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# ASSEMBLY BY PRESTRESSED OR HIGH-STRENGTH BOLT.

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**Abstract:** The analytical method provides a very good approximation of the tightening of a bolted assembly. Consequently, one can be satisfied with a fast analytical calculation to rather precisely check the tightening of a bolted assembly. A model finite element is hardly more precise at least of a point of considering total. One could still look further into these problems by determining in experiments the constraints in the screw and the part of a pre stressed bolted assembly. It would be also interesting to see more precisely, how evolves locally the constraints in the part and thescrew.

In this work, one learns how to quickly check the tightening of a pre stressed assembly bolted under static head. For that, one determines by an approach RDM the cons **Keywords:** Assembly pre stressed analyzes digital, constraints in the components of the assembly bolted.



## 1.INTRODUCTION:

What it is necessary to keep in mind: The simple design is to be searched in the choice of the devices of assemblies, this in order to have:

- Transmission of the more exact efforts
- Calculation more exact.
- Materials better utilizes.

The checking of the assemblies is done while following the transmission of each effort in each part, the resistance of an assembly being that of its weakest point. One thus may find it beneficial to carry out the same degree of security for all the elements. The number and the importance of the assemblies on building site must be as reduced as possible, insofar as the assembly and transport conditions allow it. It cannot exist of regulation or handbook; however perfect is it, which gives to the designer the possibility of carrying out a drawing without the provided rules or indications being interpreted on the basis of the personal judgment, the professional experience and technical good sense of the designer.

Control of the deformations: the assemblies by bolts requested in traction have only rather weak deformations. On the other hand the assemblies by requested bolts perpendicular to their axes have more raised deformations because of the standardized games. One can however reduce these deformations by adopting gauged bolts or many pre stressed bolts.

Dynamic stresses: the actions on the buildings are generally static, except in the event of presence of revolving machines, machines of handling, sieve, etc the vibrations can generate loosening of nuts or simply to harm the effectiveness of the bolted assemblies. One can in this case adopt devices of check nuts or assemblies by pre stressed bolts.

In the same way the assemblies by pre stressed bolts are essential in the Low temperatures: it is advisable to use bolts profiting from a warranty of energy of minimal rupture during the crash test in inflection.

Parasitic efforts in the assemblies: one must

take account of it case of seismic actions (where it is necessary to dissipate the energy of the jolts by the plastic deformations of the frame and the assemblies) and in the event of risk of ruin by tiredness.


Figure1: assembly by simple covering. An assembly by simple covering creates one bending moment parasitizes and the assembly becomes deformed:
It creates for itself parasitic internal efforts then:

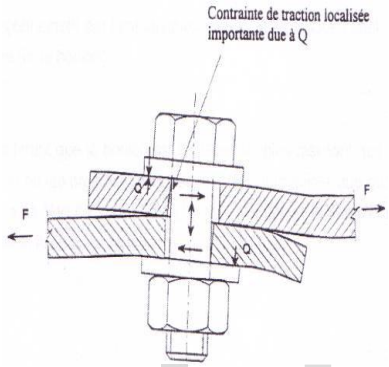



figure2: deformation of an assembly. The solution is to create symmetrical assemblies

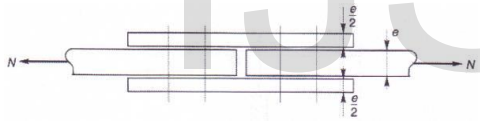



Figure3: symmetrical assemblies. Additional moment of well aware of confusing neutral axis and axis of marking in the beams lattice: it is often the case when angles are used. In example Ci below the secondary moment is of

$M = (F2 - F1)d$.....1.1

Additional moment by action leverage: to avoid that Ci should be designed sufficiently rigid soles. If the soles are not rigid enough the bolts play the part of pivot, and one has a compressive force Q at the edges external of soles (effort of lever). By balance the traction effort in the bolts is worth:

$Fb = F + Q$.....1.2

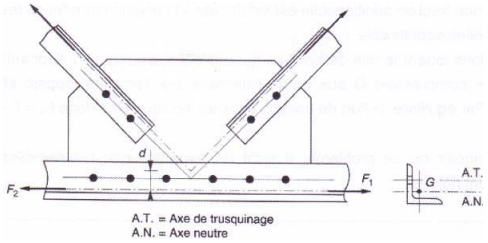



Figure4: assembly reinforced by sole.

