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TOROIDAL MAGNETIC FIELD SYSTEM FOR A 2-MA REVERSED-FIELD PINCH EXPERIMENT*

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ABSTRACT

The engineering design of the toroidal magnetic field (TF) system for a 2-MA Reversed-Field Pinch experiment (ZT-H) is described. ZT-H is designed with major radius 2.15 meters, minor radius 0.40 meters, and a peak toroidal magnetic field of 0.85 Tesla. The requirement for highly uniform fields, with spatial ripple $\leq$ 0.2% leads to a design with 72 equally spaced circular TF coils, located at minor radius 0.6 meters, carrying a maximum current of 9.0 MA. The coils are driven by a 12-MJ capacitor bank which is allowed to ring in order to aid the reversal of magnetic field.

The mechanical design emphasizes easy access to the liner and avoids the interlocking of TF and PF coil structures, thus allowing easy removal of the coil assemblies. Two alternative design concepts are considered for the TF coils. The first concept utilizes a 1-inch square prefabricated cable wound onto a rotating spool which is structurally supported by wedges and bolts fastened to the machine frame. The other concept utilizes more conventional preformed 180° segments of 2-inch conductor connected through bolted current joints.

A stress analysis is presented, based upon calculated hoop tension, centering force, and overturning moment, treating these as a combination of static loads and considering that the periodic nature of the loading causes little amplification. The load transfer of forces and moments is considered as a stress distribution resisted by the coils, support structures, wedges, and the structural shell.

INTRODUCTION

ZT-H is a proposed 2-MA reversed-field pinch (RFP) designed to make significant extrapolation of the present RFP data base toward high energy density reactor systems. An air core transformer will be used to induce a plasma current of 2-MA and a toroidal solenoid will produce a peak toroidal magnetic field of 0.85 Tesla. The physical dimensions of the plasma will be major radius 2.15 meters and minor radius $\leq$ 0.40 meters. (See Fig. 1)

Since an RFP operates at a lower value of toroidal magnetic field than a comparable Tokamak for the same plasma current, the design of an RFP typically is not limited by mechanical stresses on the toroidal field coils. The most important factor in the design of the magnetic system then becomes the allowable magnetic field errors in the plasma region. For the toroidal field system, this amounts to minimizing the radial component of the spatial field ripple and to controlling mechanical tolerances.

This paper discusses the toroidal magnetic field system for ZT-H. First the criteria for magnetic field errors are discussed, as well as the plasma physics origins of the criteria. Second, the consequences of the criteria with regard to the number of coils and radial positioning are examined, and several coil configurations are compared by calculating the magnetic field distributions numerically. Third, the magnetic

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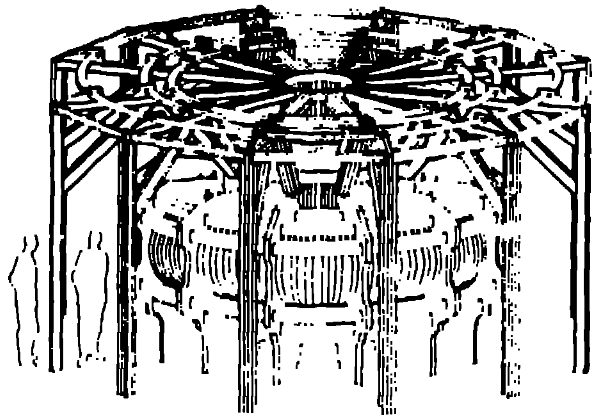

Figure 1
OVERALL VIEW OF ZT-H MACHINE STRUCTURE

field forces acting on the toroidal field coils and the supporting torus are analyzed. Next, a novel coil design is proposed whereby prefabricated coils are wound in place onto the machine by utilizing a "staging" spool next to the machine, thus eliminating the need for multiple conductor current joints in the coils. Finally, the results of a finite element stress analysis for the main support frame, bulkheads, and the supporting aluminum shell are presented.

