LCGT-SAS Note
Lower vacuum chamber tower B

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R. De Salvo, F. Raffaelli

Abstract

This note contains descriptions, calculations, analysis and a documentation of the lower vacuum chamber of tower B1 and B2.
This note is organized in sixth sections. The first section is an introduction that describes the project of the towers B1 and B2.
In the second section analytical calculations based on the Pressure European Directive (Pressure European Directive EN 13445) are reported and critical discussion on their use are tackled. The third section contains finite element analysis (Fea) analysis to verify the rigidity and stress on the chamber. The fourth section is dedicated to linear buckling analysis.
The fifth section is the nonlinear buckling and the verification of limit of gross plastic deformation (gross plastic deformation). In the sixth section the conclusions are drawn.
All the sections are linked to the appendix. In particularly the Appendix A1-A16 contains in detail the calculation based on the Pressure European Directive code for unfired pressure vessel.

Conclusions:

7.1 Conclusions.

1.1 Project description

The tower B1 and B2 contain cylindrical vacuum chamber of external diameter of 1.5 meter and total high of about 4.5 meter (see Fig. 1 and Fig. 2). We assume that the operating temperature is 20 ^{0}C. The location of the towers is inside buildings.

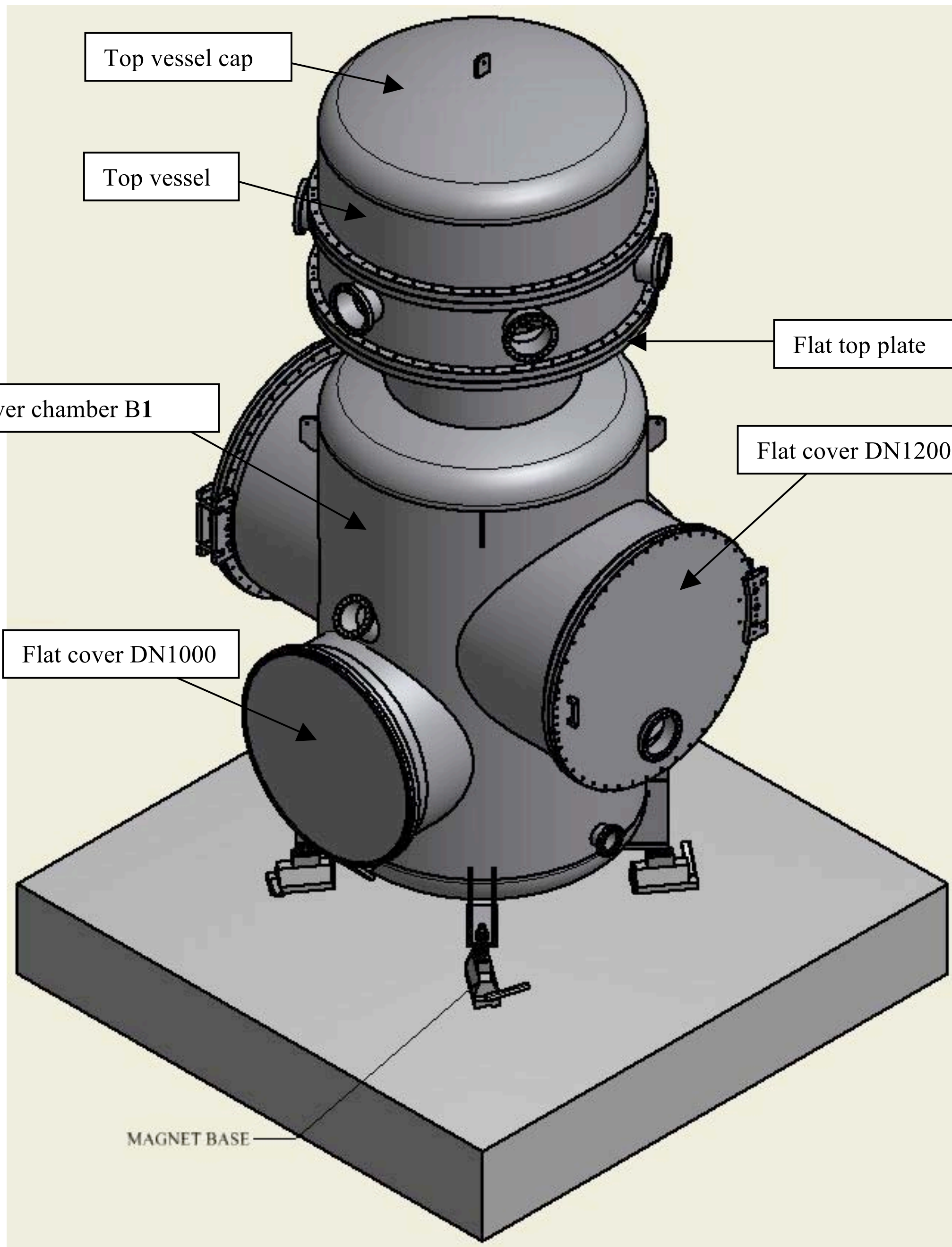


Fig.1 Tower B1 vacuum chamber

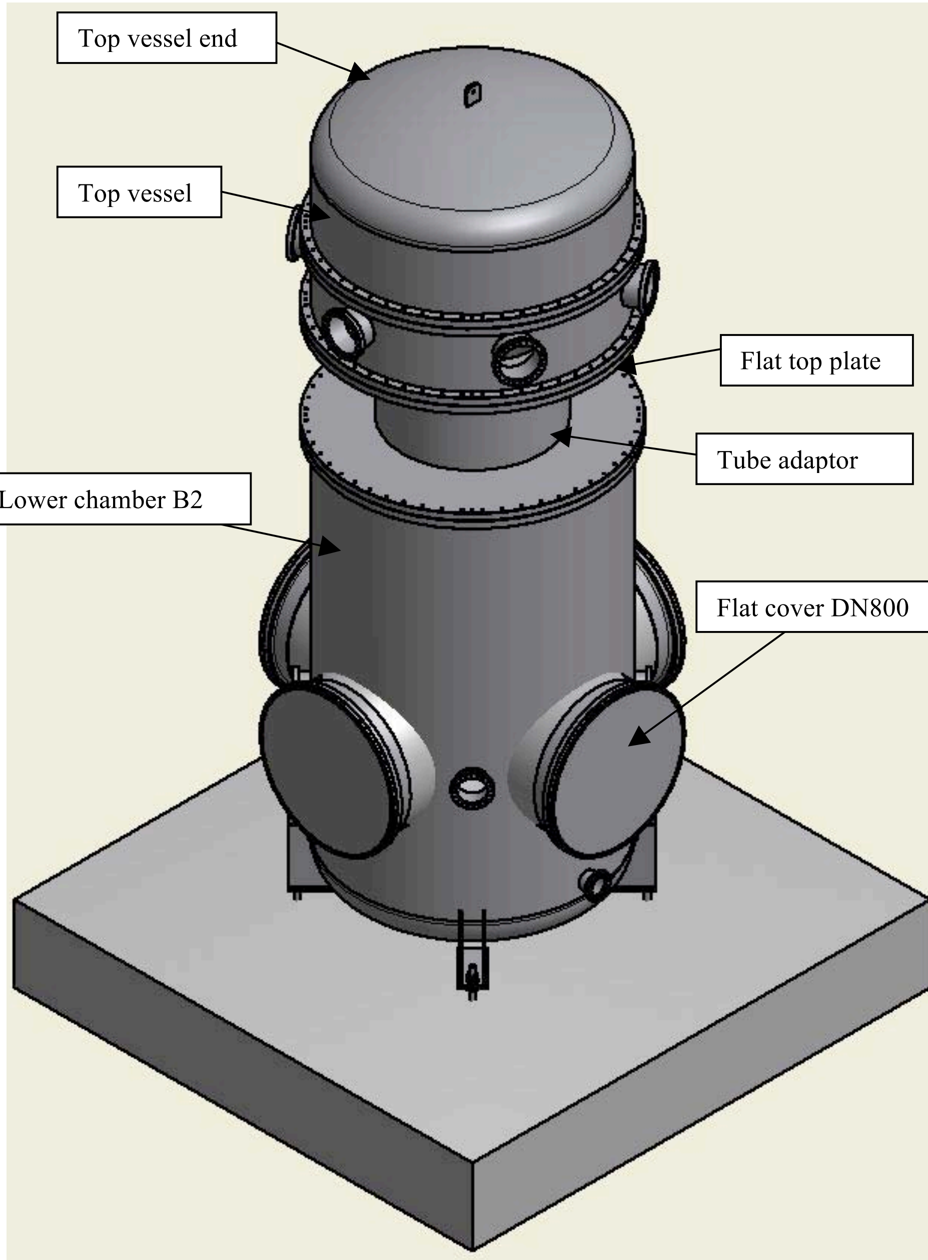


Fig.2 Tower B2 vacuum chamber

The vacuum chambers are composed of four parts flanged and bolted together. We named starting from the top: Top vessel cap; top vessel; flat plate top; tube adaptor; lower chamber B1 and B2.
The top vessel cap, top vessel and flat plate top are common to the tower B1 and tower B2. Instead the lower chamber B1, B2 and the tube adaptor are different (see Fig.1 and 2.)
The large openings required around the mean diameter of the lower chamber B1 and B2 deserve a design analysis; a design by formula or by rules of the pressure code is applied. The design analysis according the Pressure European Directive requires several

verifications considering the material plasticity behavior and the geometrical imperfection on the geometry. A gross plastic deformation analysis determines the pressure that gives a max strain of 5% in the materials. The pressure obtained is rescaled using a safety factors for the load, material and plasticity criteria used.
This gives the max admissible pressure. In our case the instability checks is a first step that must be performed (I). The Pressure European Directive procedure of the instability checks goes thru the linear buckling analysis. The shapes of the buckling modes obtained are rescaled to mimic a maximal actual construction imperfection (for instance a 1% or 0.5% of geometrical deviation from the design shape). Using this starting geometry we applied an incremental load (external pressure 20% higher of the first mode) using a model with plasticity and large deflection looking for the condition in which the deformations start becoming large for a small pressure load increment.
The value obtained is divided by the pressure load and material safety factor. In our case since we operate with vacuum the load safety factor is one and for the material used the material safety factor is γ_r=1.25.

2. Operative conditions and material properties:

The operation condition of vacuum vessel is external pressure of 0.1 Mpa at room temperature 20 ^{0}C. The material considers is type AISI 304L (1.4307). We summarize the material mechanical properties:

Minimum 0,2 % proof strength	> 180 Mpa
Minimum 1,0 % proof strength	> 215 Mpa
Minimum tensile strength	> 460 Mpa
Minimum rupture elongation	> 35%

For this material the Pressure European Directive gives maximum allowed values of the nominal design stress for pressure parts of 120 Mpa. Using the code formulas we can determine the thickness of the cylinders, the tori-spherical ends, flat covers, flanges, nozzles etc, considering the joint efficiency 1. The material safety factor in this case is 1.25.

2.1 Pressure European Directive analyses by formula.

