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Design and Analysis of the Big Dee Vacuum Vessel

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SUMMARY

Since it produced its first plasma in February 1978, Doublet III has been the world's largest operating tokamak with a non-circular cross section. The next phase in the development of the capabilities of the Doublet III facilities involves replacing its vacuum vessel by a larger, dee-shaped vessel called the Big Dee.

Replacing the vacuum vessel is possible because the toroidal field coils which enclose the vessel were designed to be demountable. Although some modifications are required for structures that attach to the vessel, the only major modifications involve the vessel, the poloidal field coils and their support structure. All other components of Doublet III will be reused for the Big Dee.

The main design requirements for the Big Dee vacuum vessel, in addition to maximum access to the plasma, are: (1) a maximum plasma volume as limited by the toroidal and poloidal field coils; (2) a high vacuum level; (3) a high toroidal electrical resistance; (4) the capability to withstand large impulsive loads due to plasma disruption; and (5) the capability to withstand thermal loads due to bakeout, discharge cleaning, and plasma operation while allowing adequate heat removal between plasma shots.

Wherever possible, the Big Dee vessel design is similar to that of the Doublet III vessel. The vessel has a sandwich wall with corrugated core. Wall and skin thicknesses are kept minimum to achieve maximum plasma volume and maximum toroidal electrical resistance, respectively. Inconel is selected for its high strength and high electrical resistivity.

The finite element stress analysis of the vessel is divided into several parts. For sizing of wall and skin thicknesses, axisymmetric models of equivalent single-walled shells are used. Static load cases include dead weight, atmospheric pressure, and discharge cleaning temperatures. The impulsive magnetic pressure due to plasma disruption is treated as a dynamic load to obtain stress and displacement time-histories of the vessel response. Detailed three-dimensional shell models are used to study typical sections of the sandwich walls. Other three-dimensional models of representative sectors of the vessel with typical ports are analyzed to determine the influence of the latter on stresses. Similar models will be used for buckling and seismic analyses.

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1. Introduction

Doublet III is the world's largest operating tokamak with a non-circular cross section. The first plasma was produced in February 1978, and by the end of 1982 over 32,000 plasma shots had been made. A detailed presentation of the engineering problems encountered in the design, construction, and operation of the machine was delivered at the 6th SMIRT Conference [1].

During its first three years of operation, Doublet III was successful in producing stable plasmas of reactor regime densities in various shapes with ohmic heating. In the past two years, the emphasis has been on dee-shaped and diverted plasmas in the upper half of the machine, with increasing levels of additional neutral beam heating; this has shown the importance of high plasma current and cross section elongation in improving plasma confinement.

Based on these promising developments, reaching reactor-like confinement conditions will require higher currents in larger dee-shaped and diverted plasmas with increased auxiliary heating. Doublet III will be shut down for 14 months in the latter part of 1984 and modified to this end. The new configuration, called Big Dee (see Fig. 1), is in the detailed design phase [2]. It has a new vacuum vessel allowing a dee-shaped plasma volume almost three times as large as in Doublet III. Big Dee is designed for a 5 MA maximum plasma current and a ratio of plasma pressure to magnetic field pressure - or beta - of 10%, both values unrivaled by any other U.S. tokamak. It retains the outstanding plasma shaping capability characteristic of Doublet III, while its larger vessel and its fewer outer poloidal field coils greatly improve plasma access for auxiliary heating and diagnostics.

This paper summarizes the conceptual design and the stress analysis of the Big Dee vacuum vessel.

2. Design of the Big Dee Vacuum Vessel

2.1. Design Requirements

The design of the new vacuum vessel must:

- Allow the largest possible dee-shaped plasma volume.
- Have a low magnetic permeability.
- Have a toroidal resistance of 0.13 mΩ or more.
- Maximize plasma access for diagnostics and auxiliary heating.
- Permit achieving a 2×10^{-6} torr-ℓ/s vacuum.
- Withstand the large impulsive magnetic load caused by a 5 MA plasma disruption in addition to the static vacuum load.
- Withstand baking and discharge cleaning up to 250°.
- Be compatible with 20 MW of plasma heating for 10-second pulses, with upgrading of the heat removal system.
- Minimize cost and construction time.

