Design optimization of first wall and breeder unit module size for the Indian HCCB blanket module

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Received 2 November 2017, revised 14 February 2018
Accepted for publication 27 February 2018
Published 2 May 2018

Abstract
The Indian test blanket module (TBM) program in ITER is one of the major steps in the Indian fusion reactor program for carrying out the R&D activities in the critical areas like design of tritium breeding blankets relevant to future Indian fusion devices (ITER relevant and DEMO). The Indian Lead–Lithium Cooled Ceramic Breeder (LLCB) blanket concept is one of the Indian DEMO relevant TBM, to be tested in ITER as a part of the TBM program. Helium-Cooled Ceramic Breeder (HCCB) is an alternative blanket concept that consists of lithium titanate ($\text{Li}_2\text{TiO}_3$) as ceramic breeder (CB) material in the form of packed pebble beds and beryllium as the neutron multiplier. Specifically, attentions are given to the optimization of first wall coolant channel design and size of breeder unit module considering coolant pressure and thermal loads for the proposed Indian HCCB blanket based on ITER relevant TBM and loading conditions. These analyses will help proceeding further in designing blankets for loads relevant to the future fusion device.

Keywords: first wall, blanket, breeder unit, thermal hydraulics, structural analysis, HCCB (helium-cooled ceramic breeder)

(Some figures may appear in colour only in the online journal)

1. Introduction
India has proposed the lead–lithium cooled ceramic breeder (LLCB) as the blanket concept to be tested in ITER [1]. The helium-cooled ceramic breeder (HCCB) is an alternative Indian solid breeder blanket concept for future fusion devices (like ITER and DEMO) [2, 3]. In this paper, the HCCB blanket based on ITER relevant test blanket modules (TBMs) has been proposed and analyzed. It is manufactured from Indian reduced activation ferritic/martensitic steel (IN-RAFMS), and uses lithium titanate ($\text{Li}_2\text{TiO}_3$) as the tritium breeder, and beryllium as the neutron multiplier. The Indian HCCB blanket concept shown in figure 1 is a modular one, having overall dimensions of $1.67 \text{ m} \times 0.462 \text{ m} \times 0.63 \text{ m}$ (poloidal × toroidal × radial). The modular structure consists of breeder unit modules, and is formed by attaching grid plates to the first wall (FW). The FW, grid plates, and top and bottom plates form cavities that accommodate the breeder units and provide structural strength to the blanket in case of accidental leak of high-pressure (8 MPa) helium gas from the cooling plates or FW into the breeder unit module. The size of the breeder unit has been optimized using different configurations so as to withstand such accidental loading.

The FW, as shown in figure 1, is a component that faces the plasma, with helium flowing inside the channels to extract the incident heat flux [4] from the plasma and heat deposit on the FW structure [2]. It is a U-shaped structure made of IN-RAFMS material, having internal coolant channels with high-pressure (8 MPa) and high-temperature ($300 ^\circ \text{C}$) helium flowing to cool the structure. The study of FW coolant channel optimization consists of analyzing helium flow with different sizes and shapes of the channel, number of flow circuits, mass flow rate, and corresponding thermal performance and hydraulic parameters. A comparison between thermo-mechanical stresses of square and
An optimization study has been done using square/circular channels 15 and 20 mm in hydraulic diameter. A comparison between thermo-mechanical stresses of square and circular channels of the same hydraulic diameter has also been done.

2.1. Parameters for FW design optimization

The various parameters required for optimization of FW design, namely the HTCs, bulk temperatures, pressure drop, and FW temperatures for various flow parameters using the correlations and analysis defined below.

2.1.1. HTCs and bulk temperature calculation. HTCs are calculated using the Gnielinski equation [7, 8] given below:

\[ Nu = \frac{1}{x} \frac{\xi (Re - 1000) Pr}{1 + 12.7 \left[ \frac{1}{8} (Pr^{2/3} - 1) \right]}, \]

\[ \xi = \frac{1}{(1.82 \log Re - 1.64)^2}, \]

where \( Nu \) is the Nusselt number, \( Pr \) is the Prandtl number, \( Re \) is the Reynolds number, and \( \xi \) is the friction factor.

