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Design and Analysis of the 50mm Collider Dipole Vacuum Vessel*

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ABSTRACT

In this study, a finite element model is used to predict the vacuum vessel's response to loads resulting from its own self-weight and shipping and handling. Structural members of the cold mass support system must behave elastically during assembly and transport of the magnets so that alignment with respect to an external reference is reproducible. This paper includes deflection and stress analyses of the vacuum vessel for the SSC 50 mm collider dipole cryostats. In addition, material selection and stress relieving techniques are discussed. Several methods of local structural reinforcement were analyzed prior to the selection of the current design. The results from these analyses are given for comparison.

INTRODUCTION

The functions of the vacuum vessel are to maintain a vacuum for low heat leak to the cold mass and to structurally support all cryostat components. The first function only requires that the vessel remain leaktight. The second function is quantified by shipping and handling load specifications and the beam tube centerline positional tolerances which are given in Table 1. Presently, one 50 mm prototype vacuum vessel has been manufactured by industry.

The design of the vacuum vessel to be used for the 50 mm collider dipole is a direct descendent of that used in the 40 mm magnets (figure 1). Along the length of the vacuum vessel, thinner pipe sections are joined with thicker connecting rings. Connecting ring positions are coincident with cold mass reentrant support post locations. Rotational adjustments about the three axes are achievable at the joint between the pipe sections and connecting ring. The most notable difference between the 40 mm and 50 mm vacuum vessel is the location of the external supports. In the 40 mm magnets, external supports were coincident with support post locations 2 and 4. The 40 mm vessel was built with a camber upward at the ends. Insertion of the cold mass into the vessel causes the camber to be leveled. Survey results of port ring locations in the 40 mm vacuum vessels before and after assembly exist and are well documented.[1]

The 50 mm vacuum vessel is designed with all port rings level and parallel to each other within 0.5 mm. External supports are located at 4.154 m from the center. The vessel is 15.024 m long. At assembly the cold mass is inserted and deflections of port rings 1, 3, and 5 are equal and consequently minimized. The vacuum vessel external support

Table 1. Prime Item Development Specifications

Beamtube Centerline Positional Tolerance: ±1.0 mm

Shipping and Handling Loads:

Vertical	2.0 g
Axial	1.5 g
Lateral	1.0 g

Factors of Safety:

Load Type	Yield F.S.	Ultimate F.S.
Shipping	1.3	1.5
Long term static	1.6	2.0
Handling	2.0	3.0

Allowable Stresses

Stress Component	Magnet Structure
Allowable membrane stresses, s_m	lesser of 33% UTS or 67% YS
Allowable membrane + bending stresses	150% s_m

locations are reinforced with 12.7 mm thick plates. After insertion of the cold mass into the vessel, the height of the beam tube is corrected by shimming at each port ring. The only built in deflection, then, is due to self-weight of the cold mass.