2.2. Design Description

The unique design of the Doublet III vessel, that is, its continuous, all-welded fabrication consisting of conical and cylindrical sections, is also used for Big Dee [3].

To maximize plasma size, the vessel must be nested close against the poloidal field coils and the vessel wall thickness must be as small as possible.

The vessel has a maximum radius of 2.46 m (96.75 in.), a minimum radius of 0.91 m (36.00 in.) and a height of 2.92 m (113.06 in.). It is supported at the outer midplane by guided supports which allow the vessel to expand and contract radially (see Fig. 2).

The walls of the vessel are 25.4 mm (1.0 in.) thick, of sandwich construction (see Fig. 3) with skin thicknesses of 2.36 mm (0.094 in.). The poloidally oriented corrugations, formed of 1.59 (0.063 in.) thick sheet, serve as passages for a cooling system and for secondary vacuum.

The material selected for the vessel is the same as in Doublet III, that is the high nickel alloy Inconel 625. It is non-magnetic and it has an electrical resistance and a mechanical strength which are both about twice that of non-magnetic stainless steel. Its yield strength is 414 MPa (60 ksi).

Table I summarizes the main parameters of the Big Dee vessel.

The vessel has approximately 130 potential port or feed-through locations accessible between the coils. The ports, being both much larger and more numerous than in the Doublet III vessel, pose some of the more difficult design problems. They also require more sophisticated stress analyses as explained in the next section.

3. Structural Analysis of the Vacuum Vessel

3.1. Design Loads and Stress Criteria

The principal static loads consists of:

- The weight of the vessel and all its attachments, about 13,600 Kg (30,000 lb.).
- The atmospheric pressure (vacuum) of 1 atm (14.7 psi) acting either on the outer or on the inner skin of the vessel walls.
- A maximum temperature difference of 100°C (180°F), between the minimum radius of the vessel at 250°C (482°F) and the maximum radius at 150°C (302°F), occurring during discharge cleaning.
- The temperature gradient across the wall of the vessel due to the heat radiated by the plasma during each shot, tempered by the vessel cooling system.
- The pressure of the coolant in the corrugations of the wall.

The main dynamic loads on the vessel are:

- The magnetic loading caused by a 5 MA plasma disruption. As the plasma current suddenly decays following the onset of some instability, a current is induced in the vessel which, combined with the magnetic field, results in a magnetic pressure. The pressure acts to implode the vessel, i.e., it acts in the same direction as the atmospheric loading (see Fig. 4a). Both the peak value and the variation with time of the pressure depend on τ, the plasma current decay time, as shown in Fig. 4b.
- Other loads including those due to seismic events, vessel misalignment with the toroidal field, and various fault conditions. Although important, these loadings do not affect the wall and skin sizing.

In general, the criteria of the ASME Boiler and Pressure Vessel Code, Section III, Division 2 for design by analysis are used as guidelines. However, to minimize permanent deformation and fatigue considerations, the vessel walls are sized, and the cooling system is designed, so that the combined mechanical and thermal normal stresses do not exceed the yield strength of Inconel 625 when local stress concentrations are disregarded.

3.2. Structural Analyses

The structural analysis of the vessel is divided into several parts, depending on the degree of detail and the types of loadings. Extensive use is made of the finite element method.

For sizing the thicknesses of the vessel walls and wall skins, the ports are first ignored and an axisymmetric finite element model is used. The sandwich wall is replaced with a single-wall thin shell of revolution with the same membrane and bending stiffnesses. The model is analyzed with the MODSAP code developed by GA Technologies Inc. The static loadings include dead weight, atmospheric loading, and discharge cleaning temperatures. Figure 5 shows the deformed shape and stress distributions for the second of these cases. The impulsive loads due to plasma disruptions are treated by the direct integration method, also using the MODSAP code. Typical deflected shapes and displacement responses are shown in Fig. 6.