Three elements are common to the Tower B1 and B2, top vessel cap, top vessel and flat plate top. In appendix A1, A2…A15 we report in detail all the calculations. In table 1 are reported for reference required thickness for the tori-spherical end, for the cylinders, for the flanges and for the flat plates not considering the reinforcements required for the presence of holes or for not full-penetration welding. Furthermore the actual thicknesses used are larger, considering that most of the welds are not continuous and that extra thickness may be required to fulfill the high leak tightness requirements. A more a rigid structure is required with respect to strength criteria requirements. Furthermore the use of large thickness are justified by several issues: vacuum welding technology impedes the use of heavy welding; the outside welds are not continuous, due to the requirements for leak searches, and to avoid the need of reinforced areas in the vicinity of the holes for the nozzles.

The flange analysis is carried out considering that max momentum is applied to preset the gasket; this is the more severe condition. The total load of the large flange DN 1500 for squishing the gasket is 117 tons, considering a presetting force of 200 newton/mm. In this case 52 screw M12 property class 70, each preloaded at 2 ton assure the presetting of the gasket, in order to minimize the bending moment on the flange itself we optimize the bolts position make them as close as possible to the gasket grove. In fact the distance between the screws axis and the average diameter of the gasket it is only 13mm.

The required and the actual thickness are reported in table 1.

	Geom. Par. (mm)	Required thickness (mm)	Actual thickness (mm)
Thori-spherical end	Ravg=1500	4	8
Cylinder top cylinder end	Ravg=746; H=543	3.5	8
Cylinder top vessel	Ravg=746; H=440	3	8
Top vessel flange	Dia=1500	46	46
Flat plate top	Dia=1500	47	45
Flange DN980	Dia=983	35	35
Lower chamber cylinder B1	Dia =1500; H=2346	5	10
Lower chamber cylinder B2	Dia=1500; H=2451	6	10
Flange 1500	Dext=1600	46	45
Flat cover DN800	Dext=880	14	30
Flat cover DN1000	Dext=1068	17	30
Flat cover DN1200	Dext=1310	21.5	35

Table 1

In table 1 we report the reference thickness calculated by formula on the assumption that are the welding have joint efficiency 1 and in most of the case we do not include the presence of holes. In table 2 we summarized the Appendixes content.

In table 2 we report the list of the checks made for the various components.

TOP VESSEL CAP	Torispherical end thickness	Appendix A1
	Cylinder thickness	Appendix A2
	Flange DN1500 thickness	Appendix A3
TOP VESSEL	Cylinder thickness	Appendix A4
	Flange DN 1500 thickness	Appendix A3
FLAT PLATE TOP	Thickness calculation	Appendix A5
TUBE ADAPTOR	Flange DN 980 thickness	Appendix A6
	Cylinder thickness	Appendix A7
	Flat plate DN 1500	Appendix A8
LOWER CHAMBER B1	Cylinder thickness DN 1500	Appendix A9
	Cylinder thickness DN1200	Appendix A10
	Flange DN 1200 thickness	Appendix A11
	Torispherical end thickness	Appendix A1
	Flange DN 980 thickness	Appendix A6
LOWER CHAMBER B2	Flange DN1500 thickness	Appendix A3
	Cylinder thickness DN 1500	Appendix A12
	Cylinder thickness DN 1000	Appendix A13
	Torispherical end thickness	Appendix A1
Flat cover	DN800	Appendix A14
Flat cover	DN1000	Appendix A15
Flat cover	DN1200	Appendix A16

Table 2

In all calculations the standard required thickness to hold vacuum considers join efficiency equal to 1 that is not our case. However this table is, in some cases, a point of references for the thickness used.

Linear Fea analsys:

3.1 Case and results of lower chambers.

We analyzed the case in which the main cylinders of the lower chambers are 10mm and 8mm, while the nozzles tubes are respectively 8mm and 5mm in thickness. Nozzle and small tube are closed by flat covers respectively 30mm and 15mm thick. We model only a quarter with the symmetry boundary conditions.

In table three we report the thickness used the maximum displacement and the Von Mises stress on top and bottom of the shell. The linear analysis tells us when we not exceeded the elastic limit with Von Mises criteria.

	Model	Cylinder Thickness (mm)	Nozzles Thickness (mm)	Max deflection (mm)	Max Von Mises stress top shell (Mpa)	Max Von Mises stress bot shell (Mpa)
Tower B1	Fine mesh	10	8	0.7	92	78
Tower B1	Fine mesh	8	5	0.9	157	134
Tower B2	fine mesh	10	8	0.75	123	123
Tower B2	Fine mesh	8	5	0.96	125	144

Table 3

In Figure 3 we show the mesh used for the two models of Tower B1 and B2.