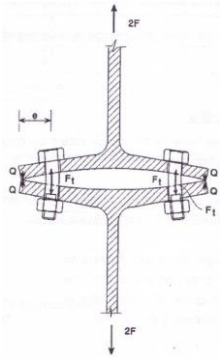



Figure5: deformation of an assembly by traction.

**2. Design of the assembly by bolt:** From the point of view of construction, there are two basic methods of the design of an assembly per bolt:

- Assembly by bolt with pin A
- Assembly by bolt with nut B

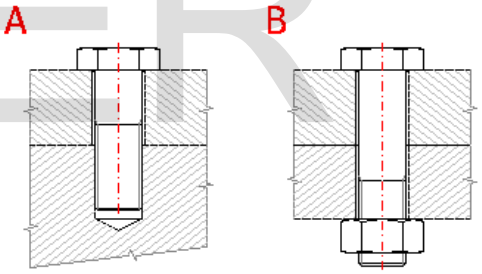



Figure 6: Assembly by bolt.

**2.1 Load of the assembly by bolt:** According to the type of load, an assembly by pre stressed bolt must answer the various requirements; this has like consequence a different method of calculating of the pre stressing of assembly. There are three various types of load for calculations of the assemblies by bolt: Load in line with bolt: The assembly by bolt is under the action of the thrust load F. Here the pre stressing of assembly ensures compactness or more precisely, the sealing of the assembly during operation. The pre stressing of the assembly must thus be enough high to ensure a sufficient residual pre stressing of the part connected after the constraint of the assembly by the active force and thus the link of force

necessary between surfaces of contact. Load perpendicular to the axis of the bolt: The assembly by bolt is under the action of the radial force Fr, acting in the plan of the parts connected. The pre stressing of assembly ensures the load capacity of shearing of the assembly using the forces of friction. The transverse force acting on the assembly must thus be transferred between the parts connected by the friction, which is due to the pre stressing of tightening of the bolts. Load combined:
The assemblies being due to a combined load

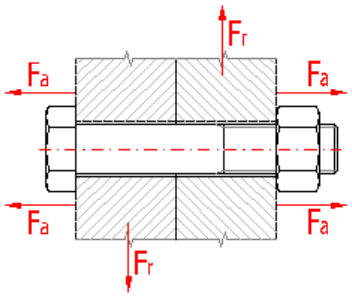



Figure7: Various types of load for calculations of the assemblies by bolt.

**2.2 Type of the load:**
This list makes it possible to the users to define the type of the load which acts on the assembly. The assembly by bolt can be conceived for the following types of load:

A- Silencer (static).
B- With pulsation.
C- Passage.
D- AlterNet asymmetrical.
E- Alternated symmetrical.

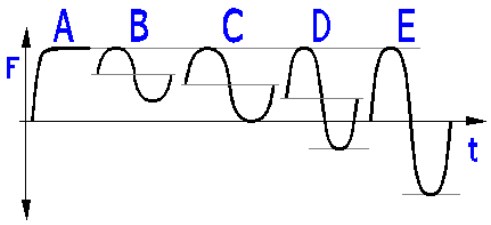



Figure 8: Type of the load which acts on the assembly.

F- With the assemblies due to a cyclic load (load B with…) it is also necessary to carry out, besides the current strength checks, controls of the bolt by the strength of tiredness.
G- Choose the desired type of the load. After choice, calculation will be modified in configuration which corresponds to the selected type of load the parameters, which do not have any meaning for the type chosen, will be hidden.

**3. Automatic design of the bolt:** In software SOLIDWORKS one will make the manual design as one can open the library of the SOLIDWORK which one goes selected among the types of
bolts this choice and called automatic design; from the toolbox SOLIDWORKS with the selected ISO standard one among the types according to:

- bolt with square collars
- bolt with cruciform print
- cap screw set split
- bolt auto—drilling machines
- bolt CHC
- bolt hexagonal
- bolt hexagonal with fine step
- bolt without head 6hollow sides
- bolt without split head

And here is an example (figure) which we applied has our simulation.

