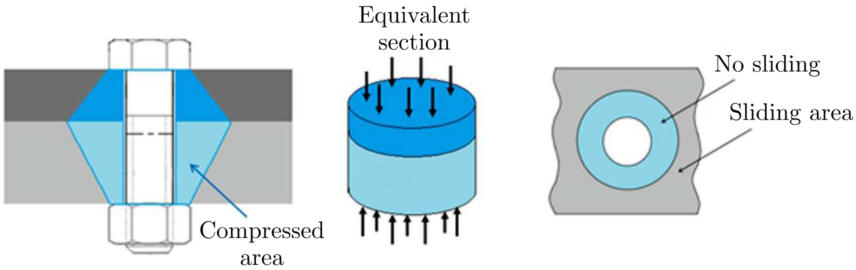

Fig. 2. Compressed area of the assembled part in bolted joints and the equivalent section

A clamping torque must be applied to the bolt to make the junction functional. This gives a rise to the called preload. Its intensity impacts the contact area dimensions. Therefore, the preload must be properly modelled. It is defined by compressive/tensile stress and torsional torque in the bolt shaft. Preloading increases fatigue resistance by reducing stress variation in the bolt. The clamping torque C is composed of three torques as follows

$$C = C_u + C_{ff} + C_{ft} \tag{2.1}$$

The C_{ff} torque is used to overcome friction of the nut threads on the bolt shaft leading then to torsional stress. The torque C_{ft} is used to overcome friction between the assembled part surface under the head/nut when turning. The useful torque C_u ensures the tension F of the bolt shaft. This conducts to clamping of the assembled parts. The exact value of this tension cannot be known due to the effect of friction. However, several formulations had been developed to estimate it. In this work, the Kellerman and Klein (1955) one is used. It is given by

$$F = \frac{C}{\frac{p}{2\pi} + 0.583d\mu_f + \frac{D}{2}\mu_t} \tag{2.2}$$

where p is the thread pitch; d is the thread flank diameter; μ_f is the friction coefficient in threads; D is the mean diameter under the head; μ_t is the friction coefficient under the head.

When clamping the bolt, the shaft is thus subjected to a tensile force and also a residual torque allowing its torsion to be demonstrated. This torque is estimated as follows

$$C_{ft} = F\Big(\frac{p}{2\pi} + 0.583d\mu_f\Big) \tag{2.3}$$

Most researches and industrials consider this torque negligible. In order to confirm this assumption, a study is carried out in this work and presented in Section 3.

3. Application of the lap-joint model

Numerical simulation is now carried out on a lap-joint structure to calculate the stress distribution at the contact interface using SDT toolkit in Matlab software (Balmés and Leclére, 2007). The lap-joint is composed of two plates of 180° of flatness and a single bolt to assemble them.

By implementing a finite element model of the bolt and the expression of tensile load described previously, the static behavior is under investigation. To have a faithful model of the bolted joint, the bolt must be efficiently modelled. The "Solid" model is considered as the most complete one (Fig. 3a). It presents a complete volume representation of all components. The contact between them is the surface-to-surface one. The issues of this model are generally related to friction in interfaces and at bolt threads (Abad *et al.*, 2012). This model presents many advantages. It is possible to transfer tensile and bending forces in the structure. The stress distribution can also be calculated in all the bolt parts. Adding to this, the modelling of contact on all interfaces is possible. More advanced studies dealing with elasto-plastic deformations in all components or pressure in the vicinity of the bolt can thus be done (Ju *et al.*, 2004). The inconvenience of this model is when treating a complex structure of large size in terms of degrees of freedom with several bolted joints. This type of model can be quickly complicated to process. Therefore, many levels of simplification had been proposed (Montgomery, 2002; Kim *et al.* 2007). The "coupled model" is another proposed alternative (Fig. 4a). It uses beam elements with equivalent stiffness placed at the bolt axis to model the shaft. The head and the nut are modeled as coupled nodes. The degrees of freedom of the head/nut are coupled with those of the assembled structures. Rigid links "RBE2" are used to connect these nodes. The beam nodes impose then rigid motion on the connected surfaces of the structure. This tends to stiffen the structure too much. The "RBE3" presents thus an alternative. The displacement of the beam node is a linear function of the surface displacements. Other models exist in literature and can be used in specific applications such as the "no bolt" model or the "hybrid" one (Montgomery, 2002).

The relevance of the RBE3 model against the Solid model is also studied. The goal is to study the influence of the model adopted on the static behavior of the bolted joint.

