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Design of Slim Tuner ver. 3.9.4

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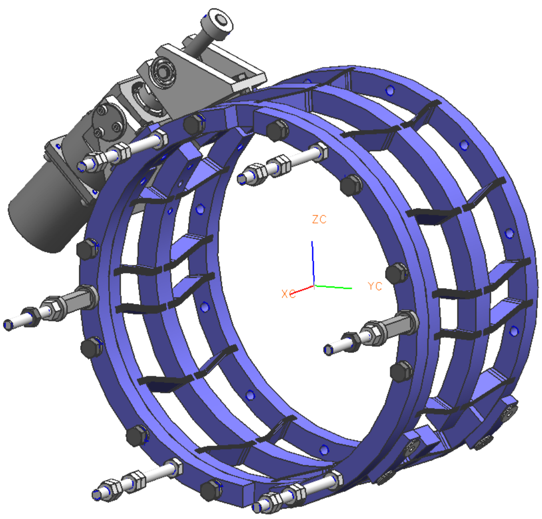

ABSTRACT

This document show the computations for the design of the slim tuner, version 3.9.4 in titanium, with a modified number and position of blades in order to increase its strength in compression, such as to satisfies the requirements needed for the certification procedure.

The tuner strength is compared with all the identified load cases that can occurs during the test, work and emergency situations.

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1 Introduction

This document shows the computations performed in order to design the blade tuner ver. 3.9.4 and to check its strength. The improvements introduced in this version are related to the blades thickness and their distribution along the tuner rings. These improvements became necessary after that the tests performed at room temperature in Milano (INFN – Milano LASA) shown a compressive strength too lower to allow to use the tuner in real conditions.

An analysis of the possible load conditions, temporary phases and ASME requirements lead us to set the following goals:

- limit compressive load higher than 12 kN when applied to the piezo positions;
- stiffness around to 30 kN, evaluated as the mean at the piezo positions;
- limit tension load higher than 20 kN when applied to the four safety bars positions;

The improvements here described have been suggested by simple analytical computations on a beam on six supports representing the original six blade packs on the half tuner rings.

The check of the new design capabilities in term of strength and stiffness is done by means of a full 3D FEM model.

2 Historical review

2.1 The blade tuner for superstructures

The main idea for the new tuner has been originally inspired by H. Kaiser who studied a possible tuner solution for the *superstructure* option in TESLA. This special design has been proposed in order to obtain a higher fill factor and simultaneously to increase the number of cells fed by a single input coupler. The superstructure option has been later on discarded, but the search for a minimization of inter-cavity distance remained as the critical point to reduce the accelerator footprint. Hence the development of a new frequency tuner able to guarantee the ILC cavity interconnection requirements has started. This new tuner has been designed to be coaxial with the cavity (see figure 1). The tuner assembly is mainly composed of two parts, the movement leverage and the bending rings. The leverage system provides the amplification of the torque of the stepper motor, dramatically reducing the total movement and increasing the tuning sensitivity. A stepping motor, similar to those used for the TTF tuner, is rigidly connected to the helium vessel and produces a rotation of the big arm in the center of the tuner. The movement of the big arm induces the rotation of the bending system that changes the cavity length.

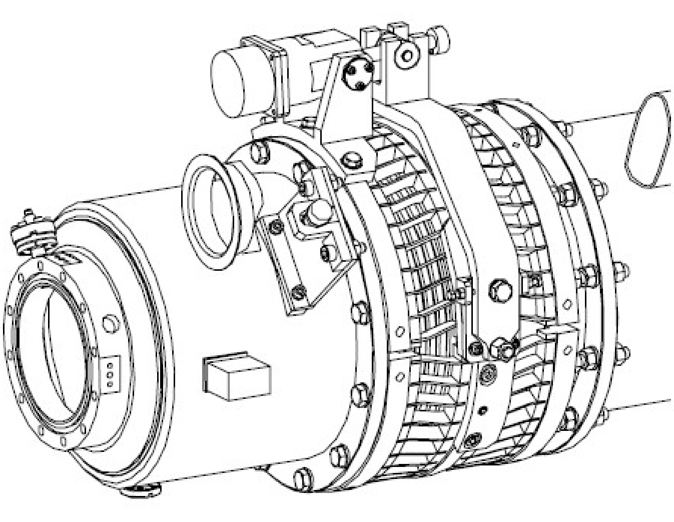


Figure 1: drawing of the original blade tuner (SuTu) assembly

This prototype model (named as original Blade Tuner or SuTu tuner, from the former SUperstructures TUner), successfully tested at DESY in 2002, actually represents the reference design for the entire Blade Tuner activity.

2.2 The blade tuner ver. 3.0.0 and the experimental tests

In the design of the version 3.0.0 coaxial tuner, an important modification has been introduced concerning the geometry of the blades, since the length and width have been adjusted so to significantly improve the tuning range. The new design makes use of longer blades allowing to increase the displacement induced on the cavity to 1.5 mm instead of 1.1 mm as for the original design.

Moreover a simplified driving system was introduced, mainly composed by the motor with its harmonic drive and a CuBe screw. The axial movement of the nut is directly transferred in the rotational one by the central rings (see figure 2).

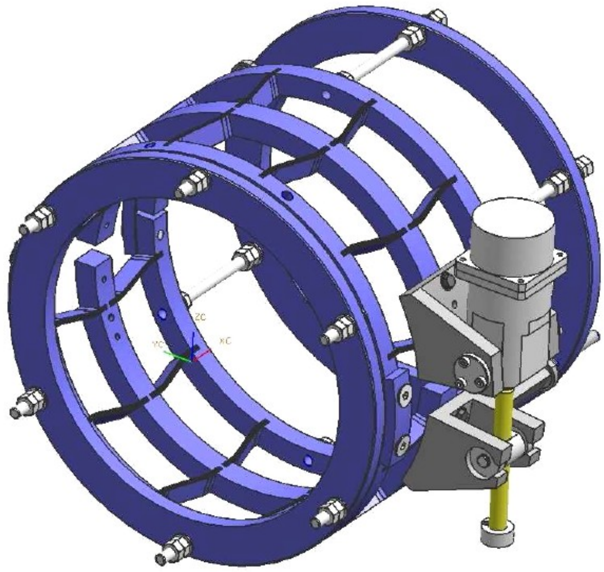


Figure 2: the new design of the Blade Tuner with the revised driving system

This tuner has been tested on Chechia and the tuning range is shown in figure 3. The tuner sensitivity, namely the frequency shift vs. screw turn, has also been computed from data and reported in figure 4.

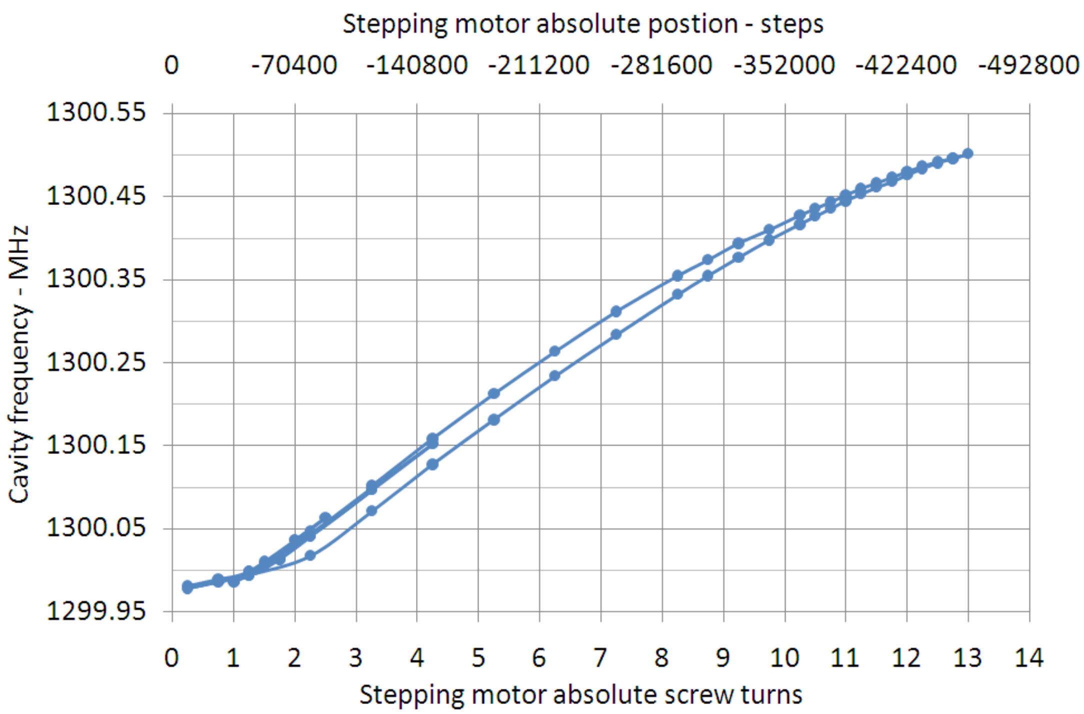


Figure 3: Blade tuner ver. 3.0.0 tuning range, 13 complete screw turns.

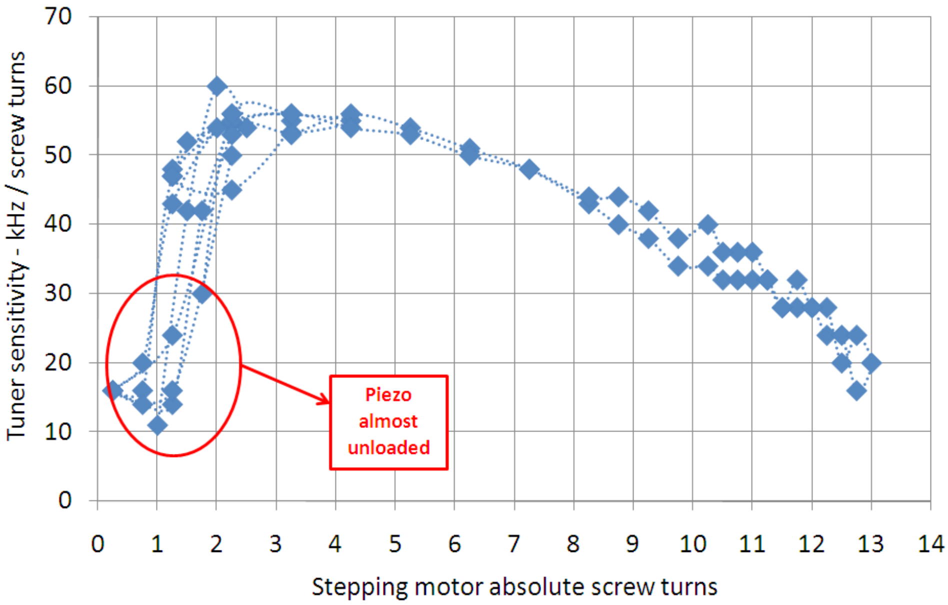


Figure 4: frequency shift vs. screw turns sensitivity for the Blade tuner ver. 3.0.0

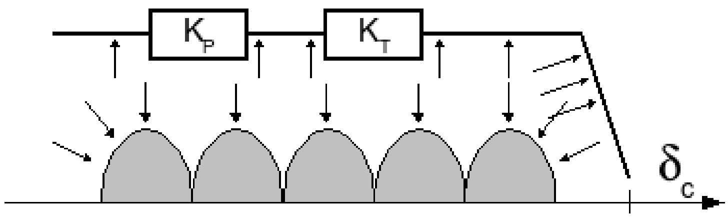
The experimental data collected confirmed that the Slim_SS Blade tuner performed the expected tuning range of 520 kHz, over 13 complete motor screw turns and confirmed also the peak sensitivity value of 50 kHz per screw turn expected by tuner design.

3 Load cases

The main loads on the tuner can arise from the pressure difference between the helium tank and the external world, or from the deformations impressed to the cavity in working conditions. The different load cases take into account both normal (tests and working phase) and emergency conditions such that required by the ASME check. The values reported in the following sections have been obtained by assuming a tuner stiffness of 30 kN/mm and the helium tank geometry of the TTF model.

3.1 Load cases in normal conditions

During the test and in working ocnditions the pressure in the cavity, helium tank and cryomodule can change as reported in table 1.



Pressure beam pipe: pressure inside the cavity

Pressure He Tank: pressure inside the helium tank and outside the cavity

Presuure isovac: pressure inside the cryomodule… or outside the helium tank

Condition	Pressure			Temp.	Max load[1]
	Beam pipe	He tank	isovac	cavity	tuner
	mbar	mbar	mbar	K	N
Start	1000 - Ar	1000	1000	300	0
Piezo preloaded	1000 - Ar	1000	1000	300	-2200
Ready to cool down	0	1000	1000	300	-3116
Cool down	0	2000	0	300 to 4	+4815
Stable 1.9 K	0	20	0	1.9	-2150

Table 1: normal load conditions that can occurs during the tests or working operations.

3.2 Load cases for the ASME / PED check

Check against the ASME or PED code is required for pressure vessels. The helium tank axial strength depends from the tuner assembly that rigidly connects the two parts of the vessel. It is therefore necessary to verify the tuner when subject to forces arising from the test conditions. These conditions are listed in table 2. The values herein reported have to be multiplied by a safety factor of 1.43, therefore during the acceptance test the pressure inside the helium tank will be of 5.8 bar and the maximum traction load on the tuner goes up to 14000 N.

Condition	Pressure			Temp.	Max load
	Beam pipe	He tank	isovac	cavity	Tuner
	mbar	mbar	mbar	K	N
Emergency	0	4000	0	300	+9630
Leak test	0	0	1000	300	-2840

Table 2: load conditions to be considered for the check of helium tank.

4 Analysis of the load distribution

The stresses in the blades strongly depends on the way the load is applied to the tuner. Two differents cases can be identified: compressive load applied by the piezo elements and tration load applied by the four safety bars. In this document the case of compressive load applied by the four bars is not considered because the vacuum test and handling should be done without the tuner installed. In any case the compression force that can be applied to the tuner by means of the four bars is higher than the limit force going through the piezo elements.

4.1 Evaluation of the force distribution on tuner ver. 3.9.0

The evaluation of the forces withstand by the blade packs is done by solving a simple model made of an hypothetic linear beam constrained by six springs (see figure 5). The spring stiffness is equal to the axial stiffness of the blade packs as determined in [1]. The beam has also boundary conditions at the ends to take into account the connections between the two half rings. This connection is supposed to be infinitely rigid and the beam stiffness is considered equal to the sum of the tuner and reinforcing rings stiffness.

The connection is described by the following relations:

$$v_1 = v_6 \qquad \phi_1 = \phi_6$$

and R and M are the connection forces between the two half rings.

