Development of a Numerical Technique for the Static Analysis of Bolted Joints by the FEM

D. Valladares, M. Carrera, L. Castejon, C. Martin

Abstract— **The use of numerical simulation tools applying the finite element method for the design and analysis of bolted joints involves a series of difficulties providing complexity to the numerical modelling of apparently simple mechanical systems. In order to solve this difficulties and to achieve accurate, effective and fast calculations, it is necessary to take into account several simplifications of the mesh models during the preprocess step. Factors such as the substitution of the bolts and nuts' threads for contact conditions with friction interaction between cylindrical surfaces, the adequate characterization of the preload existing in the different members of the joint, or the correct modelling of contact interactions between the parts suppose a series of problems that will be necessary to overcome so as to successfully face the analysis of a mechanical system with bolted joints subjected to load conditions. The aim of this paper is the development of a numerical technique for the static analysis of preloaded bolted joints by the FEM using the software ABAQUS Standard and a mesh modelling with solid type elements. Parting from a new design with bolted joints for the longitudinal beams of an innovative modular semitrailer, the different problems associated to this type of analysis were stated first. Then, the numerical technique developed has been described and all its considerations have been applied to that specific semitrailer bolted joints.**

Index Terms— **Bolted joint, preload, FEM, static analysis, semitrailer.**

I. Bolted joints. Introduction

In construction and machine design, a bolted joint consists in the assembly of two or more members through screw threaded elements or fasteners such as bolts and nuts.

Not only longitudinal loads but also transversal ones can be applied to bolted joint. However, an optimal mechanical design must try that bolts only be working under longitudinal loads because transversal loads are withstood worse by them.

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Therefore if transversal loads are considered in the design of a mechanical joint, it has to be tried that bolts don't absorb them.

In every design it must be verified that none of the elements comprising the bolted joint fails under the stresses that can appear. Depending on the way the joint works, it can fail statically in different modes:

1) Wearing out or crushing in the material of the plates appearing at the inner faces of the drill hole putting up the bolt.

2) Excessive tensile loading in the plates joined. Drill holes suppose a weakening that could lead to the failure and breakage of the part due to tearing of the material when bolts are working under shear and crushing, in the presence of transversal loads.

3) Shear stress in the stud of the bolt. This load occurs when the plates try to slide the one respect to the other.

4) Tensile stress in the stud of the bolt. This failure mode occurs in load configurations that can generate longitudinal forces in the bolts.

5) A combination of tensile and shear stress in those load cases including longitudinal and transversal loads.

6) In preloaded (or clamp loaded) joints, when transversal loads appear the joint resists by means of the frictional behaviour between the plates which is assisted by the existing preload. If the friction force limit is reached by this transversal load, the plates slide and the studs contact the drill hole inner walls, giving way to shear and crush strains.

It is also necessary to consider that a preloaded bolted joint possess an operational force limit that is not functionally allowable to overcome. Higher axial forces could make the bolt itself absorb all the tensile load; in that case, compression force would disappear at the joined plates or flanges and the joint would lose its functionality.

Additionally, those bolted joints subjected to a variable tensile loading, that is, exposed to a dynamic loading, are subjected to fatigue action [1]. Peterson noted that common failure distribution in bolts is approximately 15% under the bolt head, 20% at the end of the thread, 65% at the thread beside the nut face [2].

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II. F.E. MODELLIZATION OF BOLTED JOINTS. STATE OF THE ART

The current development of modern computers and the common use of powerful processing computers and clusters on engineering design and calculation are allowing the progressive application of the finite element method to more and more complex and large mesh models (numerically speaking) within reasonable CPU processing times and computational costs.

This is why nowadays is possible to carry out a more accurate numerical modelling of bolted joints, which allows for more realistic analyses than, for example, ten years before. Like Kim et al explain [3], early research in this field already pointed out that three-dimensional and detailed modelling were desirable for bolts so as to accurately predict the mechanical behaviour of structures assembled with bolted joints. By this way several FE models with variable complexity and detail level were developed by these authors using the software ANSYS, obtaining the best experimental-numerical correlation for those models simulating both the bolt and the nut with solid elements.