The Prandtl number for the helium is \( \sim 0.66 \). The Nusselt number has a range of validity with \( 0.5 < Pr < 1.5 \) and \( 2300 < Re < 10^6 \), and HTCs are estimated using the calculated Nusselt number. The outlet helium temperature is calculated using the energy balance equation considering head loads of 0.3 MW m\(^{-2} \) [4], heat flux on the plasma-facing surface, and 5 MW m\(^{-3} \) [2] of heat generation on the FW structure.

2.1.2. Pressure drop calculation. The pressure drop is calculated from correlations and data given in the Handbook of Hydraulic Resistance [9]. The pressure drop is calculated in one circuit based on the number of channels per circuit and number of bends per circuit. There are two types of pressure drop:

- Frictional pressure drop (\( \Delta P \)) due to the flow length of a circuit:

\[ \Delta P = \frac{\rho f v^2}{2D}, \]

where \( \rho \) is the helium density, \( f \) is the friction factor, \( l \) is the channel length, \( v \) is the fluid velocity, and \( D \) is the hydraulic diameter of the channel;

- Pressure loss (\( \Delta P \)) due to smooth and sharp bends in a circuit:

\[ \Delta P = \frac{K \rho v^2}{2}, \]

where \( K \) is the concentrated loss factor calculated from the
data and correlation in [9].

For smooth bends:
\[ K = (A1 \ast B1 \ast C1) \]
\[ + 0.0035 \ast \frac{R0}{D} \ast \delta^o \]  
(5)

A1, B1, and C1 are angle-dependent functions given in \( R0 \), the curvature radius, \( D \) is the section’s hydraulic diameter, and \( \delta^o \) is the angle in degrees.

For sharp bends:
\[ K = C1 \ast A \ast \zeta 1 \]  
(6)

C1, A, and \( \zeta 1 \) are functions based on angle.

Smooth bends are U-shaped bends in the channels of the FW structure, and sharp bends are present at the back where the two channels are interconnected in a circuit. Based on the correlations given, the pressure drop in the circuit is approximated by adding a frictional pressure drop and pressure drop due to bends.

2.1.3. FW temperature calculation using ANSYS. The FW temperature calculation is performed using ANSYS [10]. The 2D single channel ANSYS models for different configurations are shown in figure 2. The calculated HTCs and corresponding outlet temperatures, as discussed in section 2.1.1 and considered conservatively as bulk temperature, are used as boundary conditions along with thermal loads. The IN-RAFMS material properties are used in the analysis.

2.2. Inputs and consideration for optimization study

The inputs are mass flow rates (0.8, 1, 1.2, 1.4, 1.6 kg s\(^{-1}\)) and the number of circuits (16, 13, 11, 9, 7). For comparison, the mass flow rate and number of circuits are kept the same for each configuration. The total number of channels are varied for 15 and 20 mm channel sizes. This is because the total FW height is fixed. The number of channels in one circuit is varied according to the number of circuits to get close to the total number of channels. The total number of channels in the 15 and 20 mm channel cases ranges from 77–84 and 63–66, respectively. The range difference can be accommodated by adjusting the ligament width between the two channels.

2.3. Calculation results and observations

The HTC, FW temperature, and pressure drop in each circuit of the FW for all four types of channel configurations corresponding to mass flow and the number of circuits is shown in figure 3 below.

The range of velocity for the 20 mm square and circular channels is 20–116 m s\(^{-1}\), and the Reynolds number ranges from \( 7.7 \times 10^4 \) to \( 1.96 \times 10^5 \). The range of velocity for the 15 mm square and circular channel configurations is 35–206 m s\(^{-1}\), and the Reynolds number ranges from
1.03 × 10^5 to 5.97 × 10^5. The outlet temperatures for all configurations are given in Table 1.

As shown in Figure 3, for a particular mass flow, the HTC and pressure drop increases with a decreasing number of circuits; meanwhile, the temperature decreases because as the number of circuits decrease the mass flow per circuit increases. The pressure drop increases because of the increase in the number of bends and flow length for one circuit. With a similar mass flow and number of circuits, the 15 mm channel configuration has lower temperatures compared to the 20 mm configuration. We can see that with the 1 kg s⁻¹ and 16 circuits configuration for the 15 mm channel (both square and circular), the temperature is below 500 °C, which is well below the allowed temperature of 550 °C with sufficient margins. The pressure drop is also lower than 0.25 MPa.