[1] Positive forces are in traction direction

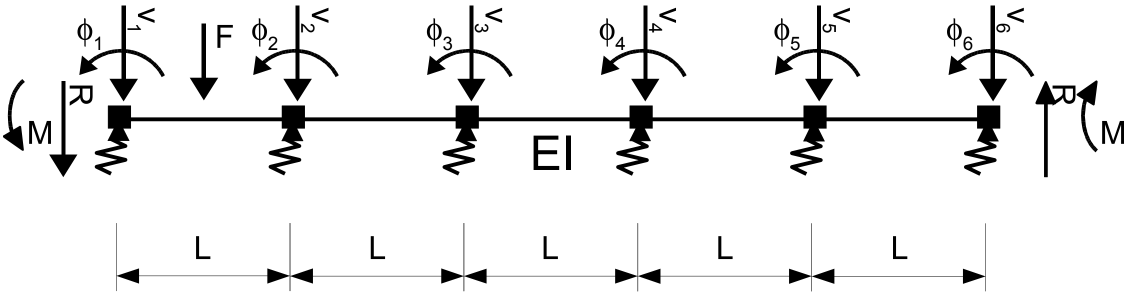
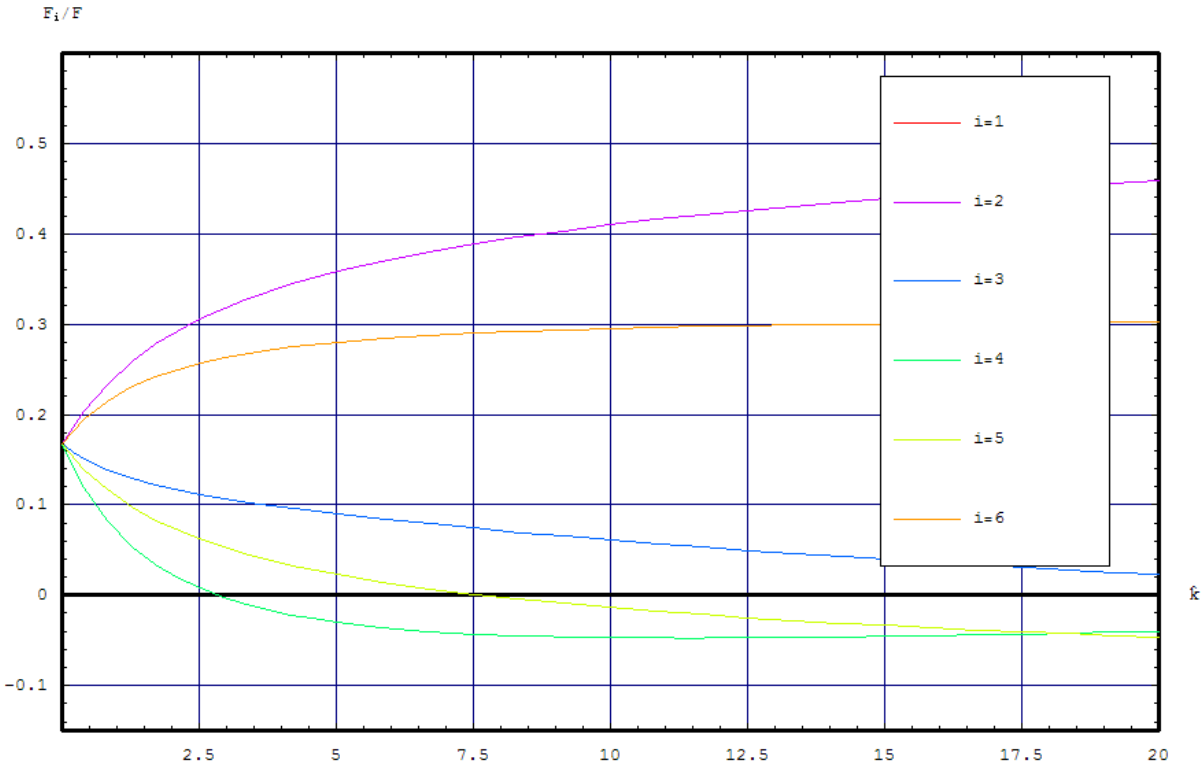


Figure 5: beam model of half tuner ring with connections at its ends

The displacement method is used to solve the problem. The solution is presented in terms of adimensionalized quantity:

$$\hat{F}=\frac{FL^2}{EI} \qquad \hat{K}=\frac{KL^3}{EI} \qquad \hat{\phi}_i=\phi \qquad \hat{v}_i=\frac{v_i}{L}$$



In this case I ≈ 16200 mm^4 and $\hat{K}\approx 5.1$

It is evident that 35% of the piezo load goes on the blade pack n. 1 and 2 and a 28% goes on blade pack n. 6. Moreover the effect of the stiffening ring is to reduce the maximum load on the blade pack n. 2 from 50% to 30%. This analysis suggest to add some blades in the region near the piezo elements.

5 Geometry

The compatibility of the new tuner design has been verified against the external constraints imposed by the ILC/XFEL cryomodule geometry. In particular the tuner has been rotated in order to avoid any interference between the lateral motor and the gas return pipe as illustrated in figure 6. As a principal consequence of this rotation the piezo element positions go to correspond to two blade packs: these packs withstand an high load and it is necessary to double them. The others blade packs remain in their original position with a reduced number of blades because the load acting on them is lower. The final configuration can therefore been seen in figure 7, where a 3D drawings of the rings assembly is reported.

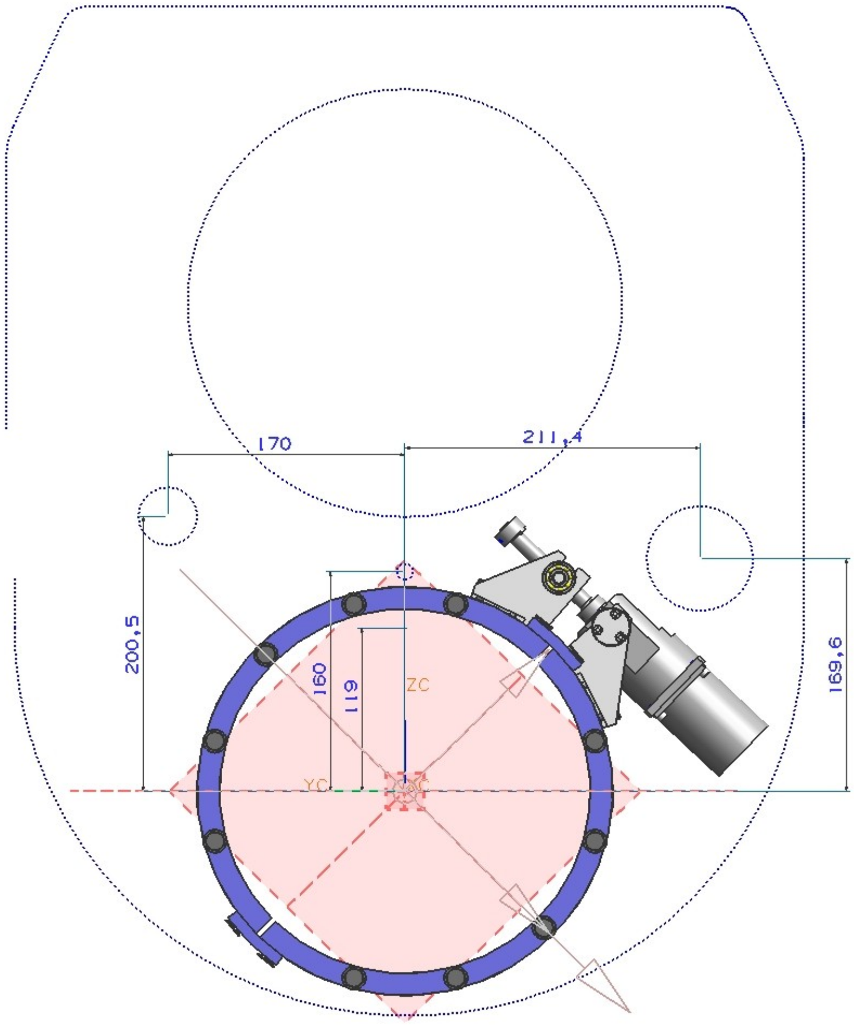


Figure 6: positioning of the blade tuner with respect to the cryomodule geometry

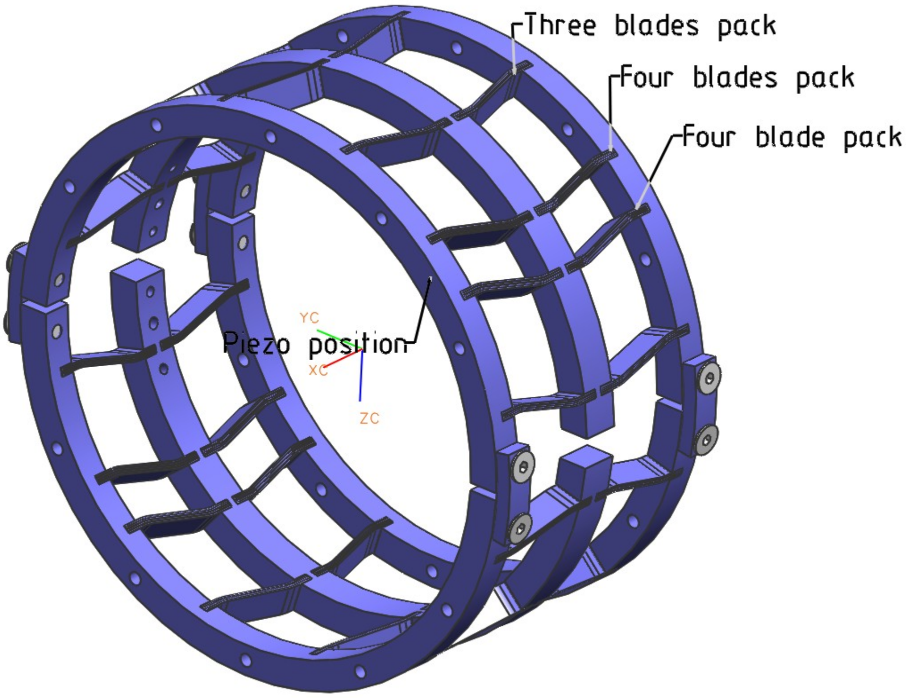


Figure 7: position of piezo and blades distribution

6 Evaluation of tuner ver. 3.9.4 strength

The evaluation of the new tuner design strength is done by means of several finite element analyses perfomed on a full 3D Ansys® model. In particular the limit load in compression and traction have been found both by means of linear buckling and non linear incremental analyses. Yield material limit and buckling phenomena have been therefore considered.

Moreover tha analyses have been performed such as to simulate the different load conditions that will occur on the tuner, allowing to obtain important informations about the forces acting on the motor shaft, on the connecting bolts and a good estimation of the whole system stiffness.

6.1 FE model

The FE model is composed of 42990 nodes, 7920 SHELL181 4 nodes elements (blade and thicknesses, type 5 to 304), 504 SOLID186 20 nodes elements (reinforcing rings, type 1 and 348), 11931 SOLID187 10 nodes elements (rings, connecting elements and motor supports, type 2 to 4 and 349 to 357). Other 32759 elements (contacts, links and beams) are used to describes contact, rigid connections and bolts.

A view of the model is reported in figure 8, in green are reported the rings in titanium grade 2 and in blu the blades in titanium grade 5. The motor supports are in Al 6082T6 while all the bolts and nuts are in stainless steel.

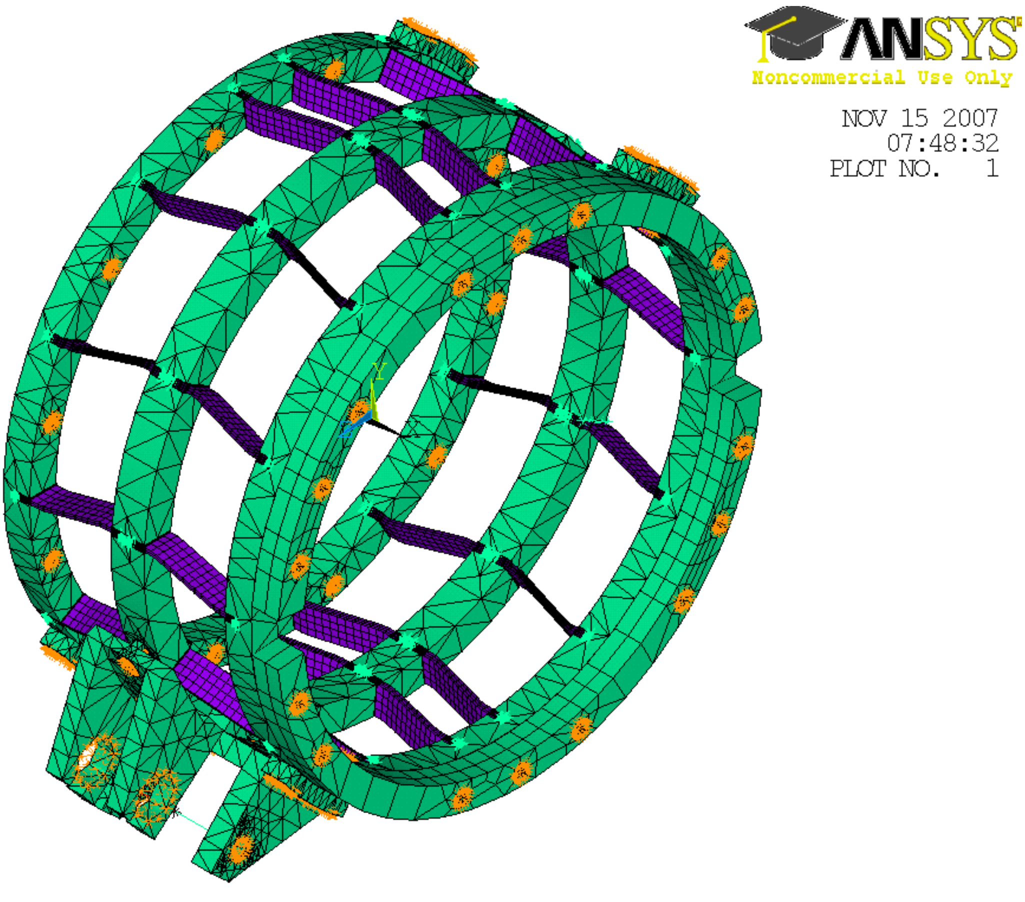


Figure 8: 3D view of the FE model with material based color code

6.2 Material properties

The material properties used in the analyses are referred to room temperature. This choice has been done because many load cases occurs at room temperature (piezo preload, vacuum in the cryostat, conditioning of tuner) and in any case the properties at cryogenic temperature are generally better than that considered.

In addition to this assumption, in order to have faster analyses, the non linearity of materials has been considered only for the blades.

Part	Material	E_y (MPa)	ν	f_y (MPa)[2]	ε_t	f_t (MPa)	ρ (kg/m³)
Blades	Ti Gr5	105000	0.37	830	10%	900	4500
Rings	Ti Gr2	105000	0.37	275	10%	900	4500
Motor supports	Al 6082T6	70000	0.32				2700
Bolts	A4	191000	0.29				

Table 3: material properties used in the FE analysis

6.3 Compression strength

From what reported in section 3, it is evident that the worst compression case can occurs during the tuning phases by stretching the cavity. In some cases, vacuum test or during handling, compression forces can occur also when the blades are in their rest position. Therefore the compression strength is evaluated both in rest and maximum deformation position.

6.3.1 Tuner in rest position: buckling load

In order to evaluate the buckling load in rest position, a preliminary analysis with a base total load of 12kN has been performed. The load has been applied to the piezo location (2x6kN) by means of a central node rigidly connected to the hole edge nodes with MPC contact. After this analysis, needed to obtain the base stress in the elements, the buckling load has been obtained for the first three modes. The results are reported in table 4, while the three modes can be seen in figures 9 - 10.

Buckling mode	Load (kN)	Description
1st	17.6	Whole tuner
2nd	21.1	Single blade
3rd	21.7	Single blade