Authors such as Wileman et al [4] or Yang et al [5], among others, have also analysed through the FEM the stiffness behaviour of the bolted region and the contact pressures appearing in the joint members.

In this context, the present paper is focused on the development of a FE numerical technique allowing to simulate static analyses of preloaded bolted joints in a functional and effective way using the software ABAQUS Standard.

III. SIMPLIFICATIONS CONSIDERED IN THE MESH MODEL

In order to achieve the above-mentioned objective, and as a first step, is necessary to carry out a description of several simplifications that will allow to produce a numerical model of a bolted joint in a fast and accurate way. While in the preprocess of the FE model it has been used the software PATRAN 2012.2., the numerical analysis has been performed with the software ABAQUS Standard. The mesh model has been created by means of first order interpolation continuum "solid" type elements C3D8R and C3D6 [6].

These types of elements are recommended for reasonable fine meshes, that is, for meshes modelled with small element sizes compared with the model dimensions. Therefore, a model with a large number of elements will be necessary so as to obtain a correct simulation. Additionally this entails to consider several simplification in the model in order to reach an adequate compromise between the model accuracy and the computational cost associated to the preprocess and the calculation.

As stated before, this paper is focussed on a preloaded bolted joint, comprising basically two joined members (such as plates or flanges) with clearance holes, a bolt and a nut. Washers have not been considered in this modelling methodology in order to achieve a simpler model, although they are usually used in practice so as to avoid non desirable stresses appearing at the bolt.

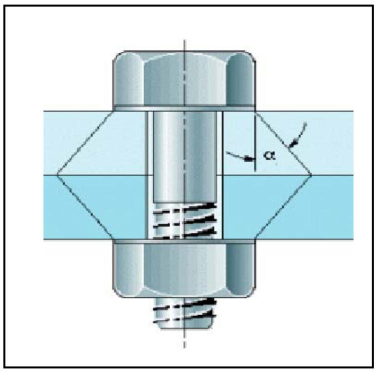

Fig. 1. Bolted joint. Stiffness region [1]

The following considerations have been taken into account for simplifying the numerical model:

1) The use of "shell" type planar elements and "beam" type elements has not been considered adequate for the accuracy level required in the bolt modelling to correctly simulate the interaction between the bolt and the assembled members. As a result of this decision, "solid" type elements were used in all the mesh model.
2) Bolt and nut threads have not been modelled, giving way to a cylindrical geometry much easier and faster to mesh. Threads modelling would entail the use of automatic mesh options, creating a finer mesh consisting of tetrahedral and wedge solid elements. This would increase the number of elements, reduce the time increment size during the analysis and worsen the model accuracy. It would also be necessary to apply a torque load for achieving the preload as well as the use of complex contact conditions between the bolt and nut threads that would increase the computational cost.
3) Contact conditions have been defined between the following surface pairs (figure 2): the base of the bolt head and the upper surface of the upper flange (part 1), the upper surface of the nut and the lower surface of the lower flange (part 2), part 1 lower surface and part 2 upper surface, the bolt stud and the inner face of clearance holes at parts 1 and 2, the bolt stud and the nut.
4) The preload (or clamp load) has been simulated by means of the application of nodal tensile forces at the end of the bolt stud and nodal compression forces at the exterior face of the nut.

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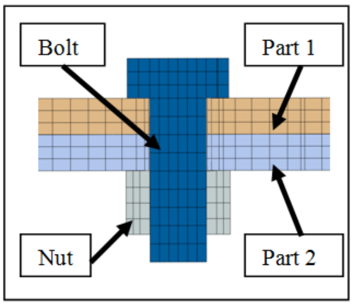

Fig. 2. FE model of the bolted joint. Cross-sectional view

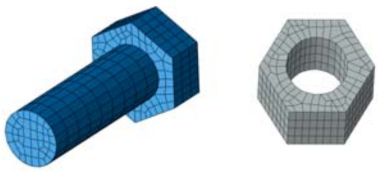

Fig. 3. Bolt FE model

IV. Contact Modelling

An adequate contact modelling between surfaces that contact in the model is essential for incorporating correct force interactions between the different parts comprising the bolted joint. Next the different contact types defined in the models are described:

- **Contact no 1: Contact between the base of the bolt head and the upper surface of the upper flange (part 1).**

The interaction between the base of the bolt head and the first part to join (in fig. 2, part 1 at the top) has been modelled by means of the definition of a "tied" contact type [6] between two surfaces, one acting as the slave surface and the other as the master surface: whereas the slave surface comprises the lower external element faces at the bottom of the bolt head, the master surface comprises the upper external element faces surrounding the drill hole at part 1.