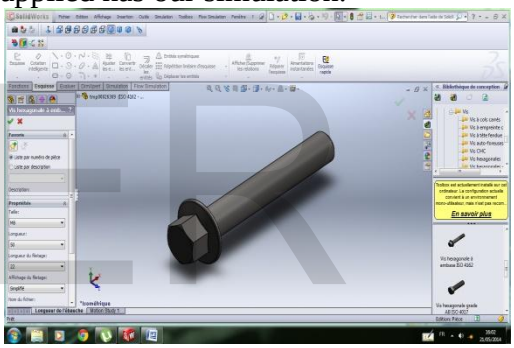


Figure 9: bolt hexagonal ISO 4162 – M8. 50. 22-N

**3.1 .type of study:**
In the machine analysis there is several type of studies like continuation:

- Statics
- Frequency
- Buckling
- Thermal
- Test of fall
- Tiredness
- Nonlinear statics
- Linear dynamics
- Design of a device under pressure

With our case of study we chose the linear static study.

**3.2. Linear static analysis**
When a loading is applied to a body, this one becomes deformed and the effects of the loading are transmitted in all the body. The external forces induce internal forces and reactions which lead the body to a state of

balance.

The linear static analysis calculates displacements, the strains, the stresses and the forces of reaction under the effect of the loadings applied.

Assumption of staticity: All the loadings are applied slowly and gradually until the desired intensity. Once this intensity reached, the loadings remain constant in time. This assumption enables us to neglect the effects of the inertial forces and damping, because induced speeds and accelerations are negligible. A variable loading in the time, which would induce inertial forces and/or of damping considerable, could justify a dynamic analysis. The dynamic loadings vary in time and, in many cases, induce inertial forces and of damping important, which cannot be neglected.

Assumption of linearity. The relation between the loadings and the induced answers is linear. If you double for example the intensity of the loading, the answer of the model (displacements, strains and stresses), will be doubled. You can apply the assumption of linearity if:

All the materials used in the model check the law of Hooke, i.e. the calculated constraint is directly proportional to the deformation. Induced displacements are sufficiently small to be unaware of the changes generated by the loading on the matrix of stiffness.

The boundary conditions do not vary during the application of the loading. The loading must be constant in time in intensity, direction and distribution. It should not change under the effect of the deformation of the model.

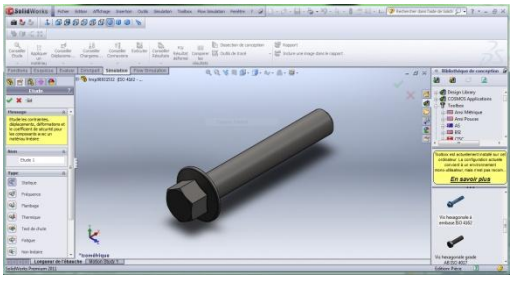


Figure 10: linear static study

### 3.3. Choice of material:

Before carrying out a study of analysis, one will define all the material properties necessary to the corresponding type of analysis. For example, the modulus of elasticity is necessary for the static studies, for our example one will choose the steel alloy. (Figure11).

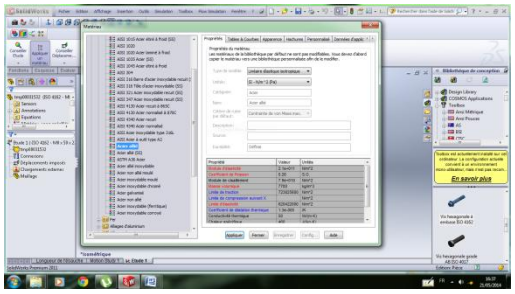


Figure11: choice of material.

