Numerical analysis of bolt Joints considering the complete tightening process and failure

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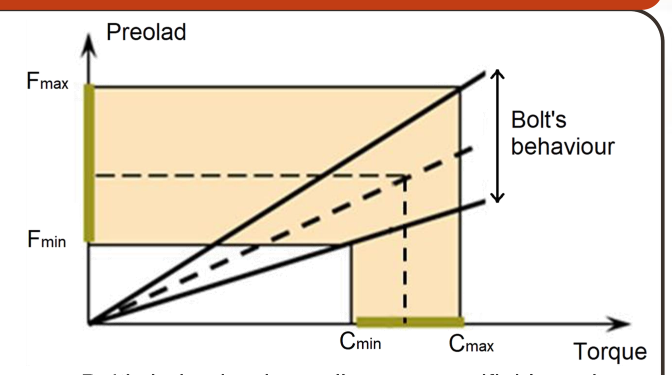
Introduction

The large number of bolted joints used in the aerospace industry, associated with the high dispersion of compressive forces during their tightening [1], motivates an in-depth study of their behaviour. The introduction of Industry 4.0 in the development of technologies drives the digitization of processes in order to be able to predict and optimize the results. In addition, an Abaqus numerical model provides more flexibility and better control of input and output data, and can overcome the limits of experimental benches. The complexity of this numerical model is due to the complex surface contact, with modelling of the threads, the strongly non-linear behaviour, with the integration of the destruction of the elements, and also the considerable relative displacements. The objective is to converge towards a robust 3D numerical model capable of simulating the clamping process until failure while representing reality as faithfully as possible.



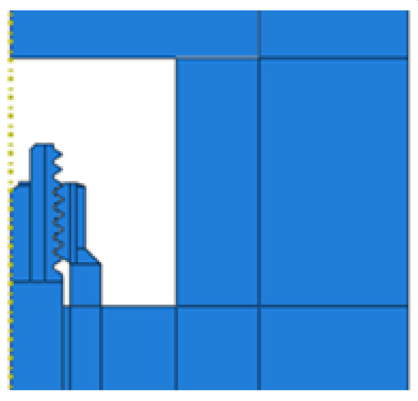
Bolt's behavior depending on quantifiable and unquantifiable parameters

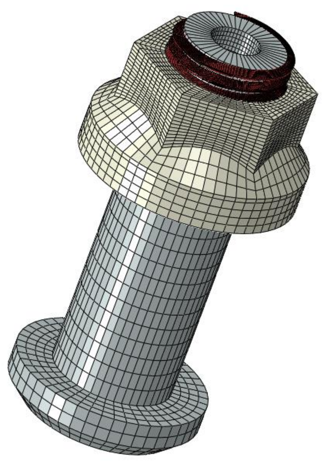
Methodology

Starting model

Axisymmetric model

Designed with parts of revolution, this model makes it possible to obtain the maximum preload at break. However the angle of the helix of the threads is not considered and thus bending of the screw is neglected.



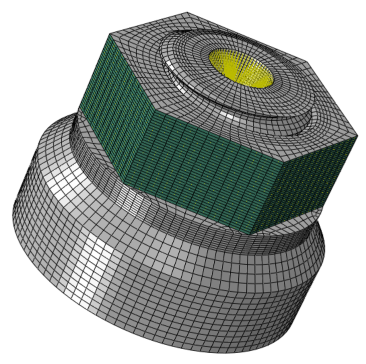


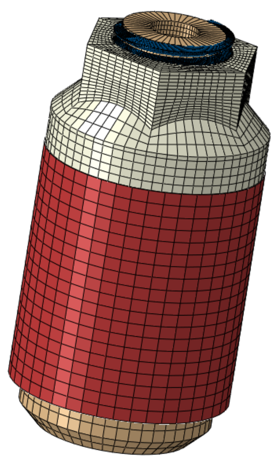
Switch to 3D

Representation of the entire screw and nut with kinematic coupling at the threads. Only an axial displacement is imposed so it is impossible to obtain the tightening torque and to take the friction in the threads into account.

Adding the rotation of the nut

The rotation of the nut is applied using a displacement constraint at the hexagon. At this stage, the response to the crushing of intermediate parts is lacking.



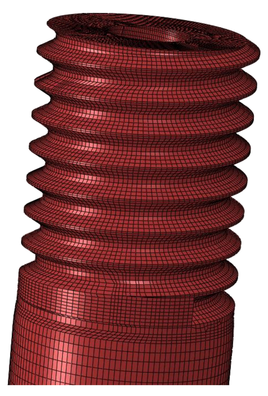


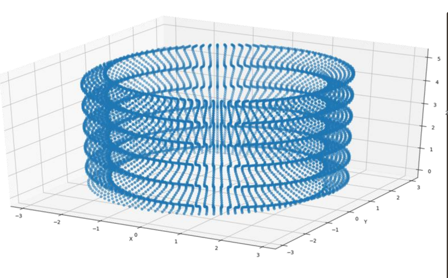
Adding the spacer

Complete tightening process and failure are considered in this model but the kinematic coupling is responsible for a discontinuity in the stress field. This discontinuity creates stress concentrations that lead to a type of failure different from that observed.