Detailed three-dimensional finite element models of typical sections of the sandwich wall (see Fig. 7) are used to determine equivalent stiffnesses, calculated local stresses, and study local loadings such as atmospheric and coolant pressure and thermal gradients across the wall. They are generated with the COCO meshing code and analyzed with the TRICO beam and shell code, both developed by the French Commissariat a l'Energie Atomique.

The great number of large ports and the presence of six discrete supports make three-dimensional calculations necessary to refine the stress analysis of the vessel walls and to size the port reinforcements. Finite element models of representative sectors of the vessel are analyzed as single-walled shells using the COCO and TRICO codes to which pre- and post-processing programs have been added by GA Technologies. The atmospheric pressure load case is being calculated at the time of this writing. Figure 8 shows the mesh.

It is also planned to calculate the detailed dynamic response to plasma disruption, and to determine the critical buckling pressure of the vessel under external pressure with similar three-dimensional model analyses. Excellent correlation with theoretical buckling pressures has already been obtained in test cases involving simple cylinders and circular tori. Finally, the response of the vessel to a 0.2 g design earthquake will be calculated with a three-dimensional finite element model to determine vessel stresses and deflections.

References

[1] PUHN, F. A., "Structural and Thermal Engineering Problems Encountered in Design, Construction, and Operation of Doublet III," 6th International Conference on Structural Mechanics in Reactor Technology, Paris, France (August 1981).

[2] DAVIS, L. G., "Design of a Dee Vacuum Vessel for Doublet III," 29th Annual Symposium of the American Vacuum Society, Baltimore, Maryland, U.S.A. (November 1982).

[3] MILLER, J. E., "Unique Design of Doublet and Big Dee Vacuum Vessels," 28th National Symposium for the American Vacuum Society, Anaheim, California, U.S.A. (1981), General Atomic Company Report GA-A16475, November 1981.

TABLE I
BIG DEE VACUUM VESSEL PARAMETERS

Inside height	2.87 m
Maximum radius	2.43 m
Minimum radius	0.91 m
Height-to-width ratio	1.91
Volume	37 m^3
Surface area	77 m^2
Cross section area	3.7 m^2
Wall thickness	2.54 cm
Inner - skin thickness	2.36 mm
Outer - skin thickness	2.36 mm
Corrugation thickness	1.59 mm
Material	Inconel 625

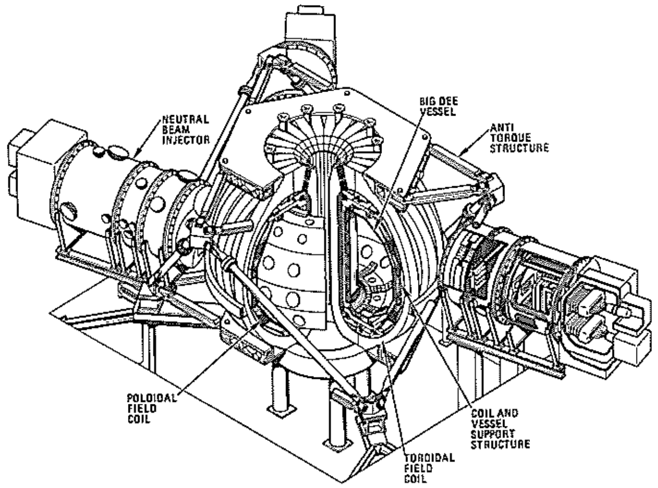

Fig. 1. Overview of the Big Dee Machine.

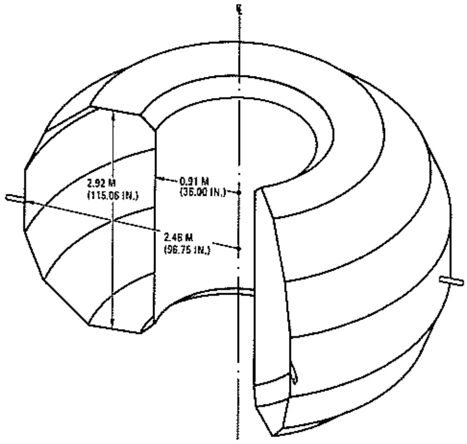

Fig. 2. Cutout view and outer dimensions of the Big Dee vacuum vessel (port openings not shown).

