Thermostructural design of the first wall/blanket for the TITAN-RFP fusion reactor

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1 INTRODUCTION

The mass power density, which is defined as the average power per unit mass within the magnet boundary, is a rough and general measure of economic competitiveness. Conn et al. (1985) have identified a target value of 100 kW(e)/tonne as a reasonable threshold for "compact" commercial fusion systems. In pursuit of this goal, Hagenson et al. (1984) and Najmabadi et al. (1987) have pointed out the inherent characteristics of the RFP toroidal confinement concept which allow it to exceed this target value. It is inevitable that the compactness of the fusion power core will introduce a unique set of design issues. These special design concerns stem from high thermal surface fluxes, high bulk energy deposition by neutrons, and a relatively short blanket structural lifetime.

In the TITAN-RFP, study Najmabadi et al. (1987) investigate a number of blanket (B) and first wall (FW) options suitable for high power density fusion reactors. Final choices were made for two designs: a high pressure aqueous blanket and a vanadium/lithium self-cooled blanket. The first design utilizes a pressurized aqueous loop containing a lithium compound dissolved in water, while the second design is based upon a self-cooled lithium-vanadium blanket. In this paper, we consider the beginning-of-life (BOL) thermostructural design and analysis of only the second concept. Radiation effects on lifetime will be considered in a separate paper. The overall design parameters of the TITAN-RFP Li/Li/V blanket are given in Table I.

A unique feature of the TITAN_RFP is the use of electric current flowing through the lithium coolant to produce the necessary toroidal magnetic field. A typical blanket module alos contains a hot shield zone, thus integrating the functions of the blanket, coil, and shield. The total blanket thickness is 30 cm and the overall thickness of the integrated blanket/coil (IBC) module is about 70 cm. Lithium is introduced to the module at 300°C and exits at an approximate temperature of 600°C.

2 THERMOMECHANICAL ANALYSIS OF THE FIRST WALL

2.1 Description of the loading and modeling.

The blanket geometries and loading conditions considered suggest that

TABLE I
OVERALL SYSTEM PARAMETERS FOR THE Li/Li/V SELF-COOLED BLANKET

Parameter	Value
14-MeV neutron wall load, MW/m^2	18.1
FW maximum surface heat flux, MW/m^2	4.1
Plasma major radius, m	3.9
Plasma minor radius, m	0.6
FW minor radius, m	0.68
Reversed toroidal field during burn, T	0.36
Poloidal field at plasma surface	5.9
Tritium breeding ratio	1.2
Neutron energy multiplication, 1-D	1.2
Total fusion power core mass, tonne	1900
Mass power density, kW(e)/tonne	526
Fusion power, MW	2528
Recovered thermal power, MW	3189
Thermal energy conversion efficiency, %	40
Gross electric power,MW	1243
Recirculating power, MW	243
Net electric power, MW	1000

both the thermal and mechanical fields will exhibit significantly smaller gradients in the poloidal and toroidal directions, as compared to the radial direction. The fluid bulk temperature and the internal coolant pressure change poloidally, however, the thermo-mechanical effect of this dependence on the first wall is small. Consequently, stresses perpendicular to the plane of a cross-section will not change significantly in the poloidal direction and a two dimensional model of the cross section at the location of the extreme value of the loading in the poloidal direction is appropriate. In this paper, axisymmetric models are used for both the thermal and the mechanical analyses. Toroidal effects have been neglected because of the fairly small toroidal angle subtended by one first wall channel, load invariance in the toroidal direction, and the fact that no load is transmitted in the toroidal direction.

Design criteria employed here involve maximum allowable primary and secondary stresses as well as surface temperatures. Since the structural material is metallic, the relevant uniaxial stress measure is the von Mises stress.

2.2 Axisymmetric Finite Element Models

Two first wall designs have been considered, a configuration with square channels and one with tubes. The tube design was preferred over the square channel design because of the more efficient heat transfer associated with round tubes. We will therefore concentrate on the tube design in this paper. A half-section of the first wall tubes is modeled using symmetry; the internal radius and the wall thickness are denoted by r and t, respectively. Finite element models used throughout this paper were set up using the ANSYS computer program, described by DeSalvo et al. (1979).