A compressive force $F = 3990\,\mathrm{N}$ is applied to the bolt shaft as a preload. This force corresponds to a clamping torque $C = 10\,\mathrm{Nm}$. A residual torque $C_t = 2.5\,\mathrm{Nm}$ is then deduced.

The application of the tensile force differs the model using solid elements from a model using the beam ones. For the solid model, the finite element layer at the bolt shaft must be removed (Fig. 3b). Then, two crowns of rigid links must be created on the free surfaces. The preload force is then applied to the master central nodes facing each other in the opposite direction.

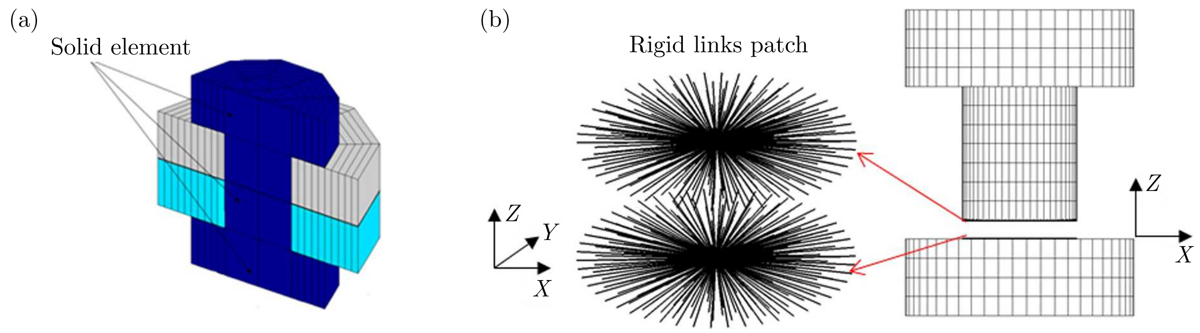

Fig. 3. Solid model: (a) finite element modelization, (b) application of the tensile force in the bolt modeled by solid elements

Two methods exist to model the compressive force in the coupled model with RBE3. The first is based on the same principle described above. However, no rigid links are added. The beam element is removed, and a force is applied in each node of the two sides in the opposite direction (Fig. 4b-config1). The second method consists of not removing any element. The compressive force is applied to the central node of the beam and the half-values are applied to each side of the head/nut (Fig. 4b-config2).

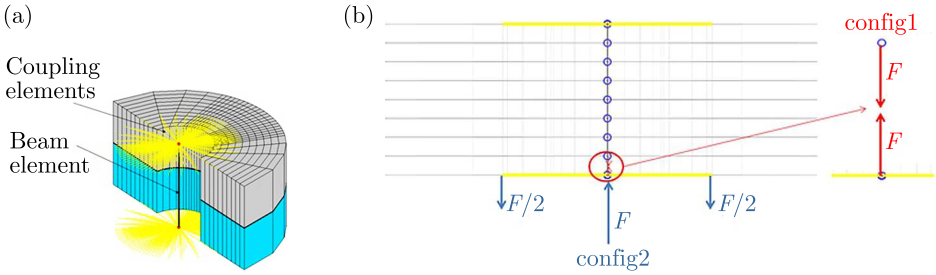

Fig. 4. RBE3 model: (a) finite element modelization, (b) application of the compressive force in the RBE3 model with beam elements

The normal stress is then calculated using the following equation

$$\{\sigma_{i3}\} = \boldsymbol{\sigma}\mathbf{V}_3 = \begin{bmatrix} \sigma_{11} & \sigma_{12} & \sigma_{13} \\ \sigma_{21} & \sigma_{22} & \sigma_{23} \\ \sigma_{31} & \sigma_{32} & \sigma_{33} \end{bmatrix} \begin{Bmatrix} 0 \\ 0 \\ 1 \end{Bmatrix} = \begin{Bmatrix} \sigma_{13} \\ \sigma_{23} \\ \sigma_{33} \end{Bmatrix} \tag{3.1}$$

where $\mathbf{V}_3$ is the normal unit vector on the surface (X, Y) and $\boldsymbol{\sigma}$ is the stress tensor.

Results are shown in Fig. 5. The resultant of static normal stresses at the Gauss points at the contact interface is presented. The residual couple C_t is not taken into account.