Table 4: buckling loads and modes description

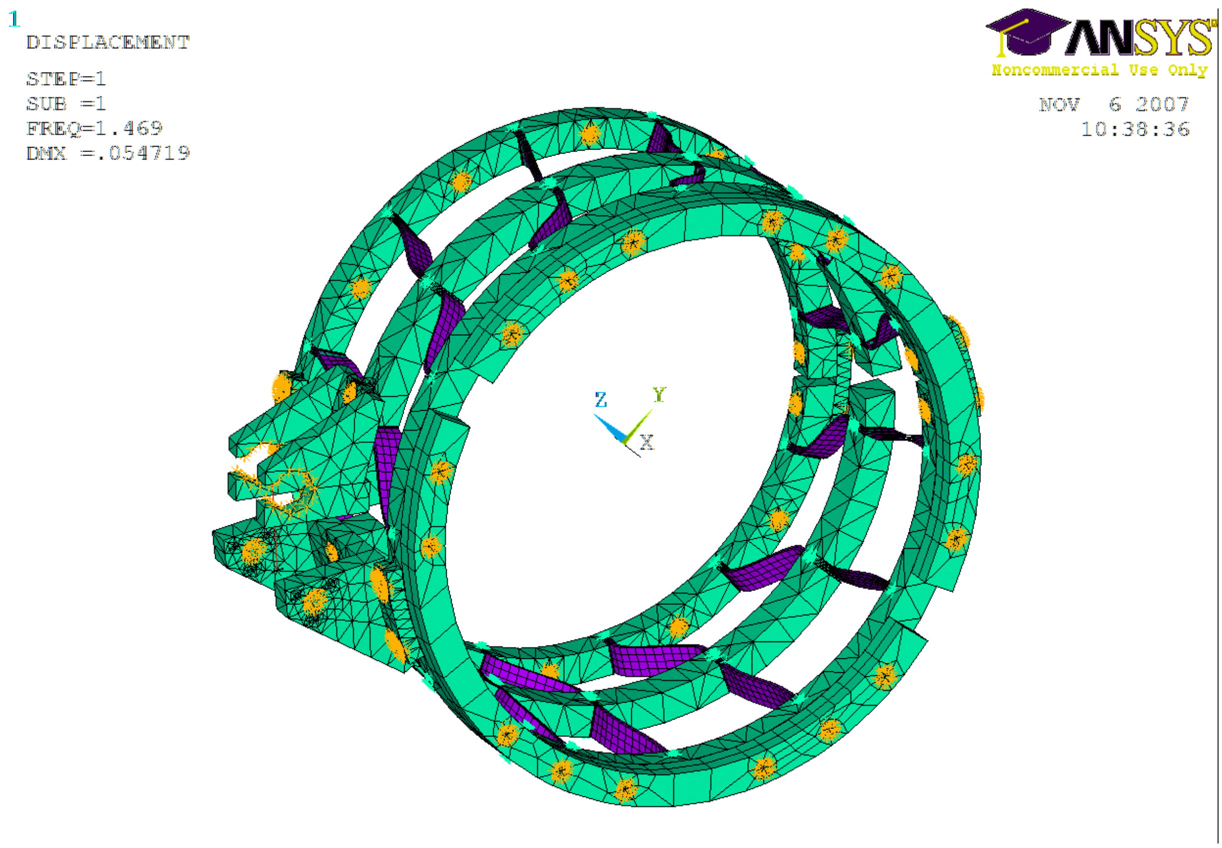


Figure 9: first buckling mode

[2] Not considered in the FE analyses

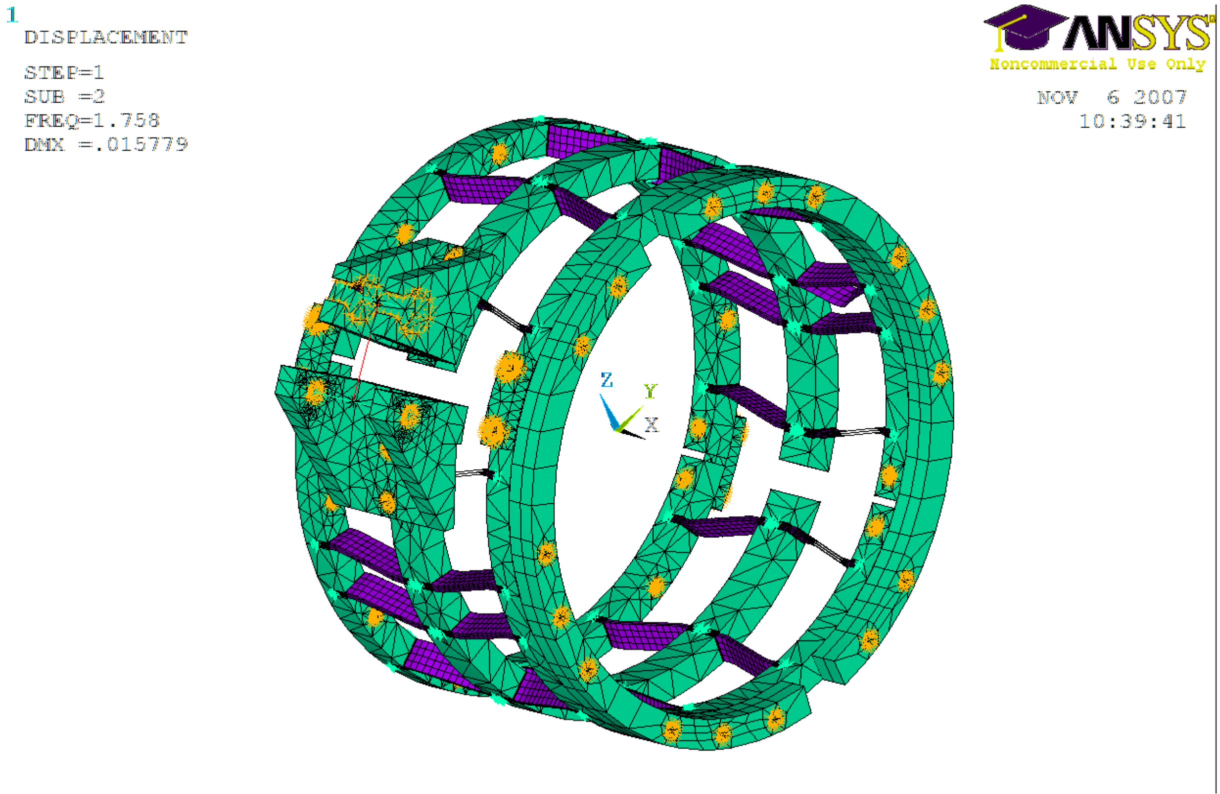


Figure 10: second buckling mode

The values reported in table 4 are limit values, an appropriate safety factor must be considered in order to obtain admissible load. Being the buckling phenomena critical and unexpected, a safety factor of 2 is considered necessary, therefore the admissible load in compression, in undeformed state, is equal to 9 kN evenly distributed at the piezo position.

6.3.2 Tuner in rest position: collapse load

An incremental non linear analysis has been performed in order to check the obtained buckling value and to obtain the collapse load and the behavior of the tuner. This analysis takes into account the material and geometric non linearities. In this case two compression forces, of the same value, have been applied to the piezo positions. The load increments between the steps is automatically chosen by the arc-length method used to drive the solution. In such a way it has been possible to follow also the unload phase that occurs after the collapse.

The final configuration, where the two halves of the tuner deforms in antisymmetric mode (see fig. 11), give a suggestion of how to check the beginning of the collapse. This can be done by observing the change of the transverse load on the CuBe screw, that normally should be equal to zero. When the two halves deforms in different direction this value steeply increase up to a value of 250N (see fig. 12). The beginning of the collapse can be therefore set to a total compression load of 10000 N. At this value the maximum stress on the blades is equal to 552 Mpa (see fig. 13), well below the yield limit of 830 Mpa.

The plastic limit on the blades is obtained for compression load of 11400 N (see fig. 14), while at the collapse load of 11600 N the maximum von Mises stress is equal to 838 MPa (see fig. 15).

The load displacement curve at the piezo position is reported in fig. 16. It can be observed that the tuner side near the motor is stiffer that the other one and the displacements at the corresponding piezo position are lower.

Finally the axial force on the driving CuBe screw is reported in fig. 17. The maximum load is equal to 2000 N, approximately 1/5 of the compression load on the tuner.