Magnetic Field Error Criterion

A simplified analytic analysis, supported by more detailed numerical field line tracing, indicates that in order to avoid magnetic islands above a given size, the radial component of axisymmetric field errors, normalized to the poloidal component of the magnetic field, should satisfy the inequality[1]

$$\frac{\delta b_{rn}}{B_\theta} \leq \frac{naq'(\Delta r/a)^2}{16} \qquad (1)$$

where δb_{rn} is the radial component of the n^{th} error field mode

B_θ is the poloidal component of the magnetic field,
n is the mode number,
a is the minor radius of the plasma,
q' is the derivative dq/dr of the q profile, and
$\Delta r/a$ is the maximum allowed radial width of the magnetic islands.

The inequality must be satisfied at the critical reversal surface, where the toroidal magnetic field reverses. However, for conservative design purposes we require that it be satisfied at the surface of the plasma.

Using numbers appropriate for ZT-H, a=.4, q'=.6, and requiring $\Delta r/a \leq 0.15$ (i.e. that the island width be less than the distance between the reversal layer and the plasma surface) the following criterion results.

$$\frac{\delta b_{rn}}{B_\theta} \approx n(.0004) \qquad (2)$$

The maximum field errors for various pertinent mode numbers are summarized in Table 1.

TABLE 1

MAXIMUM PERMISSABLE FIELD ERROR

Mode Number n	Maximum Error (b_r/B_θ) %
1	.04
12	.45
24	.90
36	1.36
48	1.81
72	2.70

The most important mode numbers in Table 1 are n = 1 and n = 12; n = 1 because the maximum permissable error is so small, and n = 12 because the main support structure for the torus consists of twelve vertical bulkheads spaced equally toroidally around the machine. (See Fig. 1) It is difficult to meet the n = 1 mode criterion, simply because of difficulties in mechanical tolerances. The n = 12 mode, while not so difficult, will appear when the machine is fired and the torus tries to flex in an n = 12 mode. In view of the difficulty in analyzing the static and dynamic problem exactly, 0.3 field error was set as a preliminary target. The following design criteria were established.

1) The radial component of the spatial ripple of the magnetic field shall be less than 30 gauss at the plasma surface. This corresponds to 0.3% of the poloidal magnetic field (1.0 Tesla) produced by a 2.0 MA plasma current at a minor radius of 60 cm.

2) The radial component of the error fields produced by the current feeds and interconnections to the TF coils shall also not exceed 30 gauss. This restriction places an upper limit on the current in the connections to the TF coils, and also places a lower limit on the total number of turns that must be used in the TF windings.

3) The error fields due to anticipated tolerances in the positioning and alignment of the TF coils shall not exceed 30 gauss.

Comparison of Several TF Coil Configurations

Several TF coil configurations were considered for ZT-H. For each configuration, the positions of the TF coils were specified and the 3-D magnetic field code EFFI was used to calculate the radial component of the spatial field ripple at the plasma surface.

Three coil configurations are illustrated in Figs. 2 and 3. Each configuration allows spacing for the accommodation of 5 inch diagnostic and/or pump ports in twelve toroidal locations. Figure 2 A shows a view of 36 "straight" coil modules equally spaced every 10 degrees around the torus with each module containing 8 turns. Figure 2 B is similar, except the coils are tapered in order to reduce the spacing between coils on the outside of the torus. Figure 3 shows 72 straight coil modules spaced every 5 degrees around the torus with each module containing 4 turns.

The spatial ripple as a function of radial position of the coils for each of these configurations is shown in Fig. 4. While there is an evident advantage to tapering the coils, there is a greater advantage to increasing the number of coil modules. The 72 coil configuration was selected for the ZT-H design, and for a coil radius of 60 cm, the radial error field is only 0.12%, which is well within the target criterion of 0.3%. This allows a margin for future design changes, and for tolerances in positioning the TF coils, as well as allowing for some uncertainty concerning the precision of the criterion itself.