The corresponding velocities in square (15 × 15 mm²) and circular (15 mm diameter) channels are 44 m s⁻¹ and 56 m s⁻¹, respectively. The maximum FW temperature is 490 °C and 497 °C for square (15 × 15 mm²) and circular (15 mm diameter) channels, respectively. The circular channel has a higher HTC, but its temperature is still higher because of the higher relative distance from the plasma-facing surface and due to the higher material volume. The pressure drop is estimated to be 0.13 and 0.21 MPa for the square and circular channels, respectively. Thus, both square (15 × 15 mm²) and circular (15 mm diameter) channels with 16 circuits and a mass flow of 1 kg s⁻¹ are optimized configurations of the FW. The outlet temperature for both configurations is around 360 °C. The 2D analysis results of these optimized configurations is shown in figure 4 below.

**Table 1.** Helium outlet temperatures for different configurations.

<table>
<thead>
<tr>
<th>Mass flow (kg s⁻¹)</th>
<th>Outlet Temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>20 × 20 mm²</td>
</tr>
<tr>
<td>0.8</td>
<td>369</td>
</tr>
<tr>
<td>1</td>
<td>355</td>
</tr>
<tr>
<td>1.2</td>
<td>346</td>
</tr>
<tr>
<td>1.4</td>
<td>339</td>
</tr>
<tr>
<td>1.6</td>
<td>335</td>
</tr>
</tbody>
</table>

Figure 3. Variation of HTC, FW temperature, and pressure drop with different configurations.
2.4. 3D thermal and thermo-structural analysis of the optimized configuration of the FW for 15 x 15 mm$^2$ square and 15 mm diameter circular channels

3D thermal and thermo-structural analysis of the optimized FW configuration using ANSYS are performed. Two models each for square (15 x 15 mm$^2$) and circular (15 mm diameter) configurations are prepared for the analysis. Here, the symmetry of the model in the radial poloidal (Y-Z) plane is used, and half of the FW box structure is modeled as shown in figure 5. The HTC and bulk temperature evaluated for the optimized configuration along with heat loads (heat flux and heat generation) are used for the 3D thermal analysis. For the thermo-structural analysis, we consider a pressure load of 8 MPa, a temperature distribution obtained from the thermal analysis, and constraints like fixing the symmetrical plane in the normal (X) direction and fixing the back side of the FW in the radial (Z) direction.

The temperature profile of the FW with square (15 x 15 mm$^2$) (maximum: 497°C) and circular (15 mm diameter) (maximum: 503°C) configurations is shown in figure 6.

The results show that the circular configuration has a higher FW temperature (503°C) due to more distance from the flowing helium surface. However, both temperatures are within the limits with sufficient margins. The results match with the 2D analysis results in section 3. Table 2 shows the maximum von Mises stress due to pressure, thermal, and combined loadings.

The von Mises stress and total deformation plots due to combined loading for square and circular channel configurations are shown in figures 7 and 8, respectively. The maximum stress in square (15 x 15 mm$^2$) and circular (15 mm diameter) configurations due to combined loading is 372 and 343 MPa, respectively. The maximum deformation is around 4 mm in both cases.

The temperature in the circular channel is higher than in the square channel, but the thermal stress is lower (343 MPa compared to 358 MPa). It is seen that bending stresses are higher in the square channel configuration due to the total stress slightly increasing. The stress difference is relatively small between square and circular channels in the frontal FW region. The results in table 2 show that both channel configurations show similar structural performance. Furthermore, the maximum von Mises stresses are less than the $S_m$ and 1.5$S_m$ limits for pressure loads and 3$S_m$ limits for combined loading [11], and thus stress linearization is not necessary. The maximum stress occurs at the center of the poloidal height of the FW bends.
Breeder unit modules are formed by attaching grid plates to the FW. The FW, grid plates, and top and bottom plates form cavities that accommodate breeder units. The grid plates provide structural strength to the blanket in case of accidental leak of high-pressure (8 MPa) helium gas from the cooling plates into the breeder module. For this analysis, various configurations are

