CONF-771029--98

A PRACTICAL BLANKET DESIGN FOR A TOROIDAL FUSION REACTOR*

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Summary

For about four years now, there have been studies underway to produce conceptual designs of large fusion power reactors. One of the components needed by any such reactor is a blanket. This blanket performs three distinct functions: 1) absorbs the fusion energy (particles and radiation) and converts it into heat; 2) converts some fertile material into tritium; and 3) helps to shield the toroidal field coils from the plasma.

The blanket described in this paper is one designed for an Elmo Bumpy Torus Reactor, a design study which has been pursued at ORNL for about eighteen months. In arriving at the blanket design, the experience gained in tokamak studies (FBX, EPR, and DEMO) at ORNL weighed rather heavily. Other tokamak studies at the University of Wisconsin, Argonne, and General Atomics were also influential as guides.

Many previous blanket designs had treated some problems (neutronics, materials, heat transfer, etc.) quite extensively, but at the expense of other problem areas. In arriving at this design, a very conscious effort was made to be sure that all problems were considered, and that the resulting design should stand as a viable concept that presented at least a solution for all the different problems. It is believed that this objective has been met.

Design Description

In a paper of this length it is not possible to defend certain materials choices or design constraints. These are listed below without explanation (see Ref. 10).

1. Stainless steel (modified 316) for all structures
2. Metalic normal lithium as the fertile material
3. High pressure gaseous helium as coolant
4. First wall temperatures limited to 400°C
5. High pressure helium coolant doubly contained for protection of blanket buckets
6. Thin first wall to handle surface energy deposition

This blanket concept will be examined for use in a tokamak, but the blanket to be described here is for an Elmo Bumpy Torus Reactor. In this reactor, the plasma region has a diameter of 2.4 meters under the coils, and a diameter of 3.4 meters in the "bumps" between coils. For either diameter the segments are divided into twelve buckets in the poloidal plane. There are nine toroidal segments between the center lines of two coils. The blanket is of uniform thickness for all segments, and this dimension is 60 cm. The bucket which will be described in detail is one in the segment under the coil.

The bucket, as shown in Fig. 1 consists of two compartments, a heavier wall rear section, and a thin walled ribbed section toward the plasma. The end walls are tapered while the side walls and top and bottom are straight. The partition plate separating the two sections is perforated so that the lithium volume

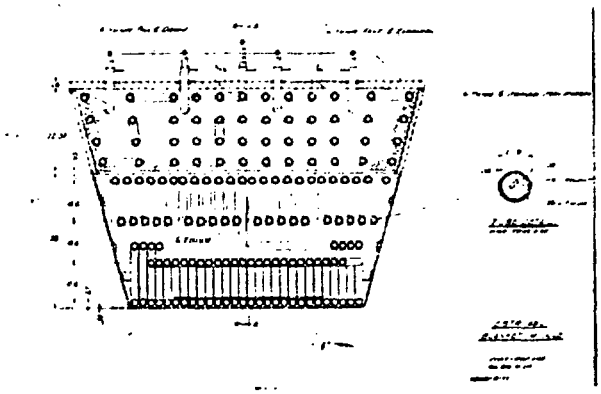

Figure 1 - Cross Section of EBTR Blanket Bucket. First Wall is at Bottom of Figure.

communicates between the compartments. The front section contains only lithium and the double walled serpentine cooling tubes. This is where most of the neutron energy is absorbed, and where the tritium is produced.

The rear compartment of the blanket bucket also contains coolant tubes and lithium, but, in addition, all space around the tubes is filled with stainless steel spheres. This portion of the blanket is primarily a gamma shield. The function of the lithium in this region is to conduct the heat from the spheres to the coolant tubes.