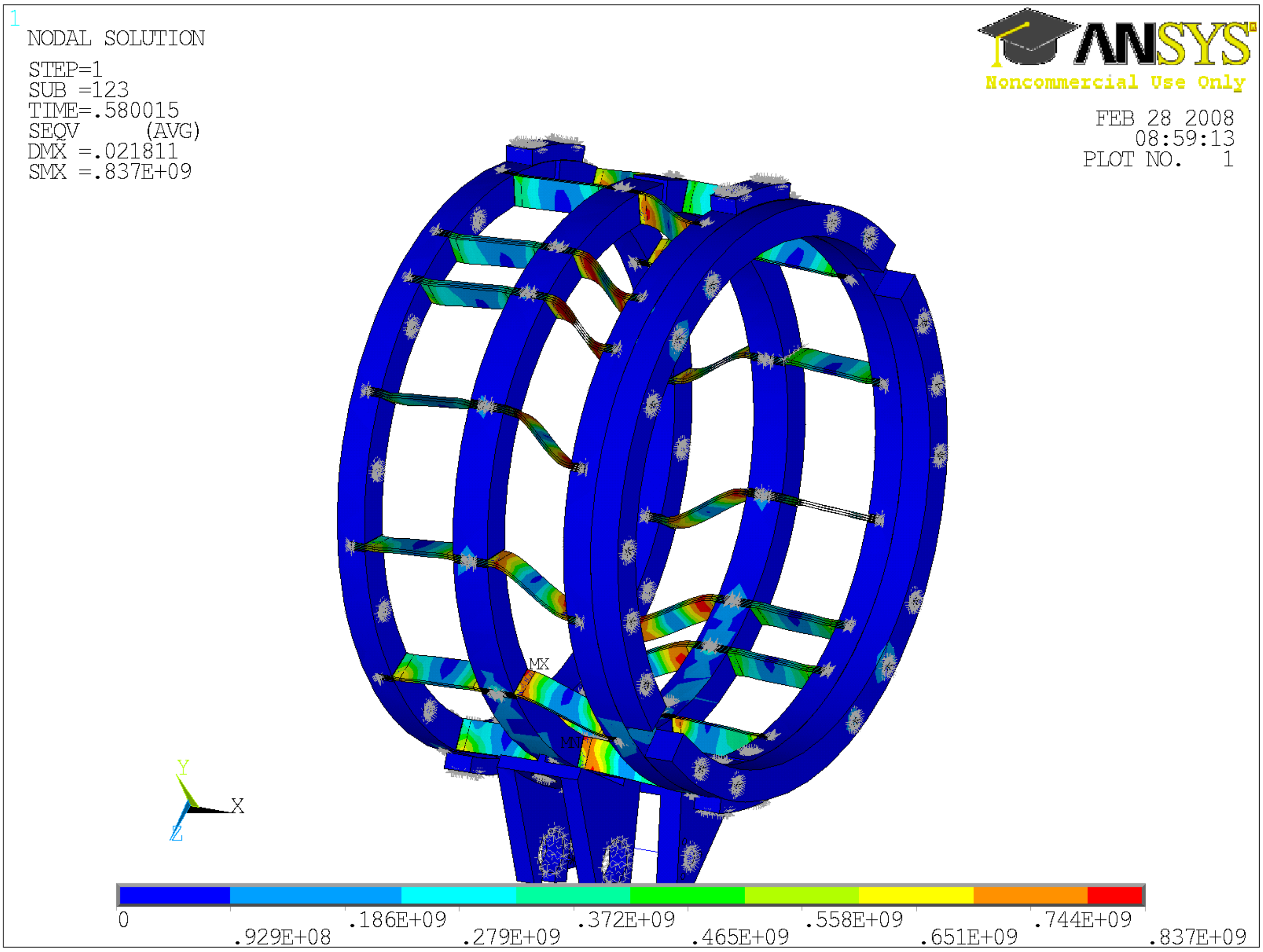


Figure 11: deformation of the tuner at the collapse

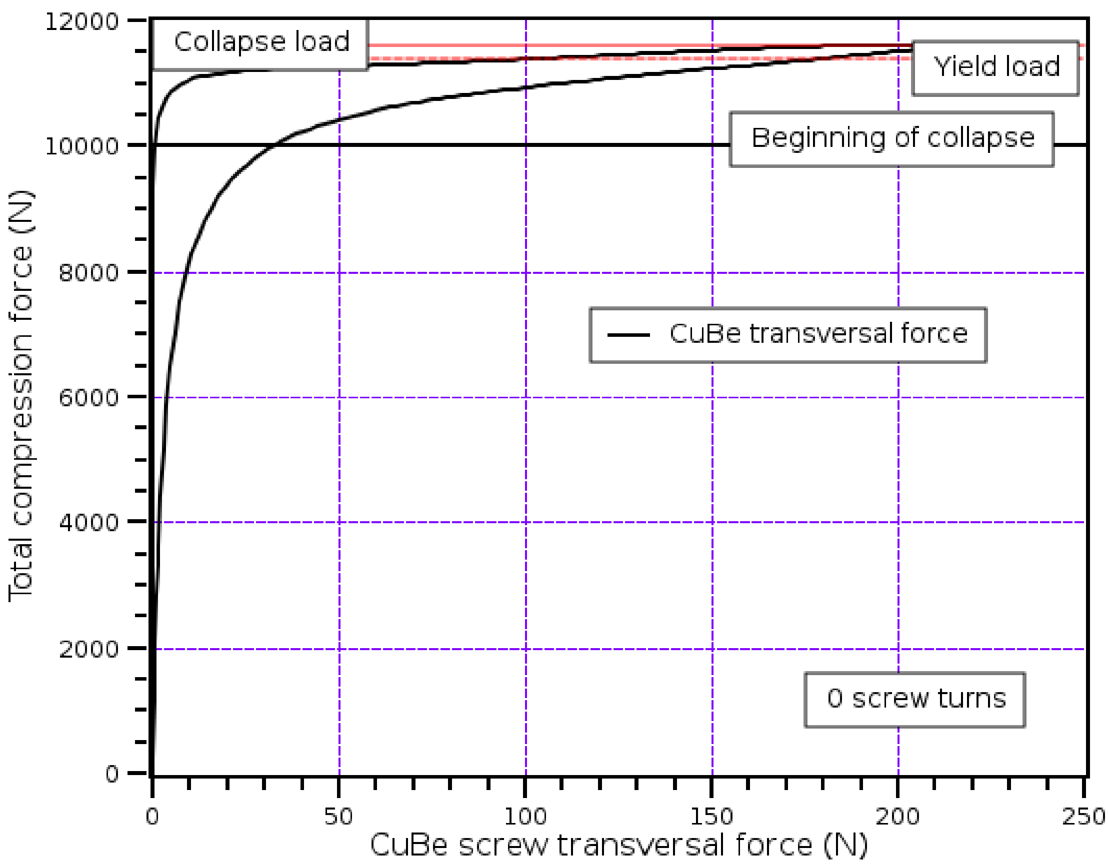


Figure 12: transversal load on the CuBe screw

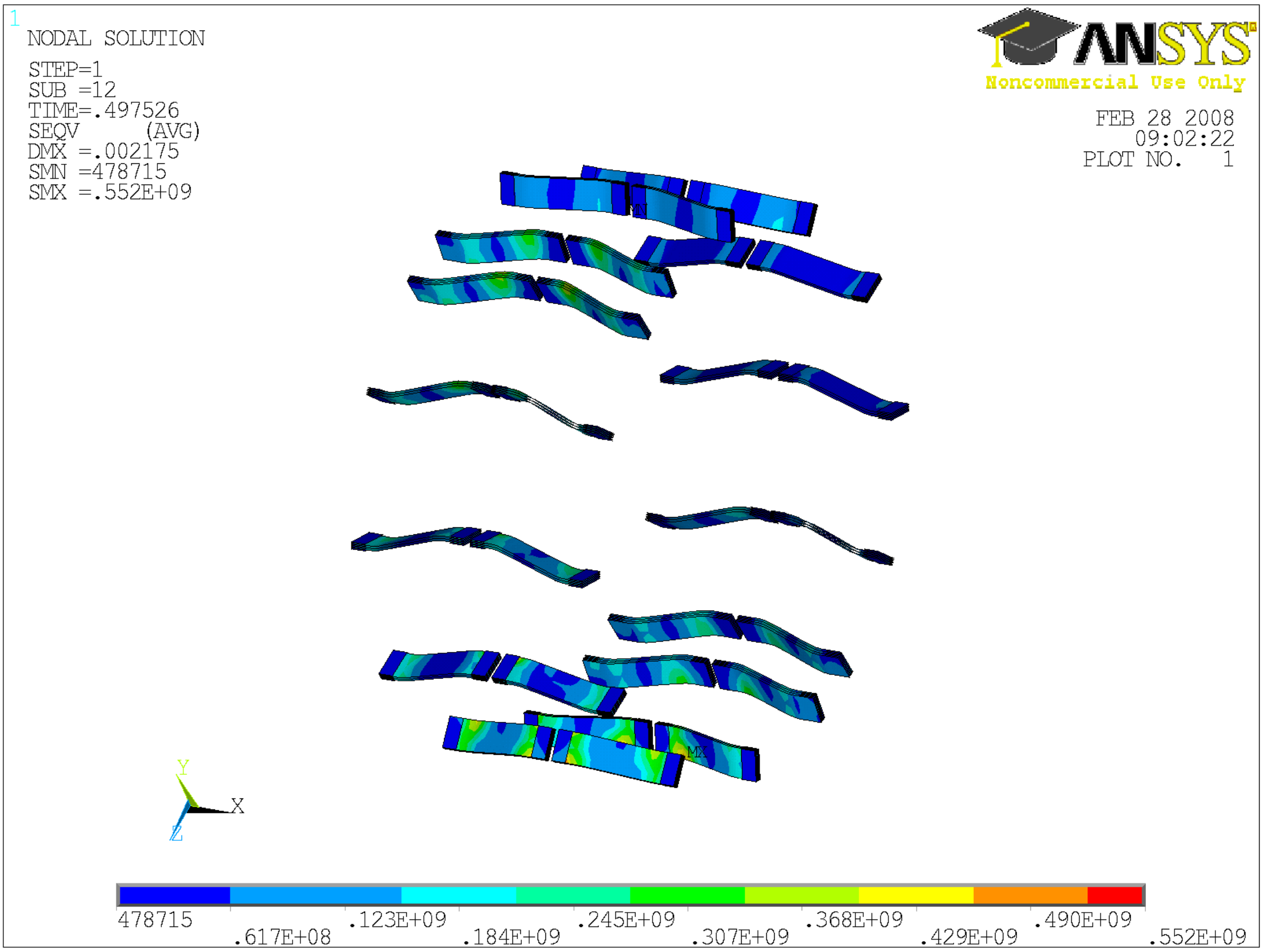


Figure 13: von Mises stresses on the blades for a compression load of 10000 N

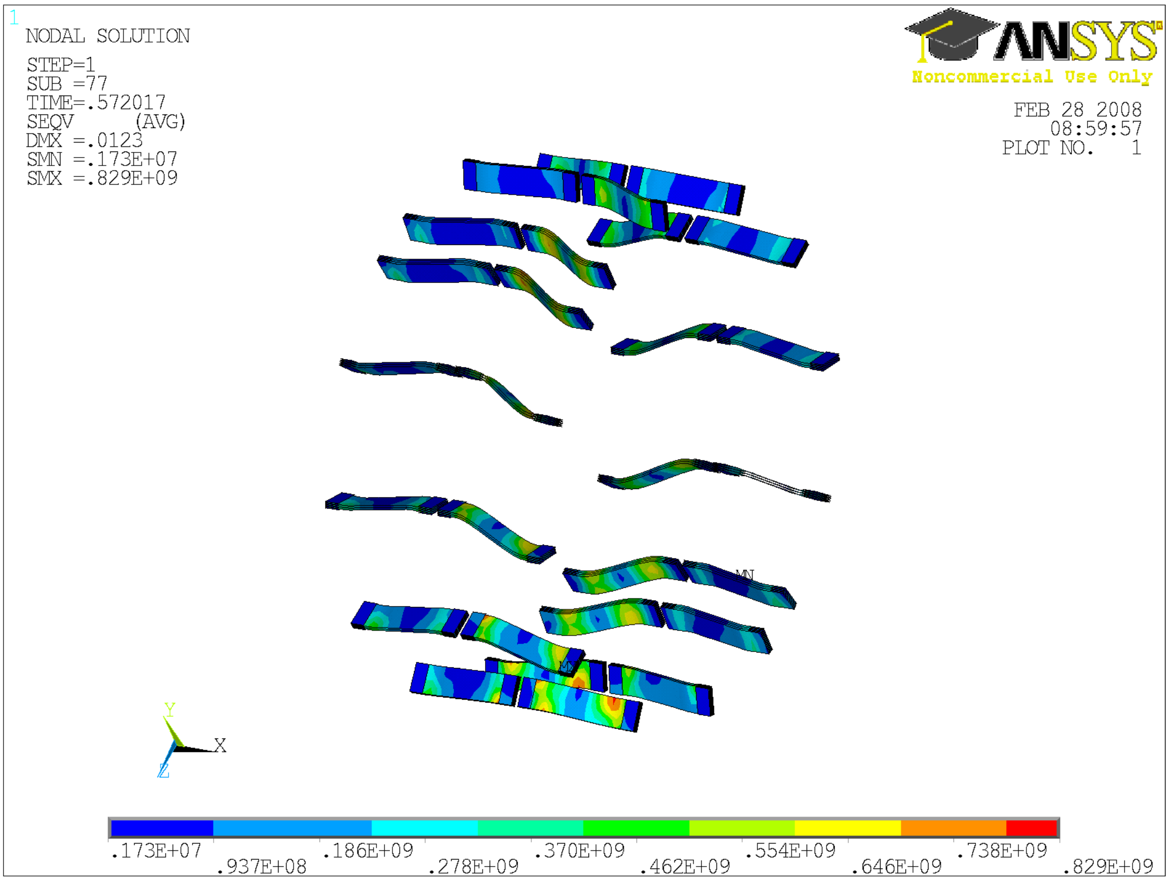


Figure 14: von Mises stresses on the blades for a compression load of 11400 N

Figure 15: von Mises stresses for a compression load of 11600 N

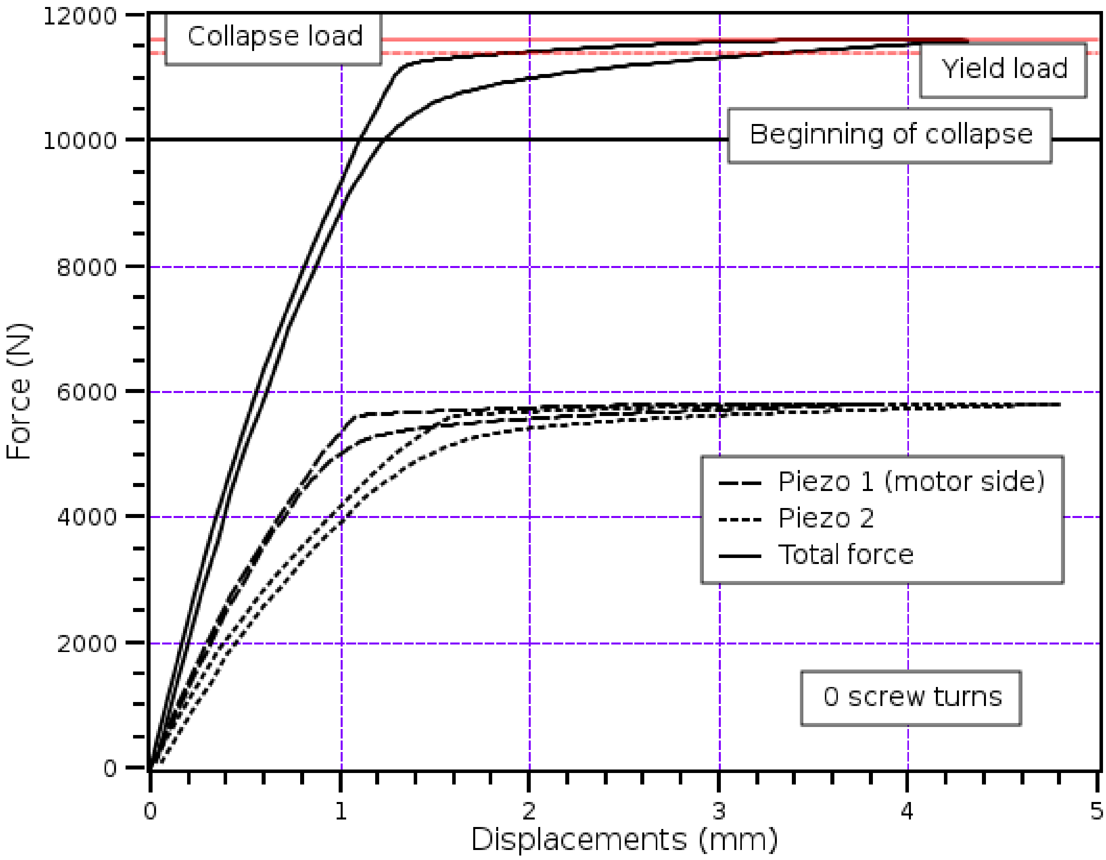


Figure 16: load-displacement curve at the piezo position

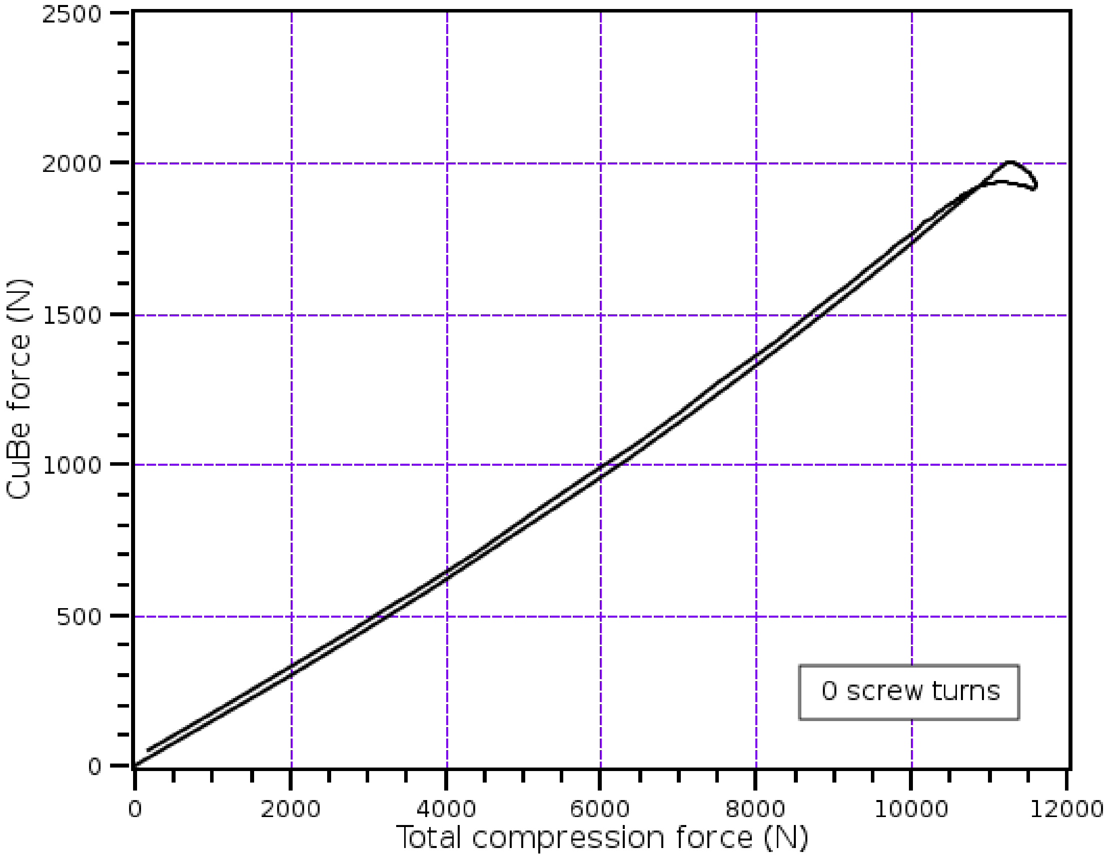


Figure 17: axial force on the driving CuBe screw

6.3.3 Tuner in working conditions: maximum stresses

During the accelerator activity, the driving of the tuner could stretch the cavity up to its maximum range. The blade deformation and the compression load arising from the cavity reaction induce some stress in the tuner that must always be lower of the yield limit in order to avoid any non-linearity. In this section the maximum stresses on the blades arising from the described case are obtained considering all the steps: preload on piezo, motor driving up to the top limit of the tuning range, an eventual axial load due to a pressure difference.

6.3.3.1 Piezo preload

During the piezo preload, a nominal force of 2x1.1 kN is applied to the tuner. In this case the displacements are reported in figure 18. In proximity of the piezo more far from the motor (piezo 1) the displacement is equal to -0.21 mm, while at the piezo 2 position the displacement is equal to -0.16 mm.

The stresses in the tuner are reported in figure 19. the maximum value of 166 MPa occurs near to the motor support pin. This is principally due to a stress concentration that depends from the model definition of constraints and contacts. In figure 20 are pointed out the stresses in the blades only, which maximum value is equal to 99 MPa and occurs at the blades near the end of the central rings and not near the piezo position.

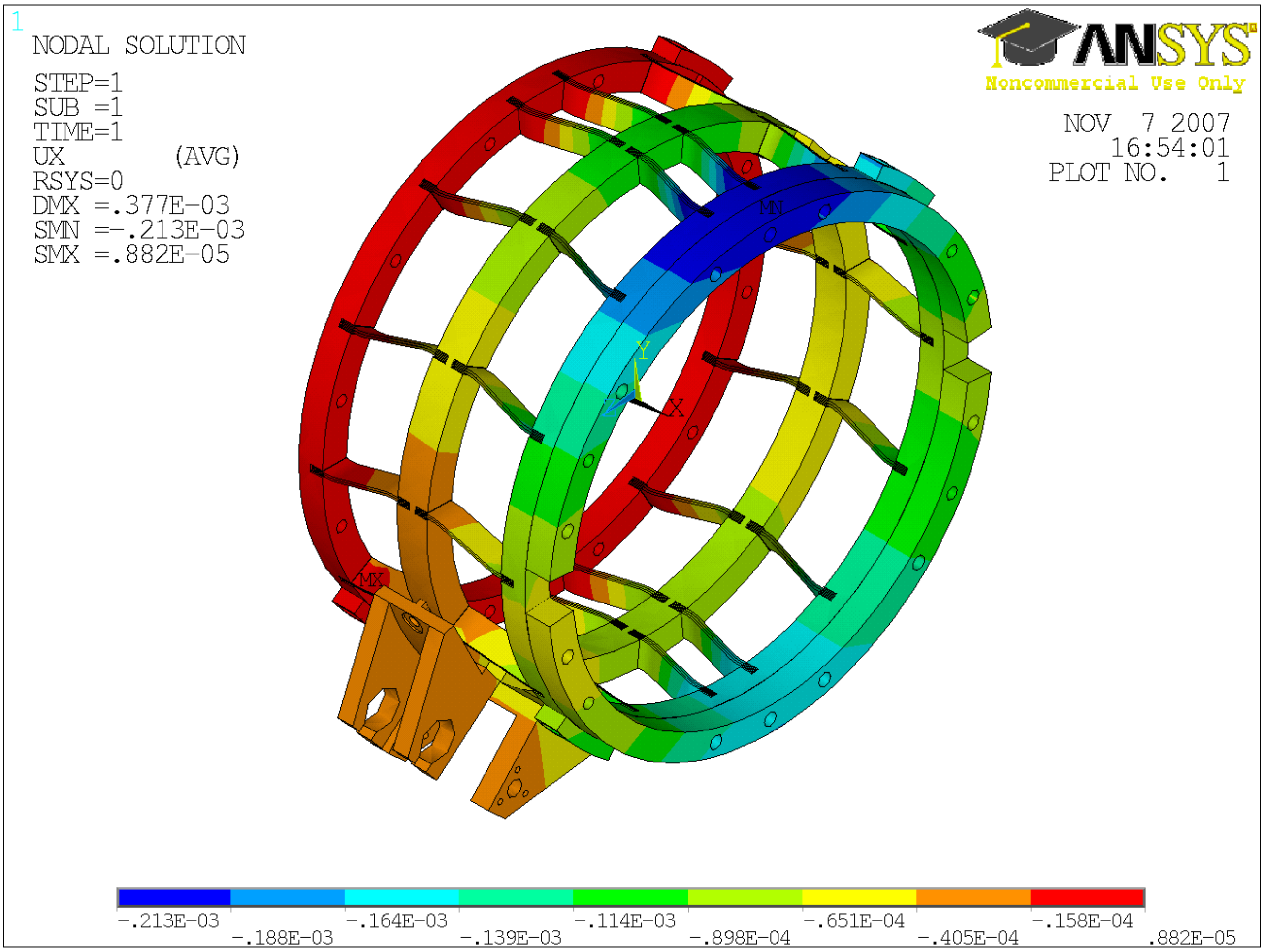


Figure 18: displacements in the tuner after the preload of piezos (2x1.1 kN)

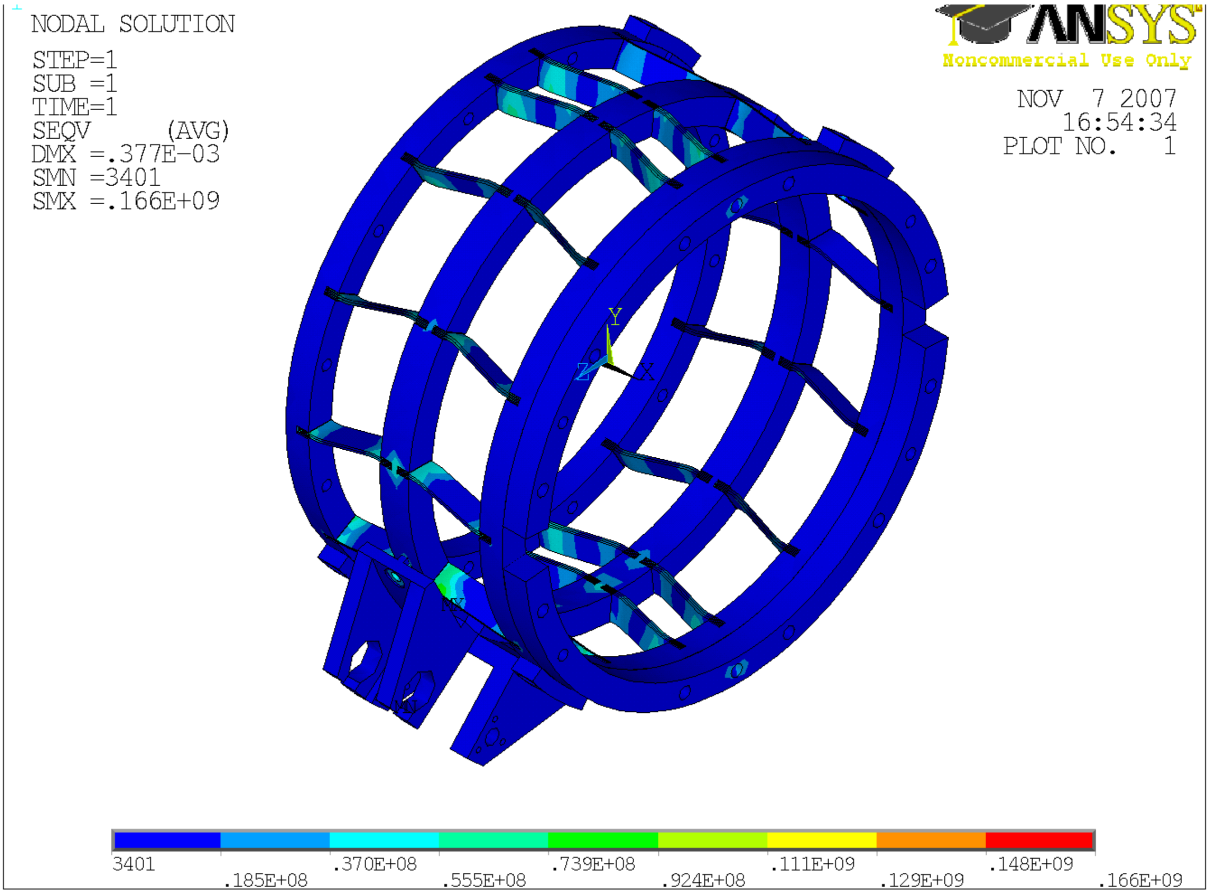


Figure 19: stresses in the tuner after the preload of piezo (2x1.1 kN)