It has been considered a drill hole diameter slightly higher than the bolt stud, according to ISO standard indications for metric thread. Figure 6 shows the elements corresponding to the master and the slave contact surfaces where lines connecting the nodes of the different surfaces show the constraints between them.

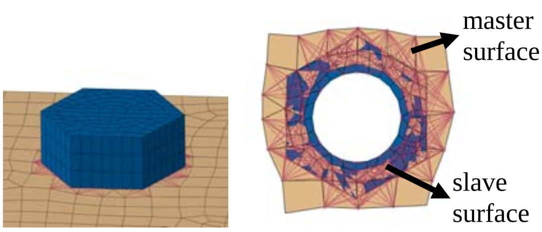

Fig. 4. Contact no 1: Tied contact surfaces, bolt-part 1

A tied contact constrains each of the nodes on the slave surface to have the same value of displacement as the point on the master surface that it contacts. This contact type allows for rapid transitions in mesh density within the model but it is very important that the tied surfaces be precisely in contact at the start of the simulation.

- **Contact no 2: Contact between the upper surface of the nut and the lower surface of the lower flange (part 2).**

The interaction between the nut and the second part to join (in fig. 2, part 2 at the bottom) has been modelled in the same way, defining a "tied" contact type between two surfaces. The slave surface comprises the upper element faces of the nut and the master surface comprises the lower external element faces surrounding the drill hole at part 2. Figure 5 shows the elements corresponding to these slave and master surfaces.

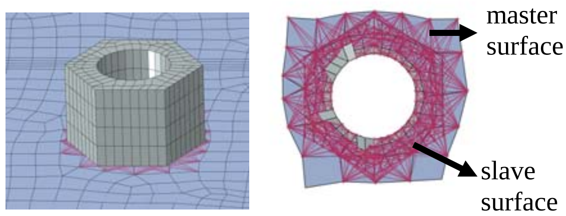

Fig. 5. Contact no 2: Tied contact surfaces, nut - part 2

Interactions due to contacts no 1 and 2 are fundamental for a correct application of the preload in the model.

- **Contact no 3: Contact between the bolt stud and the inner faces of the clearance holes at parts 1 and 2, and between the bolt stud and the inner faces of the nut hole at the treaded zone.**

These interactions have been modelled by means of a pair of contact surfaces or contact pair. In this case, for the definition of the contact pair, surfaces have been considered as deformable and a friction property has been assigned to them, with a friction coefficient of value 0.25 [1] (steel - steel, in dry conditions) and a slip tolerance of 0.02 mm between surfaces.

When a contact pair contains two surfaces, the two surfaces are not allowed to include any of the same nodes and it is necessary to choose which surface will be the slave and which will be the master. The following rules must be taken into account:

- The larger of the two surfaces should act as the master surface.
- If the surfaces are of comparable size, the surface on the stiffer body should act as the master surface.
- If the surfaces are of comparable size and stiffness, the surface with the coarser mesh should act as the master surface.

According to these rules, there have been considered a master surface defined by the exterior faces of the bolt stud and a slave surface defined by the exterior nodes located at the clearance holes of the joined parts and at the nut hole

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Figure 6 shows these two surfaces and how are they combined. For this contact type is not necessary that defined surfaces be in contact at the start of the simulation.