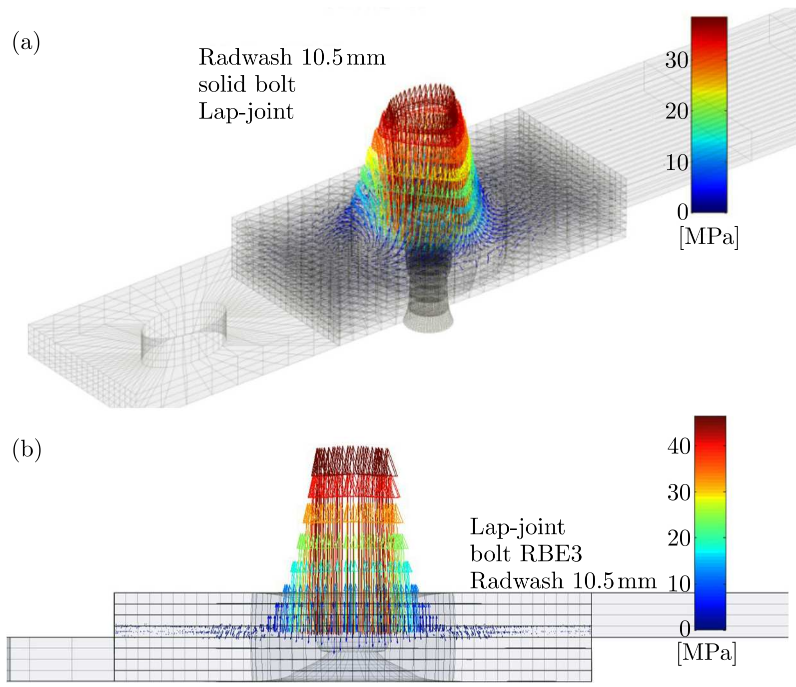

Fig. 5. Normal stress distribution: (a) solid model, (b) RBE3 model

From these results, the cohesion of the assembly is clear. It is illustrated by high values of the normal load near the bolt through the hole for the two models. A gradual decrease in the stresses is also observed in a conical form moving away from the shaft/hole. It is where the sliding can occur.

The similarity between the distribution forms in the two models is also noticed. However, the maximum values are little different. The RBE3 model presents the most important values. This can be explained by the fact that its elements less stiffen the structure giving greater stresses.

To highlight this difference in amplitudes, a comparison of the normal stress values in the three model is established (Fig. 6). The RBE2 model is added in this study. The hole is indicated by its limits at $[-3.5, 3.5]$ mm. The limits of the head/nut are indicated by lines at $[-6.5, 6.5]$ mm.

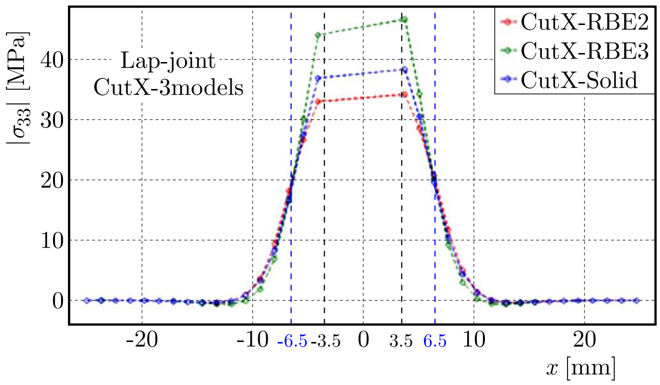

Fig. 6. Normal stress evolution on the contact interface for the Solid model, RBE3 model and RBE2 model

Amplitudes of normal stresses are different near the hole. Far from it, the curves are closer. The stiffening effect of the RBE2 model is more important than of the Solid model.

This distribution is also observed through thickness of the assembly as shown in Fig. 7. R_{hole} and R_{head} are radii of the hole and the bolt head, respectively.