### 3.4. Displacement imposed:

For volumes, this kind of imposed displacement defines all the degrees of freedom of translation in zero. For the bolts, it defines the degrees of freedom of translation and rotation in zero. For connections of bars, it defines the degrees of freedom of translation in zero. When one uses in this type imposed displacement, no geometry of reference is necessary.

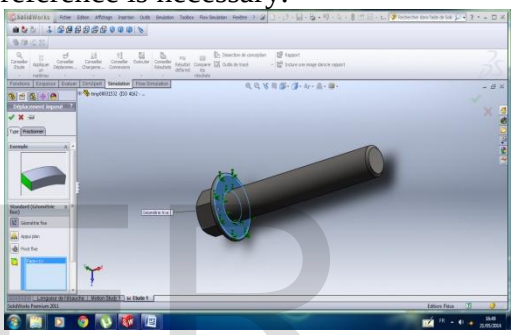


Figure12: fixed geometry.

### 3.5. Force applied or couples applied:

a) Force/Torque:

Application of the forces, the moments or the couples presenting a uniform distribution on the faces, edges, points of reference, tops and beams in any managerial staff, the executive staff of structural studies. The specified force is applied to each selected top, edge, face or beam. When we apply a force to a face or an edge, the value specifies represents only the amplitude. Thus, if we apply a normal force to an entirely cylindrical face, the vector sum of the forces applied is worthless. We can apply non uniform forces to the faces, edges of hulls and beams. A non uniform force is defined by a value and a distribution. The value of the force is equal to the sum of all the absolute values of the forces applied to each face. The distribution of the force is described by the coefficients of a polynomial of the 2nd degree according to axes X and Y of the frame of reference of reference, as described for the variable pressure.

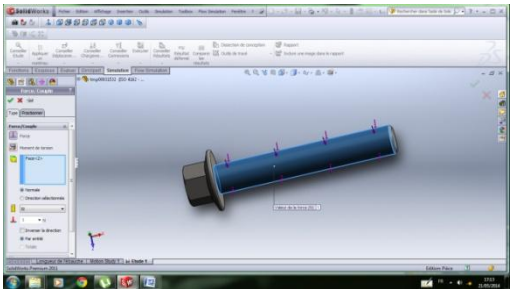


Figure 13: application of the loads on the bolt.

### 3.6.le grid:

During the grid of a part or one assembly using volume elements, the software generates one of the following types of elements, according to the active options of grid for the study:

· Grid average quality. The automatic mesh generates volume linear tetrahedral elements.

· Grid top quality. The automatic mesh generates volume parabolic tetrahedral elements.

The linear elements are also called first order elements or of a lower nature. The parabolic elements are also called elements of the second order or a higher nature.

A linear tetrahedral element is defined by 4 nodes in the tops and 6 right edges which connect them. A parabolic tetrahedral element is defined by 4 nodes in the median tops, 6 nodes and 6 edges. A schematized representation of the linear and parabolic tetrahedral elements volume is given in the following figures

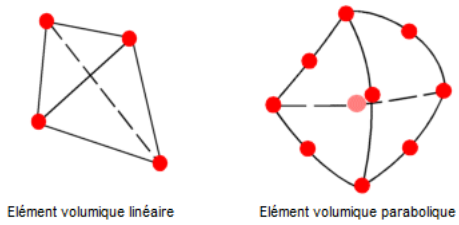



Figure 14: choice of elements for grid.

For the structural problems, each node of a volume element has 3 degrees of freedom representing the translations in the three orthogonal directions of space. The software uses directions X, Y and Z of the total Cartesian frame of reference to formulate the problem (figure 15).

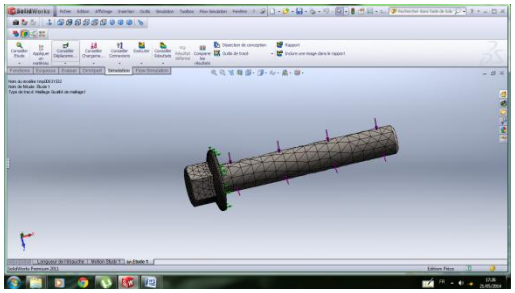


Figure15: linear mesh

### 3.7. Design and geometry of the bolt:

Sometimes it is suitable, from the technological point of view or construction, to use a special bolt in the assembly instead of an ordinary prismatic bolt with various cross sections. For example where necessary of a precise assembly of the parts using the adapted bolts or using a bolt connected to a stem shortened to reduce the influence of the additional bending stresses. Flexible bolts with a special treatment are also frequently used for the assemblies exposed to a variable load.

