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Deformation of the Cryostat Cold Vessel under the Load of the EM Calorimeter

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Abstract

We study the deformation of the barrel cryostat after insertion of the EM calorimeter, before the end flanges have been mounted. We find that the stresses are modest and that the ledge supporting the EM calorimeter deforms by 1-2 mm.

1 Introduction

During installation of the EM calorimeter into the cryostat the stress distribution is somewhat different from that in the final setup. We sketch the final assembled barrel cryostat in a side view in Figure 1. The cryostat consists of a warm vessel which is in contact with air, and a cold vessel which is in contact with the Liquid Argon. Both warm and cold vessel have an outer shell and an inner shell, and both have heads which join these two.

Initially only the warm and cold outer shell are assembled, and then the calorimeter, with the presampler already attached, is installed. These cryostat parts are shown in Figure 2 in a side view and in Figure 3 in an end view. In Figure 3 one can also identify the ledge on which the calorimeter rests. Neither of the heads are in place, and thus they also do not provide any stiffening of the cold or warm shell[1].

In this note we discuss only the deformation of the cold shell after the insertion of the EM calorimeter. This study, followed later on by a study of the outer warm shell, should serve as a background to the design of the installation procedures. It replaces an earlier study[2] which studied the same problem on a preliminary design of the cryostat.

2 The cold outer shell of the cryostat

The cold shell of the cryostat, shown in Figure 1 and again in Figure 2, is a 30 mm thick Aluminum cylinder of 2.14 m inner radius and 5.7 m length. It has to withstand an inner pressure of 2.5 bar when filed with Liquid Argon, and has to support in addition the EM calorimeter which weighs about 130 tons. At both of its ends there is a support ring, followed by a region designed for attaching the signal feedthroughs and the cold vessel head.

For our purpose it suffices to calculate the deformation of that part of the cold shell which is highlighted in Figure 3. We ignore the slight strengthening effect of the region where the feedthroughs are mounted, and pretend that it is not there. In this note the cold shell ends at the support ring, which is the place where the cold shell is suspended from the warm shell as is shown clearest in Figure 3. At each end of the shell a suspension rod - really an Inconel 718 rod of CQ. 5 cm diameter - is wound around the support ring, thus gently cradling the whole outer cold shell both before and after the calorimeter is installed. This support ring is essential; it prevents large distortions of the shell. We have chosen a width of 170 mm which we find to be adequate. The Inconel rod acts as a "sling,", and if it was really a frictionless and infinitely flexible sling, the analysis would be simpler. In reality one has to take into account that the "sling" is a very stiff rod which had been previously bent into approximately the right shape, and that the coefficient of friction between the support ring and the Inconel rod is of the order of unity. Luckily we find (see Section 6) that the rod is adequately flexible for our purposes, and that friction has only a minor effect on the deformations.

As mentioned above, and shown in Figure 1, the end region of the cold shell has to support the feedthroughs and accommodate the cable connections. This widening of the cold shell makes it impossible to insert it into the warm shell as a single piece; thus the cold shell must consist of two pieces, each inserted from opposite sides and tied together at the center by a flange. This flange serves another useful purpose: it stiffens the cold shell at the center[3]. For our calculations we assume this flange to consist of a single piece of material, 8 cm thick in the z-direction and 10 cm radially.

For most of our calculations we use a coordinate system where the z-axis is the beam axis, and y is "up." We define $x = y = z = 0$ as the nominal center of the barrel cryostat.

We also assume throughout that the cryostat material is 5083 Aluminum with a Young's modulus of 7.15 10^4 MPA = 1.037 10^7 p.s.i. All calculations described in this note were produced using ALGOR software (produced by Algor Inc, Pittsburgh, Pa, USA). This is a relatively simple but user friendly finite element software package. It is strictly linear, and thus cannot properly simulate friction or materials stressed beyond the yield point.

In most of our calculations the EM calorimeter itself is supposed to have a mass of 130 metric tons; it is supported by a ledge made of 1 cm thick Aluminum plates welded to the cold shell, shown most clearly in Figure 3. The calorimeter weight is transmitted to the ledge by the individual rings which we assume to be at z = 40.4, 82.4, 127.7, 178.1, 235.6 and 286.8 cm. However, the program we use allows application of forces only at "nodes" which join the individual elements forming the shell; we always chose the closest node to apply the force and we have checked that the deformations are not sensitive to such small variations in exactly where the load is applied. One should also note that these are the support points only after both halves of the EM calorimeter are in their final resting place inside the cryostat. During installation the shell supports only part of the calorimeter weight, and the support points are constantly moving. However, the final deformations are quite small; it is also likely that the EM calorimeter will rest on some kind of a rail which is made of stainless steel, a much hardier material.