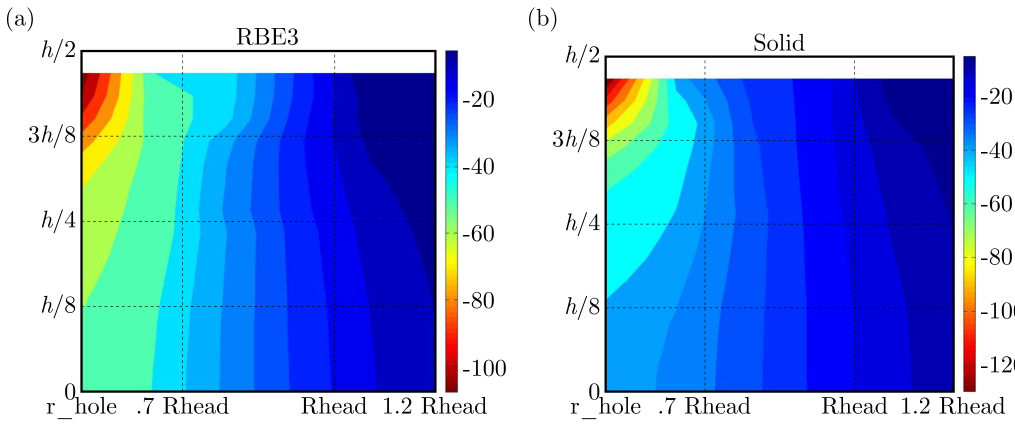

Fig. 7. Normal stress evolution in the assembly thickness: (a) RBE3 model, (b) Solid model

One can then conclude from the results above that with the simplified model (RBE3), comparable results can be obtained with a complete one (Solid) while reducing the computational time.

The influence of residual torque (2.3) on modeling of the preload is now studied. Normal stresses and shear stresses in the contact interface are calculated by introducing the residual torque in addition to the compressive force. Almost the same values were found for normal stresses. This can be explained by the fact that shear stresses are negligible compared to normal ones. Thus, the shear stress distribution is presented (Fig. 8). The maximum value of these stresses is 1 MPa. This value is too small compared to normal stresses of 40 MPa. It can, therefore, be neglected.

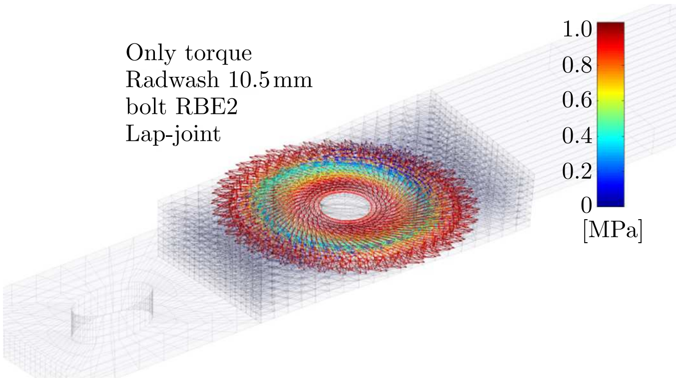

Fig. 8. Distribution of shear stresses on the interface of contact

4. Experimental validation and discussion

As seen previously, a part of the assembly in a bolted joint must be under functional pressure to maintain cohesion. The pressure-free zones do not affect the mechanical functionality of the junction, but they can be a source of damping in the connection. This topic has increased the interest of researchers for the past decade (Gaul and Becker, 2010; Goyder *et al.*, 2013). The introduction of a viscoelastic material in the junction interface is a relevant solution found to enhance damping in global bolted structures (Hammami *et al.*, 2016; Liao *et al.*, 2017).

A testbed has been created to study structures assembled by multiple bolted joints (ten M6 bolts). A viscoelastic material (Smactane50) is introduced in the junction interfaces to study generated damping as shown in Fig. 9a,b. After dismantling the assembly, dark zones are observed on the viscoelastic layers (Fig. 9c). These zones are the actual contact areas in the assembly. The joints present a geometric defect consisting of an inclination by a 1° angle and, thus, the structures are not perfectly flat at the interfaces. One can deduce, despite the reduction of the contact area due to geometric defects, that the functional area persists under the bolt head.

(a) (b) (c)

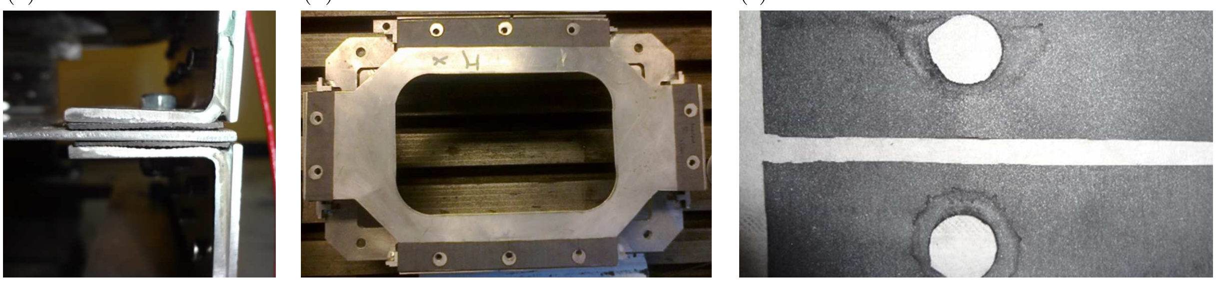

Fig. 9. Experimental test: (a) bolted joint, (b) bolted joints interfaces, (c) pressure areas in bolted joints