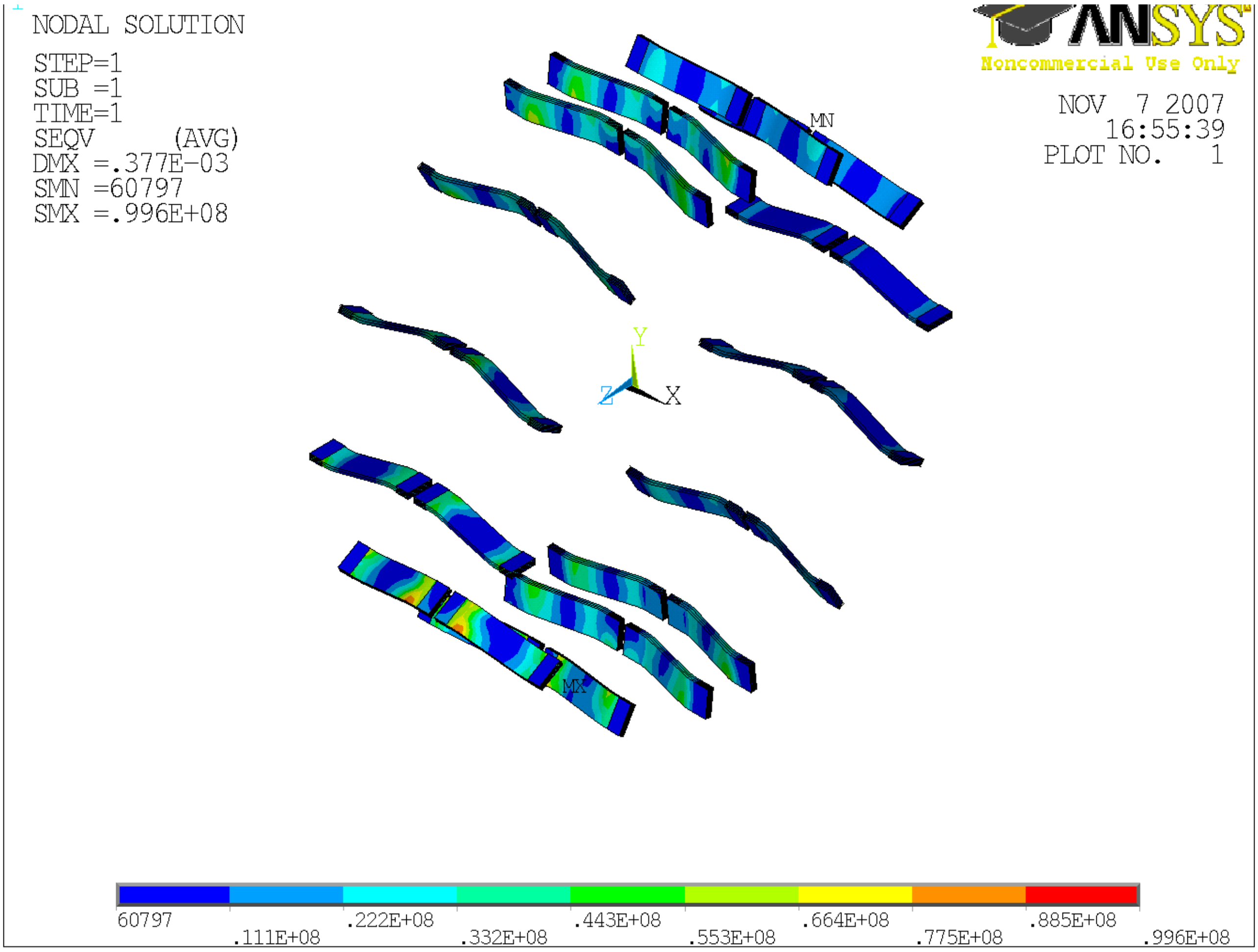


Figure 20: stresses in the blades after the preload of piezo (2x1.1 kN)

6.3.3.2 Driving of the tuner up to its maximum range

From the piezo preload step, a second analysis has been performed taking into account the cavity stiffness, in order to evaluate the behavior of tuner during normal working conditions. In particular it has been possible to obtain:

- the maximum stresses on the blades;
- the maximum force on the motor shaft;
- the load-displacement curve (shown in section 8)
- the base load from which to do a buckling analysis in deformed state

At the maximum tuning range (corresponding approximately to 14 screw turns) the stresses are as reported in figure 21. The maximum stress occurs in the blade pack near the central ring ends, motor side. At this deformation level some plastic strains develop, and this can be seen also at 13 screw turns (figure 22). The maximum deformation level without plastic strains correspond to a 12 turns of the driving screw. The stresses at this level are reported in figure 23: the maximum value of 791 MPa is lower than the material limit of 830 MPa.

From these analyses the maximum tuning range is limited by the strength of the blades to 12 screw turns that corresponds to a maximum elongation of 1.8 mm at piezo 1 and 1.9 mm at piezo 2 (see section 8).

The axial force on the driving screw is reported in figure 24. The maximum value at the maximum elongation is equal to 1600 N, almost two time the value foreseen for the blade tuner ver 3.0.0 that has been tested in Chechia (September 2007).

The loads on the two piezos are reported in figure 25. They are always lower than their blocking force.

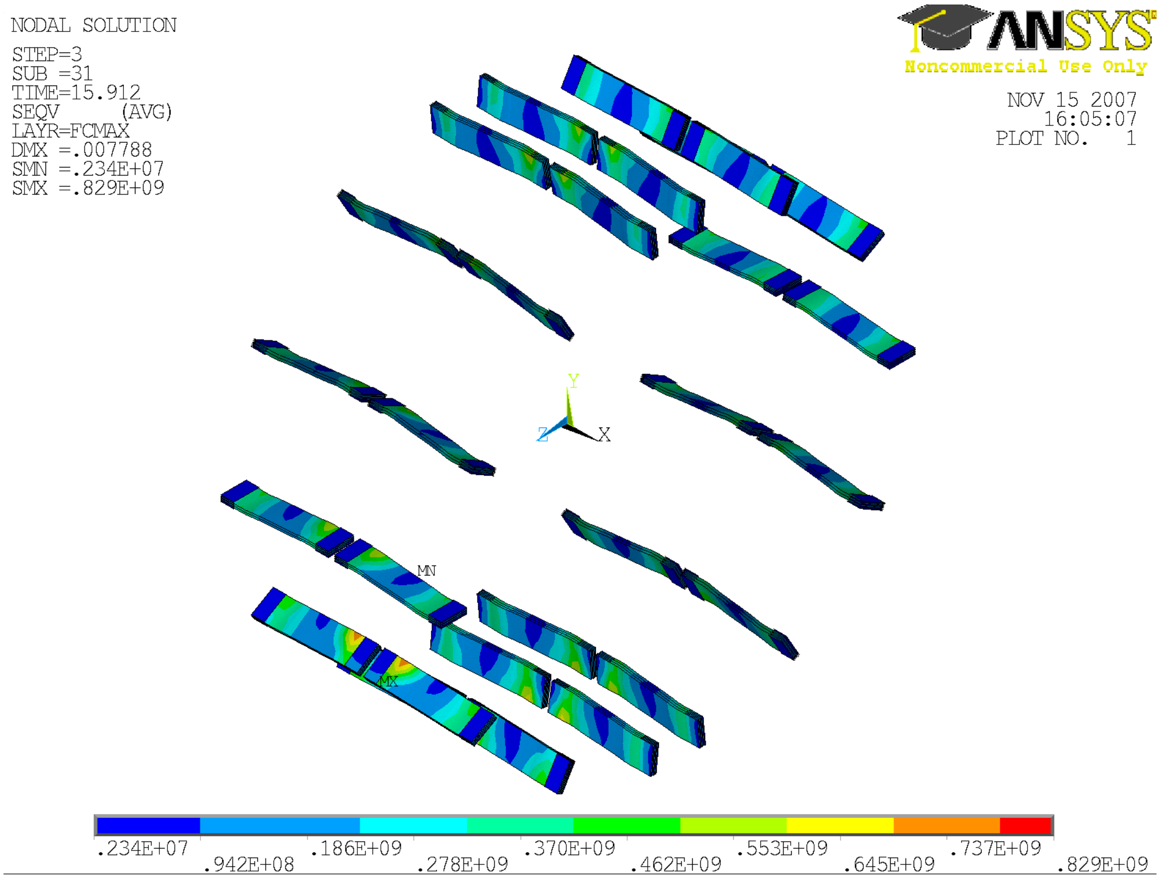


Figure 21: von mises stresses in the blades in working conditions after 14 screw turns

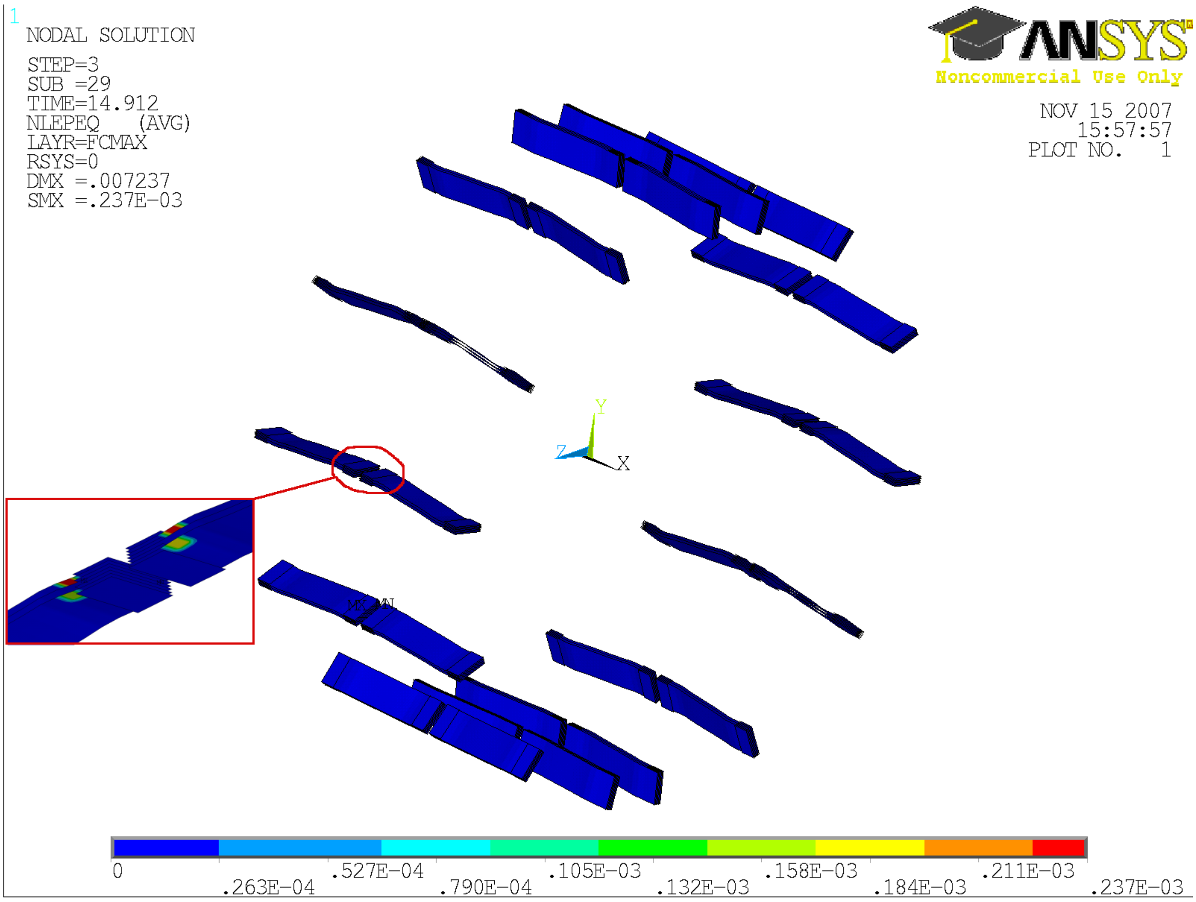


Figure 22: plastic deformations in the blades in working conditions after 13 screw turns

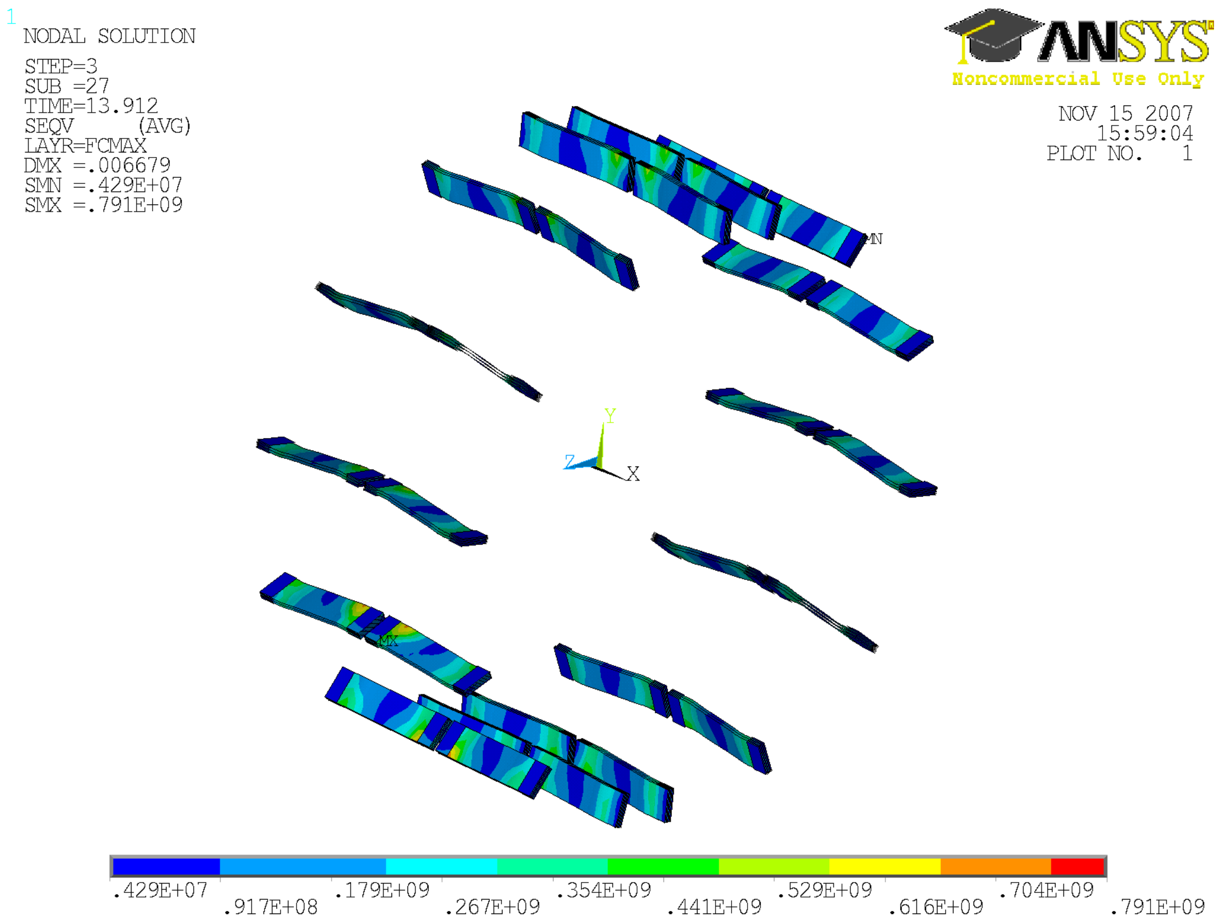


Figure 23: von mises stresses in the blades at working conditions after 12 screw turns