Figure 3

Figures 4 to 7 illustrate where stress localizes in the four cases of table 1, using the fine mesh model.

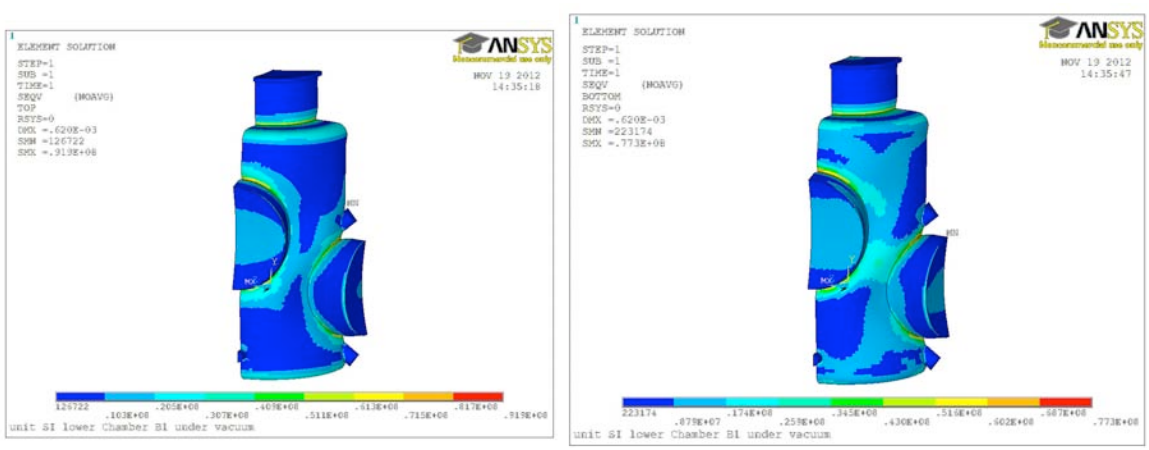


Figure 4

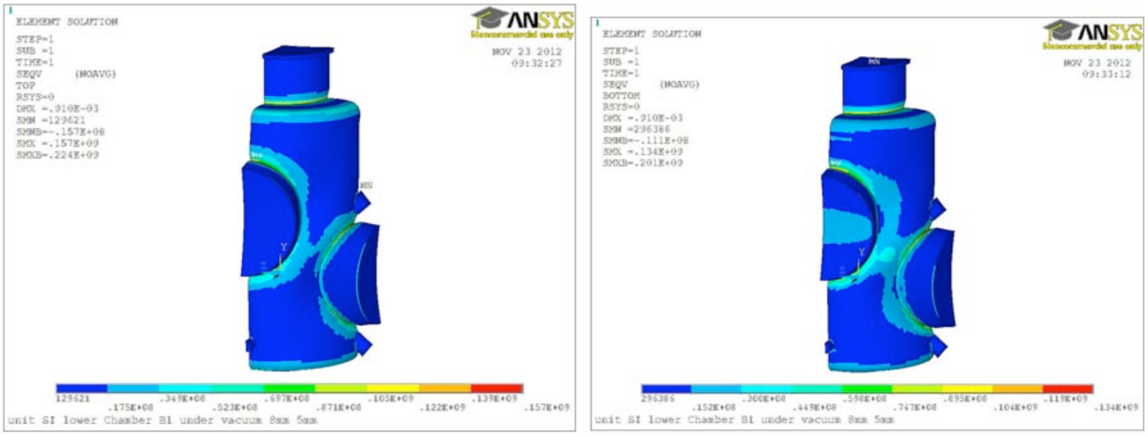


Figure 5

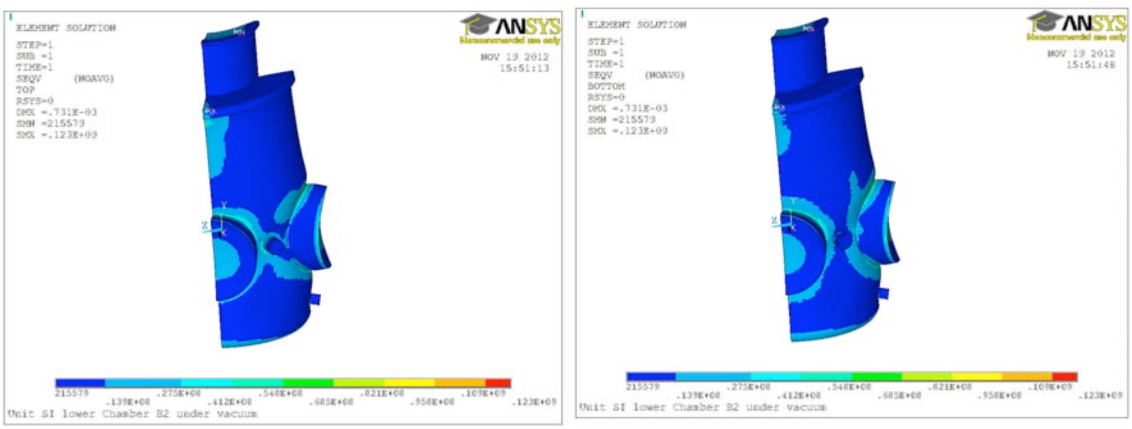


Figure 6

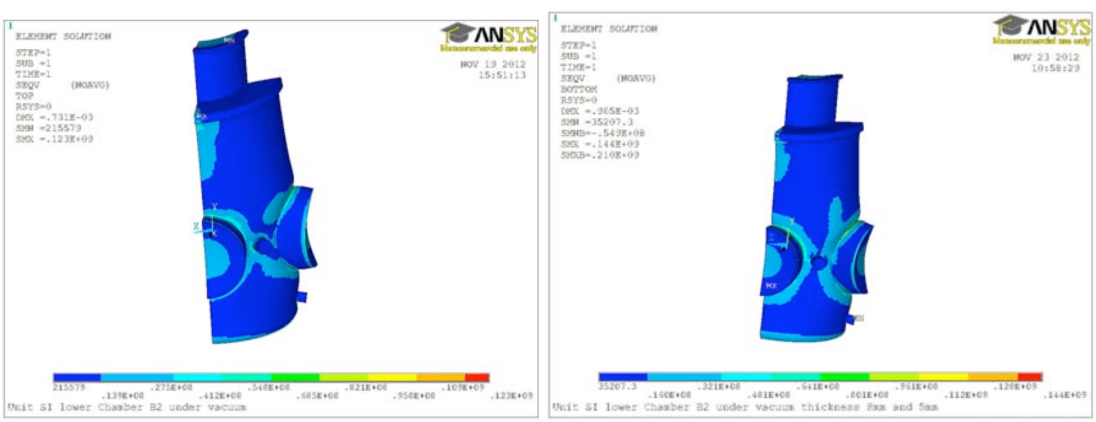


Figure 7

We note that the maximum stress is, not surprisingly, localized around the intersection between the nozzles and the main cylinder. These regions are coincident with the welds area in which there is a lack of material due to the type of weld used. In addition, residual internal stresses are present in these areas, due the welding process. These regions are called technical (HAZ) the heat-affected zone.

3.2 Flat top plate

A Fea model of the flat plate is made to verify the stress imposed and its stiffness. For simplicity the plate is considered restrained at the external diameter e loaded with 1 bar of pressure plus three forces of 10000 Newton are applied in correspondence of the radius of holes staggered at 120 degree, the points from where the type-A chamber is suspended.