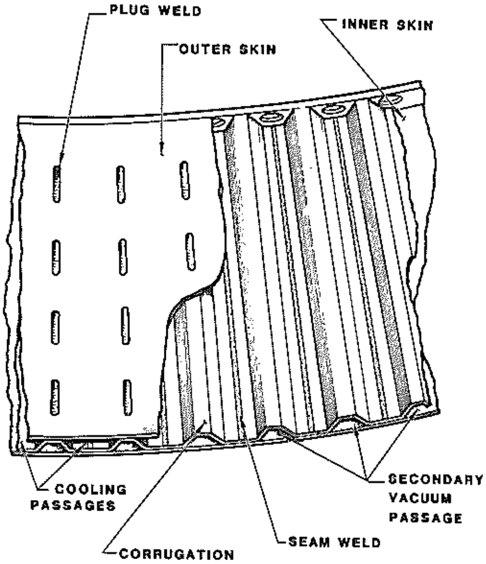

Fig. 3. Typical vessel wall construction.

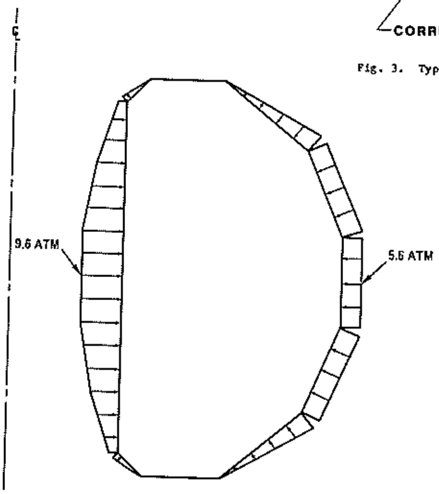

Fig. 4a. Peak pressure distribution due to the instantaneous (τ = 0 ms) disruption of a 5 MA dee-shaped plasma.

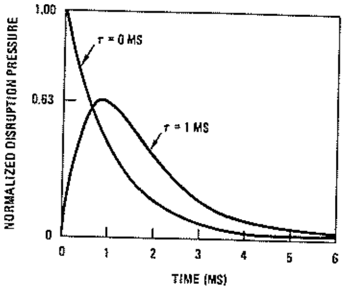

Fig. 4b. Time-variation of the normalized disruption pressure as a function of plasma current decay time τ.

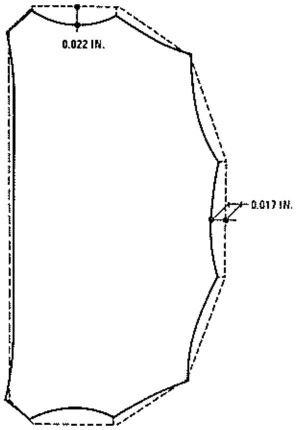

Fig. 5a. Axisymmetric analysis of the vessel under atmospheric loading - deformed shape.

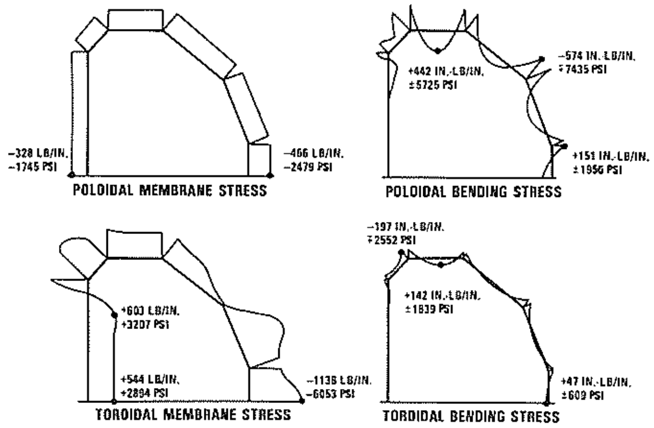

Fig. 5b. Axisymmetric analysis of the vessel under atmospheric loading - stress distributions.