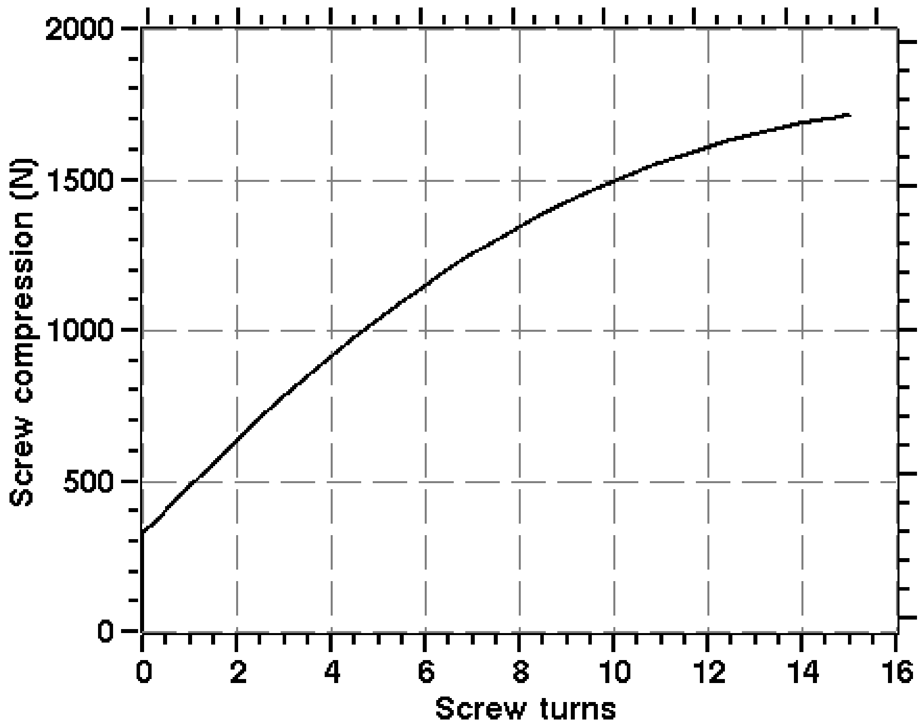


Figure 24: CuBe axial force (Valu in N) versus screw turns (time - 2)

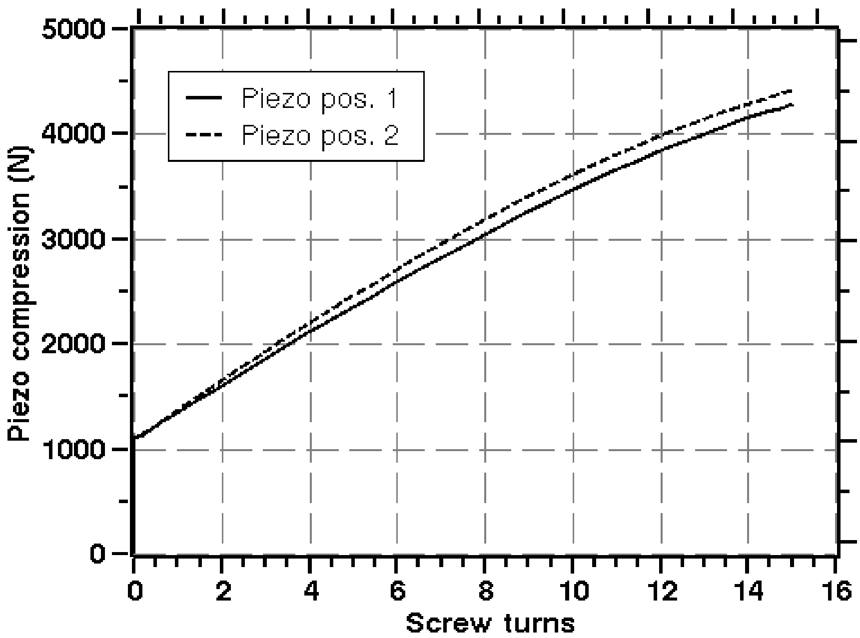


Figure 25: compression force on piezo elements

6.3.4 Tuner in working conditions: collapse load

The buckling load could be obtained, as for the rest position case, in working conditions, for example at the maximum admissible elongation level. Neverthless the obtained value should be affected by a non negligible error consequence of the fact that the stresses in the blades are higher than the yield limit. A better approach is that to analyse the collapse behavior of the tuner by applying a compression load after the deformation due to the motor drive. An analysis that take into account both material and geometrical non linearities is therefore used in order to obtain the limit load of the tuner in compression starting from the maximum elongation position (12 screw turns). The maximum load obtained is a collapse value, and a proper safety factor has to be applied to it in order to obtain a safe limit. It will be shown that before the collapse no plastic deformations will occur, therefore the tuning capabilities will not be affected by an additional compressive load.

The complete force - displacement curve (motor drive and following compression load) at the piezo position is reported in figure 26. The collapse load obtained is equal to 12500 N, 4700 N higher than the compression applied to the tuner by the cavity deformations and the piezo preload. Therefore, applying a safety factor of 1.5 to the additional compressive load, a safety value of 10900 N is obtained. If an higher admissible load is requested a safety device must be used.

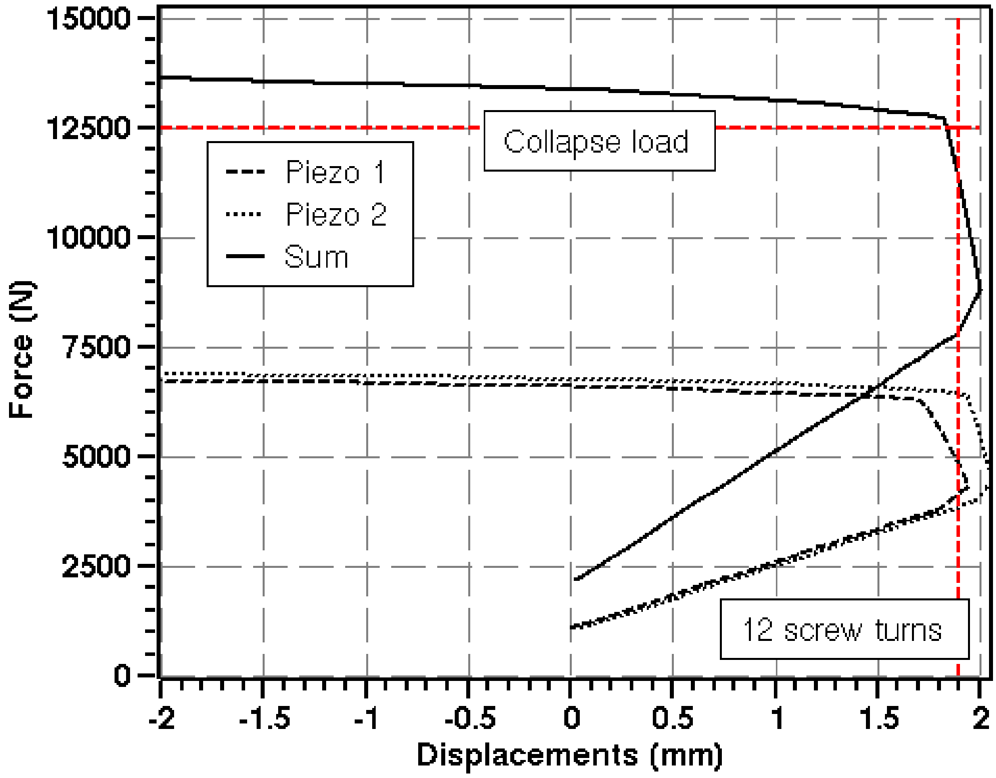


Figure 26: forces - piezo displacements plot

The displacements, stresses and deformations at the last converged step (collapsed configuration) are reported in figures 27-29. The same quantities at the collapse load (F = 12500 N) are reported in figures 30-32. From figure 32 it is evident that the blades does not exhibit plastic strains, therefore are not damaged by the additional compressive load.

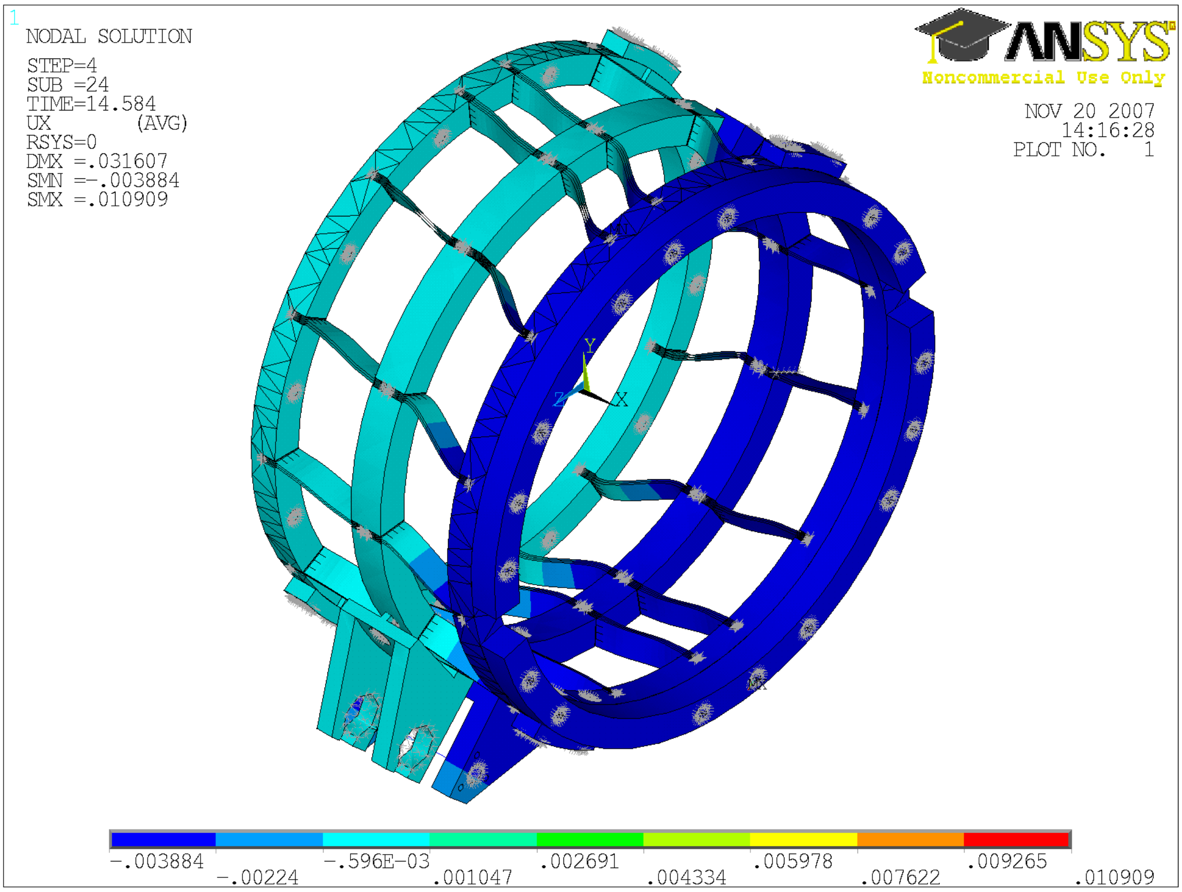


Figure 27: displacement of the collapsed tuner subject to a compression load in the extended position