There are four helium circuits in parallel to handle the necessary volume of coolant. Each circuit is approximately the same length, each one being about 33.5 meters. The tubes are formed as individual coils, and the coils are then nested, and the center partition plate, which is in two pieces, is tack welded into place around the tubes. The shallow bottom pan is located on the bottom and held to the tubes by pins which run thru holes in webs over top of bottom row of tubes. The heavy box, without the top on, is next slipped down over the tubes and welded to the partition plate.

The thin reinforced walls for the bottom section are made in two pieces. These two halves are slipped around the tubes and welded to each other, to the heavy top section, and to the up-turned lip of the bottom pan. Stainless steel spheres are put in the top compartment around the coolant tubes. The back cover plate is next put down over the ends of the tubes and the final welds are made.

In order to contain the high pressure helium and prevent damage to the blanket buckets, the helium tubes consist of concentric tubes. The inner tube is wound with a .06 cm diameter wire with a pitch of one turn per foot. After the outer tube is slipped over the wrapped inner tube, the concentric pair is run thru a swaging die to effect a tight fit. The tubes each have a wall thickness of .08 cm. The inner tube has an I.D. of 1.75 cm, the concentric assembly has an O.D. of 2.19 cm after having been swaged tight.

The inner tube extends beyond the outer one at

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each end of the tube, and the annulus between tubes is sealed off by welding. A capillary monitoring tube connects this annulus to the lithium expansion tank for the blanket. After the four coolant tubes have been formed into coils, they are heated in a furnace; the annulus between tubes is evacuated, and the annulus is filled with lithium.

The purpose of the stagnant lithium in the annulus is to provide good thermal conductivity to the helium. Having stagnant gas in this annulus would impair heat transfer intolerably. Expulsion of this lithium into the expansion tank provides a means of detecting a helium leak in the inner tube.

In addition to the helium cooling tubes, each blanket bucket has a dip tube and a vent tube for filling and emptying lithium from the blanket. These tubes are connected in series for any blanket segment, and are connected to an expansion tank to allow for volumetric changes with temperature. It is planned to drain the blanket modules periodically for batch processing to recover tritium.

An assembly of twelve buckets on a mounting frame for one blanket segment is shown in Fig. 2. The

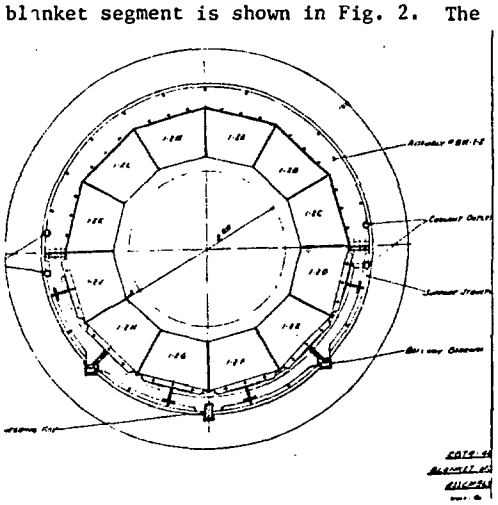

Figure 2 - Elevation Showing Twelve Blanket Buckets Assembled in a Mounting Frame to Make a Single Toroidal Segment.

buckets are bolted onto this frame by means of the mounting lugs on the tops of the buckets. The cooling manifold is integral with and part of the support frame. The blanket is supported off the shield modules which surround the blanket. The center blanket segment of one module is located on a grounded post. The four segments on either side are installed with approximately 3/8" clearance between segments. As the blanket heats and expands, these gaps close, but no vacuum tight seal results. The gas conductance thru these joints is negligible for a pressure differential of approximately 10^{-3} torr.

Actual removal and replacement of the blanket segments is accomplished by a vertical lift with an overhead crane. The center segment is removed, the two adjacent ones are then removed. Finally, the assembly of three segments inserted under each coil is first moved out from under the coils, and is then lifted out. Replacement takes place in the reverse order of the removal.