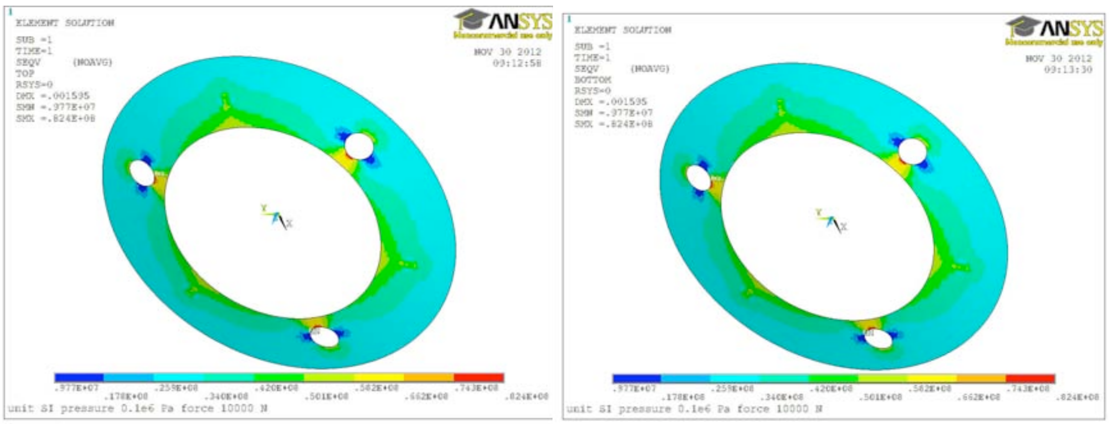


Figure 8

In figure 8 are illustrated the stress with the Von Mises criteria on the top and bottom of the shell. The max displacement is 1.5 mm and the maximum stress is 82 Mpa. We have to consider that the real max displacement will be less since at the center the plate is connected with a flange to a cylinder that provides additional stiffening. We neglect all the blind holes that are necessary to connect it to the lower flange. However we respected a rule for the thread blind hole (see par. 10.6.1.2) that say the thickness of the material exceeding the bolt hole depth is at least 50 % of the bolt diameter.

<u>Eulerian Buckling:</u>

4.1 Case and results lower chambers.

It is necessary to evaluate the linear buckling that gives an upper limit for the collapse pressure. This analysis, as mention above, is not conservative; the collapse will occur for a lower pressure load than the one listed in table. However this study let us understand how the chamber will collapse and gives a first order of magnitude of the collapse pressure. In table four we report the first four modes of linear buckling values for the models considered.

Tower	Model	Cylinder Thickness (mm)	Nozzles Thickness (mm)	Buckling Eulerian Mode 1	Buckling Eulerian Mode 2	Buckling Eulerian Mode 3	Buckling Eulerian Mode 4
Tower B1	Fine mesh	10	8	28.8	37.1	43.1	46.1
Tower B1	Fine mesh	8	5	16.8	22.1	22.3	24.328
Tower B2	Fine mesh	10	8	26.7	32.5	39.0	47.7
Tower B2	Fine mesh	8	5	15.8	17.4	22.4	25.0

Table 4

Figures 8 and 9 show the linear buckling mode for the tower B1 respectively for the thickness combinations of 10-8mm and 8-5mm respectively. In figure 10 and 11 the linear buckling mode for tower B2 respectively for the same thickness combinations.

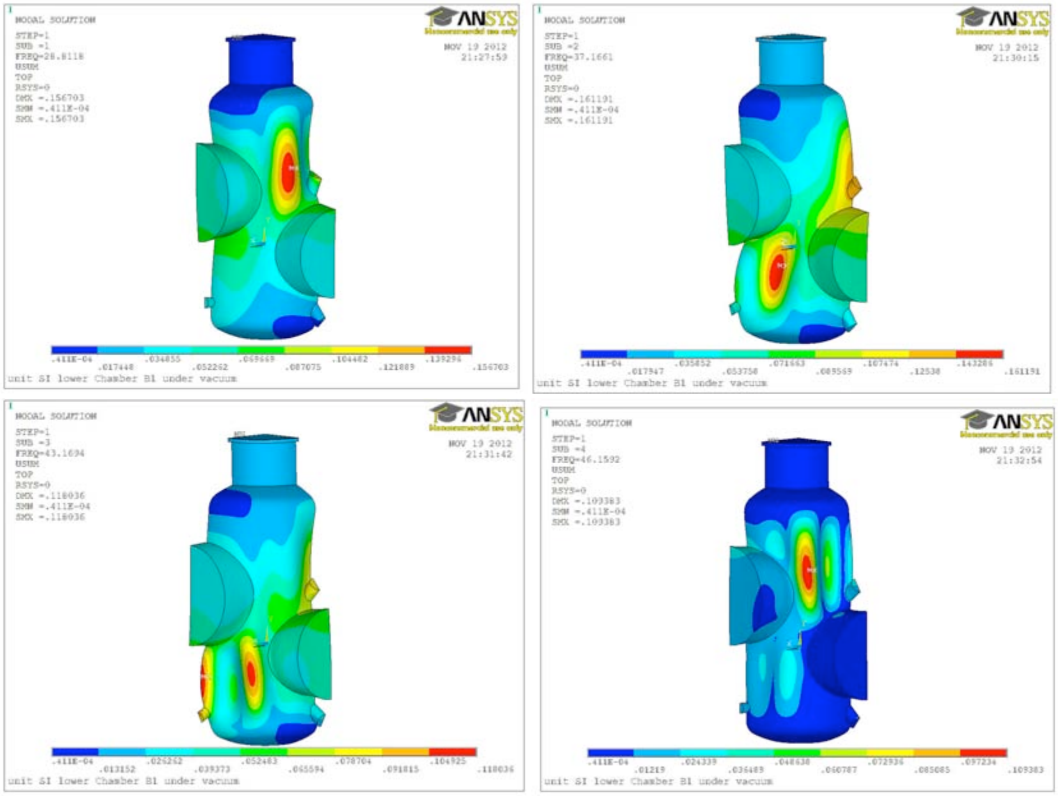


Figure 8 (Tower B1)

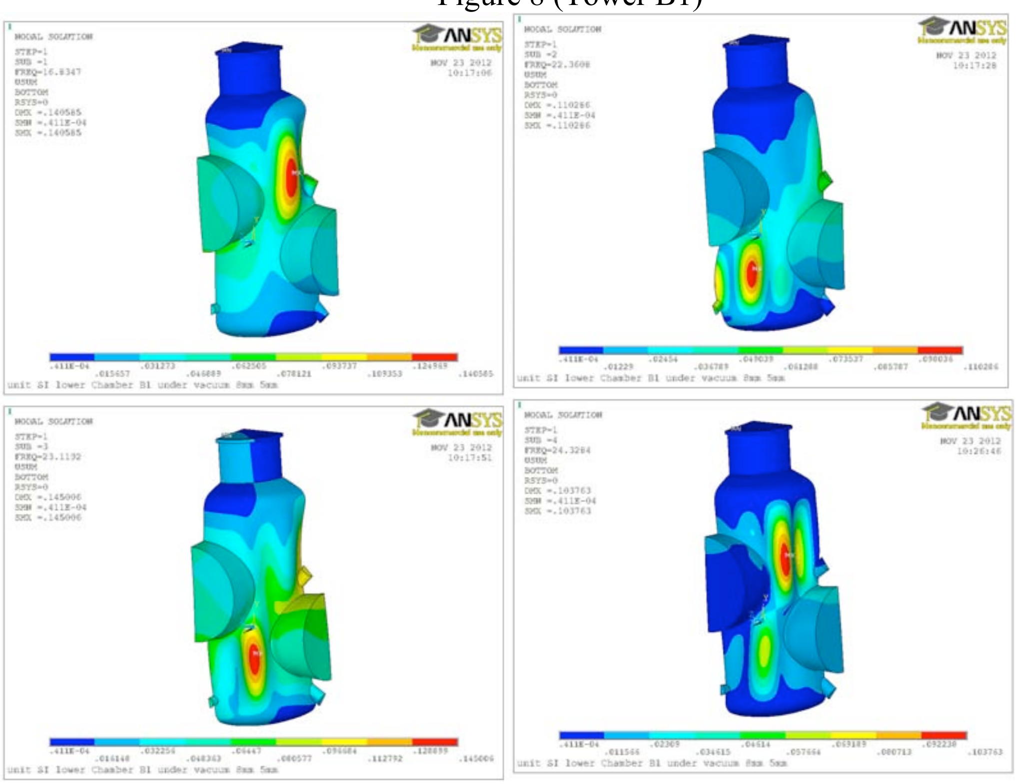


Figure 9 (Tower B1)