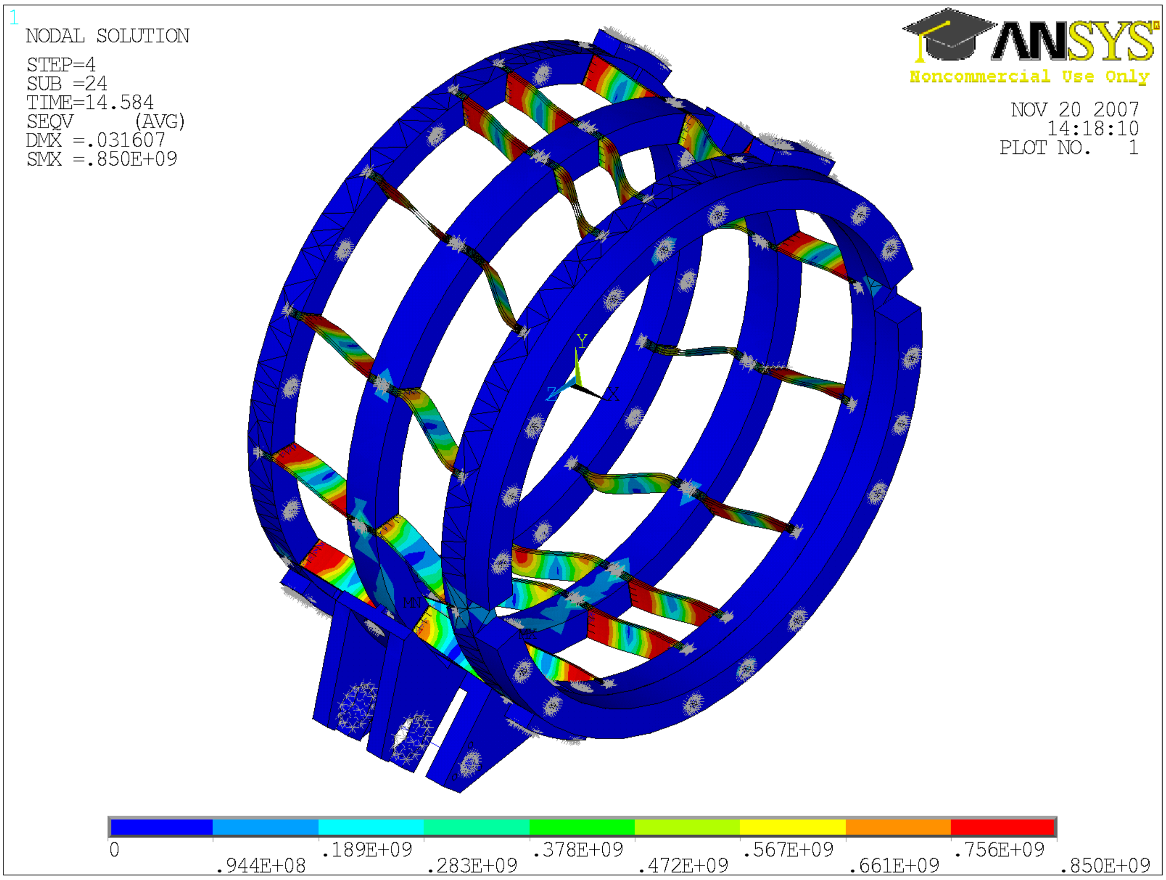


Figure 28: von Mises stresses in the collapsed tuner

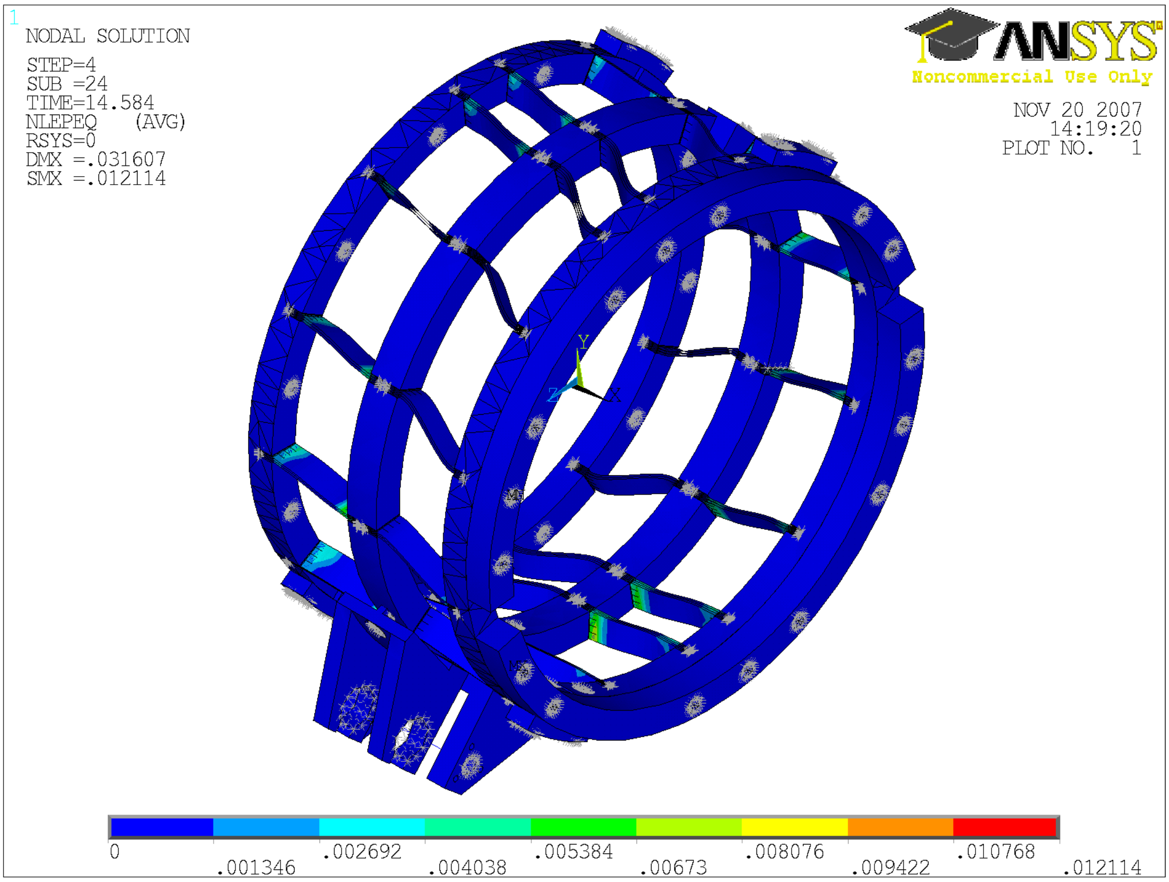


Figure 29: equivalent plastic strains in the collapsed tuner

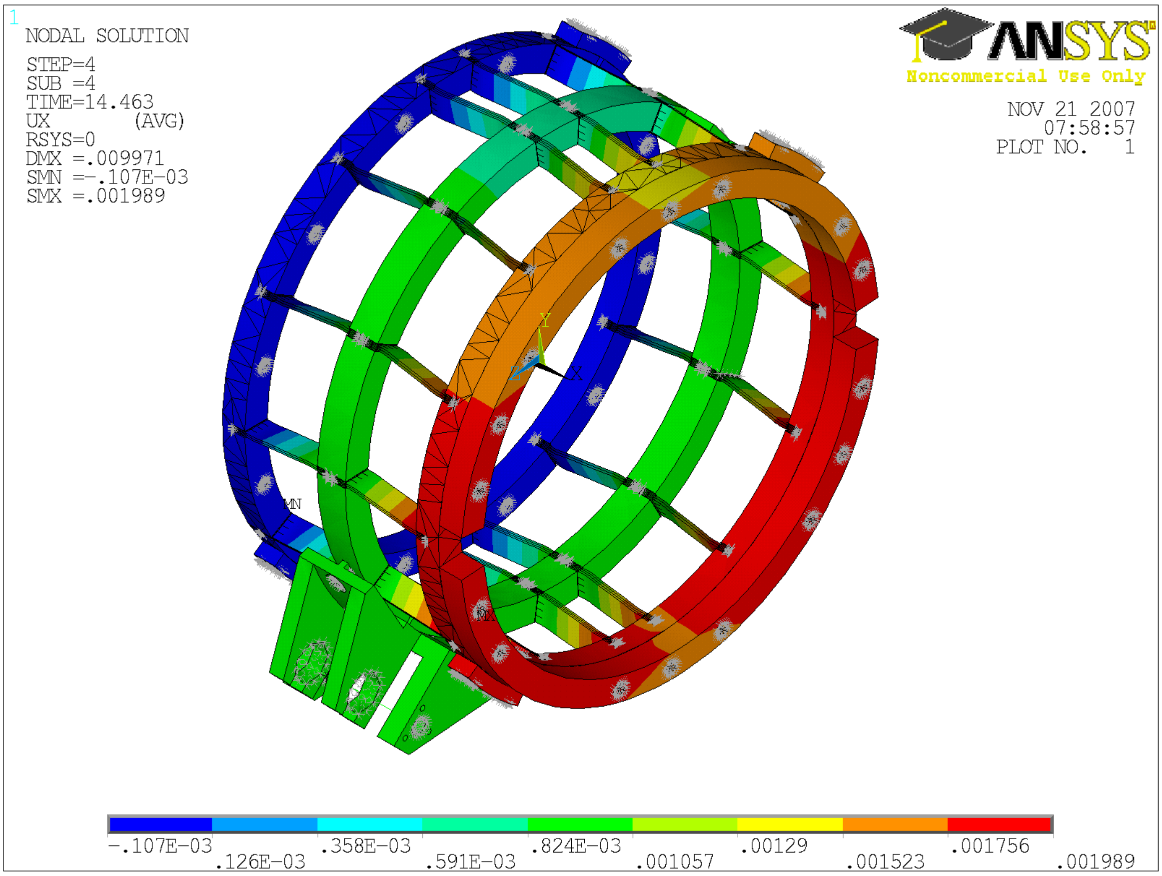


Figure 30: displacement of the tuner at the limit load level (12500 N)

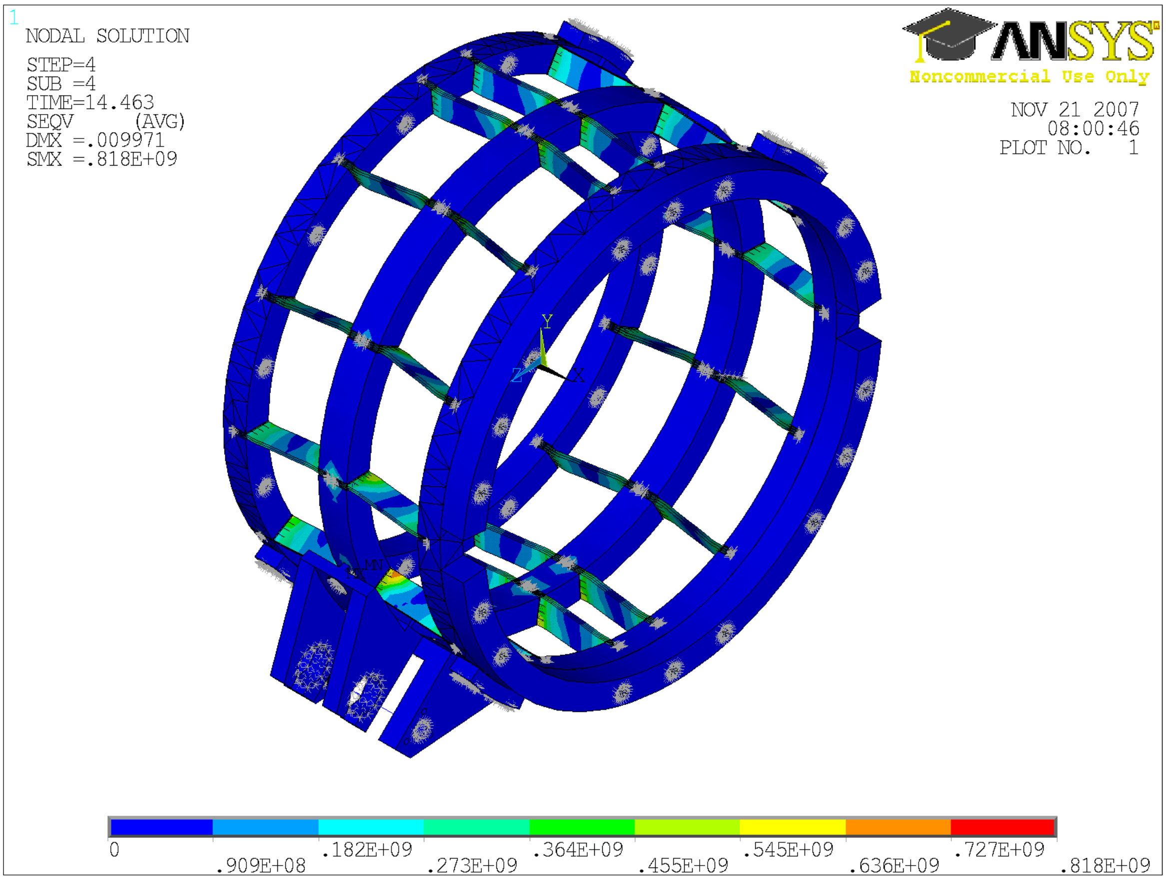


Figure 31: von Mises stresses in the tuner at the limit load level (12500 N)

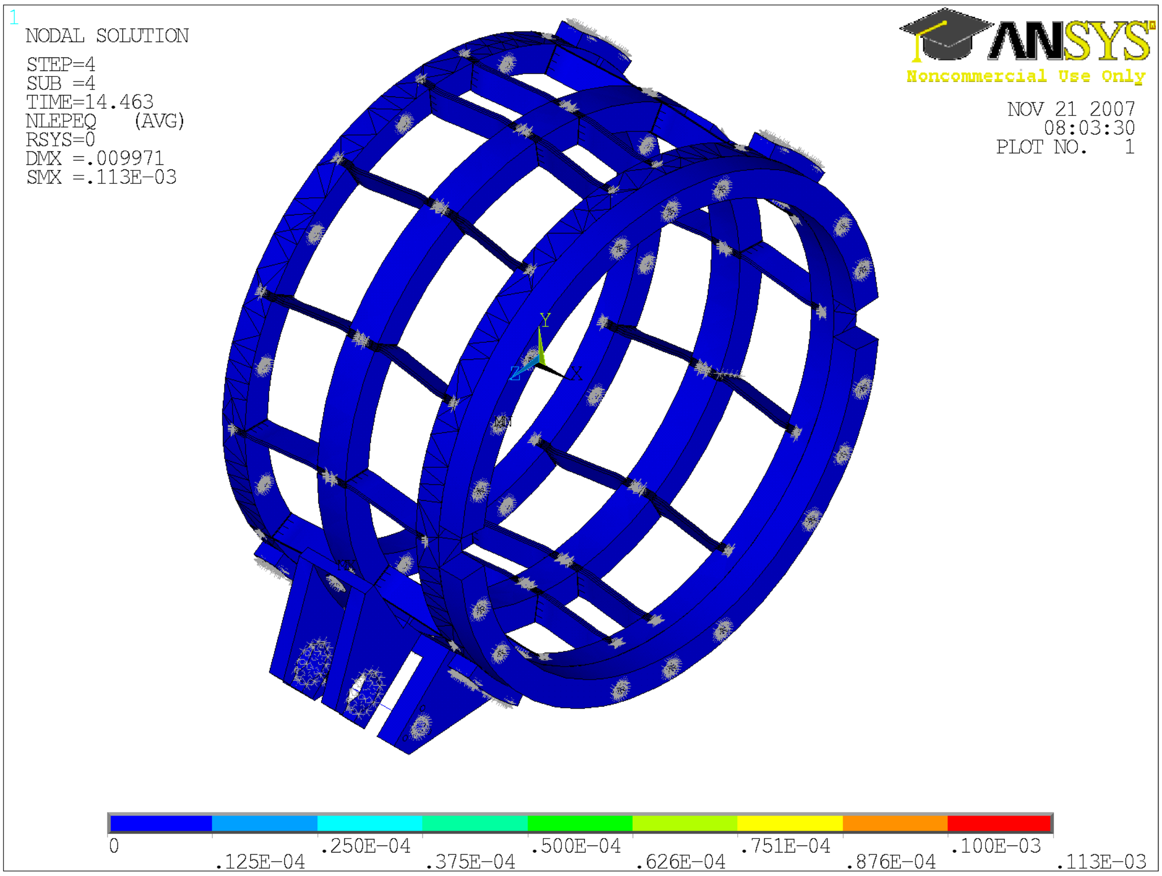


Figure 32: equivalent plastic strains in the tuner at the limit load level

6.4 Tensile strength

In some cases, like the certification test or during the cool down, the tuner can be subject to tensile forces. These forces are applied by means of the four safety bars that have to be properly assembled in order to avoid any permanent deformation to the cavity.

The maximum tensile force that the tuner can withstand has been determined by means of a finite element analysis in its undeformed state. The results, in terms of total forces versus the four bars displacement, are reported in figure 33. It can be observed that:

- the displacement at node 21919 is higher than that of other nodes. This is a consequence of a non complete symmetry of the motor and bolts position (see also figure 34);
- the re-alignment of the blades has a stiffening effect
- the solution stopped at a total force of 19000 N, due probably to some contact algorithm problems. In any case it can be noted from figure 35 that the stress level is near the yield point, therefore the maximum tensile force can be assumed equal to 19 kN.

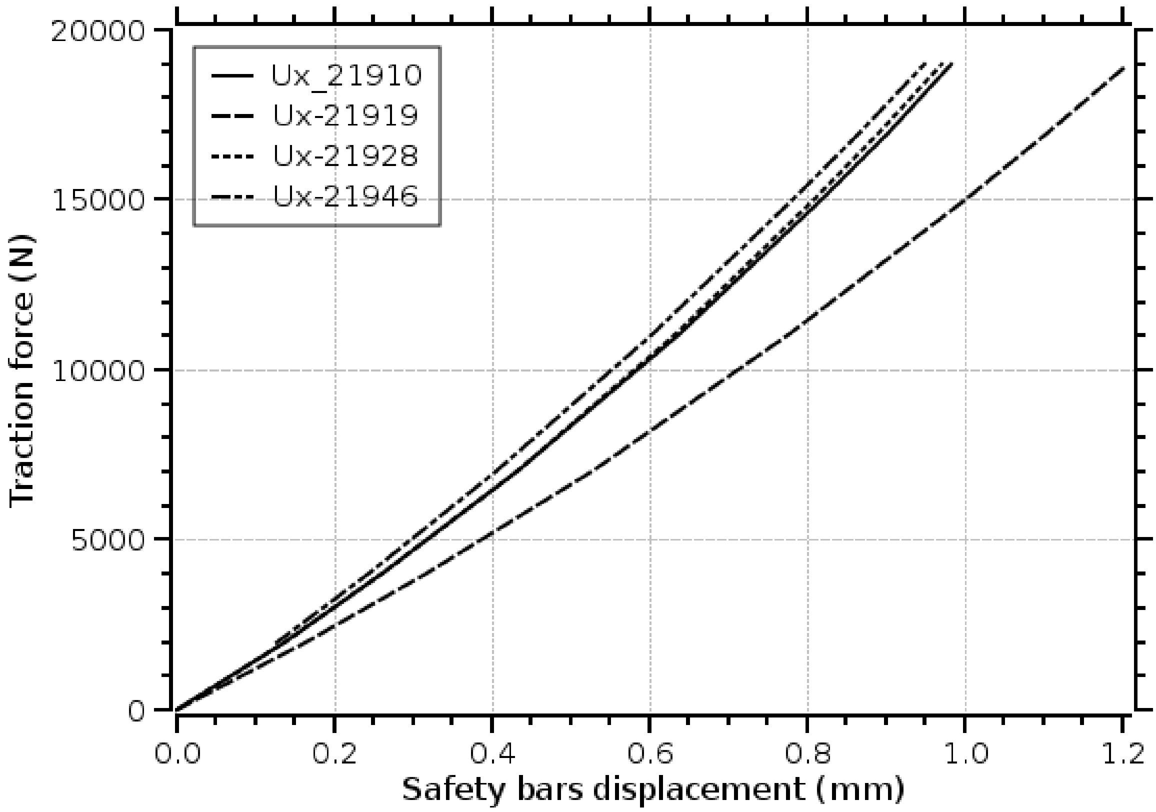


Figure 33: tuner elongation at the four safety bars position

The compression force on the driving screw is reported as a function of the tensile load in figure 36.