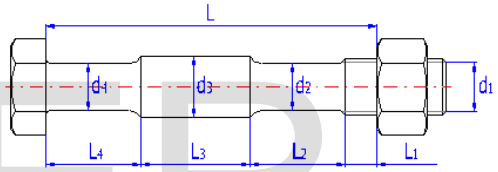



Figure 16: bolt with shortened stem.

### 4. Design of surfaces of contact under the heads (nuts) of the bolts:

For the assemblies by bolt, three types of basic designs of surfaces of contact below heads (nuts) of the bolts are used.

A- Circular surface of contact:

The most normal case more running and for the contact in the assemblies by bolt with normal nuts if surfaces of contact are perpendicular to the axis of the bolt.

B- Conical surface of contact: Special use for the assemblies by bolt where the precise centering of the part connected against the axis of the bolt is required. This requires a special conical nut and a conical seat in the hole for the bolt, consequently a greater requirement of precision in the production.

C- Spherical surface of contact.

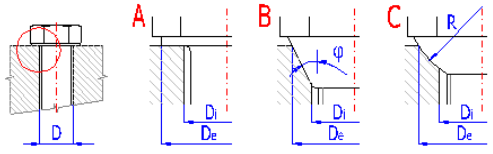


Figure 17: Different contact surfaces. Special use in the assemblies where the perpendicularity of surfaces of contact to the axis of the bolt and consequently its additional bending strain can be envisaged. This requires a special ball nut and a spherical seat in the hole for the bolt. What is very demanding for the technology of production?

**4.1. Pre stressed , power struggles and operational diagram of the assembly:** In this paragraph we can find the power struggles acting in the assembly by designed pre stressed bolt. The power struggle in an assembly by bolt entirely in charge for pre stressing of support given is calculated in the last part. The power struggles are illustrated in the image in the lower part of this paragraph.

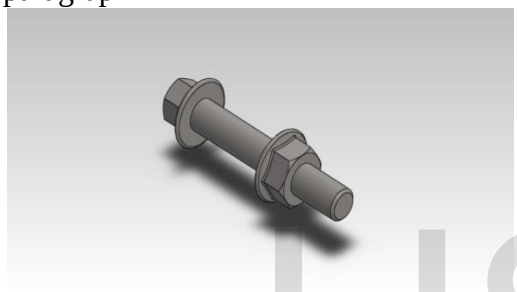


Figure 18: Assembly bolt screw nut.

a) Assumptions:

· Touch SCREW fix.

· Tightening applied between face FEMALE SCREW.

· Force applied to the threaded body.

b) Information on the model:

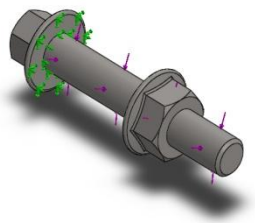


figure19: Name of the model: Assemblage1 Current configuration: Assembly bolt screw nut

Table1: Model Information

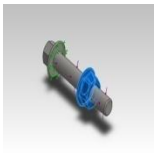
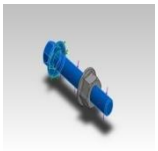
| Name of the document and reference | Treaty like | Volumetric properties |
|---|---|---|
|  EndChamfer | Volume body | Mass: 0.0172446 lb Volume: 0.0619907 in^3 Density: 0.27818 lb/in^3 Weight: 0.0172329 lbf |
|  FlangeHexFillet | Volume body | Mass: 0.0586841 lb Volume: 0.210957 in^3 Density: 0.27818 lb/in^3 Weight: 0.0586443 lbf |