Field Errors from Current Feeds

The error field from the current feeds to the TF coils can be estimated by assuming that the error field is predominantly a dipole, and integrating the dipole

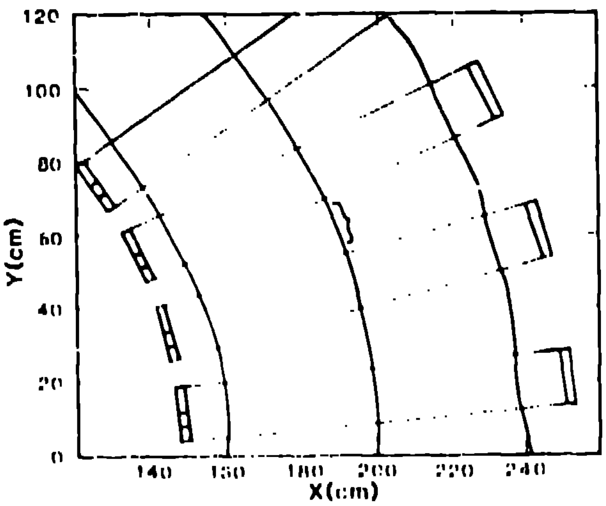

Figure 2 A

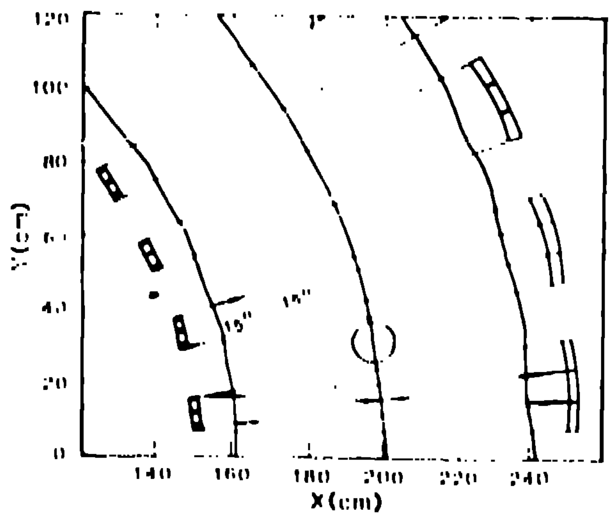

Figure 2 B
Two configurations of 36 TF coils
(A) Straight coils (B) Tapered Coils

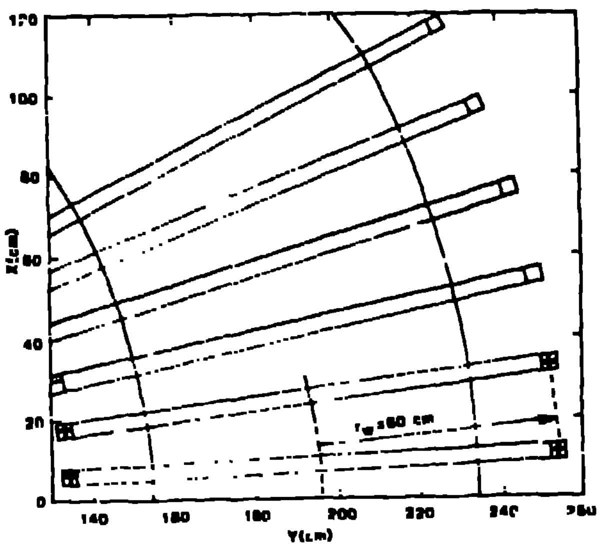

Figure 3
TF Coils Arranged as 72 Coil Modules
Showing coil locations and magnetic flux lines

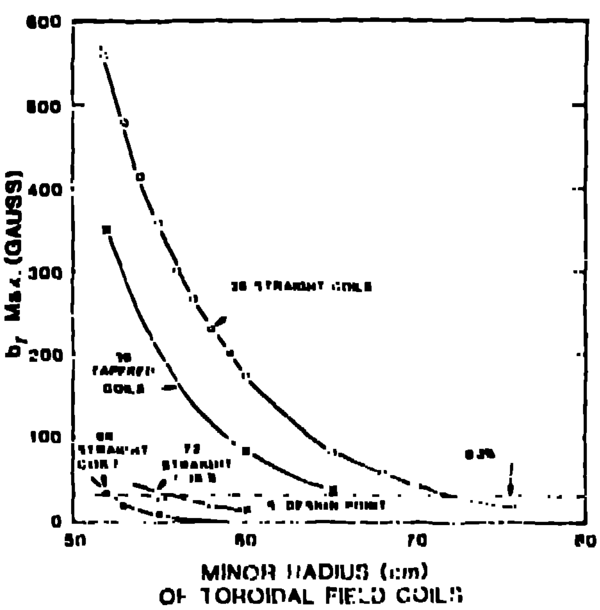

Figure 4
Maximum Field Error vs Radial Location of TF Coils

contribution over the length of the current feeds. The result is a formula for the net field error,

$$B = \frac{\mu_o I_{TOT} \delta}{8\pi NM a^2} \qquad (3)$$

where I_{TOT} = total poloidal ampere turns
N = number of turns per module
M = number of coil modules
a = closest distance to plasma
δ = spacing between current feeds
NM = total number of TF turns