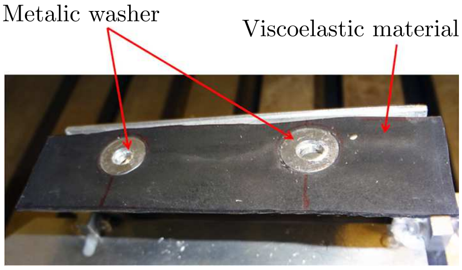

Fig. 10. Introduction of a metallic washer in the bolted joint with a viscoelastic material

To ensure functional performance of a bolted joint, one can thus add a metallic washer in the stressed zone in the assembly as shown in Fig. 10. The estimation of its dimensions can thus be done by the numerical study presented in previous Sections and associated to the distribution of normal stresses in the assembly.

The influence of the dimension of the washer diameter on the damping levels is illustrated in Fig. 11b. The evolution of modal damping is presented for the first two modal frequencies when scanning a range of the storage modulus [7.2 MPa, 10 GPa] and for a loss factor value of $\eta = 1$. These parameters describe the viscoelasticity of the material introduced in the joint interface. The washer and parts to be assembled are made of Aluminum with Young's modulus of 72 GPa and Poisson's ratio $\nu = 0.3$. Two values of the washer radius are compared: 4.11 mm and 7.11 mm. The last one corresponds to radius of the area that must be under pressure.

The overall structure modeled with the finite element method presents an important number of degrees of freedom (Fig. 11a). The RBE3 model is thus used to model the bolts in the structure since it is the best compromise between the relevance and processing cost.

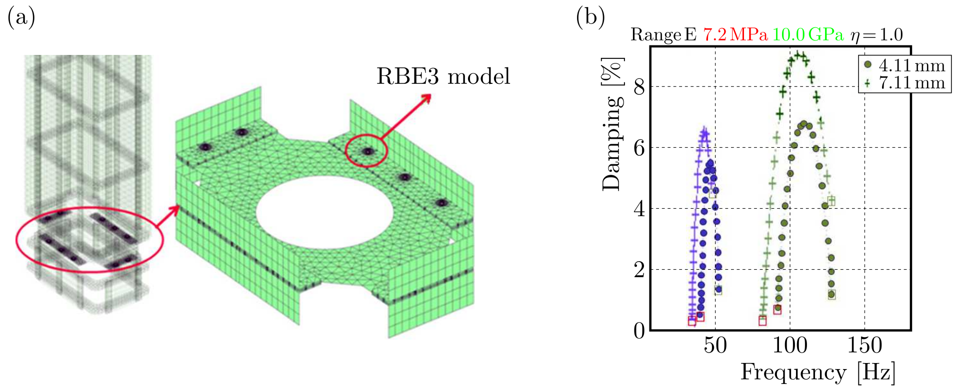

Fig. 11. Assembly with multiple bolted joints: (a) FEM model, (b) evolution of modal damping for two values of washer radius

It is deduced that when the washer radius decreases (from 7.11 mm to 4.11 mm), the damping increases and, conversely, the frequencies decrease which is reasonable due to the viscoelastic behavior of the material. However, the functional performance of the junction is no longer optimal and can thus generate undesirable consequences.

5. Conclusion

Through this paper, an analysis approach based on static investigation on behavior of bolted joints is carried out to guide the design of mechanical assemblies with multiple junctions. The distribution of stresses on the interface of contact of bolted joints is studied. This type of junction becomes functional when clamping the bolt. An area of compressed parts under the head/nut defines thus the contact area where no sliding is allowed. A numerical simulation is then presented. The normal stresses distribution on the interface is illustrated. It presents a concentration near the shaft edges and a decrease in amplitudes far from it. The finite element method is used to establish numerical simulations. A comparison between Solid, RBE2 and RBE3 models is presented. In-depth studies can be carried out based on this static analysis. The dynamic one is thus established to highlight the influence of dimension of a metallic washer, introduced in the bolted joint and made of a viscoelastic material on the damping evolution. Experimental results recorded on a testbed with multiple bolted joints have validate the static analysis presented in this work.

