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ADVANCED TOROIDAL FACILITY VACUUM VESSEL STRESS ANALYSES*

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Abstract

The complex geometry of the Advanced Toroidal Facility (ATF) vacuum vessel required special analysis techniques in investigating the structural behavior of the design. The response of a large-scale finite element model was found for transportation and operational loading. Several computer codes and systems, including the National Magnetic Fusion Energy Computer Center Cray machines, were implemented in accomplishing these analyses. The work combined complex methods that taxed the limits of both the codes and the computer systems involved. Using MSC/NASTRAN cyclic-symmetry solutions permitted using only 1/12 of the vessel geometry to mathematically analyze the entire vessel. This allowed the greater detail and accuracy demanded by the complex geometry of the vessel. Critical buckling-pressure analyses were performed with the same model. The development, results, and problems encountered in performing these analyses are described.

Introduction

The ATF vacuum vessel is constructed of 0.25-in.-wall 304 stainless steel, with irregular-shape ports and bends and twisting clearance troughs for the helical coils (Fig. 1). The complex design made it apparent that a sophisticated analytical method would be needed. Several MSC/NASTRAN structural analyses of the ATF vacuum vessel were therefore performed during various stages of vessel design. The results from these analyses were used to confirm the structural integrity of the vessel during the design process. [MSC/NASTRAN is a large-scale, general-purpose computer code that solves a wide variety of engineering problems by the finite element method. MSC/NASTRAN is a version of the NASTRAN general-purpose structural-analysis program that was developed and is maintained by the MacNeal-Schwendler Corporation (MSC). NASTRAN is a registered trademark of the National Aeronautics and Space Administration (NASA).]

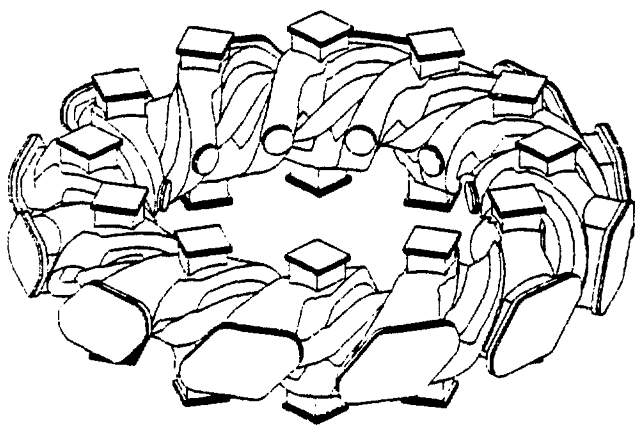

Fig. 1. Isometric diagram of the ATF vacuum vessel.

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During the conceptual phase, a model was developed by the Lockheed Missiles and Space Company under contract with the Oak Ridge National Laboratory (ORNL). This model represented the basic geometry of the vessel at that time, that is, a 30° segment of the torus with no ports and with double helical troughs of the same geometry as the helical coils that wrap around the vessel (Fig. 2). Since rotational symmetry exists every 30°, the cyclic-symmetry technique was used to achieve a smaller, more refined grid that more accurately models the tortuous geometry of the vessel.

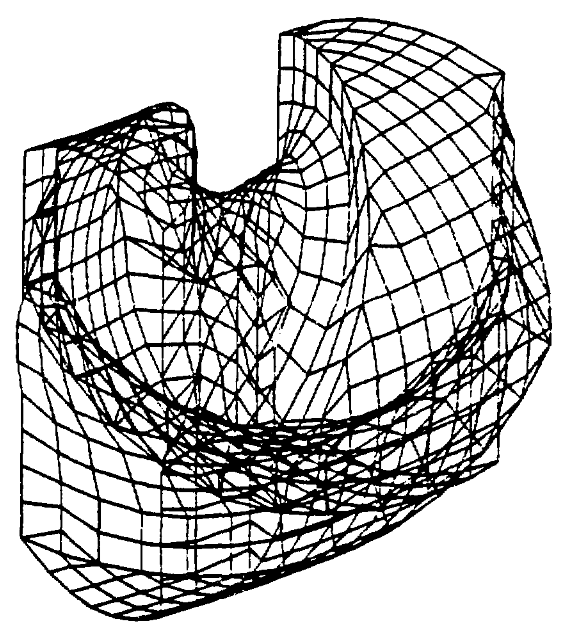

Fig. 2. Lockheed model of the ATF vacuum vessel, representing a 30° segment of the torus with no ports and with double helical troughs.