The formula can be used to determine the minimum number of total turns necessary to hold the error field below a given value:

$$NM > \frac{\mu_o I_{TOT} \delta}{8\pi B a^2}. \qquad (4)$$

For ZT-H, the peak reverse current will be $I_{TOT} = 5 \times 10^6$ ampere-turns, for which NM > 104 turns.

The error field from the interconnections between the coil modules is approximately the net field from two antiparallel currents,

$$B = \frac{\mu_o I}{2\pi} (1/R_1 - 1/R_2) = \frac{\mu_o I_{TOT} \delta}{2\pi NM a^2}, \qquad (5)$$

where δ = spacing between interconnect and return
a = distance between current path and plasma surface.

By fixing the number of turns, Eq. (5) can be used to determine how far the interconnects should be from the plasma;

$$a^2 \geq \frac{\mu_o I_{TOT} / NM}{2\pi B} \delta . \qquad (6)$$

For example, if NM = 288 turns, then a ≥ 0.25 meters. For this case, the interconnects could be made at 6 cm outside of the coil radius.

The final design configuration for ZT-H consists of 72 coil modules, radius 60 cm, evenly spaced every 5 degrees around the torus, each coil module containing 4 turns, for a total of 288 turns, and with the interconnects between modules being made at a minor radius of 66 cm or greater.

Design Description of the Toroidal Windings Support Structure and Loads

The cross section of each toroidal coil turn was found to be 0.986 in² turn based on a total toroidal current of 8.5 MA, a current density of 30 kA in², and 288 turns.

Because of the short (0.2 second) equivalent square wave and the 100 second repetition rate, it is not necessary to cool the toroidal conductor. The current joint was tested at 30,000 amps for 20 seconds duration, and then repeated for several hours at 100 second intervals with a resulting average temperature rise on the order of 8-10°C. Computations bear out values of temperatures of less than 5°C between pulses for 0.2 second equivalent square wave.

Placement of the 288 copper turns is dictated by magnetic field error specifications. They must be uniformly spaced along the circumference of the torus to achieve the specified error field. The azimuthal and radial positions of the toroidal coils are further influenced by the need for access to the shell and liner, and the space allocated to vacuum ports and diagnostic tubulations. Satisfying these design conditions results in 72 evenly spaced locations for the toroidal windings, and requires adjacent 2 × 2 turn coils. Each turn will be prefabricated from 15 strips of 1/16" thick × 1"-wide Formvar coated copper to be overwrapped with insulation and used as a 1" square cable. Because of the differential length developed between inner and outer copper strips, only the first 2" of the copper strip will be soldered into a current joint to allow the remainder of the 25-foot long strips to be wound into a two-turn coil. The ends of the 15 strips are then trimmed to suit, and the other current joint is formed and soldered. A threaded sleeve insert is silver brazed into each end of the 1" square copper cable, so that the beginning and end of each two-turn copper strap can be bolted to the next coil in series or to the incoming power lead (see Fig. 5). Transferring the prefabricated cable from a wooden spool onto the machine coil form yields easy machine assembly and reduces the number of current joints.

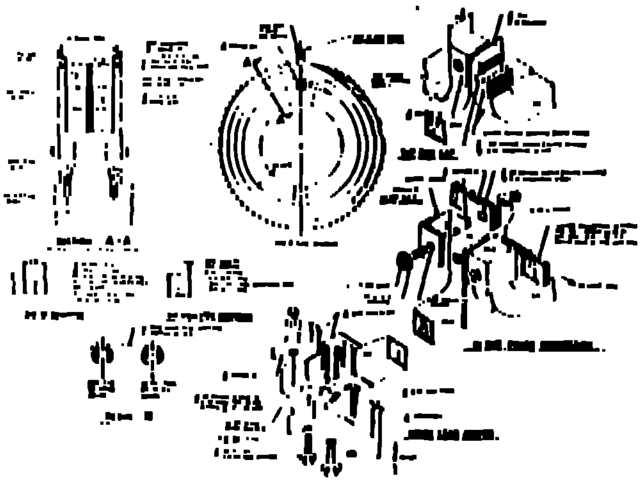

Figure 5.
Typical Four Turn Toroidal Coil Assembly.