Neutronics Performance

Neutronics calculations were carried out to assess the capability of the lithium-laden blanket assembly for recovering the kinetic energy of D-T fusion neutrons and secondary gamma rays in the form of heat and for breeding tritium. Also, the capability of the blanket and shield assembly were evaluated in terms of their combined shielding capability for protecting the magnet assembly from the intense plasma radiation.

The calculations were performed in a one-dimensional cylindrical geometry using the discrete ordinates code ANSIN[1] with a P_3-S_{12} approximation. The neutron- and gamma-ray transport cross sections were taken from the DLC-37[2] library and collapsed to a 35-neutron and 21-gamma-ray energy group subset. All of the nuclear responses were obtained using the latest available ENDF/B-IV data. The reactor was modelled from detailed engineering drawings[3] and consists of a 60-cm-thick blanket assembly consisting of lithium in a stainless steel-316 structure and cooled with high pressure helium. The blanket surrounds a 100-cm radius plasma region and a 20-cm cold zone. The gamma shield is 55-cm thick and is separated from the blanket by a 30-cm thick maintenance void. The shield is composed of 65% SS-316-35% water and supported by stainless steel structure. Surrounding the blanket and shield is a magnet assembly consisting of Nb-Ti/Cu coil and supported by stainless steel. The overall radial dimension of the coil is 85 cm. All of the data were normalized to a first-wall neutron loading of $1.4 MW/m^2$.

The blanket assembly recovers 95.9% of the neutron and secondary gamma-ray energy in the form of heat. Approximately 4% of the energy is deposited in the shield and the remainder, 7×10^{-4}%, is deposited in the magnet assembly. The total heating rate in the reactor is 1.47×10^5 Watts per cm of plasma length. The peak heating rate in the blanket is 13.5 w/cm^3 and drops off to 0.34 w/cm^3 at the front of the shield. The shield reduces the nuclear heating by nearly four orders of magnitude and at the front of the magnet assembly the peak heating is down to 5.8×10^{-5} w/cm^3.

Natural lithium is used as the fertile material in the blanket. The tritium breeding ratio is 1.29 tritium nuclei per incident neutron with 56% of the breeding resulting from neutron reactions with ^6Li.

The radiation damage in the stainless steel structural components at various locations in the reactor is summarized in Table 1.

A detailed discussion of the nuclear performance of the EBTR may be found in Ref. 4.

Thermal Hydraulics

The thermal hydraulic design space for the blanket module previously described was examined in detail. Specifically, a parametric study was performed to determine all the relevant hydraulic parameters, such as heat transfer coefficients, pumping power, etc.

In addition, calculations were made to determine the operating temperatures of the blanket structure. In this respect, the design goals were as follows:

1. Limit maximum temperatures anywhere in the stainless steel structure to approximately 600°C.

2. Limit the temperatures in areas experiencing the most severe radiation damage (first wall) to approximately 400°C.

3. Minimize thermal stresses by arranging the coolant tubes to provide favorable temperature gradients in the blanket structure.

Neutron Wall Loading = 1.47 MW/m^2			
Location	Atomic Displacement Rate[a] (DPA/Year)	Hydrogen Gas Production Rate (APPM/Year)[b]	Helium Gas Production Rate (APPM/Year)
First Structural Wall	20.9	913.	276.
Center of Blanket (Zone 28)	4.62	127.	34.7
Rear of Blanket	7.70×10^{-1}	7.43	1.91
Front of Shield	2.16×10^{-1}	1.45	3.62×10^{-1}
Rear of Shield	4.92×10^{-5}	1.11×10^{-3}	2.74×10^{-4}
Front of Magnet Coil	3.51×10^{-5}	5.13×10^{-4}	1.26×10^{-4}
Rear of Magnet Coil	6.26×10^{-9}	7.76×10^{-10}	1.67×10^{-10}

[a]Based on an effective displacement energy of 40 eV.
[b]Atom parts per million per year.

Method of Analysis & Results

The fundamental energy exchange volume of the blanket is the module shown in Fig. 1. From the neutronics data described in Ref. 4, the poloidal variation of the volumetric heat generation is a known quantity for such a module.