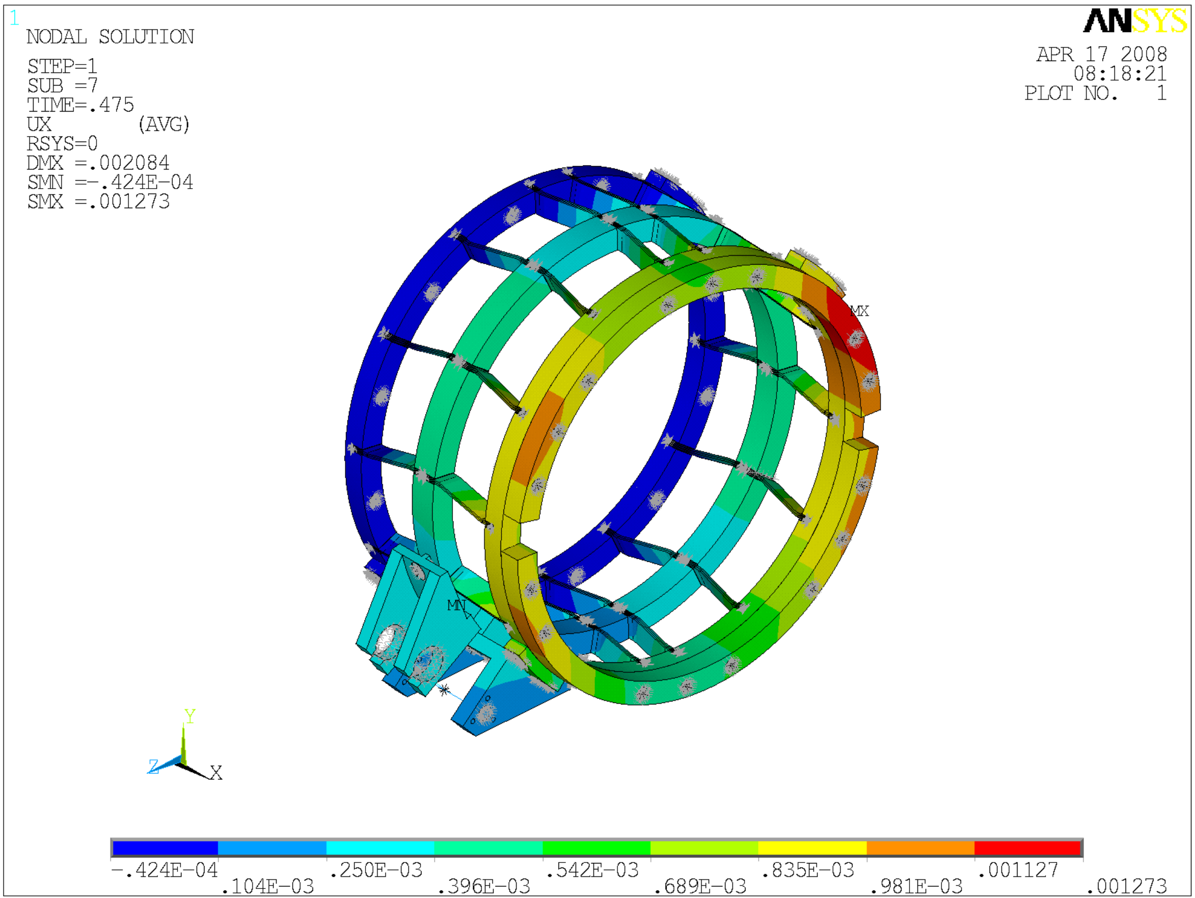


Figure 34: tuner displacements for a tensile force of 19000 N

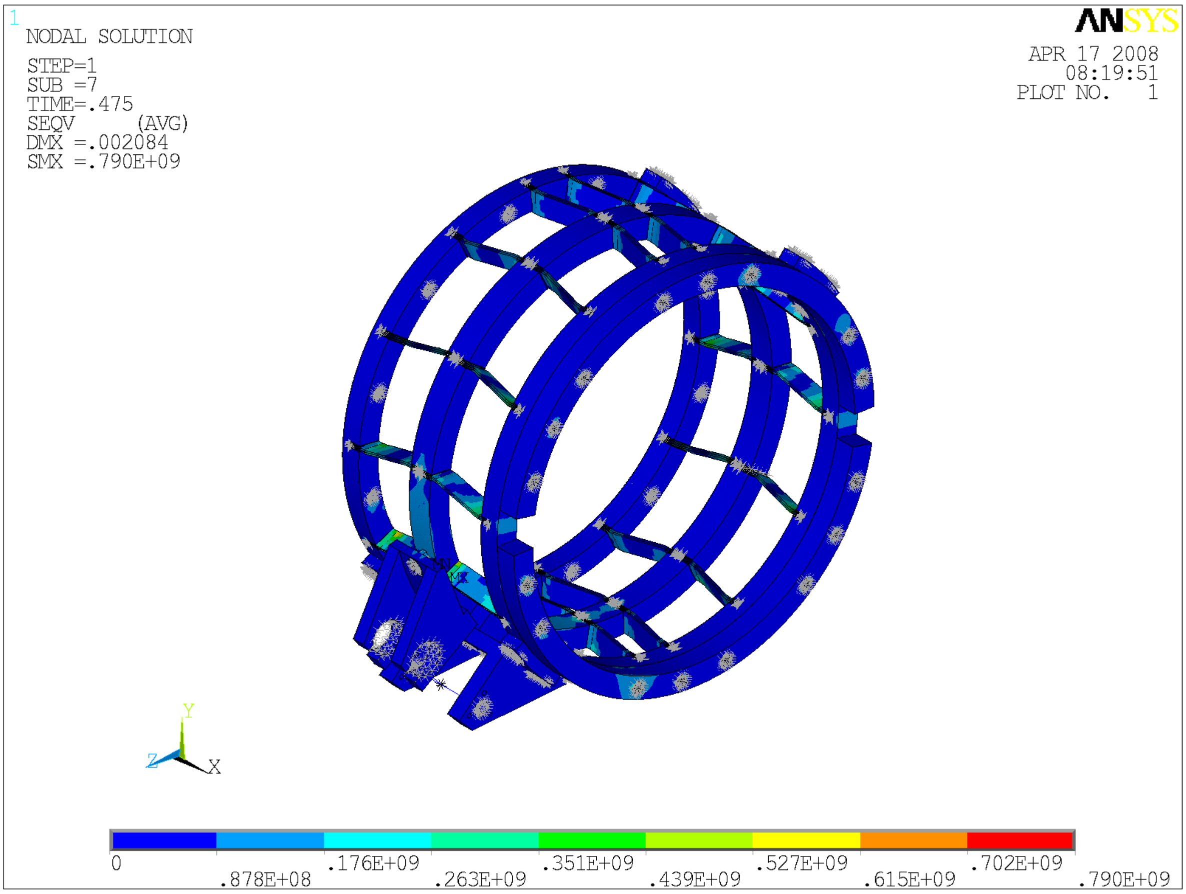


Figure 35: von Mises stress for a tensile force of 19000 N

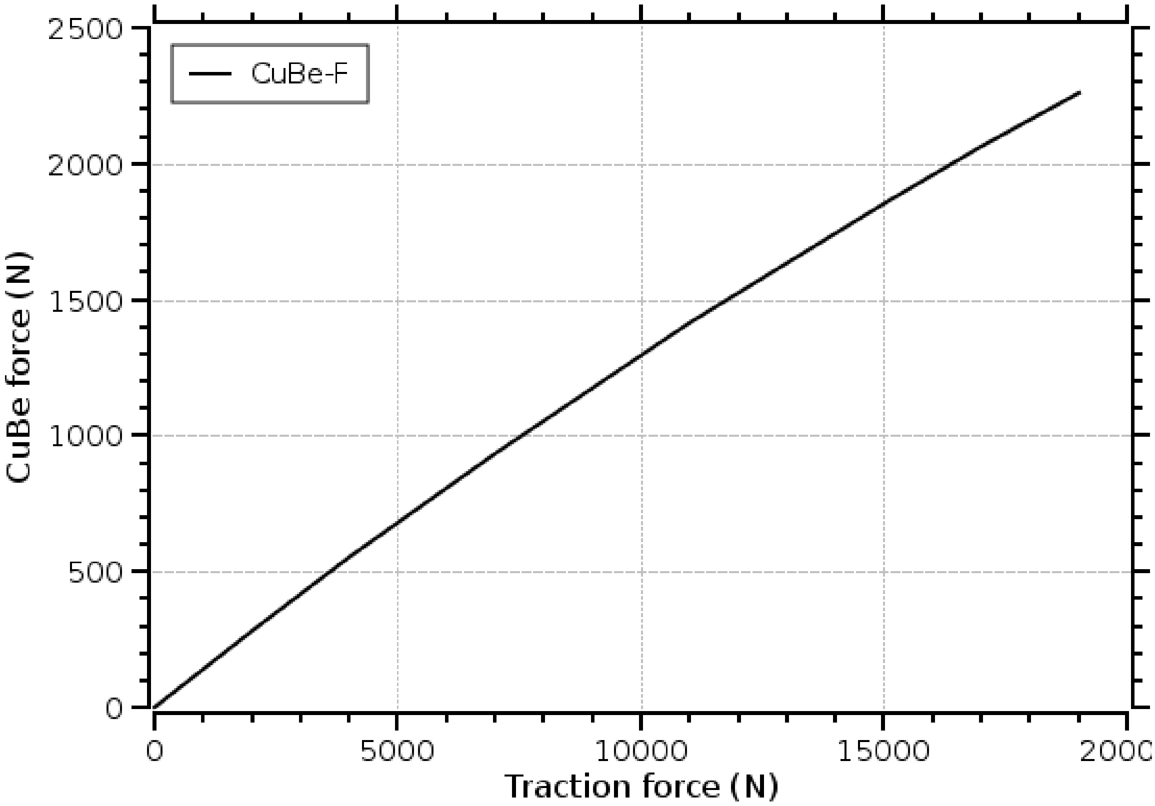


Figure 36: compression force on the driving screw

7 Check of bolts and motor requirements

7.1 Motor requirements

The torque required at the driving screw is obtained by means of the principle of virtual work neglecting the frictions:

$$M\,\delta\,\theta = F\,\delta\,L \rightarrow M = \frac{F\,\delta\,L}{\delta\,\theta} = \frac{2000\,1.5}{2\,\pi} = 478\,Nmm$$

The force of 2000 N is an higher approximation of the maximum value reported in figure 24, while the screw pitch of 1.5 mm is the maximum foreseen for this driving mechanism.

The value obtained is lower than the nominal value of 30 Nm of the Phytron motor equipped with a 100:1 gearbox. In the hypothesis of a current of 1.0 A, and not 2.5 A, the available torque should be not less than 10 Nm, always higher than the required one.

The friction expected is equal to 2.5 Nm (scaling from a DESY request to Phytron), therefore the total required torque should be not higher than 3 Nm.

7.2 Bolts

8 Evaluation of tuner ver. 3.9.4 stiffness

Being the behavior of the tuner non linear due to the varying of the blade slope, the stiffness change during the operation and must be evaluated step by step. A good estimation of the axial stiffness can be obtained observing the different elongation between the unloaded and loaded case at the same driven level (see figure 37). This difference is reported in figure 38. The stiffness values reported in figure 39 are obtained by the ratio between the piezo reaction and the corresponding displacement. The values obtained for a number of screw turns higher than 12 are obviously wrong, because plastic deformations occurs and a different approach must be used in order to evaluate the overall tuner stiffness.

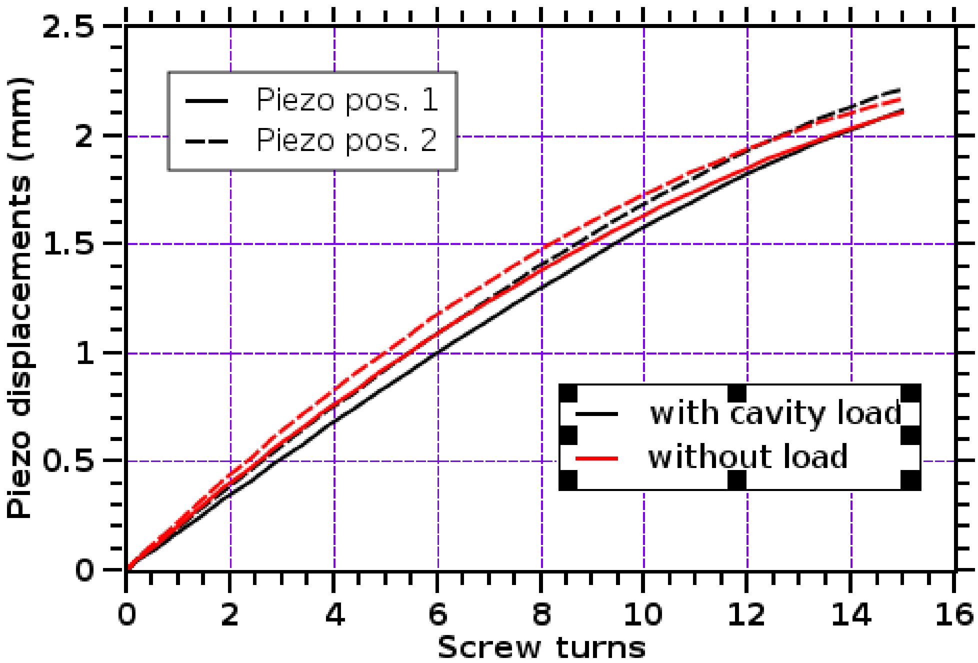


Figure 37: displacements at the piezo positions with and without cavity reaction

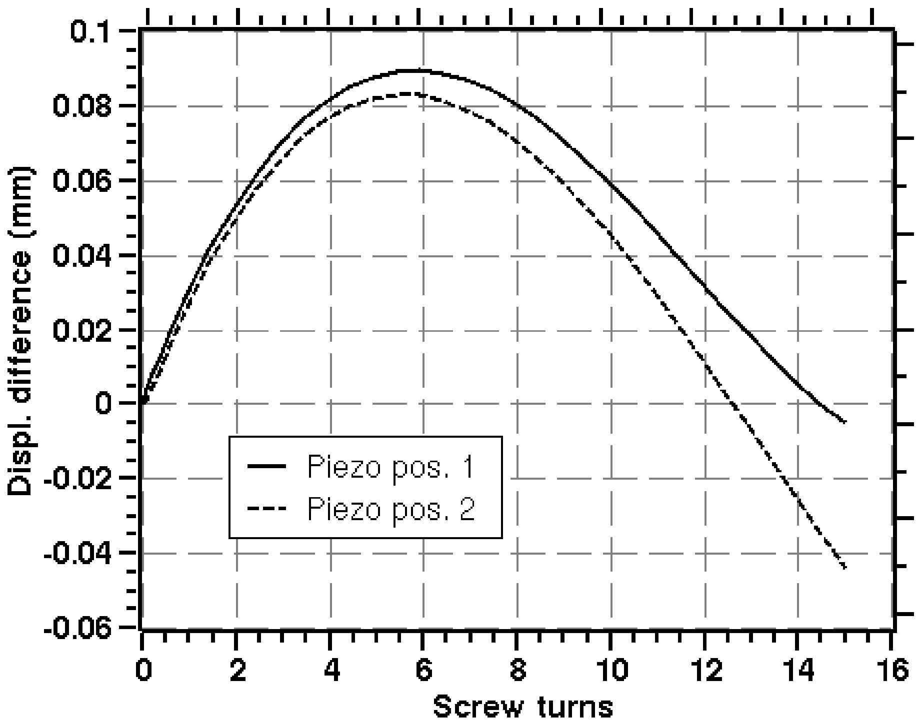


Figure 38: difference of displacements at piezo position between the loaded and unloaded case

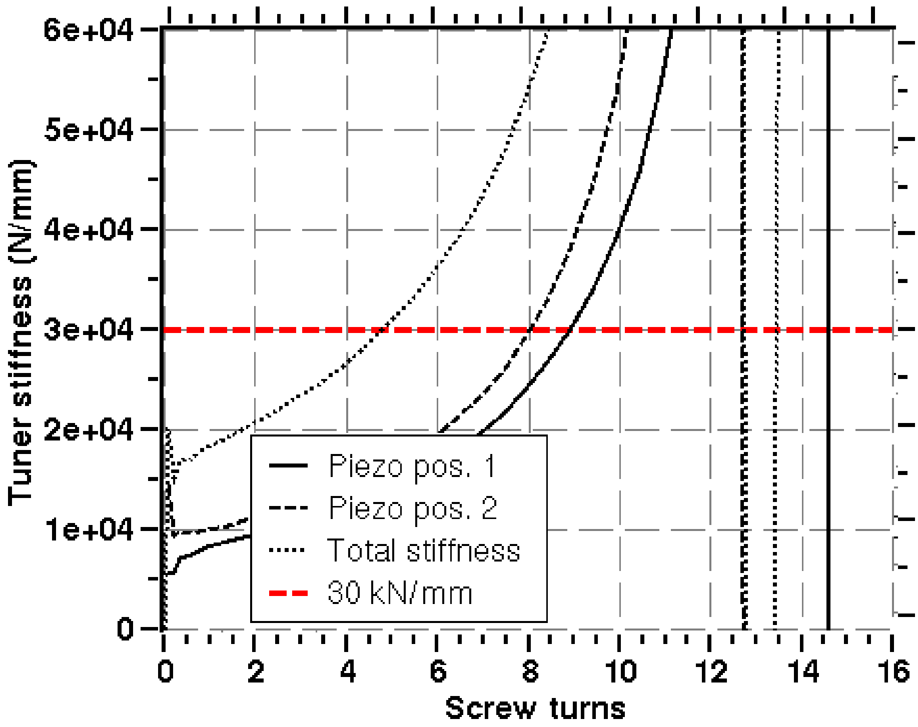


Figure 39: approximated stiffness as a function of driven position

9 Conclusions

The main parameters of the actual design are reported in table .

Tuner ver. 3.9.4	Tuner characteristic	Required value	Margin factor
Tuning range (no hysteresis)	0 – 500 kHz		
Tuning range[3] (some hysteresis)	0 – 600 kHz		
Max compression strength[4]	7800 + 3100 N	7800 + 1.1 * 2840 N	1.0
Max traction strength	16000 N	13771 N	1.16
Compression stiffness	15 – 100 kN/mm		
Mean sensitivity[5]	3 Hz/step		
Torque[6]	10 Nm	3 Nm	3.3

[3] With plastic deformations limited to the blade packs near the motor
[4] This is composed of the fixed part due to the cavity deformation and a variable part due to external pressure
[5] With a screw pitch of 1.5 mm and a 100:1 gear box
[6] See section 7.1 for the assumptions

Bibliography

[1] *Evaluation of stress in the blades of coaxial tuner*, Nicola Panzeri, INFN Milano – LASA, internal note, 2006-11-21

[2]