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References

1. Abad J., Franco J.M., Celorrio R., Lezáun L., 2012, Design of experiments and energy dissipation analysis for a contact mechanics 3D model of frictional bolted lap joints, *Advances in Engineering Software*, **45**, 42-53
2. Balmès E., Leclère J.-M., 2007, *Viscoelastic Vibration Toolbox*, User's Guide Etienne Version 1.0
3. Crocombe A.D., Wang R., Richardson G., Underwood C.I., 2006, Estimating the energy dissipated in a bolted spacecraft at resonance, *Computers and Structures*, **84**, 5-6, 340-350
4. Gaul L., Becker J., 2010, Damping prediction of structures with bolted joints, *Shock and Vibration*, **17**, 359-371
5. Giannella V., Sepe R., Citarella R., Armentani E., 2021, FEM modelling approaches of bolt connections for the dynamic analyses of an automotive engine, *Applied Science*, **11**, 4343
6. Gould H.H., Mikic B.B., 1972, Areas of contact and pressure distribution in bolted joints, *Journal of Engineering for Industry*, **94**, 864-870
7. Goyder H., Ind P., Brown D., 2013, Measurement of damping due to bolted joints, *International Design Engineering Technical Conferences and Computers and Information in Engineering Conference*, ASME
8. Guillot J., 2007, Modélisation et calcul des assemblages vissés. Généralités, *Techniques de l'ingénieur, L'expertise téchnique et scientifique de référence*
9. Guzas E., Behan K., Davis J., 2015, 3D finite element modeling of single bolt connections under static and dynamic tension loading, *Shock and Vibration*, **2015**, 2, 1-12
10. Hammami C., Balmes E., Guskov M., 2016, Numerical design and test on an assembled structure of a bolted joint with viscoelastic damping, *Mechanical Systems and Signal Processing*, **70-71**, 714-724
11. Ibrahim A.M., 2020, *On the Effective Finite Element Simplification of Bolted Joints: Static and Modal Analyses*, Rochester Institute of Technology
12. Ireman T., 1998, Three-dimensional stress analysis of bolted single-lap composite joints, *Composite Structures*, **43**, 3, 195-216
13. Ju S., Fan C., Wu G.H., 2004, Three-dimensional finite elements of steel bolted connections, *Engineering Structures*, **26**, 3, 403-413
14. Kellermann R., Klein C., 1955, Untersuchungen über den Einfluss der Reibung auf Vorspannung und Anzugsmoment von Schraubenverbindungen, *Konstruction*, **7**, 2, 54-68
15. Kim J., Yoon J.-C., Kang B.-S., 2007, Finite element analysis and modeling of structure with bolted joints, *Applied Mathematical Modelling*, **31**, 5, 895-911
16. Lehnhoff T.F., Ko K.I., McKay M.L., 1994, Member stiffness and contact pressure distribution of bolted joints, *Journal of Mechanical Design*, **116**, 550-557
17. Li P., Li W., Wei P., Wang Q., 2020, Research on finite element analysis and modelling of bolted joint, *IOP Conference Series: Materials Science and Engineering*, **892**, 012084
18. Liao X., Zhang J., Xu X., Zhang C., 2017, Dynamic response analysis in bolted joint structure with viscoelastic layer and experimental investigations, *Journal of Vibroengineering*, **19**, 1585-1596

19. Mantelli M.B.H., Milanez F.H., Pereira E.N., Fletcher L.S., 2010, Statistical model for pressure distribution of bolted joints, *Journal of Thermophysics and Heat Transfer*, **24**, 432-437

20. Marshall M.B., Lewis R., Dwyer-Joyce R., 2006, Characterisation of contact pressure distribution in bolted joints, *Strain*, **42**, 31-43

21. Montgomery J., 2002, Methods for modeling bolts in the bolted joint, [In:] *ANSYS User's Conference*

22. Naik R.A., Crews J.H., 1986, Stress analysis method for a clearance-fit bolt under bearing loads, *AIAA Jpurnal*, **24**, 1348-1353

23. Rasmussen J., 1978, A two body contact problem with friction, *Euromech Colloquium*, 115-120

24. Reid J.D., Hiser N.R., 2005, Detailed modeling of bolted joints with slippage, *Finite Elements in Analysis and Design*, **41**, 547-562

25. Rotscher F., 1927, *Die Maschinellente*, Springer

26. Wang R., Crocombe A., Richardson G., Underwood C., 2005, Modelling of damping in small satellite structures incorporating bolted joints, *Conference on Small Satellites*, SSC05-IX-7

27. Żyliński B., Buczkowski R., 2010, Analysis of bolt joint using the finite element method, *Archive of Mechanical Engineering*, **LVII**, 3, 275-292

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