Deleting the coupling

The automatic mesh of Abaqus no longer converges. A manual mesh is necessary but it is complicated and redundant. This is not compatible with a parametric model because of its unacceptable effect on robustness.



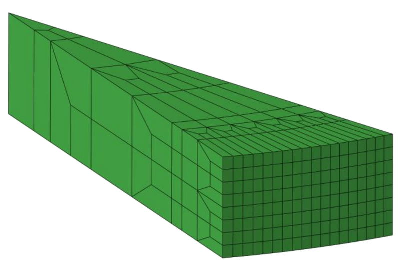


Screw thread creation routine

To overcome the mesh problem, a mesh creation routine is used to modify an existing cylindrical mesh [2]. For a good mesh density in the threads, this method implies using a fine mesh on the whole screw and the whole nut, which leads to a mesh of several hundred thousand elements.

Optimization of the mesh

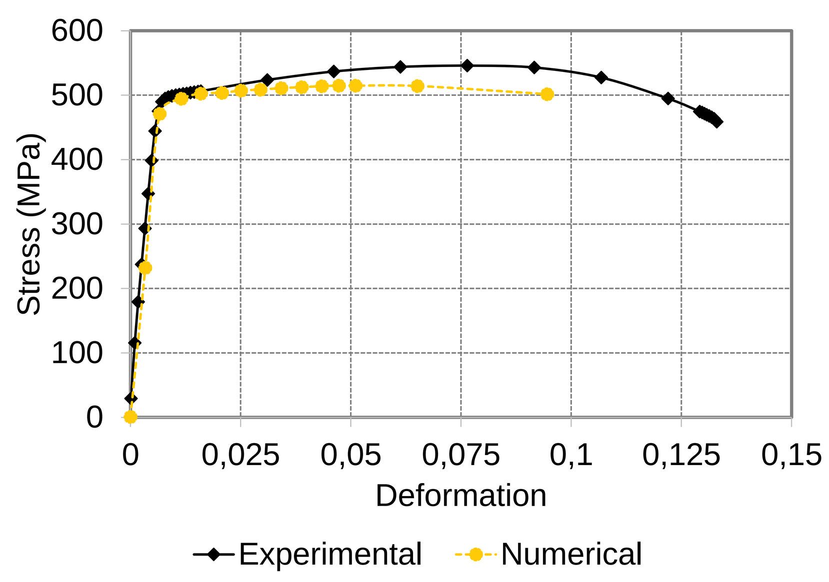
Mesh gradients are implemented axially and radially in order to refine the mesh without exploding the computation time. This gives a ratio of 114 between the number of elements in the threads and inside of the screw.