The coil support form is made from two parts: a flange-ring bolted to the shell; and the coil support form. The coil support form can be rotated until each of the four TF-coil turns are wound, and can then be securely bolted to the flange ring. The inside leads of the two-turn coils are electrically connected to each other through a recess in the I.D. of the coil form, thereby creating a four-turn coil. The leads at the coil O.D. are connected in series to the adjacent four turn coil, and the final coil lead starts the return (see Fig. 6). The Lorentz forces at the leads are resisted by clamps, and by means of mechanically tieing the incoming and return power lines.

Magnetic forces on the toroidal field coils cause hoop tension, a centering force and a toppling moment. These forces were treated as a combination of static loads, and the periodic nature of loading construed to be such as to cause very little amplification.

The vertical and radial forces, or outward magnetic pressure bring about hoop tension in the coils. Because the magnetic forces are stronger on the inside than the outside of the machine, they cause a centering force. This can be seen from the equation $B_T = \mu_0 NI/2\pi R$, indicating that the toroidal magnetic field increases with decreasing major radius of the machine. The formula for the radial force per unit length of conductor is:

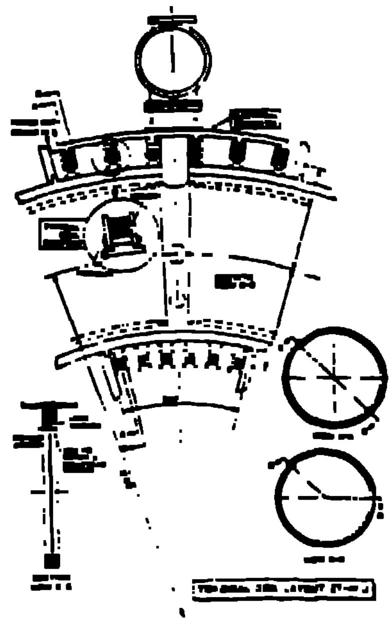

Figure 6
Arrangement of Toroidal Coils and Power Leads.

$$f = I B_0/(1 + \frac{a}{R}\cos\theta) \qquad (7)$$

where:

I_0 = 139 kA/coil
B_0 = 1 Tesla
a/R = 0.3

On the outside of the machine (θ = 0), f is 105 kN/m (600 lbs/in) and on the inside of the machine (θ = 180°), 198 kN/m (1130 lbs/in).

Integrating these forces over the entire coil results in a net centering force of 89 kN (20,000 pounds). Due to the interaction of vertical field (B_v) and the poloidal current (I_p), each coil is subjected to a torque = $\pi r^2 I_p B_v$, or the equivalent of the toppling moment tending to overturn the coil (see Fig. 7). For B_v = 0.11 T, the torque is 25 kN·m (16,000 ft-lbs). The resulting maximum force per unit length at the top and bottom of the coil, f = 256 lbs/in.

Hoop and Centering Force Stresses

The hoop load, which is based on a hoop force of 800 lbs/in, is distributed over the copper windings and the tension banding of the coil insulation, and restrained by the tightly fitting clips along the coil circumference. Assuming the copper turns to be on a low friction interface with respect to each other, and by comparing the relative stiffness of copper and clips on the basis of their radial deflections, we find the copper to be subjected to a bending stress of 8,700 psi and hoop stress of 4,100 psi. The selection of 10% cold worked copper with an allowable yield stress of 20 ksi will provide a 2:1 safety factor. The thickness of the angle clips will be 1/8", and the fibers of the material so oriented that they oppose flexure so as to arrive at a safe margin for the bending stress.