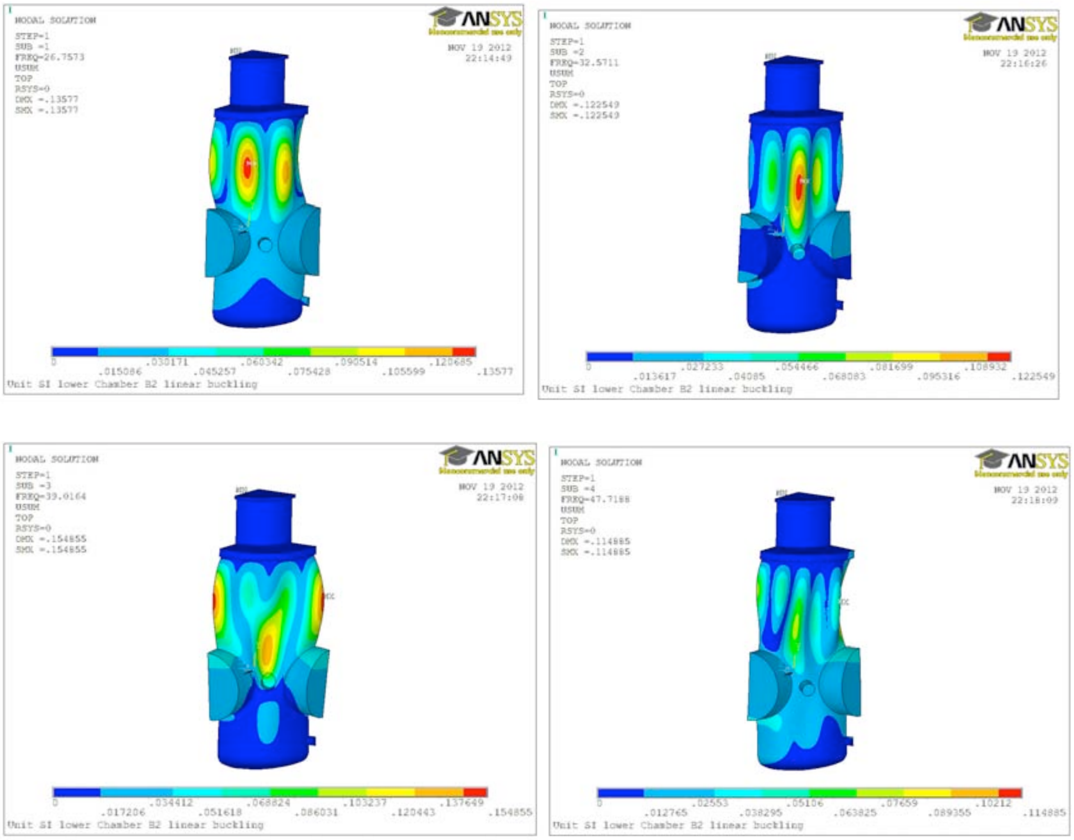


Figure 10 Tower B2

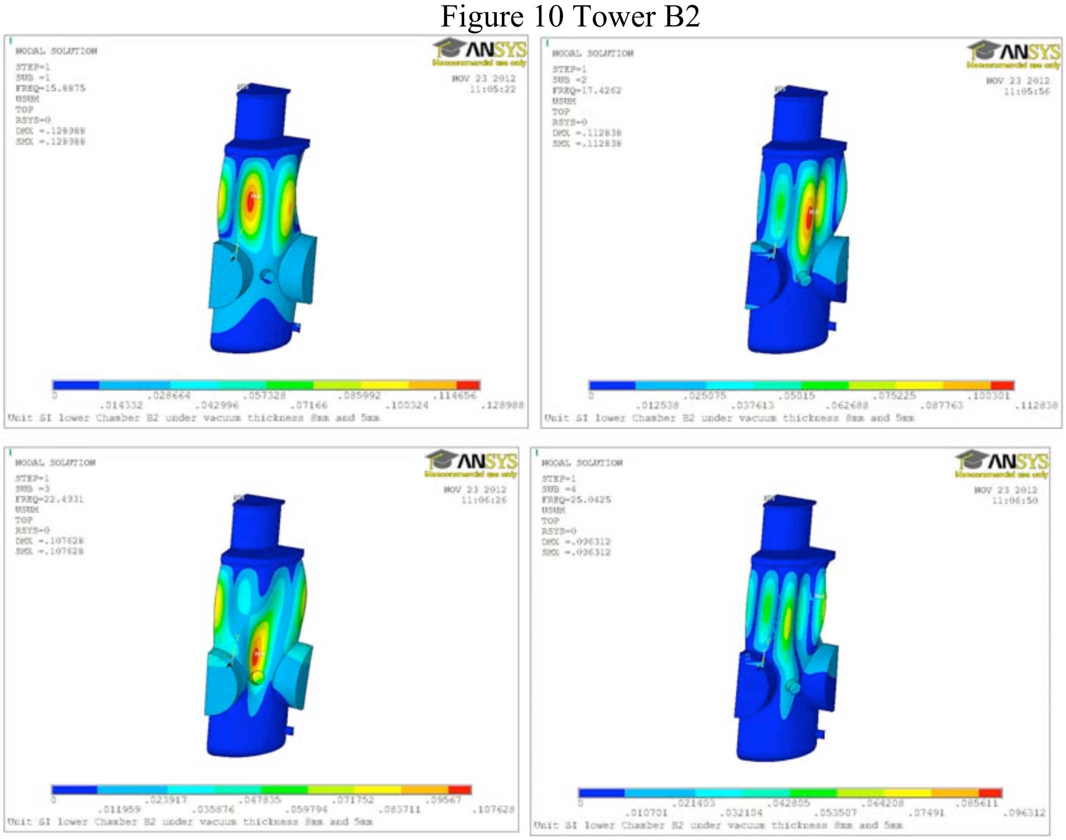


Figure 11 (Tower B2)

The values obtained are so high that any instability will occur after the strength failure occur. This suggests that the correct analysis to perform is the gross plastic deformation. The maximum allowable strain condition (5%) is reached much earlier in the welding region. Not surprisingly the final max pressure will be much lower of what found with the linear bucking analysis using the criteria mentions above.

Non linear Buckling and gross plastic deformation evaluation:

5.1 Case and results.

To evaluate the stress on the welds, we use a linear combination of the first four modes of the Eulerian buckling to generate an initial geometry with relevant imperfections (suitable to initiate early collapse).

Using weight factors of 0.1 and 0.05, we introduce into the model a geometrical construction deformation of 15mm and 8mm respectively. This geometry will be used as a starting point to perform a non-linear analysis that considers large defections and plasticity. The material plasticity behavior follows the fig. 13.