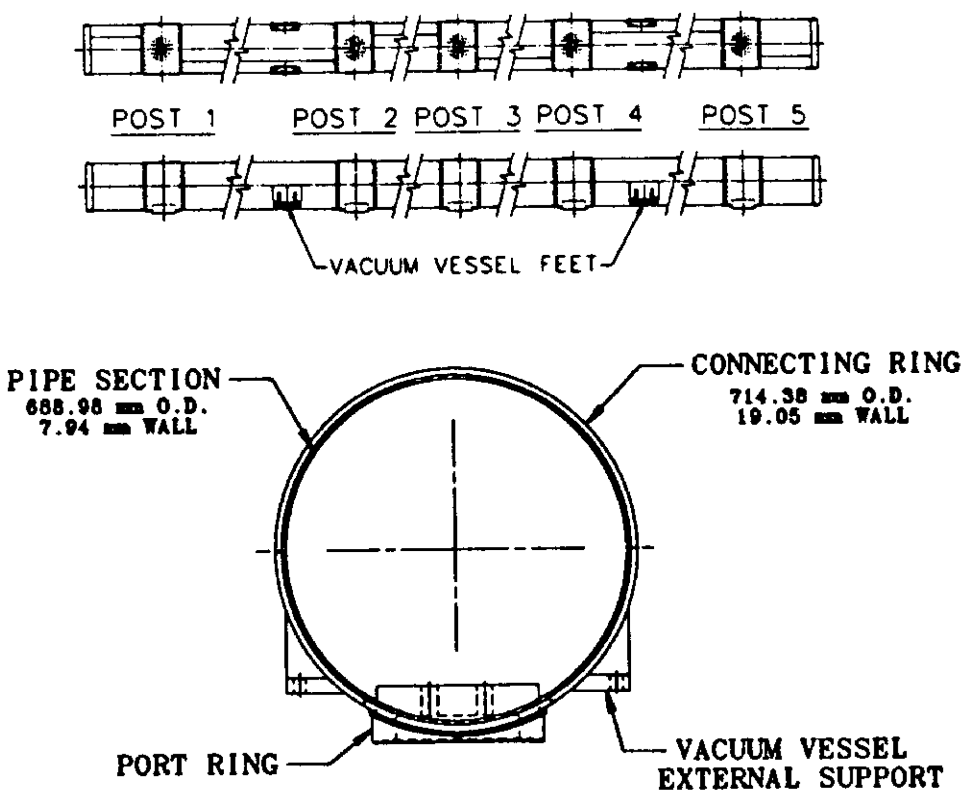


Figure 1. SSC 50 mm Collider Dipole Vacuum Vessel

The optimum spacing between five support posts for the 50 mm cold mass is 3.175 m for a temperature of 4 K. Deflections at the cold mass ends and midpoint between supports, as calculated by a beam model, are 1.5 mm, 0.38 mm, and 0.14 mm for 3, 4 and 5 cold mass supports respectively. The stiffness of the cold mass in this model is due solely to the moment of inertia of the cold mass skin. Experience in the 40 mm program showed that actual cold mass deflections were 20 percent less than beam model predictions.[2] The additional stiffness is due to the yoke laminations. Thus, the result of 0.14 mm cold mass deflection is conservative and within the allowable beam tube centerline positional tolerance of ±1.0 mm.[3]

MODEL

Deflection and stress analysis of the vacuum vessel under static weight and shipping and handling loads is performed using ANSYS finite element software. Model parameters are listed in Table 2. Deflection of the vacuum vessel can be approximated using beam elements. In the beam model (figure 2) the cold mass is coupled to the top of support post 3 (anchor position) in all translational degrees of freedom. All other cold mass/support post connections are free to slide longitudinally and rotate about the three axes. The vacuum vessel is fixed at the external supports in all translational degrees of freedom and in rotation about a line parallel to its axis. Vertical and lateral loads are imposed by applying acceleration. Axial loads are distributed from the center post through the tie bars as described in reference 4. All load cases include the weight of the vacuum vessel and cold mass.

Stresses in the vacuum vessel are more accurately determined with shell elements. The models are shown in figure 3. Stress concentrations occur near the port rings and at the external supports. Mesh size is reduced to approximately 12 x 12 mm in these areas. It is important to note the orientation of the model with respect to gravity in these figures. Results are plotted so that the regions of highest stress are visible. Gravity is always directed along the negative z axis. Because the vacuum vessel supports shear, both bending and membrane capabilities are included in the stiffness of the shell elements. Forces caused by the weight and acceleration of the 11430 kg cold mass are applied via point loads on support posts. Support posts are modeled by a rigid cylindrical framework of beam elements. Gravitational loads on the vessel due to self-weight and shipping and handling are imposed by fixing the vessel at the external support locations and applying an acceleration in one of the specified directions.

In order to reduce the shell element model size, symmetric boundary conditions and correspondingly scaled loads are used wherever possible (figure 3). A model one quarter the size of the actual vacuum vessel is sufficient to characterize stress contours under vertical loads. Half size models are necessary to produce all stress contours under axial and lateral loads. The axially loaded model has a vertical plane of symmetry located along the length through the beam line. The laterally loaded model has a vertical plane of symmetry at the center of the vessel.

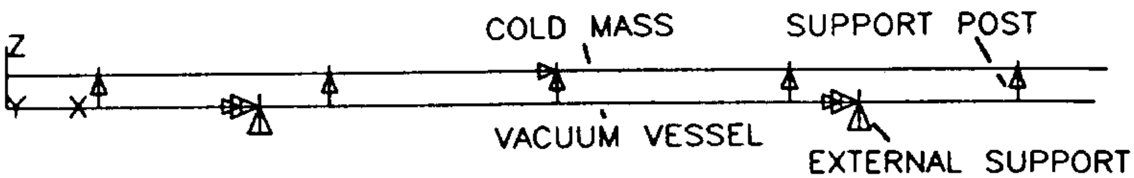


Figure 2. Beam Model

Table 2. Model Parameters

	50 mm
Cold Mass	
No. support posts	5
Support post spacing	3.180 m
Weight (approximate)	104,200 N
Length	15.253 m
Shell	
Outer diameter	340.00 mm
Shell thickness	4.95 mm
Cross sectional area	.0886 m^2
Moment of inertia	.0000703 m^4
Vacuum Vessel	
No. external supports	2
External support spacing	8.308 m
Weight (approximate)	19,445 N
Length	15.024 m
Material	A516
Yield stress	262 MPa
Ultimate stress	551 MPa
Pipe section	
Outer diameter	685.8 mm
Wall thickness	7.94 mm
Cross sectional area	.0169 m^2
Moment of inertia	.000971 m^4
Connecting ring	
Outer diameter	714.38 mm
Wall thickness	22.22 mm
Cross sectional area	.0483 m^2
Moment of inertia	.00290 m^4