The term "cyclic symmetry" is used to include rotational symmetry, dihedral symmetry, and axisymmetry. Since reflective symmetry can be thought of as a special case of dihedral symmetry, it can also be included. The input data and the cost of the computation can be reduced by taking advantage of symmetry. The input data are reduced from the fundamental model (the symmetrical subregion) through the description of its boundary conditions to an analysis set. A solution for each symmetry condition is then obtained by using the appropriate boundary conditions. The results of the individual symmetry conditions are then combined to obtain a solution for the complete structure. The method and its usage are discussed in Refs. 1 and 2. Initially, a static solution (rigid format 47) was performed by using the 30° cyclic-symmetric model with a loading of 1 atm external pressure caused by vacuum. As the design matured and ports were defined, the port geometry was added to the model (Fig. 3), and the analysis was repeated. This second iteration included not only the static run but also an eigenvalue solution for buckling (rigid format 77).

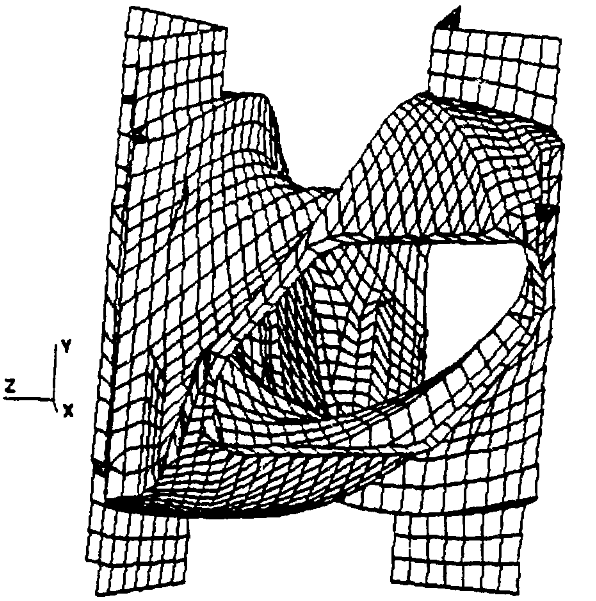

Fig. 3. MSC/NASTRAN model of the ATF vacuum vessel with port geometry.

Theory

There are two classes of buckling: linear and nonlinear. An example of the difference in the two forms can be easily demonstrated in the following procedure:

Place a perfectly straight piece of lumber, 1 in. × 8 in. × 16 ft long, in a vertical compressive test machine with both ends effectively built in. Since the lumber is perfectly straight, the stresses will remain extremely low until the Euler load (about 1070 lb) is reached. Just before the Euler load is attained, the stress is only 134 psi, and the allowable stress is 1000 psi. Then, however, the deflection goes to infinity, and thus the board linearly buckles. In reality, however, the board is not perfectly straight, and the eccentricity will cause the board to bow out (into its first-mode buckled shape), enormously increasing the stress due to bending, which causes it to buckle nonlinearly drastically below its Euler load.

Fortunately, MSC/NASTRAN allows us to determine either linear or nonlinear buckling. In the case of the 16-ft board, the eccentricities of the board centerline are measured and modeled into the finite element model. Then, using rigid format 66 (geometric and material nonlinear solution), one would enter the stress-strain curve for the wood as input. Next, one applies the load in small increments and notes the total applied load when the matrix goes singular. That total load would be defined as the nonlinear buckling load.

It can be shown that the eccentric board will deform into its fundamental mode and fail (from excessive stress and strain) nonlinearly in its fundamental eigenvalue shape.[3]

For the ATF vacuum vessel, the linear or nonlinear buckling loads can be determined as follows:

1. Run the normal modes and eigenvalue buckling analysis, using rigid format 77 for cyclic symmetry.
2. Using the fundamental (lowest eigenvalue) mode shape, impose the deformed shape and boundary conditions on one segment (in our case, a 30° segment) of the ATF vacuum vessel, and run the geometric and material nonlinear rigid-format case (rigid format 66) on the one segment to obtain the load at which the matrix goes singular or reaches excessive plastic deformation. If this load is lower than the fundamental eigenvalue, the vessel will fail nonlinearly, as the board did. (In our case, the nonlinear load was almost identical to the linear eigenvalue solution.)