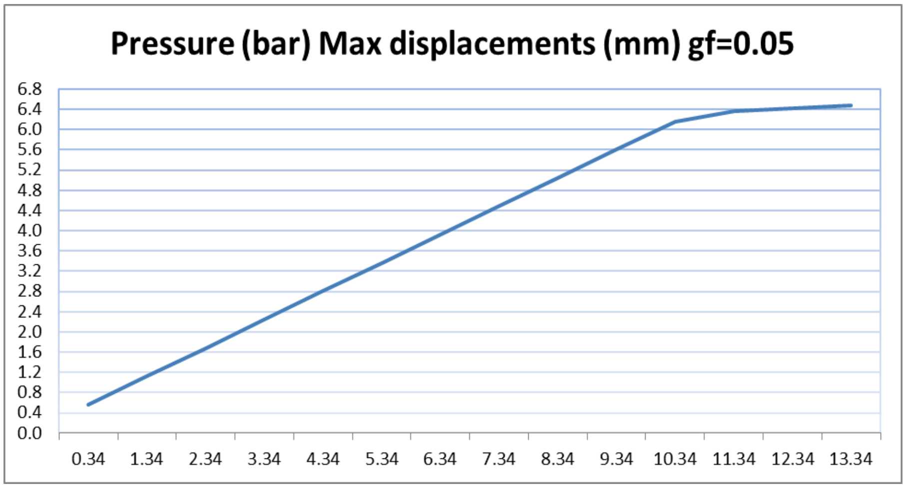
Figure 12 (TowerB1)

At the pressure in which the model starts becoming instable we need to verify that the maximum strain must remain under 5% at all points.

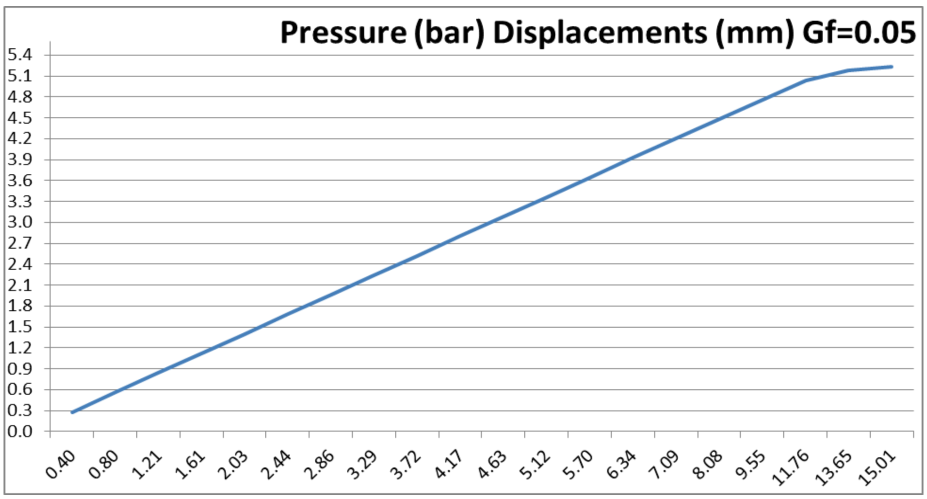
Two or three models were made and the critical pressure is ranging from 5.04 to 5.43 (bar) for the tower B1 considered (thickness of 10mm main cylinder and 8mm nozzles). The value obtained must be divided by a load safety factor that in our vacuum case is one (atmospheric pressure does not fluctuate), by a standard material safety factor that for the stainless steel is 1.25 and multiply by a factor $\sqrt{3}/2$ for the use of Von Mises instead of Tresca criteria.

Finally we conclude that the allowable external pressure for the tower B1 is 3.4 (bar), well in excess of atmospheric pressure.

A plot shows the displacements versus the applied pressure in bar, until plastic regime is reached.



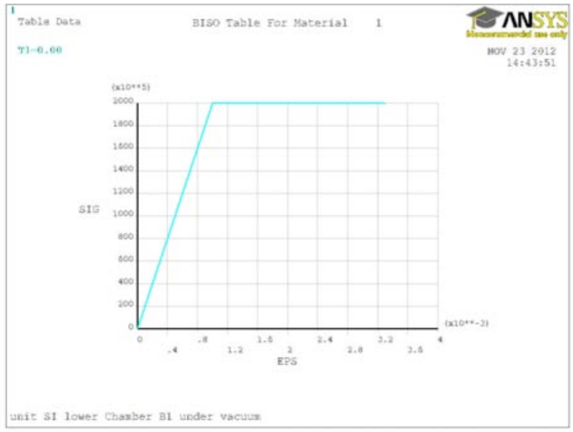
Plot of max displacemnt versus pressure tower B1



Plot of max displacemnt versus pressure tower B2

Model	Geometry correction factor	Pressure (Mpa)	Principal strain top S1	Principal strain top S3	Principal strain Bot S1	Principal strain Bot S3	Max Deflection (mm)
TowerB1	0.05	0.543	0.049	-0.030	0.059	-0.043	11.6
Tower B1	0.1	0.504	0.034	-0.020	0.041	-0.030	9.4
Tower B2	0.05	0.523	0.042	-0.032	0.034	-0.022	15
Tower B2	0.1	0.519	0.037	-0.029	0.031	-0.020	17