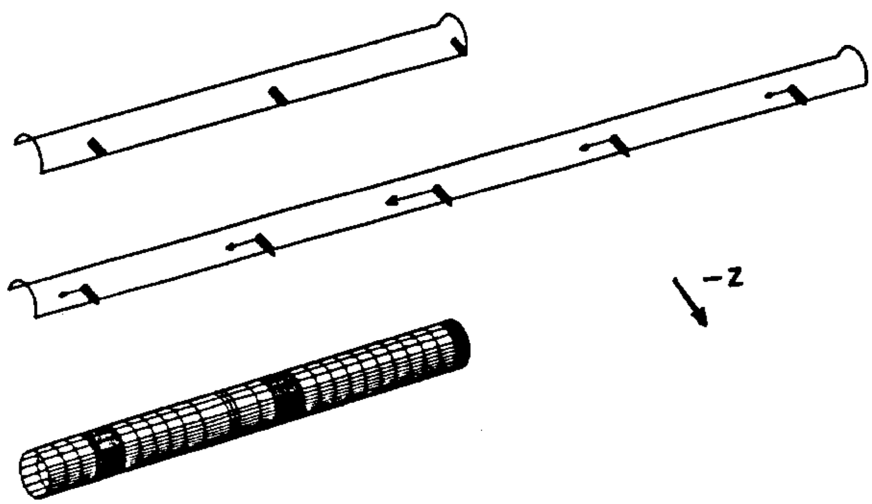


Figure 3a. Shell model - vertically loaded
Figure 3b. Shell model - axially loaded
Figure 3c. Shell model - laterally loaded

FINITE ELEMENT RESULTS

Results from the shell model indicate areas of stress concentration and local cross sectional deformation. These results can be verified by comparison with free body diagrams and simpler finite element models. Moment diagrams as calculated with the beam model are shown in figure 4 for all load cases. Deflected shapes as calculated with the beam model are given in figures 5 and Table 3.

Vertical loading is the simplest case to verify. Maximum moments as determined from hand calculations and from the finite element beam model are comparable within 1.8 percent. The weight of the vessel was neglected in hand calculations. Deflection of the beam and shell models are similar. The deflection of the shell model is derived by making a slice along a vertical plane down the length of the vacuum vessel and then averaging the deflections of the upper and lower line of intersection. The shell cross section deforms from circular by as much as 0.5 mm. Axially loaded models also show similar deflected shapes. The deflections are plotted together in figures 5b and 6. The unsymmetric curve arises because the moment due to the cold mass weight and cold mass axial load oppose each other at one end and work in the same direction at the other end. For lateral loading, the deflections are determined by intersecting a horizontal plane along the axis of the vessel. The horizontal contour is similar to that of the vertically cut vessel under self-weight. For all load cases there is agreement between deflection results of the beam model and the more detailed shell model.

If stresses exceed the yield strength of the material, magnet alignment will be compromised. The most severe loads imposed on the vacuum vessel are due to shipping and handling. For all load cases, the stresses are below the specified allowable stress at continuous and welded sections. Maximum stresses from the finite element shell model are shown in Table 4. The yielding criteria used for evaluation of stress in the shell finite element model is defined by Von Mises' equivalent stress.

$$\sigma_e = (\sqrt{2})^{-1}\ [(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6((\tau xy)^2 + (\tau yz)^2 + (\tau zx)^2)]^{1/2}$$

Stress contours are in the regions of maximum stress shown in figures 7a, 7b, and 7c.