For each module, an energy balance between the blanket module and its helium coolant requires that:

$$\dot{Q} = \dot{M}\, c_p\, \Delta T \qquad (1)$$

where $\dot{Q}$ = total rate at which energy is deposited into the module due to both alpha and neutron deposition.

$\dot{M}$ = total helium mass flow rate through a module

c_p = specific heat of the helium coolant

ΔT = total coolant temperature change through the module

In Equation 1, $\dot{Q}$ is obtained by integration of the neutronics data of Ref. 4 over the module volume. The ΔT for every case considered was 415°C, and c_p is a constant for helium over the temperature range considered. Hence, Equation 1 can be solved for $\dot{M}$.

Once the total mass flow into a module is established, it is necessary to establish how the temperature of the coolant varied as it traversed the complicated path shown in Fig. 1. This is done by applying Equation 2 to each tube as it traverses a small length increment through the module, i.e.

$$\Delta t = \dot{q}/c_p\, \dot{m} \qquad (2)$$

where Δt = temperature change through an increment in tubing length

$\dot{q}$ = rate at which energy is deposited locally into that increment of tubing

c_p = same as Equation 1

$\dot{m}$ = coolant mass flow rate through that tube

Once the incremental changes in the coolant temperature and the mass flow are established, the average thermophysical properties of the coolant can be determined for each increment in tubing length. Likewise, all the incremental thermal-hydraulic parameters, such as the heat transfer coefficient, pressure drop, pumping power, etc., can be calculated in each increment by the standard equations and methods.

In Figure 3, some of the results of a large number of hydraulic calculations done by this method are shown. In this figure, the effects of wall loading, tubing size, and inlet pressure are parameterized with respect to the ratio of pumping power used, to thermal power extracted.

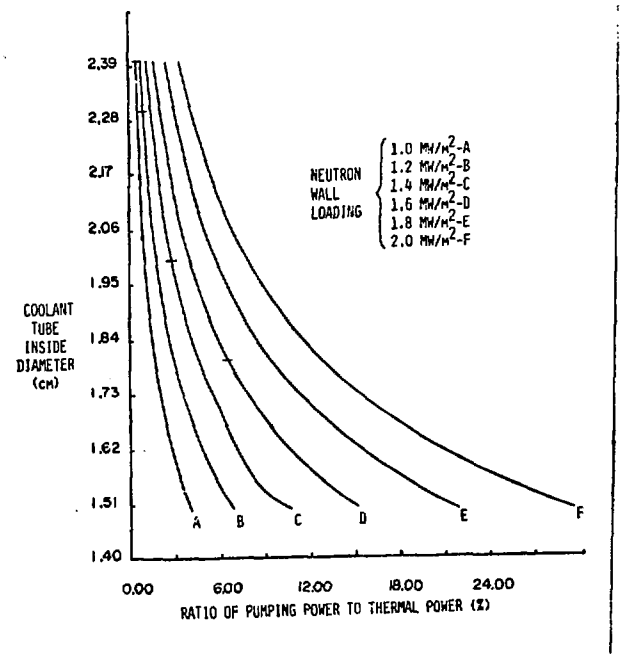

Figure 3 - Hydraulic Parameters for: P_{inlet}=1000 psig T_{in} = 66°C, ΔT = 415°C

Also indicated on Figure 1 is a series of horizontal lines intersecting each of the curves of constant neutron wall loading. This line indicates the maximum diameter tube that can be used and still maintain the first wall maximum temperature at less than 400°C. For instance, suppose that the operating wall loading is 1.4 MW/m^2, and the module inlet pressure is 1000 psig. From Fig. 3, one should note that depending on what size tubing is used in the module, the pumping power ratio can vary from 1.5% to 10.5%. Obviously, one would like to minimize the pumping power ratio; however, as the tube size increases, the heat transfer coefficient decreases, which increases the operating temperature of the material around the tube. Therefore, the horizontal line on each of the constant wall loading curves, indicates the tube diameter that results in a minimum pumping power for this design that

will still maintain design structural temperatures. Summarizing similar calculations at different inlet pressures and 1.4 MW/m^2, these data indicate:

@ 1000 psig; P_p/P_{th} = 2.8% for D = 2.0 cm

@ 1250 psig; P_p/P_{th} = 1.8% for D = 2.0 cm

@ 1500 psig; P_p/P_{th} = 1.3% for D = 2.0 cm

One should note that there are no horizontal lines on either the 1.8 or 2.0 MW/m^2 curves. This simply means that at these power levels, the 400°C first wall temperatures cannot be achieved. There is just too much material in between the outside of the first wall and the cooling passage to allow this to be realized in this design configuration. To operate at wall loading higher than 1.6 MW/m^2, the 400°C maximum temperature would have to be relaxed somewhat.

The other important purpose of this analysis was to establish the thermal gradients in the module structure. This information is used in the structural analysis described subsequently.

Briefly, the temperature fields in the module were calculated using the ORNL Heating V conduction code. The input to this code was generated from the hydraulic analysis previously described and the internal heat generation rates described in Ref. 4.

Of primary concern were the structural temperatures in the module side plates and front wall plate. In Figure 4, the temperatures in the side plates and the first wall plate are shown. These temperatures are for a module exposed to a neutron wall loading of 1.4 MW/m^2. As shown in this figure, the maximum first wall temperature is approximately 400°C. The maximum structural temperature is about 575°C, and occurs at the very top of the module (not shown). The structural implications of the gradients shown in these figures will now be discussed.

Thermal Stress Analysis

This discussion is concerned with the thermal stresses in a typical blanket module described above. In order to facilitate the analysis, the blanket geometry was modeled as a thin-wall rectangular vessel being depicted in Figure 5. Thermal stresses were then examined based on the results of the thermal hydraulic analysis given in Fig. 4.

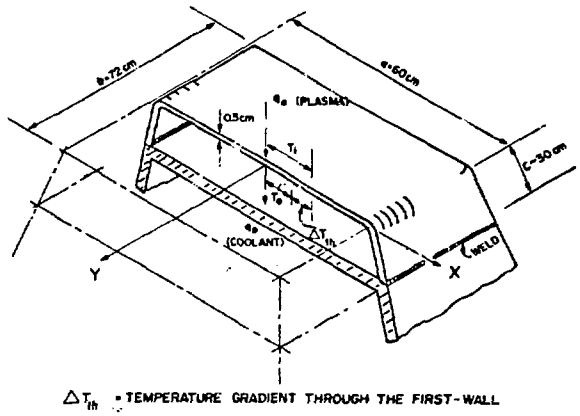

Figure 5 - Idealized Blanket Structural Model

In such a thin-wall vessel, two categories of thermal stresses are produced under a non-uniform distribution of temperature. In the first category, thermal stresses are produced by a temperature gradient through the wall thickness. Because the distribution of the thermal stresses is in proportion to the distance from the middle surface of the wall, if the temperature gradient varies linearly, the net effect is a bending moment.