Table 5



Material model Fig. 13

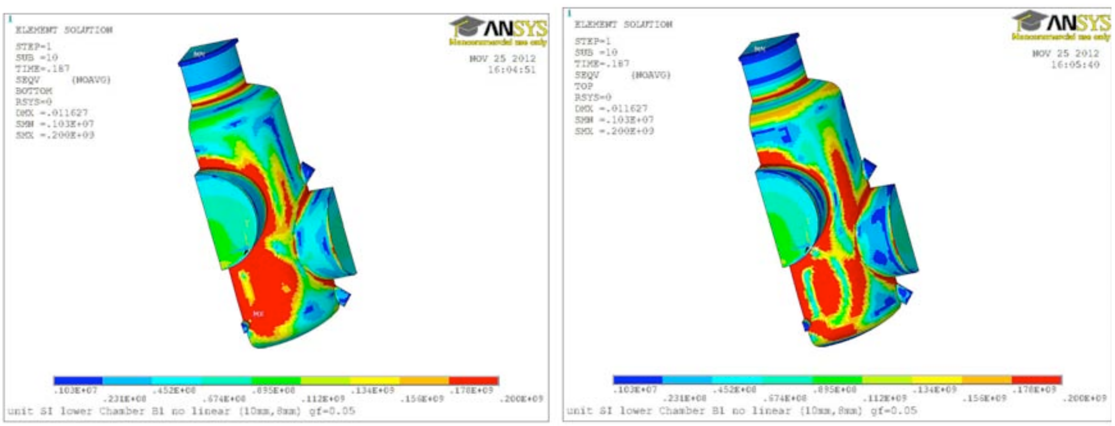


Fig. 14 Gf=0.05 stress Von Mises top and bottom tower B1

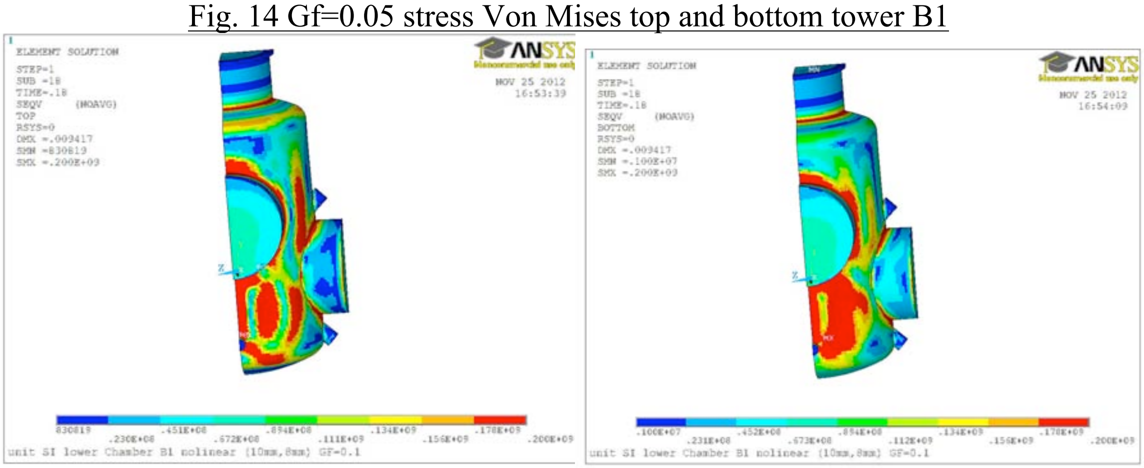


Fig. 15 Gf=0.1 stress Von Mises top and bottom tower B1

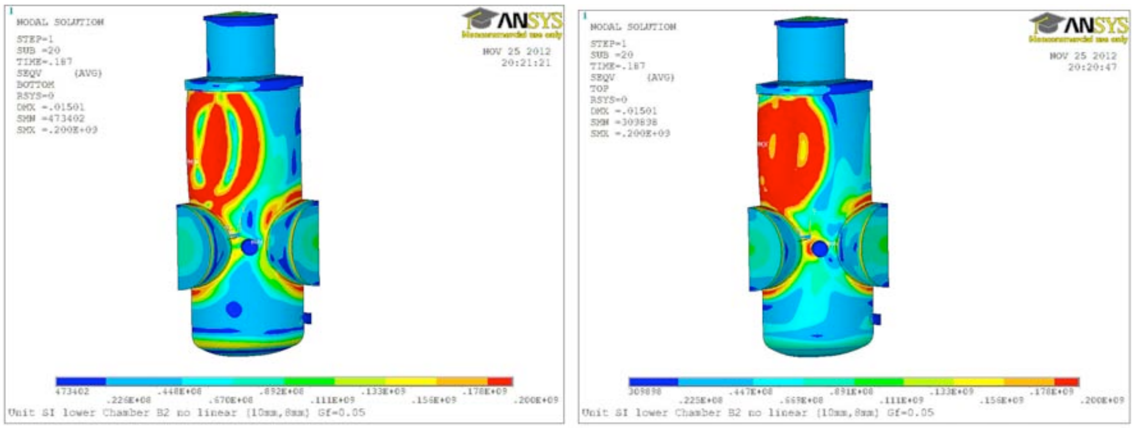


Fig.16 Gf=0.05 stress Von Mises top and bottom tower B2

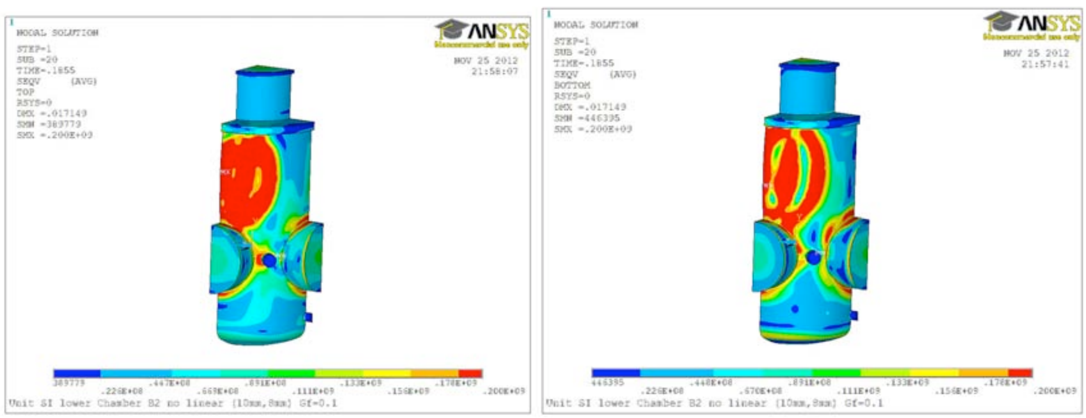


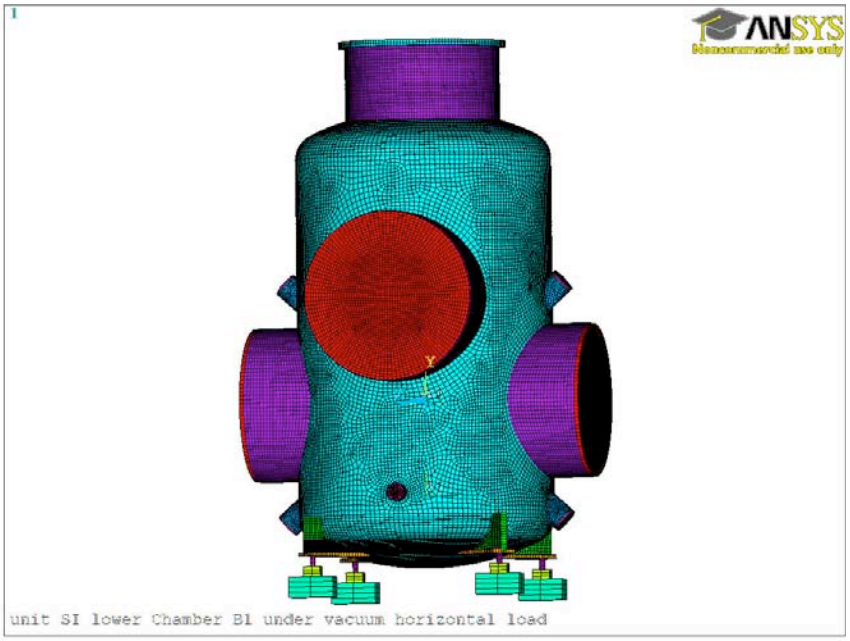
Fig.17 Gf=0.1 stress Von Mises top and bottom tower B2

The allowable pressure for the tower B2 is 3.4 (bar). The read region in Fig. 14,15,16,17, are the area in which we reach the plasticity at 200 Mpa. In table five are reported for each geometrical correction factors and pressure value the max principal strains on top and bottom shell and the max deflections.

Horizontal load on lower chamber B1:

6.1 Case and results of horizontal load applied lowers chamber B1.

In this chapter we analyzed a load condition caused by vacuum applied on only one of the nozzle of the beam line of tower B1. This condition, in presence of baffles, causes an unbalance horizontal force that must be absorbed by the feet reactions. From this analysis we calculate the feet reactions for a large imbalance, in order to evaluate the maximum allowable force unbalance. Using a full geometrical model of the lower chamber B1 adding a vertical load of 11,500 Newton due to the weight of the upper part that is not modeled in this section. When the gravity is applied the total vertical reaction is 36,011Newton. The model is loaded with the vacuum pressure, except for one 900 mm diameter cover on the beam line. The figure 18 shows the geometrical FEA model of this condition with the four feet. The final reaction results are the combination of the gravity plus transversal load (see Fig. 19). In the model Z is the horizontal axis in the direction of the beam; Y is the vertical axis and X is the transversal direction (see table 6).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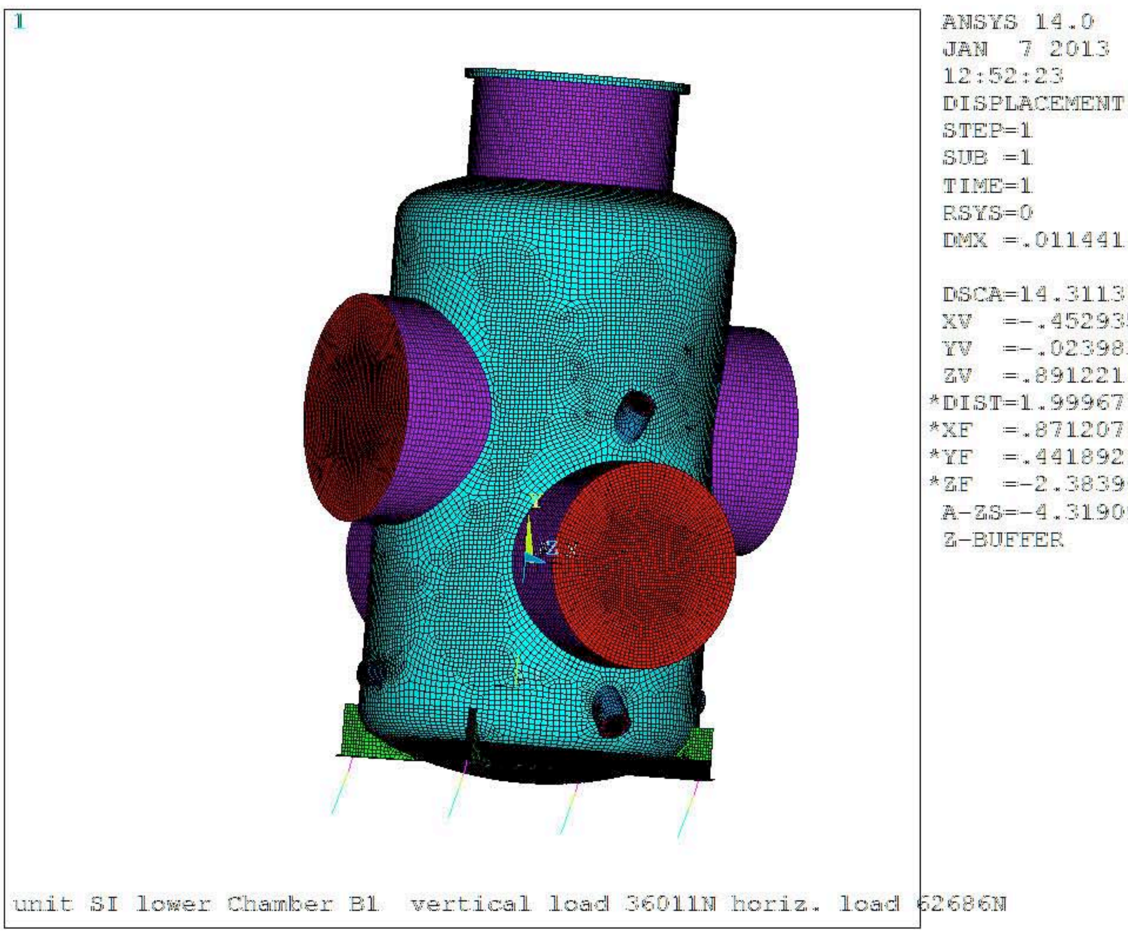
unit SI lower Chamber B1 under vacuum horizontal load

Fig 18.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*XF =.871207
*YF =.441892
*ZF =-2.3839
A-ZS=-4.3190
Z-BUFFER
unit SI lower Chamber B1 vertical load 36011N horiz. load 62686N

Fig. 19

Foot Number	FX (transversal) (Newton)	FY (vertical) (Newton)	FZ (longitudinal) (Newton)
1	-91.38	-22321	-15831
2	17.378	41816	-15503
3	282.73	-22323	-15780
4	-8.7223	41817	-15572
Total	0.7774e-5	38989	-62686

Table 6 Reaction forces with 900 mm effective diameter bellow

A vertical reaction of about 4 Ton is pulling up two feet from the floor and a shear reaction of 1.5 Ton against the total unbalance load of 6.2 Ton. The maximum displacement (at the top of the chamber, due to bolt stretching) is 11mm, as shown in figure 19. Clearly one sided vacuum cannot be applied because the stress applied on the feet would exceed the hold of the magnetic feet MHM-IT500 which has a nominal hold of only 5,000N vertically and transversal hold of 0.3 5,000N = 1,500N.

The maximum allowable transversal forces are calculated first evaluating the lift force on the far feet and then the slippage limit.

The maximum transversal force unbalance that would impose a lift force on the two feet under traction balancing the nominal hold of the magnetic feet is 28,000 N, calculated in table 7 and 8. This limit still leaves a large safety factor due to the fact that the true hold of the magnetic feet is at least 3 times larger than their nominal hold, each foot under traction would still be held down by a net force about 10,000 N.

Foot Number	FY (vertical) (Newton)	FZ (longitudinal) (Newton)
1-3	-22320 -32050 (pull) 9750 (grav)	-15800
2-4	41820 32070 (pull) 9750 (grav)	-15500
Total	39000 (grav)	-62700

Table 7 Forces breakdown (excluding magnetic hold)

Foot Number	FY (vertical) (Newton)	FZ (longitudinal) (Newton)
1-3	-4560 -16300 (pull) 9750 (grav)	-7055
2-4	24070 14320 (pull) 9750 (grav)	-6920
Total	39000 (grav)	-28000

Table 8 Forces applied on the feet with 28,000 N transversal load.

The feet's cumulative transversal slippage holding force is calculated multiplying the weight of the tank, plus the hold of the feet, by the slippage factor 0.3, i.e.
0.3 *(39,000+4*5,000)N = 17,700N. It is already a smaller limit than the maximum transversal force allowed in table 8. One have to keep in consideration, though, that in this case the transversal pull would be held mostly by the inner feet (2 and 4) which are under compression of 24,070N each, while the outer feet (1 and 3) would be compressed to ground only by the hold due to the safety factor of the feet, i.e. 10,000N each, and risk to slip.
In Table 9 the transversal holding of the separate feet is calculated for a transversal load of 15,000N considering only the nominal hold of the magnetic feet, and then considering their safety factor in the last column.

Foot Number	FY (vertical) (Newton)	FZ (longitudinal) (Newton)	Nominal longitudinal hold	Longitudinal hold including safety factor
1-3	2085 -7665 (pull) 9750 (grav)	-3780	2125	5125
2-4	17420 7670(pull) 9750 (grav)	-3720	6726	9726
Total	39000 (grav)	-15000	17700	29700

Table 9 Forces applied on the feet with 15,000 N transversal load.

Foot Number	FY (vertical) (Newton)	FZ (longitudinal) (Newton)	Nominal longitudinal hold	Longitudinal hold including safety factor
1-3	4640 -5110 (pull) 9750 (grav)	-2520	2892	5886
2-4	14863 5133(pull) 9750 (grav)	-2480	5959	8959
Total	39000 (grav)	-10000	17700	29690

Table 10 Forces applied on the feet with 10,000 N transversal load.

A maximum transversal force of 15,000 N is allowable, but the feet 1 and 3 risk to slip and an additional stiffening plate linking the four feet is advisable if this limit stress is to be approached. A transversal force of 10,000N is allowable and safe without risk of slipping, even without stiffening plate, as illustrated in table 10.

Conclusions:

7.1 Conclusions.

The results of this study are that the solution of 10mm cylinder thickness with the nozzles tube 8mm thick is pursuable, considering that the outside welds cannot be continuous due to the UHV leak-testing requirements. With the welding sizes specified on the drawings no x rays check is needed after the welding. However the process of welding qualification, visual inspection and leak test must be carried out. The reduction of the wall thickness of the lower chambers of main cylinder and nozzle respectively to 8mm and 5mm thick can be pursued at the condition that the welding of the nozzles are full penetration welds, in violation of the standard UHV vessel requirements.

The forces applied on the vacuum tanks by vacuum must be mostly balanced.

Transversal forces of up to 1 ton are acceptable on the vacuum tanks. These forces include those generated by bellows of different diameters on the two sides of each chamber and forces from not aligned ducts on the two sides.

Transversal forces of up to 1.5 tons are acceptable with an additional stiffening plate at the level of the magnetic feet of the structure.