We have found the modelling of the support by a sling to be quite CPU time consuming. For many calculations we have therefore simplified the problem by assuming the support rings to be fixed in the horizontal plane. We investigate separately in Section 5 the effect of this simplification.

The stresses in the shell or support rings are generally quite modest. Of course, wherever one applies "point loads" or "point constraints" in a finite element analysis program, one always produces a singularity of the stresses at that point. This is an artifact of the software, and for determining the real stresses at such points one has to be careful to spread the loads properly. However, away from such singularities the program predicts the stresses correctly. These never exceeded 50-55 MPa in all the studies described here - less than 20% of the ultimate tensile strength of the material.

In this note we first describe an alternate way of supporting the cold shell, and show why we rejected it. We then discuss several of the calculations which we have done on the proposed arrangement.

3 Cold Vessel Supported By Pads From Below

This was an earlier study; we wanted to see whether one can support the cold shell from underneath by putting large area pads between the cold shell end support ring and the warm shell.The situation is sketched in Figure 4. This would have been the ideal solution as far as the warm shell is concerned: the load is transmitted directly from the cold shell to the warm shell and to the tile calorimeter support. Thus the warm shell would have to carry only the feedthroughs and its own weight. Specifically the assumption was that the support ring of the cold shell is supported by two pads each of area 6 cm x 40 cm. The calorimeter load was assumed to be 140 metric tons, and distributed as described above.

The results were disappointing. The whole vessel flattens out into an oval shape, the top of the support ring moving by 16 mm downwards. The shell itself deforms by a similar amount. Making the support ring thicker in radial direction helps, but not enough: even a 25 cm wide ring (instead of the original 17 cm) only reduces the sag on top to 7 mm - still too much for a satisfactory design. This design was therefore rejected as not adequate for our purposes.

4 Evaluation of Baseline Design

Here we are modeling the actual baseline setup, with one important exception. Instead of using the sling support, the cold shell is "constrained" at the support ring at the position where the sling would first touch the outer rim of the support ring - that is at the four points at $z = \pm$ 291 cm, $y = 0$, and $x = \pm$ 231 cm. This produces a singular stress distribution at the support points, but realistic deformations and stresses everywhere else.

The deformations of the support ring as well as of the shelf itself have a maximum of 2-3 mm. Figure 5 identifies all points where the deformation in any direction is larger than 3.0 mm. The overall deformation gives the shell total a "pear-like" shape: below the median plane the shell bows out, above the median plane it moves inwards. The support rings and (to a lesser extent) the center flange constrain the deformation, which is largest roughly halfway between the two - at z = 116 cm. Table I lists the maximum deformations occurring at the central flange, on the shell itself and on the support ring.

The exact location of the ledge supporting the calorimeter is not yet well defined. Luckily the deformation of the ledge, as well as overall deformations of the cold vessel shell itself, are only weakly dependent on the exact location of the shell. We have compared the deformations of the vessel with a shell at exactly $y = 0$ and at y = -5.6 cm, and find differences on the 0.05-0.1 mm scale. Table II lists the deformations if the ledge is at y = -5.6 cm, which we estimate to be close to the final design value.

The ledge actually slopes somewhat; we show the deformation at a point 5 cm away from the shell, the "support line" of the calorimeter by the ledge, as well as at the point where the ledge is welded to the shell. One can also notice in Table II "indentations" where the load is applied: These indentations can also be seen in Figure 6 which shows the ledge deformations scaled up leg a factor 200. These indentations - of the order of 0.1-0.15 mm - are a good approximation to what will happen if the calorimeter load at each rib support is distributed over an area of our mesh size 8cm x 5 cm.

In order to check the reliability of our calculation, we have calculated the deformations several ways, using different simulations allowed by the software. In some of them the vessel shell consisted of "plates" – 2-dimensional objects of a defined stiffness. In some calculations the shell was made up from "bricks" – 3-dimensional objects. We did the calculation modelling the full shell, or modelling only a quarter-shell obtained by cutting at $z = 0$ and at $x = 0$ and using appropriate boundary conditions. All these calculations yielded consistent results at the 5% level.

5 Torque On Shell Due To Calorimeter Load

The cold shell supports the EM calorimeter at a ledge attached to the cold shell, and the support point is at some distance from the shell itself. This distance is not well defined, and there is a local torque on the shell which is proportional to the distance from the shell to the load point. The load position will also change the deformations, and thus the sag of the calorimeter ledge itself. We describe here a study of this effect.