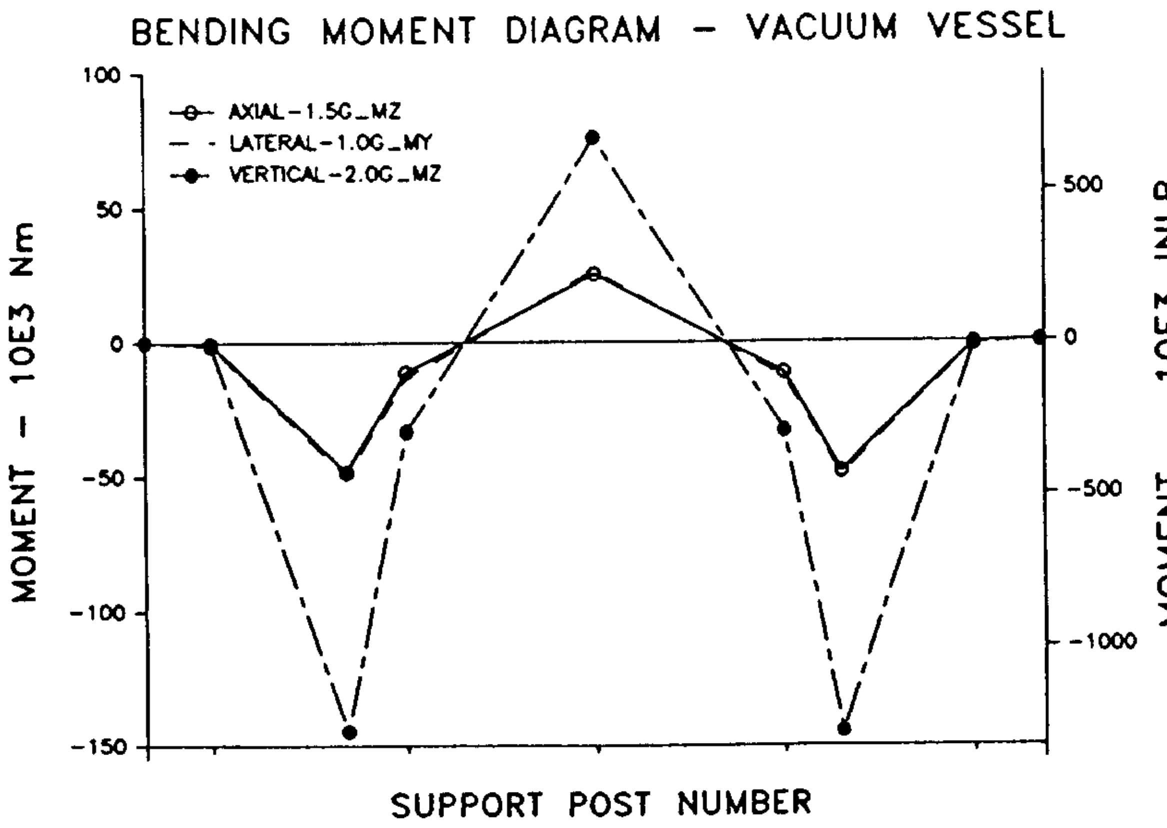


Figure 4. Vacuum Vessel Bending Moment Diagrams

Table 3. Vertical Deflections of the Vacuum Vessel (mm) - Shell Model

Load type	Direction	Support Post Number 1	2	3	4	5
Self-weight	DZ	-.76	-.23	-.74	-.23	-.76
2.0 g vertical	DZ	-2.209	-.69	-2.21	-.69	-2.29
1.5 g axial	DZ	-.64	-.28	-.74	-.23	-.94
1.0 g lateral	DY	-.79	-.15	-1.01	-.15	-.79

Under 2.0G vertical loads, a maximum stress of 107.0 MPa (15.5 ksi) occurs coincident with the maximum moment at the external supports. The stress has been distributed throughout the 12.7 mm reinforcing plates. The anchor position port ring/connecting ring joint, under shear and bending, reaches a stress of 105.0 MPa (15.0 ksi).

In the 1.5G axially loaded case, a maximum stress of 50.4 MPa (7.31ksi) is seen at the anchor position port ring. The post transmits a moment about the z-axis placing one side of the adjacent connecting ring in tension and the other side in compression. Equivalent stress evaluates the highest combined stress state, thus values of σ_e are equal on either side of the post. The second largest value of stress of 35.9 MPa (5.20 ksi) is seen at the external supports.

Under 1.0G lateral loads, the bending moment placed on the vessel about the y-axis is nearly identical to the moment about the z-axis under self-weight. The maximum stress of 126.9 MPa (18.4 ksi), however, is much higher than that in the vertical case. The local difference is due to the additional moment created as the lateral force of the cold mass acts at a height of .368 m above the port ring. This increases the stress in material adjacent to the connecting ring. The beam model does not lend itself to evaluating this effect of the combined loads.

BUCKLING CRITERIA

Buckling becomes a concern during magnet handling because the vessel will be lifted at locations other than the external supports. The further the support is from the external supports ("D" on figure 8), the larger the bending moment on the vacuum vessel shell. This is true for loads in all directions. The proposed length "D" for tunnel operations is 0.91 m (away from the center) of each external support. Figures 8 presents the results of the buckling calculations for the 50 mm vacuum vessel. For all load cases, except 2.0G vertical for D greater than 0.61 m, the maximum vacuum vessel moment is less than the moment at which the correlation predicts buckling will occur for the thinner pipe section (worst case).