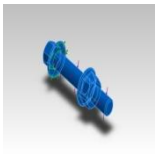
Table2: Properties of the study:

| | |
|---|---|
| Name of study | Study 2 |
| Type of analysis | Statics |
| Type of grid | Volume grid |
| Heating effects: | Activated (E) |
| Thermal option | To include thermal loadings |
| Temperature of worthless deformation | 25 Centigrade |
| To include the pressure of the fluid Calculated by Solid Works Flow Simulation | Disabled (E) |
| Type of solvor | FFE More |
| Stiffening stress: | Disabled (E) |
| Low stiffness: | Disabled (E) |
| Inertial relieving: | Disabled (E) |
| Options of incompatible interdependent contact | Automatic |
| Great displacement | Disabled (E) |
| To check the external forces | Activated (E) |
| Friction | Disabled(E) |
| Adaptive method: | Disabled (E) |

Table3: Units

| | |
|---|---|
| System of units: | IF (MKS) |
| Length/Displacement | mm |
| Temperature | Centigrade |
| Angular velocity | rad/ Dry |
| Forced /pressure | N/m^2 |

Table4: Properties of material:

| Reference of the model | Properties | Components |
|---|---|---|
|  | Name: Steel alloy<br>Type of model: Linear isotropic rubber band<br>Criterion of ruin by default: Constraint of max.von Mises Elastic limit: 6.20422e+008 N/m^2<br>Limit of traction: 7.23826e+008 N/m^2<br>Modulus of elasticity: 2.1e+011 N/m^2<br>Poisson's ratio: 0.28<br>Density: 7700 kg/m^3<br>Modulus of rigidity: 7.9e+010 N/m^2<br>Thermal dilation coefficient: 1.3e-005 /Kelvin | Volume body 1 (End Chamfer) (Hexagon Flange Nut ISO - 4161 - M8 - N-1), volume Body 1 (Flange Hex Fillet) (ISO 4162 - M8 X 50 X 22-N-1) |

Data of the curve: N/A

Table5: External actions:

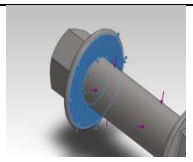
| Name of displacement imposes | Image of imposed displacement | Details of imposed displacement |
|---|---|---|
| Fixed-1 |  | Entities: 1face (S)<br>Type: Fixed geometry |

Table6: Resulting forces

| Components | X | Y | Z | Resultant |
|---|---|---|---|---|
| Force of reaction (NR) | 1572.51 | 0.00014383 | -.00108089 | 1572.51 |
| Moment of reaction (Nm) | 0 | 0 | 0 | 0 |

Table7: loading Force

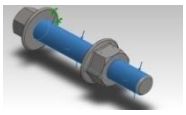
| Name of the loading | Image of the loading | Details of the loading |
|---|---|---|
| Force-1 |  | Entity:1face (S)<br>Type: Normal force<br>Value:500000N |

Table8: Information on the contact:

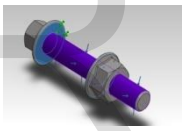
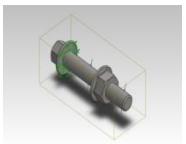
| Contact | Image of the contact | Properties of the contact |
|---|---|---|
| Contact between whole-1 |  | Type: Contact between pair tight adjustment<br>Entities:3face (S)<br>Advanced: Surface on surface |
| Total contact |  | Type: Inter dependent<br>Components:1 component (S)<br>Options: Compatible mesh |

Table9: Information on mesh:

| | |
|---|---|
| Type of grid | Volume grid |
| Mesh used: | Grid based on the curve |
| Points of Jacobien | With the nodes |
| Size of maximum element | 3.29658 mm |
| Size of minimum element | 0.659317 mm |
| Quality of grid | High |
| To re-mesh the parts in failure with an incompatible grid | Disabled (E) |

Table10: Information on mesh – Details:

| | |
|---|---|
| Full number of nodes | **22825** |
| Full number of elements | **13905** |
| Maximum aspect ratio | **81.485** |
| % of elements having an aspect ratio < 3 | **93.4** |
| % of elements having an aspect ratio > 10 | **0.813** |
| % of distorted elements (Jacobien): | **0** |