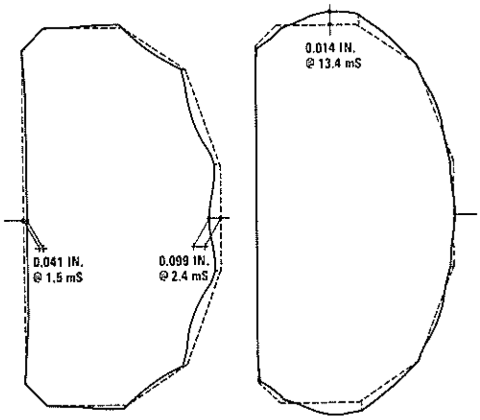

Fig. 6a. Axisymmetric analysis of the vessel during a 5 MA, 1 ms plasma disruption - maximum deflections.

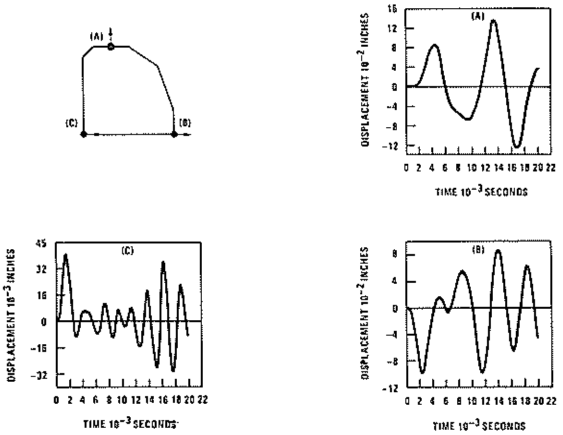

Fig. 6b. Axisymmetric analysis of the vessel during a 5 MA, 1 ms plasma disruption - displacement response at selected locations.

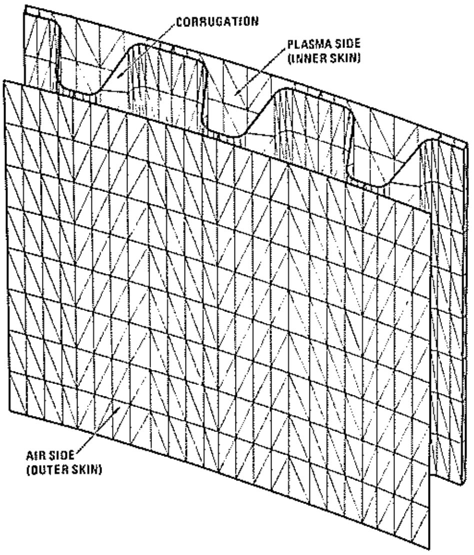

Fig. 7. Detailed 3-D model of a section of the vessel inner wall.

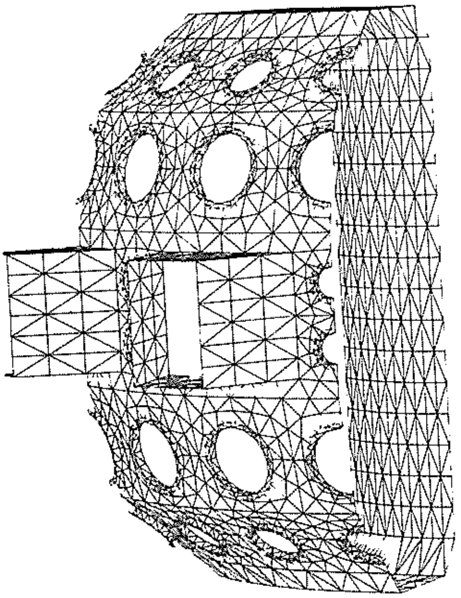

Fig. 8. Detailed 3-D model of a typical sector of the vessel with ports.