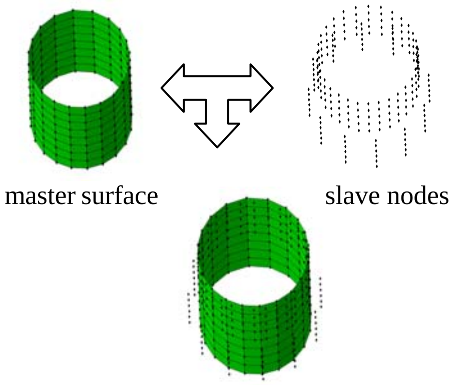

Fig. 6. Contact no 3: Between the bolt stud and the hole exterior nodes of part 1 and 2 and the nut

As can be observed at figure 6 there is a clearance distance between the bolt stud and the hole nodes. This dimension is standardized and depends on the bolt diameter (1 mm in this example with M14 bolts). On the other hand, between the bolt stud and the nut nodes there is no gap because this contact correspond to the threaded joint that generates the fastening.

It can be pointed out that ABAQUS allows the definition of this contact including in the same slave surface the clearance hole nodes and the nut threaded hole nodes.

- **Contact no 4: Contact between the joined parts.**

Finally, the interaction between the surfaces in contact of the joined members must be defined (part 1 and part 2 in this case). The contact type used is the same as no 3 contact: contact pair between deformable surfaces with friction properties assigned (0.25 for friction coefficient and 0.02 mm for the slip tolerance admitted in the analysis). It has been defined a master surface comprising the upper faces of the solid elements at the rear part (part 2) and a slave surface comprising the lower nodes at the upper part (part 1). Special care must be put in positioning the slave nodes respect to the master surface in the case of parts with coincident edges: It could be necessary to move the slave nodes at these coincident nodes a small distance in the model (for instance 0.05 mm) so as to avoid convergence problems during the implicit analysis

V. PRELOAD AND LOAD MODELIZATION

Concerning the preload modelling, two amplitude curves have been considered so as to apply independently and consecutively in the analysis in the first place the preload, and in the second place those loads due to forces applied to the joint. Therefore, and as defined in figure 7, during an ABAQUS Standard step spanning from 0 at the start to 1 at the end, the preload has been applied increasing uniformly from the start point to point 0.5 in the step. From this point on and until the end of the step, the preload has been maintained with the 100% of its value.

External forces loading the bolted joint have been applied starting from point 0.5 of the step, increasing linearly to value 1 at the end of the step. By this way at the end of the analysis can be simulated an overlapping of 100% of the preload with 100% of the external load.

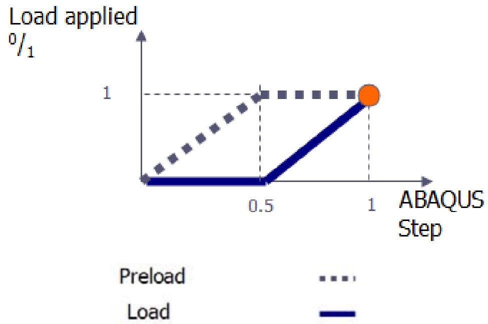

Fig. 7. Preload and load modelling in ABAQUS Standard

The preload has been modelled as two opposite forces shared out equally between the following nodes:

- Tensile force acting on all the nodes at the end planar surface of the bolt stud.
- Compression force acting on all the nodes at the most exterior planar surface of the nut

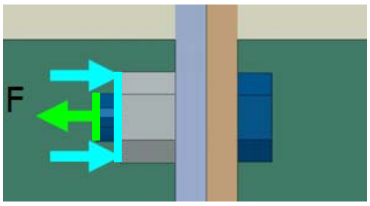

Fig. 8. Preload applied to nodes located in bolt and nut surfaces

The preload value necessary for the joint can be calculated by means of the following equations [1]:

$$\begin{cases} F_i = 0.75 \cdot F_p & \text{(1) Equation for non permanent joints} \\ F_i = 0.9 \cdot F_p & \text{(2) Equation for permanent joints} \end{cases}$$

$$F_p = A_t \cdot S_P \qquad \text{(3) Equation for the proof load}$$

Where:

F_i = Preload

F_p = Proof load

A_t = Stressed area in the bolt

S_p = Proof strength

VI. NUMERICAL TECHNIQUE DEVELOPED

Taking into account all the considerations described in points III, IV and V, it can be defined a numerical technique for the

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static FE simulation of preloaded bolted joints in ABAQUS Standard.