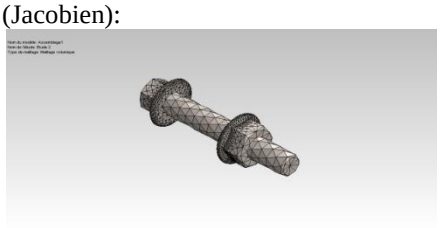


Table11: Resulting forces:
Forces of reaction:

| Together of selections | Units | Sum X | Sum Y | Sum Z | resulting |
|---|---|---|---|---|---|
| Entier model | N | 1572.51 | 0.00014383 | -0.00108089 | 1572.51 |

Table12: Moments of reaction:

| Together of selections | Units | Sum X | Sum Y | Sum Z | resulting |
|---|---|---|---|---|---|
| Entier model | N.m | 0 | 0 | 0 | 0 |

Table13: Contraintes

| Name | Type | Min | Max |
|---|---|---|---|
| Contraintes1 | VON: constraint of Von Mises | 0 N/m^2 Node: 1 | 4.97941e+008 N/m^2 Node: 13613 |

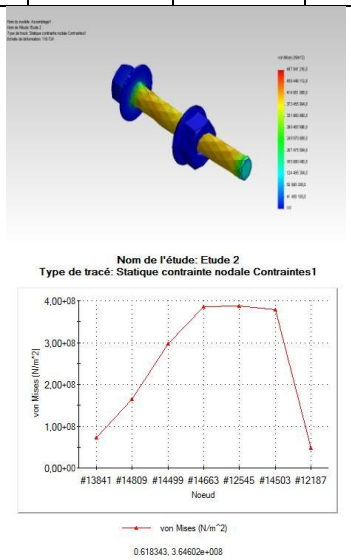



Figure2 : Assembly 1- Study 2- constraint- constraints1

Table14: displacement

| Name | Type | Min | Max |
|---|---|---|---|
| Déplacements1 | URES: resulting displacement | 0 mm Node: 1 | 0.0551651mm Node: 22260 |

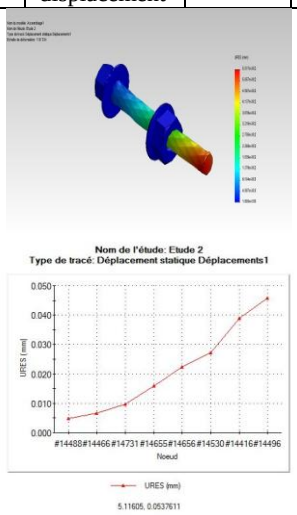



Figure21 : Assembly 1- Study 2- Diplacements - Diplacements 1

Table15 : Déformations

| Name | Type | Min | Max |
|---|---|---|---|
| Déformations1 | ESTRN: Equivalent deformation | 0 Element: 1 | 0.00182335 Element: 12191 |

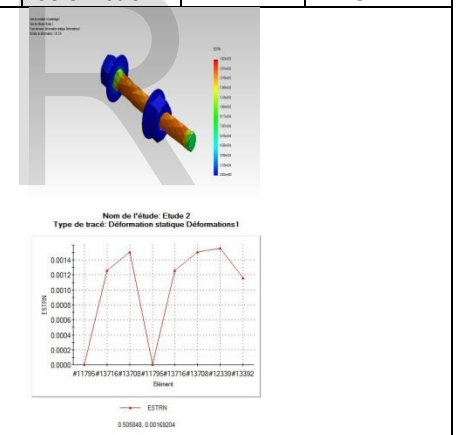



Figure22 : Assembly 1- Study 2- Déformations - Déformations1

**Conclusion:**

For a loading of 500000 NR one notes that the value of the constraint reaches a maximum value of 4x10+8 N/m2 and then will fall towards a minimal value of 0.5x10+8N/m2ce which implies that this assembly does not resist the maximum value. For this the displacement reaches the maximum value of 0.045mm. And with the node #13708 there is a deformation which reaches the maximum value and then will fall up to zero. What gives a contracting of the diameter of the Screw and the nut will slightly move what localization of

game between the parts means

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