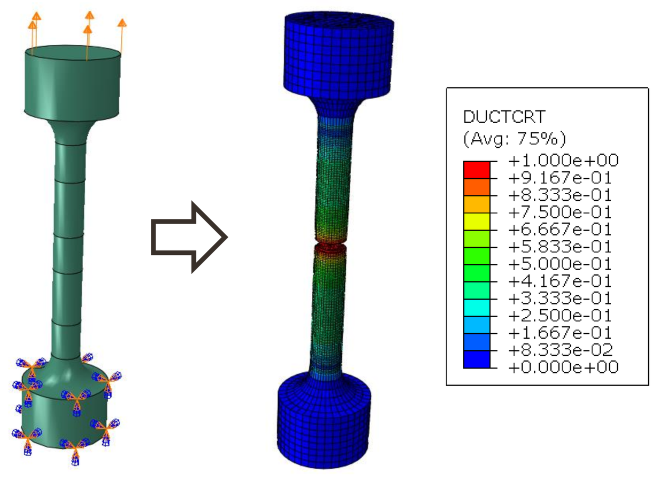


Actual model

Results

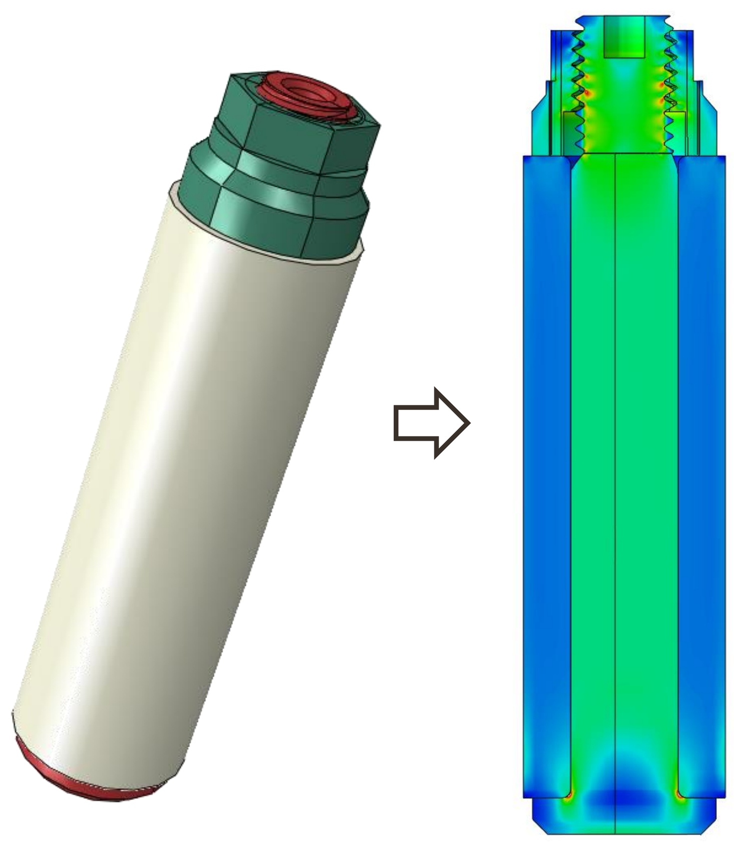
Verification of the damage law





- Solver : Explicit
- Element : C3D8R
- Damage law : Ductile Damage
- Material : Elastic & Plastic

Behaviour analysis



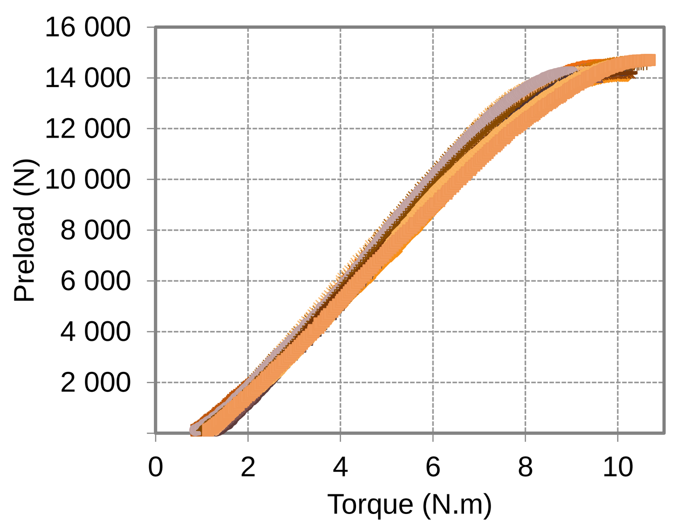
The stress field is consistent with the behaviour of the bolted connection :

- The helical thread of the screw leads to bending of the screw
- The most heavily loaded thread is the first one [3]
- Rupture occurs at the level of the nut threads

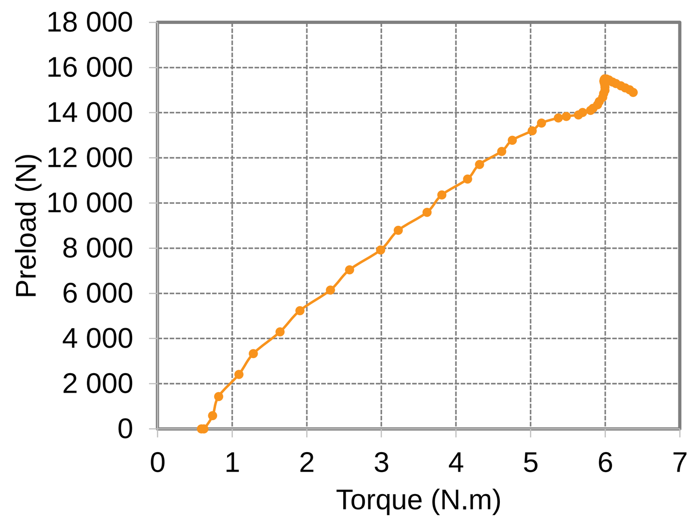
The heterogeneous distribution of stress and therefore of the contact is not sufficiently representative of reality. This is due to the complexity of the contact: the coarser the mesh, the more this phenomenon is amplified [4][5][6].

Correlation with experimental results

Experimentally, the rupture occurs on average at 13 881 N with a standard deviation of 486 N.



Numerically, the rupture occurs at 15 498 N.



Conclusion

The methodology shown is similar to an incremental improvement. It avoids the superposition of error sources and thus controls the convergence of the calculations better when the model becomes more complex. As the model is very sensitive due to its complexity, numerous tests on the choice of elements, the choice of solver and the size of the mesh were indispensable (even though they are not mentioned above).

Finally, the model gives a higher clamping strength than the one measured experimentally. This discrepancy is due to the ovalization of the nut, which has not yet been taken into account in this numerical model. The ovalization of the nut plasticizes the threads locally, so it will change the whole stress field and the plastic deformation field, both linked to the damage law. However, the global behaviour of the model is verified at several points with a damage law corresponding to real tests and phenomena validated in the literature.

In the search for improvement, work on friction will provide a better estimation of the tightening torque [7] and work on adaptive meshing could solve the recurrent problems of distortion on the elements in contact in the threads and make the model even more robust.

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