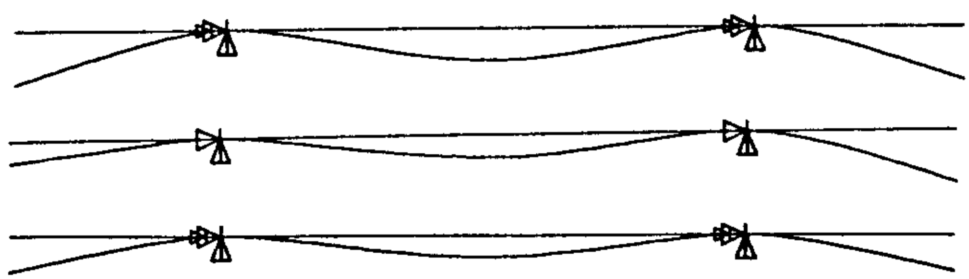


Figure 5. Vacuum Vessel Deflections: Vertical, Axial, Lateral Loads

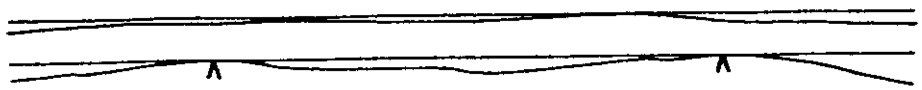


Figure 6. Vacuum Vessel Deflections - Shell Model - Axial Load

No closed form solution was found to predict buckling of a long cylindrical vessel under combined transverse shear and bending loads. However, experimental work has provided a correlation for the load at which buckling occurs.[5] The correlation regards the transition from buckling failure by shear to buckling failure by bending for thin-walled tubes. The failure criteria is satisfied when, for some angle θ_m in the cross section:

$$\frac{M \sin \theta_m}{\pi r^2 t S_b} + \frac{V \cos \theta_m}{\pi r t S_v} = 1$$

where V = applied shear
M = applied moment
r = outer radius
t = wall thickness
S_b = critical stress for bending load
S_v = critical stress for shear load

This equation is most applicable to (M/Vr) ratios less than 7. Values of (M/Vr) are determined from finite element beam models and are given for all three load cases in Table 5. The equation is solved by determining the angle at which the sum is maximum. This angle is a function of the ratios (M/Vr) and (S_b/S_v). Determination of S_b and S_v are described in reference 5. The allowable stress is expressed as a percentage of critical stress for a cylinder under pure bending. Critical stress S_b as determined experimentally is available for certain ratios of r/t and L/r.[6] Critical moments for the 50 mm vacuum vessel geometries were extrapolated from this data.

Table 4. Shell Model Stresses

	Equivalent Stress MPa / (ksi)
Static Load	
1.0 G - vertical	36.75 / (5.33)
Shipping and Handling Loads	
2.0 G - vertical	107.00 / (15.50)
1.5 G - axial	50.41 / (7.31)
1.0 G - lateral	126.86 / (18.40)