Measuring the eccentricities was not necessary since the tortuous load path (caused by the ports and helical troughs) in the vessel of the ATF ensures built-in eccentricities.

Results

The results of the Lockheed model (solution 47) bore a maximum principal stress of 8100 psi, a maximum displacement of 0.027 in., and a buckling load at 11.5 times the applied vacuum pressure load of 15 psi. These results were acceptable at the time of their completion since the exact geometry of the ports had yet to be finalized.

The analysis for the solution 77 model required a team effort to prepare the model and to run the analysis within the schedule. The modeling task was a collective effort done in series. The geometry was prepared and a static analysis was completed by using simulated boundary conditions on the 1/12 vessel model. The work was done on the ORNL Fusion Energy Division's EBV VAX 11/780 PDA PATRAN-G, Ver. 1.6, an interactive preprocessor and postprocessor program supplied by PDA Engineering Software Products Division. The authors' task was to apply their knowledge of the MSC/NASTRAN finite element code to perform the cyclic-symmetry analyses, achieve an accurate boundary response, and perform the buckling analysis (solution 77).

After a few additions the model was translated to a MSC/NASTRAN, Ver. 64, bulk data deck from the PATRAN data base by using the PATNAS translator. There were two reasons for this. First, the size of the PATRAN file had become immense, on the order of 80K blocks on the VAX; this afforded little room on the disk for further processing. Future models of this size may have problems using PATRAN because of disk restrictions imposed by undedicated time-sharing systems. Second, since PATRAN could not produce or translate many of the necessary MSC/NASTRAN cards for the analyses, this path was inevitable—specifically, in respect to the cards necessary to perform cyclic-symmetry analyses and the one-way translation restriction of the PLOAD (pressure load) card, PATRAN to MSC/NASTRAN only.

After the necessary MSC/NASTRAN cards were assembled, the first run was attempted on the EBV VAX 11/780. As expected with a model of this size, there were a few model errors to overcome, but the main problems were inherent to the type of solution used. Correct definitions of the boundary points and coordinate systems are examples.

The greatest deficiency was the VAX itself. Because of the size of the model, a cyclic static run (solution 47) would take 55 to 60 h of CPU time in roughly 6 d wall clock time. This is very inefficient and approaches the average time between crashes on the VAX when everything is lost. The solution to this situation was to use a larger computer.

The MFE Cray 1A became the likely candidate. The cyclic-symmetry buckling analysis using MSC/NASTRAN, Ver. 63, Solution 77, was conducted by using the CRAY "C" machine at Lawrence Livermore National Laboratory. Without generating postprocessing output files, the run was successfully completed in approximately 22 min of CPU time in roughly 24 h wall clock time, the long clock time being due to low priority. While generating the postprocessing output file for PATRAN (for011.dat), the same run took approximately 30 min of CPU time in roughly the same wall clock time. These times are given for usage at normal working hours in Oak Ridge. Better wall clock time can be achieved by getting special authorization to run at very high priority, by working on weekends, or by working from 12 a.m. until 7 a.m. during the week.

The results of the analyses with the ports included are as follows:

1. Maximum principal stress of 14,742 psi, located at the lower left corner of the large port where the vessel and flange join at 15 psi external pressure.

2. Maximum total displacement of approximately 0.040 in. A maximum x displacement of 0.040 in., a maximum y displacement of 0.025 in., and a maximum z displacement of 0.023 in.
3. A buckling-pressure load of 6.92 times the applied vacuum load of 15 psi.

The difference between the earlier Lockheed results and the results given here can be directly attributed to the addition of the ports and a true boundary condition. The recent model with the ports does not have the added rigidity of port covers. Stress concentration due to the port geometries contributes to the higher stresses in these regions. The vessel should withstand the design load of 1 atm vacuum. The eigenvalue of approximately 7 implies a buckling pressure of 7 times the applied vacuum load. That value was obtained from the 0 harmonic in the eigenvalue analysis. At a later time, investigation of the other modes may be of interest.

Acknowledgments

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[2] *MSC/NASTRAN Theoretical Manual*, section 4.5, MacNeal-Schwendler Corporation, Los Angeles, Calif., 1985.

[3] *MSC/NASTRAN Application Manual*, section 5, p. 1, MacNeal-Schwendler Corporation, Los Angeles, Calif., August 1982.

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