2.3 Comparison of the Results with Approximate Analytical Models

The in-plane pressure stresses exhibit the same axial symmetry within the cross section of the tube in both plane strain and axisymmetric analyses. The effect of poloidal curvature appears only as a different value of the constant poloidal stress. The axisymmetric numerical results are found to be higher than those of the approximate analytical model. This is attributed to the radial expansion of the tubes and coupling with the in-plane stresses, giving higher out-of-plane stresses. However, the equivalent stresses are lower in the axisymmetric case, since the stress state becomes closer to a hydrostatic state.

The analytical heat transfer solution for uniform heat flux results in a higher temperature drop across the wall of the tube. In the finite element model, the more realistic treatment of one-sided heating yields a smaller amount of heat deposited in the structure. Heat conduction to the cooler back segment of the tube across the tube wall thus lowers the temperature drop at the point of normal heat flux. Fig. 2. shows the temperature contours illustrating the asymmetry in the thermal field.

The axisymmetric and the plane strain results are found to be significantly different for thermal stresses. The constraint on the out-of-plane strain in the plane strain model gives large thermal stresses in that direction. Unrestrained radial expansion in the axisymmetric model is therefore more appropriate for the description of thermal stresses.

2.4 Parametric Study and Geometry Selection

To study the many interrelated parameters, a system code was developed by Hasan (1987) for the lithium cooled TITAN-1 design. This code selects an allowable first wall and blanket design satisfying efficiency, maximum temperature and stress limits. The heat transfer and stress analysis models are the same plane models discussed above. Therefore, the designs selected by the code are conservative. Fig. 1. shows the sensitivity of the maximum coolant pressure and the pressure stresses to the first wall tube radius, r, and the wall thickness, t. The first wall is cooled at the threshold velocity for turbulent flow until the maximum structure temperature limit can be maintained. When the thickness is too large, the temperature limit is reached unless the flow velocity is raised above the turbulence threshold. This, in turn, increases the pressure drops.

The allowable primary stress is taken as 100 MPa for the V-3Ti-1Si alloy, thus, as seen in Fig. 1, the whole range of geometric parameters represents acceptable designs. The minimum pressure stress occurs around t=1.3 mm for all radii considered.

Results from the thermomechanical analysis are shown on Fig. 2. and Fig.3.

When the structure temperature reaches the design limit the coolant velocity is increased and fluid bulk temperature is decreased. Thermal stresses continue to rise, since the total temperature drop across the first wall increases. The allowable primary stress is taken as 100 MPa for the V-3Ti-1Si alloy, thus, as seen in Fig. 1, the whole range of geometric parameters represents acceptable designs. The minimum pressure stress occurs around t=1.3 mm for all radii considered.

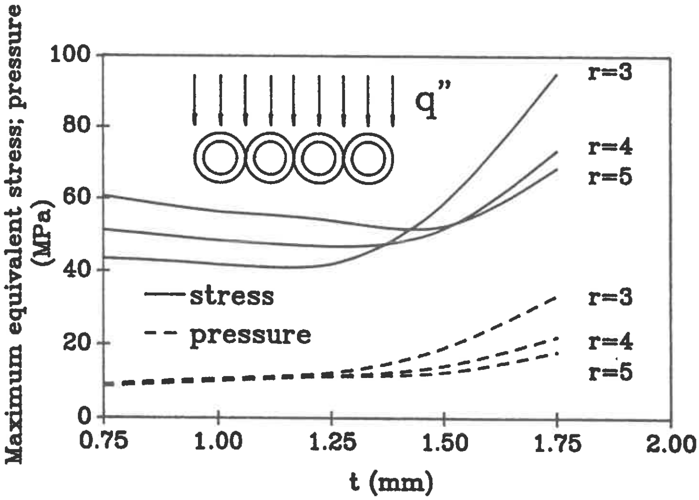

Fig. 1. Coolant Pressures and Pressure Stresses in the First Wall.

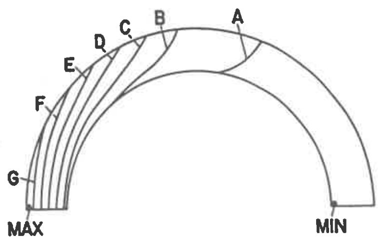

Fig. 2. Temperature Contours
A= 400°C B= 440°C C= 480°C
D= 520°C E= 560°C F= 600°C
G= 640°C MAX= 677°C MIN= 395°C

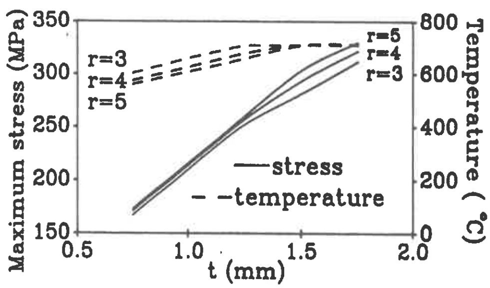

Fig. 3. Maximum Temperatures and Thermal Stresses in the First Wall.