A ledge which extended 10 cm from the center of the shell (thus 8.5 cm from the inner surface of the 30 mm thick shell) was subdivided so that the calorimeter load could be applied at different points. The z position (along the beam line) of each of the load point was not varied, but the load was applied at distances 1.0, 3.5, 6.0 and 8.5 cm radially inwards from the shell surface.

We found that the overall deformations, as well as the deformations of the ledge itself, are not very sensitive to the exact position of the load on the ledge. Figure 7 illustrates this point; we plot the sag of the point where the ledge is welded to the shell, as well as the sag of the edge of the ledge, in function of the distance between shell and load points. The deformations are shown at a value of z where they are maximal. One notes that the motion of the shell itself - of the point where the shell meets the ledge - hardly varies at all as the load point is moved away from the shell surface. The only effect of the extra torque is that the shell twists somewhat more, and therefore the tip of the ledge sags about 0.6 mm more when the load is at the actual tip. In Table II we list the maximum deformations of the shell; they occur at approx. $\pm 20°$ from the horizontal, at $z \approx$ 110-130 cm. One notes that below the median plane the deformation grows rapidly as the load distance to the shell increases.

We did this calculation in two ways. First we used a model where the outer edge of the support rings was constrained. We then repeated the calculation without an actual constraint, but supplying the supporting force as a constant force/unit length along the underside of the support ring. As is discussed in the next section, the first model corresponds to the situation where the friction between the Inconel support rod and the support ring is very large, while the second model represents the opposite limit where friction can be neglected. Both calculations showed consistent results.

6 Sling Support Versus Localized Support Of Support Rings

The calculations described in Sections 4 and 5 are not fully realistic in the sense that we usually simulate the support of the cold shell by fixing its position at four points - the points where the support rings cross the horizontal median plane. In reality the cold vessel is to be supported by a "sling"; this means it is nestled in a properly preshaped Inconel rod whose ultimate tensile stress is 1400 MPA - more than twice that of Stainless Steel[4].

Such a sling can produce a poorly defined situation: During loading the Inconel rod will stretch, and during cooling the support ring (made out of Aluminum) is contracting differently from the Inconel support rod. During these deformations the cold shell support ring and the Inconel sling may or may not slide past each other. If there is no sliding at all, the situation is very close to that where the cold shell is constrained at four points. On the other hand, a frictionless, infinitely flexible sling will produce a constant linear pressure (force/unit length) along the circumference where it touches the cold shell support ring[5]. We have investigated these effects, by comparing several methods constraining the motion of the support ring. For simplicity we model only the support ring, and simulate the calorimeter load by applying a force of 35 metric tons at the two points where the calorimeter support ledge meets this end flange[6]. We then show the comparison of three situations:

(a) a support ring constrained at two points along its circumference in the horizontal plane

(b) a support ring supported by an "ideal sling" which leads to a constant linear pressure along its circumference, and

(c) A support ring which is joined by "gap elements" to a real sling. Such gap elements are used in Algor so simulate the possibility of surfaces sliding across each other. They have a well defined linear resistance to compression, but no resistance to shear stresses.

For technical reasons having to do with the gap elements we had to change the coordinate system for this section: now z is "up", x is along the beam axis and y is sidewise. We summarize the results in Figures 8-10 and in Table IV . All three figures exaggerate the deformations by a factor 50.

a) The deformations in the constrained model, which we call the "infinite friction model," are shown in Figure 8, together with the undeformed shape of the support ring. One notices the typical pear-shaped deformation; comparison with the next two figures shows that this constraint produces a maximal deformation. This situation models the limit in which the very large friction between the sling and the support ring "freezes" the two together at $z = 0$.

b) In Figure 9 we show the "ideal sling;" if there is no friction at all, the sling can act only by a normal force which has to be constant over the bottom half of the support ring. The force per unit length is equal to

$$P = T/R \tag{1}$$

where T is the (constant) tension the sling and R is the outer radius of the support ring. In a Finite Element analysis, where forces have to applied at nodes, there is some ambiguity as to where one stops applying the linear force; we have applied the full force on all nodes below the horizontal median plane, and half the full force to the node at the horizontal line.