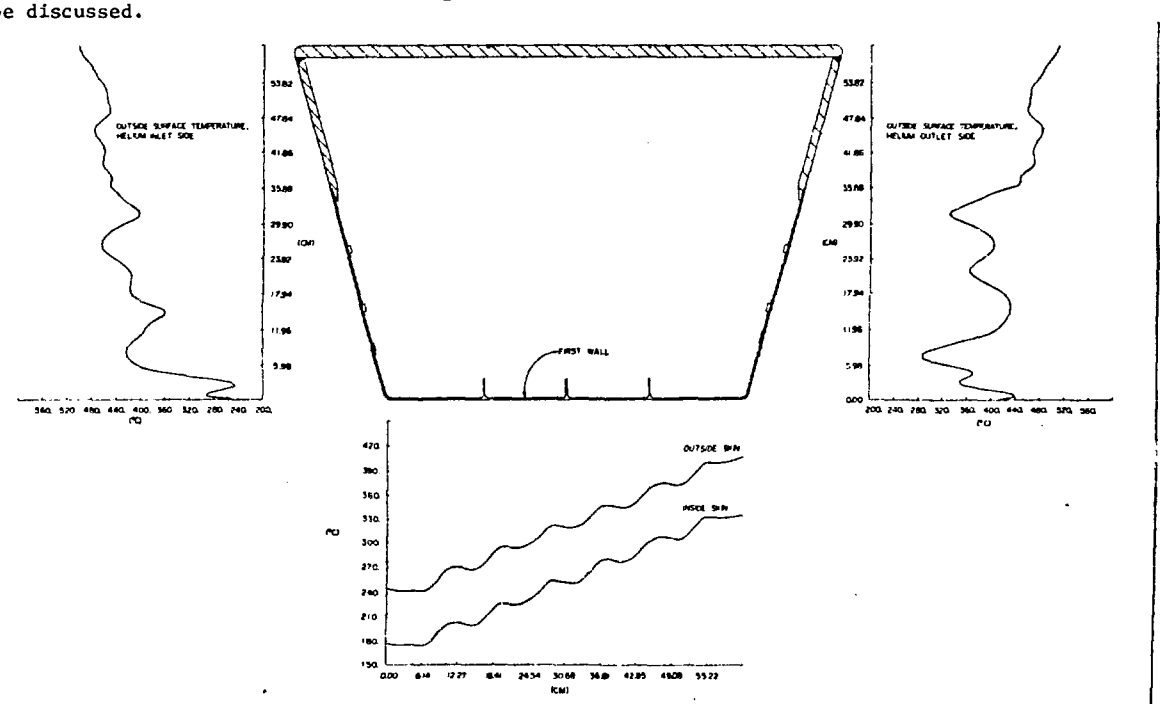

Figure 4 - Steady State Temperature Distribution in the Blanket Bucket Structure at a Neutron Wall Loading of 1.4 MW/m^2.

Thermal stresses in the second category are closely associated with differential thermal expansion in the structure. In a thin wall structure, these stresses are assumed to be uniformly distributed over the cross-section of the thickness as membrane stresses.

The distribution of thermal bending stresses subjected to a uniform temperature drops between the inner and outer surfaces of the first wall, as delineated in Fig. 4, results in thermal bending stresses that are reasonably uniform in the central area of the first wall. Partially because of the modeling, a sharp rise in thermal stress occurs at the vertex corner where the maximum stress amounts to about 140 MPa (20,300 psi). Thermal stresses decrease rapidly as the distance from the vertex increases. A reduction in the maximum stress will occur in the actual structure in which corner joints are all blunted. Sharp edges and vertices are thus avoided in the first wall region.

In addition to the thermal bending stresses, the first wall could suffer high thermal membrane stresses incurred by differential thermal expansion in the middle surface of the wall. The severity of the thermal membrane stresses depends in substantial measure on the arrangement of cooling curcuits and the temperature rise in the coolant stream. It can be demonstrated by the theory of elasticity that there will be no thermal stresses occurring in an unconstrained plate, if the distribution of temperature varies linearly everywhere in the plate. Thermal stress occurs where temperature varies from the linear distribution and the magnitude of the stress increases in proportion to the temperature fluctuation. In the region of the free edge of the plate, the thermal stress will relax considerably.

The cooling circuits of the blanket, as shown in Fig. 2, are essentially arranged to minimize the thermal membrane stresses in the first wall. Results shown in Fig. 4 indicate that the changes of temperature are not continuously linear, but are approximately stepwise linear.