The steps necessary for applying this technique are listed next:

1) Modelling of the joint members with solid type elements. A minimum of three elements is recommended for modelling correctly the thickness in plates or flanges.

2) Modelling of bolts and nuts with a fine mesh of small size solid type elements.

3) Modelling of the steel mechanical behaviours with an isotropic elastic-plastic material model corresponding to the Mises classic metal plasticity [6]. The steel strain-stress curves introduced in ABAQUS have been approximated as bilinear curves taking into account the material modulus of elasticity, the yield strength, the ultimate strength, the elongation at break and the Poisson modulus.

4) Assignment of material properties for each component (bolts, nuts, flanges, etc).

5) Definition of contact conditions no 1, 2, 3 and 4.

6) Preload application.

7) Loads (force, displacement, temperature, etc) and boundary conditions application.

8) Definition of amplitude curves for the preload and the load application during the analysis.

9) Calculation in ABAQUS Standard.

VII. Analysis of Bolted Joints in the Longitudinal Beam of a Modular Semitrailer

This technique has been applied to the calculation of the bolted joints for the longitudinal beams of a modular three-axle semitrailer which has been divided in two halves: a front part and a rear part. Concretely a FE model of one of its longitudinal beams has been created, considering a total length measured from the position of the rear diapress to the end of the vehicle.

There have been considered the following cantilever load cases so as to assess the performance of the suggested design:

- Load case no 1: Total payload of 27,000 kg, uniformly distributed on the longitudinal beams. For only one beam and at the rear zone simulated this would be equivalent to a distributed vertical load of 25,604 N

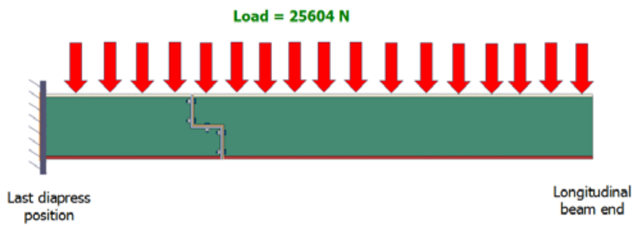

Fig. 9. Load case no 1

- Load case no 2: Forklift truck (model STILL FM-X-25) track load corresponding to its maximum weight at the front axle (5734 kg) applied at the end of the longitudinal beam. For only one beam this is equivalent to a vertical load of 28125 N applied at the rear end.

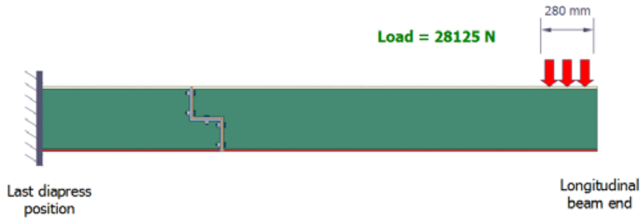

Fig. 10. Load case no 2

Fig. 11. Forklift truck STILL FM-X-25

Front part Rear part

Fig. 12. Longitudinal beam. Front and rear parts simulated

Results for load case no 1:

A maximum vertical displacement of 4.9 mm was obtained at the end of the longitudinal beam.

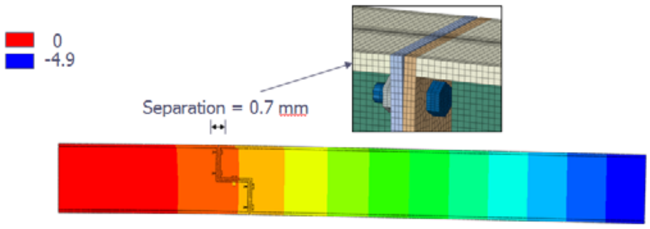

Fig. 13. Load case no 1. Vertical displacement (mm)

Concerning the bolted flanges made of high strength steel with yield stress 519 MPa, Von Mises stresses were:

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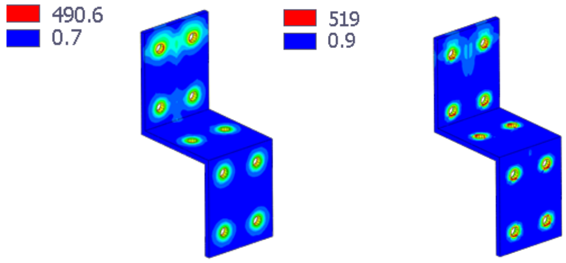

Fig. 14. Load case no 1. Von Mises stress (MPa) at flanges

In the same way figure 15 shows Von Mises stresses obtained at bolts and nuts made of 12.9 quality steel (quenched and tempered) [7].

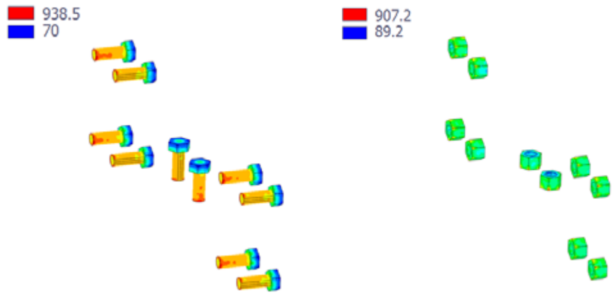

Fig. 15. Load case no 1. Von Mises stress (MPa) at bolts and nuts.

Results for load case no 2:

In this case, a greater maximum vertical displacement of value 15.6 mm was obtained at the end of the longitudinal beam.

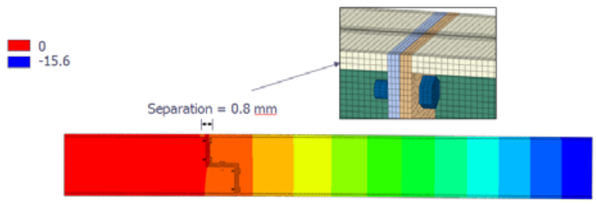

Fig. 16. Load case no 2. Vertical displacement (mm)

Von Mises stresses at the bolted joints were the following:

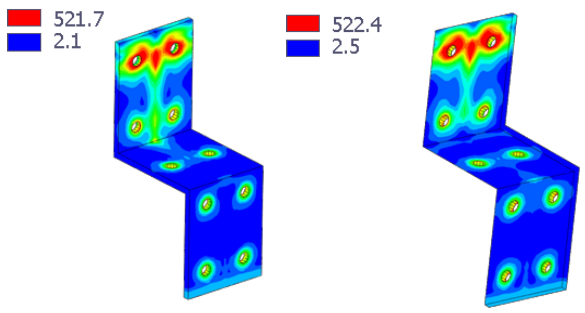

Fig. 17. Load case no 2. Von Mises stress (MPa). Flanges

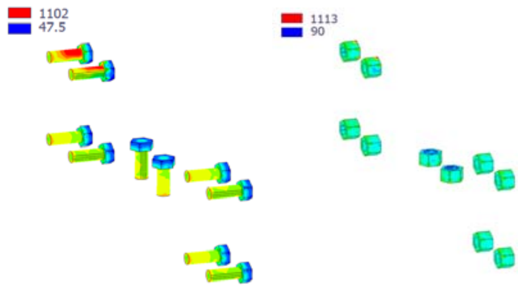

Fig. 18. Load case no 2. Von Mises stress (MPa). Bolts and nuts

In figure 17 both flanges show higher Von Mises stresses than the previous load case, reaching values higher than the yield stress at the top joints. Figure 18 shows that bolts located at the top are the most stressed ones, reaching their yield stress of 1100 MPa.

VIII. Concluding Remarks

The present paper has developed a FE numerical technique allowing to simulate static analyses of preloaded bolted joints using the software ABAQUS Standard. Effectiveness and functionality were taken into account as key factors.

This technique has been applied in the analysis of the bolted joints assembling the longitudinal beams of a new design of modular semitrailer. Due to high values of Von Mises stress reached at the flanges and at several bolt studs, it has been considered necessary to change some dimensions and materials in the proposed design. This would improve the behaviour of the bolted joints.

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