The net centering force on each T. F. coil is transferred from the coil support form to the shell. The calculated value of 95 in⁴ for the support ring moment of inertia results in a centering force stress

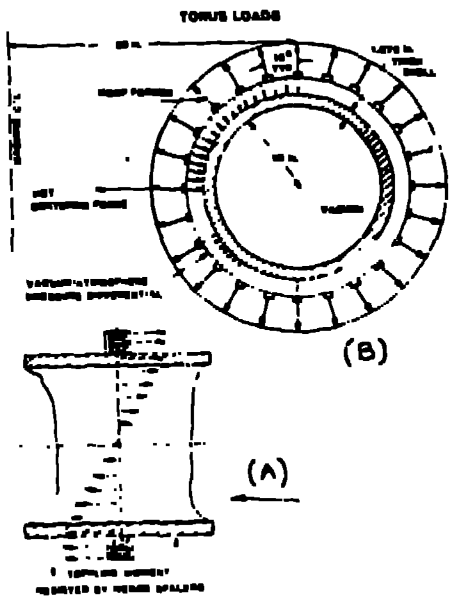

Figure 7
(A) Toppling Moment Acting on Each Toroidal Coil
(B) Pressure Differential Hoop and Centering Force Diagram.

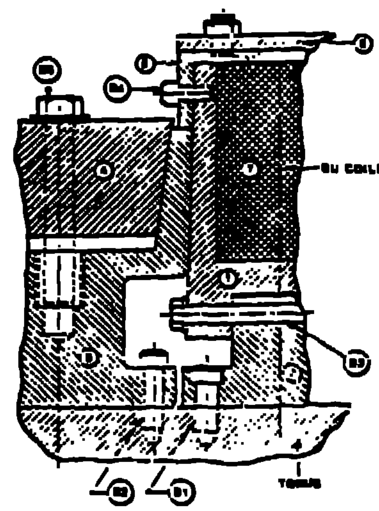

Figure 8
Preloaded Wedge Assembly to Resist Overturning Moment.

of 3,000 psi. Every detail and bolt in the coil support form subjected to radial centering loads was investigated for shear and bending stresses. The results assume all of the radial load to be first resisted by the coil support, and then transferred to the toroidal shell.

Stresses Due to the Toppling Moment

Preloaded spacer assemblies will be installed between adjacent toroidal coils to resist the effect of the toppling moment (see Fig. 8). The emphasis will be on placing six o. these wedge spacers along both the top and bottom circumference of the torus. It is understood, however, that all spacers must be in place and preloaded to accomplish the bracing against the uni-directional force along the major axis of the torus. The 236 lb/in force resulting from the overturning moment, occurs at the top and bottom of the TF Coil, creating lateral bending and shear stresses. The lateral bending stress on the support form side flanges is on the order of 7,000 psi. As in the case of the centering force, these loads must first be resisted by the coil support form, then transfered to other structural elements. The toppling moment loads are absorbed by the wedges and the toroidal shell.

A finite combined element analysis was performed to investigate the effect of forces from magnetic and vacuum loads on the main support frame, bulkheads and the aluminum shell (see Fig. 9). The analysis concentrated on the differential pressure loading, the toroidal coil centering force and the toppling moment which cause stresses on the shell in both the toroidal and poloidal direction. For sake of clarity it deals separately with either the toroidal shell or the upper support structure, but considers common load condi

Buckling analysis of the vacuum torus, consisting of shell and bellows liner, was conducted by considering: a) the pressure differential of 15 psi between

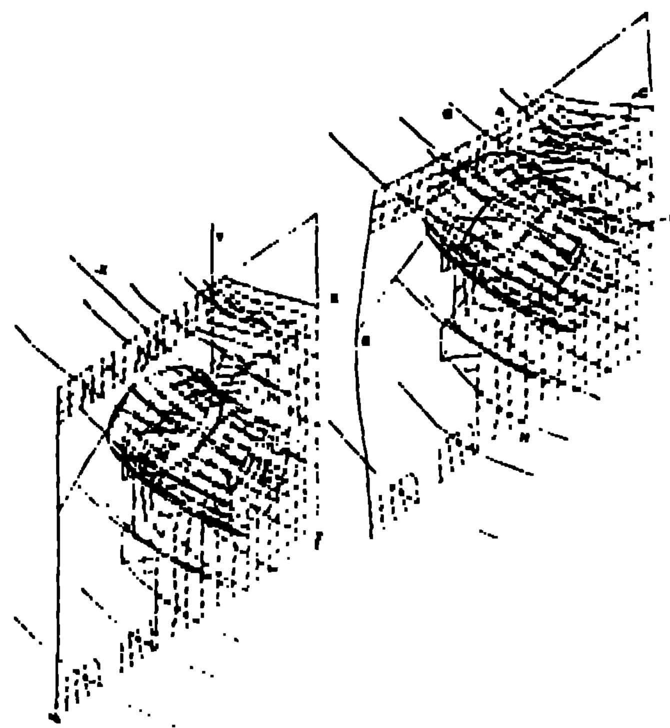

Figure 9.
Scaled Deflected Shape of Finite Element Model.