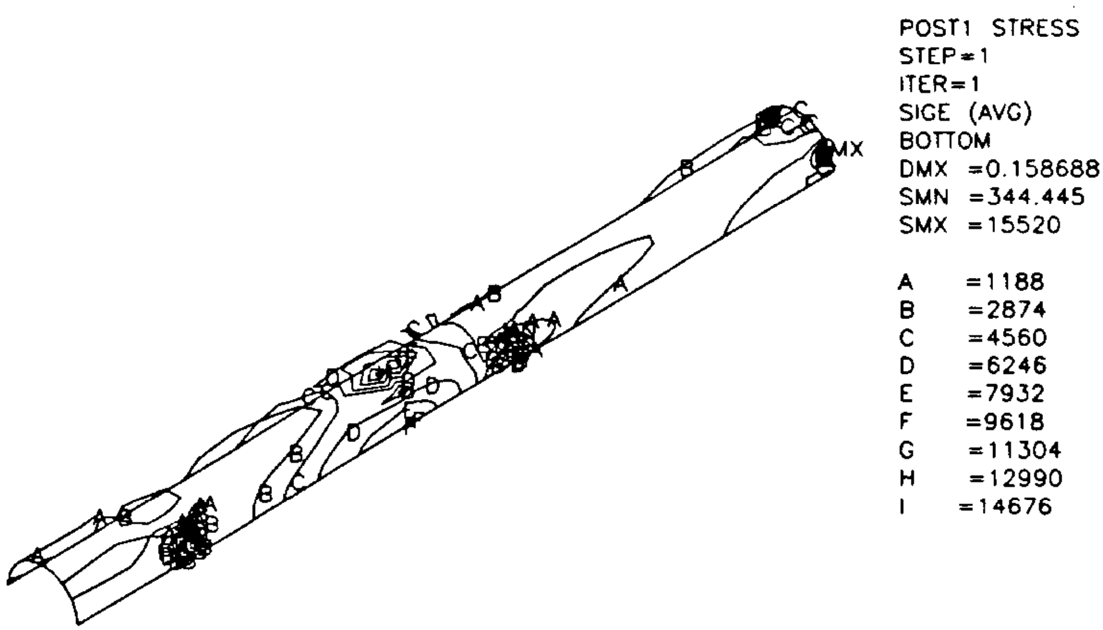


Figure 7a. Maximum Vacuum Vessel Stress - 1.0 G Vertical - External Support

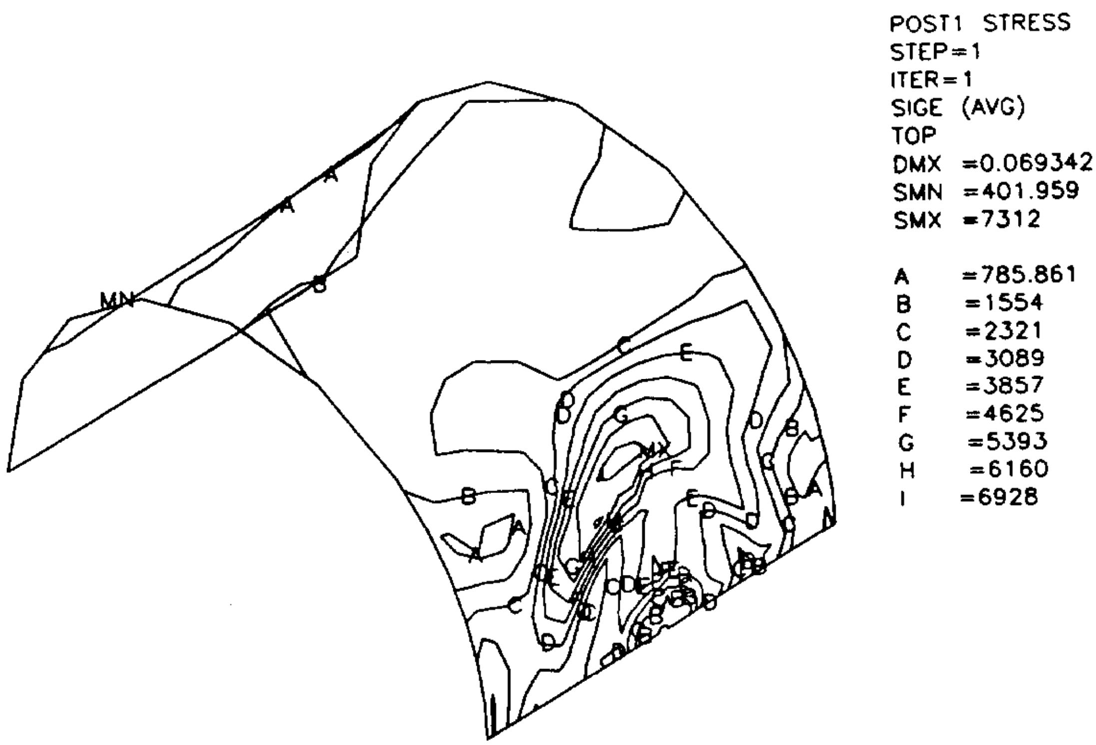


Figure 7b. Maximum Vacuum Vessel Stress - 1.5 G Axial - Port Ring #3

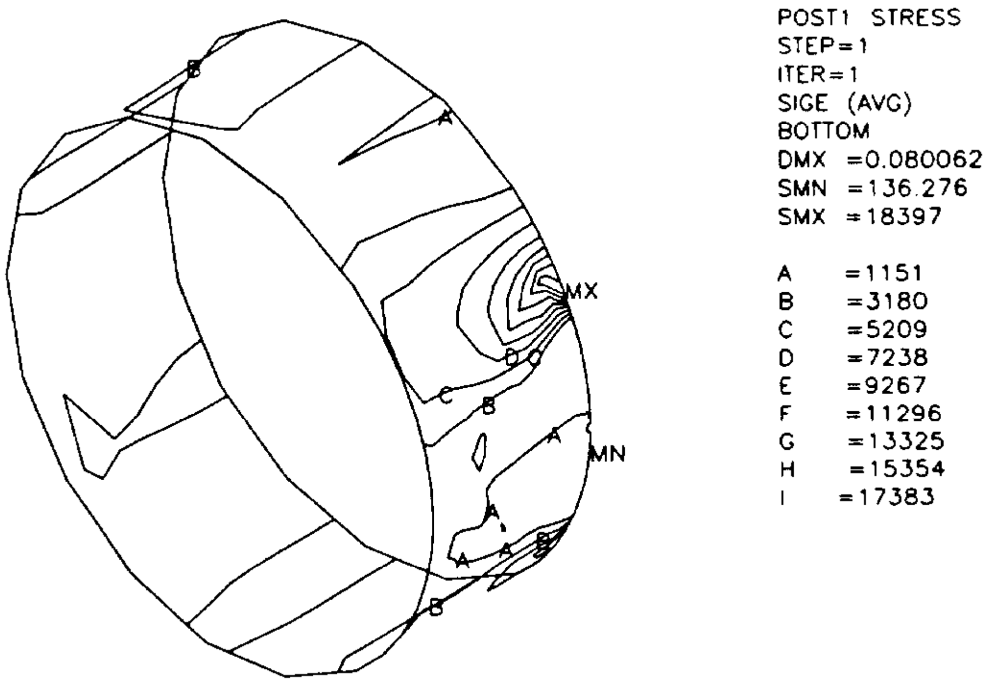
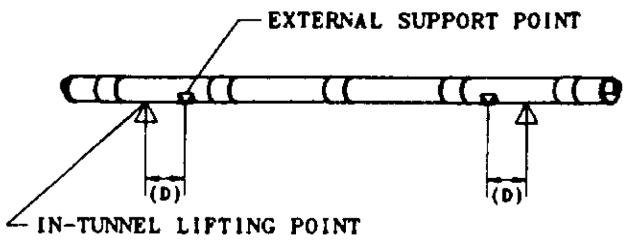
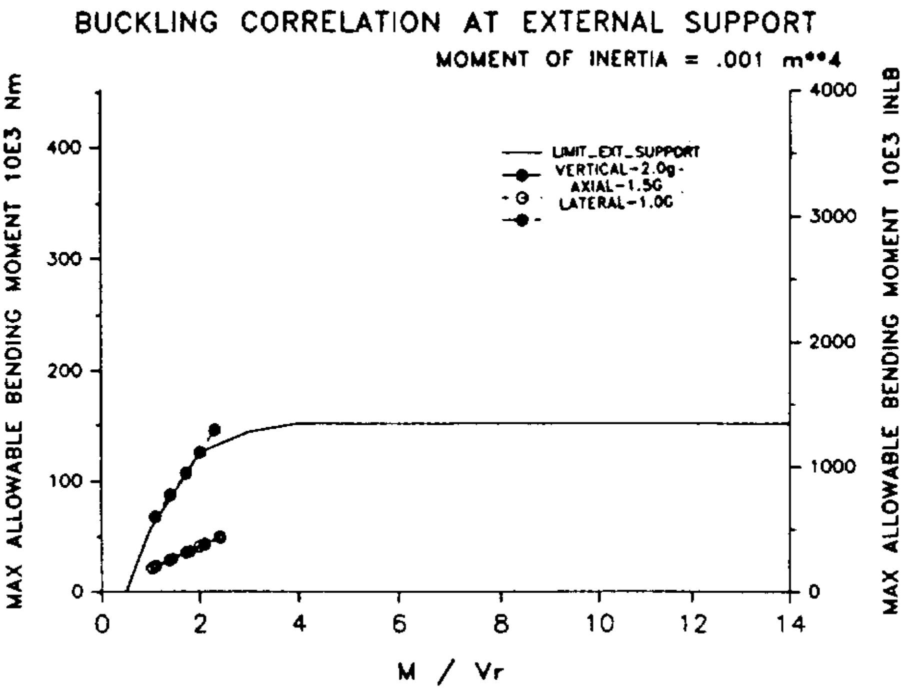


Figure 7c. Maximum Vacuum Vessel Stress - 1.0 G Lateral - Port Ring #3

Table 5. (M / Vr) Ratios Determined by Beam Model

Load Type	Location	Distance "D" (m) 0.00	0.30	0.61	0.91	1.22
Vertical 2.0g	External Support	2.3	2.0	1.7	1.4	1.1
	Post 3	3.4	5.7	8.0	10.3	12.7
Axial 1.5g	External Support	2.4	2.0	1.7	1.4	1.0
	Post 3	2.5	4.8	7.1	9.4	11.8
Lateral 1.0g	External Support	2.4	2.1	1.8	1.4	1.1
	Post 3	3.5	6.0	8.5	10.2	12.5