Peak stresses occur correspondingly at those locations where the temperature varies most from the average linear temperature line. The maximum stresses occur in the Y direction (σ_y = 31 Mpa = 4500 psi) and are all in the plane that passes through Y=0; the trend of stress relaxation toward the edges of the plate is significant. Peak stresses in the X direction are distributed along the edges normal to the Y-axis.

Similar efforts to obtain the ideal linear temperature in the module side walls was not as successful. Results of thermal hydraulic analysis show that temperature drop across the thickness of the side walls will be negligible, but these walls will experience a rather severe radial distribution of temperature, as shown in Fig. 4. This implies that higher thermal stresses will occur in the side walls. An approximate analysis was made based on the distribution of temperature given in Fig. 4 as an extreme design condition. The maximum thermal membrane stress at the outlet side of the wall will be approximately 116 MPa (16,800 psi) at 430°C and there will be a similar system of thermal stresses occurring on the inlet side at substantially similar temperatures. The side walls are also under the influence of the thermal bending moments from the first wall. The maximum bending stress at the point of the maximum membrane stress in the side wall was found to be 122 Mpa (17,650 psi).

One requirement for the acceptance of a design is that the calculated stress intensities in any parts of components shall not exceed specified allowable limits given by recognized reliable code. The fusion reactor components must at least include the effects of fast neutron irradiation and others. Such codes are unfortunately not in existence. The design analysis studies presented in this section are, therefore, guided by the ASME Boiler and Pressure Vessel Code, Section III,[5] where the maximum temperature is limited to 427°C. For components which are to experience temperature above those now provided for in Section III, ASME Code Case 1592[6] and RDT Standards F9-4T[7] and F9-5T[8] are to be supplemental.

The values of stress intensities derived from the combination of dead weight and internal pressure are shown in Table 2 as $P_m + P_b$. For this stress category, we can safely say that these values are in all cases below the allowable limits. The stress intensities in the stress category of $P_m + P_b + Q$ for the first wall are also below the specified limits with a margin of safety of approximately 2. For the assessment of the stress condition in the side walls, the more stringent rules given in Code Case 1592 may be required because the metal temperature exceeds the allowable limit specified in Section III.

The following is a summary of the stresses and stress intensities derived for three critical points in the blanket module.

TABLE 2

	First Wall		Side Wall
LOCATION	Center	Corner	Outlet Side
TEMP., °C	320	400	431
P_m, psi	150	150	150
P_b, psi	±1650	±1650	±800
Q_1, psi	4500	0	16,800
Q_2, psi	±17,900	±20,300	±17,690
$P_m + P_b$	1800	1800	950
$P_m + P_b + Q$	24,200	22,100	35,400

Where P_m = membrane stress due to dead weight and pressure,

P_b = bending stress due to dead weight and pressure,

Q_1 = thermal membrane stress, and

Q_2 = thermal bending stress

Fusion power reactors must be guarded from thermal cyclic fatigue and radiation damage. For concepts with short burn times, high cycle lifetime is required. Others with longer burn times, the interaction of creep and fatigue is important. Based on the fatigue design curves that are currently being proposed[9] for inclusion in Section III, the allowable stress intensity of 316 stainless steel for cyclic lifetime 4 to 5 x 10^5 cycles is approximately 345 MPa (50,000 psi). Again, this will provide some degree of safety margin to cover uncertainties concerning the radiation effects on fatigue lifetime.

Conclusions

A blanket using stainless steel structural material, metalic lithium as fertile material and high pressure helium coolant can be used for toroidal fusion reactors up to about 2 MW/m^2 wall loading.

For these loadings, temperatures, thermal stresses and pumping powers are acceptable. Fabrication by standard practices is possible. Modular construction permits fabrication and testing of reasonable sized units.

For tokamak application, the effect of thermal cycling on wall life indicates that 4 to 5 x 10^5 cycles are acceptable.

The segmented blanket, when used in conjunction with a vacuum building permits remote maintenance to be effected with acceptable difficulty.

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