3 BLANKET THERMOMECHANICAL ANALYSIS

Two configurations have been considered for the blanket, an array of tubes and an array of self-supporting rounded square channels. The blanket structure is subjected to heat generation rate and internal coolant pressure. Based on arguments given in section 2.1, axisymmetric finite element models were set up for the analysis of the blanket structure.

3.1 Analysis of the Blanket Tubes

For the tubes, results from the system code indicate low blanket coolant pressure, in the range of one to two MPa's. Accordingly, the blanket tubes will have a large radius to thickness ratio, and a simple one dimensional analysis is sufficient. This has been verified by comparison to finite element results.

The blanket coolant bulk exit temperature was fixed at 680 °C to allow the highest possible first wall exit temperature. Further constraints on the blanket design are imposed from neutronics and shielding. It was found that approximately 30% structure fraction is required for adequate energy multiplication. By changing the positioning and the cross-section geometry of the tubes, the structure fraction can be controlled. Analysis has shown that 2.5 mm thickness and 2.5 cm internal tube radius satisfies the design requirements. As the volumetric heating decreases towards the back of the blanket, the wall thickness can be increased and the inner radius decreased to optimize the blanket.

3.2 Analysis of square blanket channels

Blanket channels offer the advantage of a lower void fraction as compared to tubes. This improves both neutron energy multiplication and OH coil shielding. Therefore, the size of the blanket and hot shield can therefore be smaller.

Pressure is the main factor determining the size of the blanket channels. Fig. 4. shows the contours of the primary equivalent stresses, displaying stress concentrations at the corners. Fig. 5.

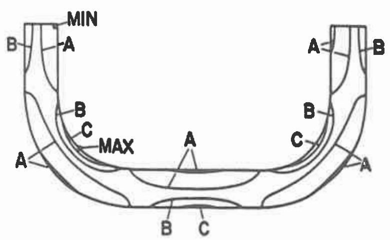

Fig. 4. Contours of the Equivalent Primary stresses MIN= 2 MPa, MAX= 40 MPa, A= 10 MPa, B= 20 MPa, C= 30 MPa

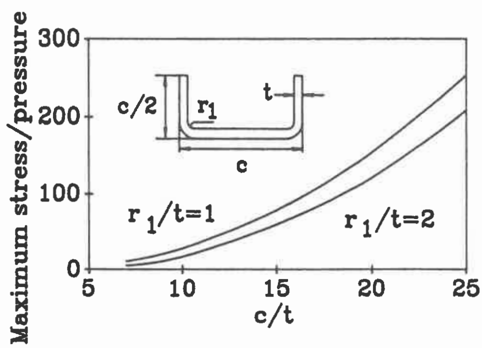

Fig. 5. Geometry of the Blanket Channels, Sensitivity of the Pressure Stresses to the Geometry

demonstrates the dependence of the primary stresses on the channel geometry. With 100 MPa allowable stress and 2-4 MPa coolant pressures the range of allowable normalized loads is 25-50. Increasing channel size results in higher bending stresses at the corner. The radius of the fillet acts in the opposite direction, larger radius relieves stress concentration. Channel design proceeds as follows: first the coolant pressure and the allowable primary stress give the appropriate aspect ratio from Fig. 5. The wall thickness is then independently determined by the heat transfer calculations based on maximum allowable wall temperature. From this, the size is completely determined.

CONCLUSIONS

The geometry of the first wall is dictated by the surface heat flux, the coolant temperature and the maximum allowable structure temperature. We found that a reasonable design could be found up to 4 MW/m^2 heat flux wall load.

The thickness of the blanket tubes is determined by the internal heat generation rate, a minimum coolant bulk temperature set by the requirement on thermal efficiency as well as the temperature limit on the structure. The radius of the tubes is given by the required structure fraction.

The aspect ratio of the rectangular channel design is determined by the pressure stress concentration at the corners. The wall thickness is limited by the heat generation rate and the fluid bulk temperature. These two constraints define the allowable range of designs. Further analysis will address the effect of the poloidally varying coolant temperature and the effect of the irradiation environment.

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