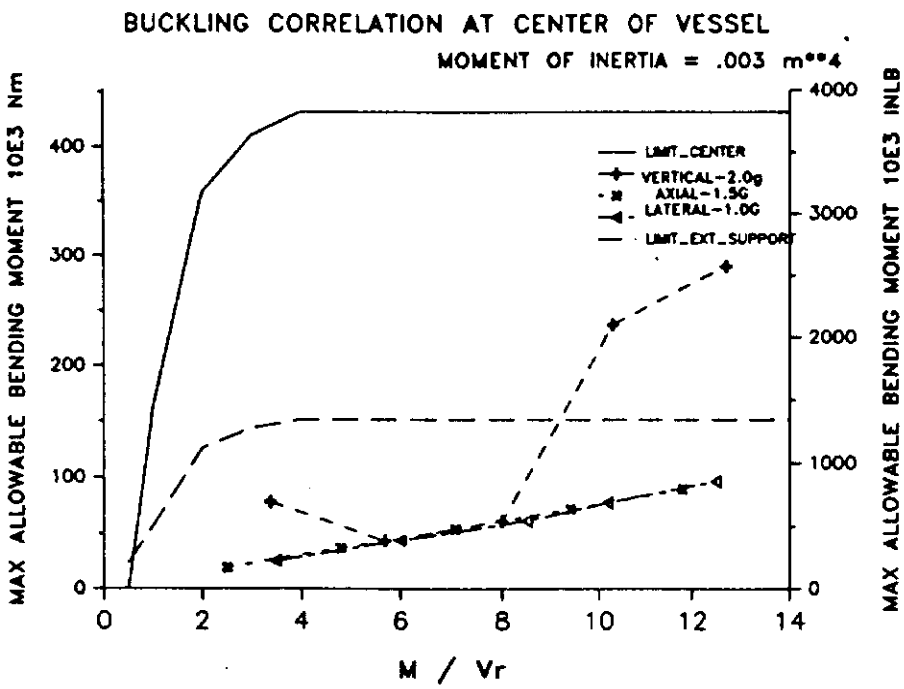


Figure 8. Magnet Handling - Buckling Correlation

FABRICATION EXPERIENCE

To date, one 50 mm vacuum vessel has been fabricated. Although the amount of experience with 50 mm vessel is limited, fabricators of the vacuum vessel conclude that assembly accuracy could be improved most if pipe sections and connecting rings were perfectly round. The current design calls for rolled and welded pipe sections having a maximum out-of-round condition of ±3.18 mm. Tighter tolerances are specified on the ends of the pipe which are inserted into connecting rings. During assembly, rotational adjustments were necessary at the connecting ring/pipe section joint. The pipe sections were secured by shimming the circumference. The disadvantage of this practice is that some regions of the joint receive thicker shims than others resulting in increased heat transfer during welding. Uneven weld shrinkage then skews the axis of the vessel.

Other methods of pipe fabrication include spiral welding, horizontal centrifugal casting, and extrusion. Of these three processes, centrifugal casting and extrusion were cost prohibitive for the quantities involved. Spirally welded pipe was found to have a larger out-of-roundness tolerance than rolled and welded pipe. Centrifugal casting yields an improved roundness tolerance of 1.27 mm for a 22.2 mm wall section.

A second problem occurs when heat from welding distorts the port ring as it is secured to the connecting ring. The upper surface of the port ring has a flatness tolerance of 0.08 mm for accurate positioning of the support post. Surveys of the first vessel showed typical flatness of 0.25 mm. It should be noted that the 50 mm prototypes are fabricated with 1020 carbon steel. The material specified for the 50 mm design is A516 steel. The latter is a pressure vessel quality plate. Its notch toughness is higher than that of 1020.

STRESS RELIEVING

Long term stability of the vacuum vessel is important for maintaining alignment of the magnets in the tunnel. The design presented here contains welded joints and any such joint under constant load will relieve residual welding stresses over time. The same holds true for the base metal, but to a lesser extent. Whether the collider dipole vacuum vessel utilized in the SSC is fabricated similarly to the design presented here or not, one may assume that welding will be used to join critically aligned components.

Several methods of stress relieving were investigated for the 50 mm prototypes. The cost of building, fixturing, and fine tuning the process is impractical for prototype quantities; however, information gathered to date is presented here. The first method investigated was stress relieving the entire vessel in a furnace. The fixturing will allow free expansion longitudinally and radially . A cost effective furnace temperature rate must be determined so that thermal stresses are not more detrimental than the original welding stresses. A second method for stress relieving is through vibration. Several commercial methods exist. The method viewed as most promising at this point is vibration at sub-resonant frequencies (approximately 80 percent of the resonant frequency). This is performed during the welding operation. The theory is that as the weld cools, grain growth is interrupted by vibration thus resulting in smaller grain size.[9] As the metal is loaded, dislocations occur at higher stress. Presently, verification of the merits of vibration stress relieving is under study by the Department of Energy. There are several advantages to this method in that it can be performed as the vessel is manufactured. It is also low in cost compared to furnace stress relieving. Difficulties may exist in accurately determining the frequency set point of the force input for a weldment which is under construction. The circumferential welds in the 50 mm vacuum vessel are made on a three step schedule. Secondly, the effectiveness of the process is highly dependent on the amount of damping present in the part itself and in the welding fixture. Measurements are necessary to determine amplitude as a function of circumferential distance from the force input.

OTHER DESIGNS

In pursuing the use of centrifugally casted pipe, finite element analysis was performed on a vacuum vessel model with constant 12.7 mm wall thickness. (A 22.2 mm wall suggested by several centrifugal casters as an approximate minimum wall thickness after machining.) Lateral shipping and handling loads were considered because this is the case of maximum stress in the current design. The maximum stress for this case was 237 MPa (47.4 ksi). One can predict that under vertical loads the thicker wall at the external support will produce stresses lower than those of the current design. One inherent difficulty exists in a vacuum vessel fabricated as one continuous pipe. Any welding which is done on the one side of the vessel will tend to bow the vessel unless it is balanced by equal amounts of welding on the other side.

CONCLUSIONS

The SSC 50 mm Collider Dipole vacuum vessel meets strength and vacuum requirements as designed. Maximum stresses as calculated with finite element analysis are below the yield stress with a factor of safety greater than 2.0 as specified in the PIDS.[3] The deflection of the cold mass under self-weight is 0.14 mm when the cryostat is assembled. This deflection along with tolerances of the beam tube position within the cold mass are the only sources for deviation of the beam tube centerline with respect to the vacuum vessel external supports.

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