TABLE 2

EQUIVALENT LINE LOADS DUE TO VACUUM, CENTERING FORCE AND OVERTURNING MOMENT.

Load	Equivalent External Pressure (psi)	Equivalent Line Load (lb/in.)
Vacuum	15	250
Centering Force	225	5,700
Toppling Wedges	30	500
Total	270	6,450

the vacuum and pressure side of the liner, b) the toroidal coil centering force and, c) the transformation of the toroidal coil overturning moment into wedge preload forces on the torus.

The centering force (20,000 lbs) and the toppling moment (36,000 ft-lbs) are converted into an equivalent external pressure or line load acting on the shell. The external pressure was used to calculate overall torus buckling, and the line load for nonsymmetric buckling evaluation.

Buckling Calculation

The theoretical buckling calculation was performed in four ways. The first is based on a long cylinder the same diameter as the minor diameter of the torus. The theoretical buckling pressure is from:

$$P_{cr} = \frac{E\,t^3\,(n^2-1)}{12\,r^3\,(1-\nu^2)} \; . \tag{8}$$

Two methods of computing the buckling pressure of the torus were used:

$$P_{cr} = 0.1738E \left(\frac{\left(\frac{t}{r}\right)^2}{\frac{R}{r}\,(1-\nu^2)^2}\right)^{1/3} \; , \tag{9}$$

and

$$P_{cr} = \frac{A\,E\,t^2}{r^2} \; , \tag{10}$$

And finally, the buckling of the curved beam subjected to a centering line load force was computed.

$$q_{cr} = \frac{3EI}{R^3} \; , \tag{11}$$

where
- E = Young's modulus;
- ν = Poisson's ratio;
- t = wall thickness;
- r = radius of cylinder or minor radius of torus;
- R = major radius of torus;
- I = moment of inertia of beam cross section, the minor cross section of the torus in this case;
- n = buckling mode number.

Results of Toroidal Shell Analysis

The torus was modeled without joints and diagnostic ports (unconservative), and did not consider stiffening supports at 30° intervals (conservative). The loads on the 1.875-in. thick shell are summarized in Table 2 and the results of the buckling analysis are shown in Table 3. The differences between theoretical buckling and design buckling are based on an imperfection factor of two and a safety factor of two. As can be seen from the table, the equivalent applied load is between 1/3 and 1/4 of the design load and therefore shows the torus design to be conservative. The shell stresses were also computed from a bulkhead finite element model resulting in an actual stress of 7,000 psi, which compared to the yield stress of 30,000 psi reconfirms a 4:1 safety margin.

Alternate Toroidal Coil Design and Consequence

Consideration was given to the use of 1" square solid copper bar pre-shaped to form 180° segments, joined by tongue and groove bolted connections. This would be a more desirable approach from a stress point of view, the copper having the capability to carry more load, thereby transferring less load to adjacent structural elements. However, the consequence would be a greater number of current joints, and the difficulty of building four-turn coils, two turns high with proper insulation between turns. During assembly of the machine it would be difficult to torque bolted joints, which by necessity would have to be off-set from one another. The bolted arrangement of eight 180° turns to form a four-turn coil would be more expensive and pose more space limitations on diagnostic access.

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TABLE 3

Failure Mode	Analysis Geometry	Theoretical Buckling Load	Design Buckling Load	Equivalent Applied Load
Local Buckling	Long cylinder	3250 psi	810 psi	270 psi
Torus Buckling	Torus	3650 psi	910 psi	270 psi
Local and Axisymmetric buckling of torus[4]	Torus	3400 psi	850 psi	270 psi
Antisymmetric	Curved beam	141,000 lb/in.	35,000 lb/in.	6450 lb/in.