The deformations are smaller here than if the ring is completely constrained at two points, and in particular the tilt of the ledge is smaller. We note that this model simulates a perfectly flexible sling which is suspended infinitely high above the shell; in addition there is no friction between the sling and the support ring.

c) Figure 10 shows the deformation when we use an actual sling, and tie it to the support ring by "gap elements" which have no shear strength. The length of the sling was chosen to be realistic - we assume that it comes from the support points on the warm shell structure. The sling has also a finite flexibility governed by Young's modulus of Inconel. Thus this model depicts a frictionless sling of finite length and finite flexibility. Comparing Figures 9 and 10, one sees very similar deformations, which are somewhat smaller than in the case of infinite friction.

There is no obviously unique way to compare the deformations in the three cases. In case (a) two points of the ring are fixed; in case (b) we actually fixed the top of the ring and checked that the forces due to that constraint are negligible[7]. In case (c) the sling itself sags by $\approx$ 11 mm and so all points of the vessel would move downwards even if the support ring was perfectly rigid. We have therefore somewhat arbitrarily defined the points on the support ring which are "fixed:" in the y-direction (horizontally) we assume that the center of the ring has not moved - this is possible since the center is a symmetry point in the horizontal direction. In the vertical z-direction we assume that the outer edge of the support ring has not moved at the horizontal median plane $z = 0$; this point is identified as X in Figures 8-10 and is the point which indeed is constrained in case (a), as well as in all the calculations in Section 4 and 5.

We then list in Table V the relative shifts of the five points A,B,C,D and E which are also indicated in Figures 8-10, namely top and bottom, two points slightly above and below the horizontal plane where the deformation is maximal, and finally the tip of the ledge.

The main difference between the infinite friction model A and the other two models is that this model maximizes the torque between the load and the shell support. This shows up in a larger tilt of the ledge and a greater deformation below the horizontal median plane.

The Inconel sling is not perfectly flexible, and thus has to be bent into approximately the correct shape. We investigated the question whether imperfections in this preshaping will affect the deformations in the shell. We find that the Inconel rod is sufficiently flexible to deform back into touching the support ring everywhere. We tried even an extreme kink in the rod, which lifts it 1 cm off the ring and prevents it from touching the ring over a distance of 35 cm. Even such a large kink leads to negligible changes in the ring deformation. Under the load the rod largely straightens out the kink and just barely misses touching the support ring everywhere.

7 Conclusions

This study shows that while the stresses in the cold shell due to the EM calorimeter are quite reasonable, the deformations of the supporting ledge are of the order of 1-2 mm and thus one should compensate for them when designing the details of the suport rail for the EM Calorimeter. Attention also will have to be given to the overall sag of the cold shell after the calorimeter is loaded.

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References

[1] One should be aware if the heads are later welded on, they are welded to an already distorted shell. Thus any deformations which were there before are frozen in.

[2] P. Vedrine and P. Giovannoni, Saclay note E-572-N-900. (1994)

[3] Of course, it makes the vacuum problem more difficult. One will have to seal weld this area after all bolts have been tightened. The bolt elasticity has to keep them tight during and after the cooldown.

[4] Young's modulus of Inconel 718 is 2.0 10^5 MPA, and its design strength by US ASME code (Tensile/4) is 358 MPA. Its shrinkage between 293-77k is 0.22%, while that of Aluminum is 0.38%.

[5] There is the possibility that the sliding might be asymmetric, and therefore the vessel could shift and/or rotate during loading or during cooldown. Therefore the "sling" has to be clamped to the support ring at the bottom.

[6] The whole calorimeter load is now concentrated at two points of the support ring, and the stiffening effect of the shell itself is completely ignored. Thus the absolute values of the shifts in Section V should not be compared to those in e.g. Section 3. Only the relative differences

between the various setups discussed in this section have an objective meaning.

[7] Rounding errors in the calculation make it impossible to obtain a situation where all the external forces cancel exactly; one always needs some fixed point which will pick up these residual small forces. Failure to do so makes the structure drift away during the calculation. As long as these constraint forces are small, it does not matter whether the artificial constraint is absolute or elastic.

Table I. We list the points of maximum deformation on the central flange, the shell itself and the support ring. The coordinate center origin is at the center of the cold shell; y is "up", z is along the beam line and a positive CX implies an "outwards" deformation.

In the shell itself:

	DX = 2.87 mm	DY = -1.59 mm	DZ = 0 mm
at	X = 199.77 cm	Y = -76.7 cm	Z = 116 cm

On central flange:

	DX = 2.72 mm	DY = -1.65 mm	DZ = 0.0 mm
at	X = 181.76 cm	Y = -92.6 cm	Z = 4 cm

On support ring:

	DX = 1.59 mm	DY = -0.802 mm	DZ = 0.39 mm
at	X = 208.29 cm	Y = -92.74 cm	Z = 291 cm