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Pressure load maximum von Mises (MPa)</th>
<th>Thermal load maximum von Mises (MPa)</th>
<th>Combined load maximum von Mises (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Square (15 × 15 mm²)</td>
<td>56</td>
<td>358</td>
<td>372</td>
</tr>
<tr>
<td>Circular (15 mm diameter)</td>
<td>37</td>
<td>343</td>
<td>343</td>
</tr>
</tbody>
</table>

3. Optimization of breeder module unit size [12]

Breeder unit modules are formed by attaching grid plates to the FW. The FW, grid plates, and top and bottom plates form cavities that accommodate breeder units. The grid plates provide structural strength to the blanket in case of accidental leak of high-pressure (8 MPa) helium gas from the cooling plates into the breeder module. For this analysis, various configurations are
obtained by changing the number of grid plates while keeping the outer dimensions constant; thus, we get the optimized breeder unit dimensions based on structural analysis. This optimization helps reduce the structural material and thus increase the breeding material. The analysis is performed by applying internal pressure on the inside surface of the cavity for one of the breeder units and by checking for stresses in ANSYS. The stress on the FW and grid plates should be within the acceptable limits for In-box loss of coolant accident (LOCA) criteria (Category IV event) [4] inside the breeder unit so as to maintain integrity of the blanket module during an accident.

3.1. Analysis with different configurations

Different configurations of breeder modules are obtained by dividing toroidally and poloidally using grid plates. Initially, four configurations consisting of $4 \times 2$, $5 \times 2$, $6 \times 2$, and $7 \times 2$ numbers of breeder unit modules are analyzed. The ANSYS model of one such configuration is shown in figure 9.

For simplification of optimization, instead of modeling channels in FW, an equivalent thickness considering the volume of material in FW has been used. A pressure load of 8 MPa is applied in one of the cavities of the breeder unit since the accident is anticipated in one module only. The back side of the module is given a fixed boundary condition.

Von Mises stress plots for the four different configurations are shown in figure 10, and the maximum values are tabulated in table 3.

The location of maximum stress is the corner of grid plate to FW joint. In-box LOCA is considered to be a category IV event, and is classified under the level D criteria of RCC-MR 2007 [11] design rules.

The following criteria are to be satisfied as per RCC-MR:

- Primary membrane stress: $P_m \leq S_m^d$;
- Primary local membrane stress: $P_l \leq 1.5S_m^d$;
- Primary membrane plus bending stress: $P_m + P_b \leq 1.5S_m^d$;
- $S_m^d = \min \{2.4S_m, 0.7R_m \}$, $S_m = 141$ MPa, $R_m = 376$ MPa at $550^\circ$C, $S_m^d = 263$ MPa, and $1.5 S_m^d = 394$ MPa.
Stress linearization has been performed to estimate the membrane and membrane plus bending stress at the center of the grid plate and at the corner of the FW–grid plate corner joint, as shown in figure 11. The paths are located at the radial center of the box structure.

Tables 4 and 5 show the stress linearization results along with safety margins for stresses at the middle of the grid plate and at the FW–grid plate corner joint, respectively. The stresses for 6 × 2 and 7 × 2 configurations are within the required limits.

As a next step toward optimization, an attempt with configurations avoiding vertical grid plates was made with a higher thickness of horizontal grid plates. This configuration has the advantage of simplifying the flow mechanism in manifolds because of the removal of vertical grid plates, which require active cooling. The thickness required for a single horizontal grid plate of higher thickness is calculated using formulas for stress on flat plates, which are given in [13]. It is found that a 35 mm thick horizontal grid plate is sufficient to withstand the stress. The 6 × 1 configuration is selected for the structural analysis to estimate the stress levels. The von Mises stress plot for this configuration is shown in figure 12, which reveals the maximum von Mises stress of 560 MPa.

Stress linearization is performed to estimate the membrane and membrane plus bending stress at the center of the grid plate and at the corner of the FW–grid plate corner joint. Table 6 shows the stress linearization results along with safety margins for stresses at the middle of the grid plate and at the FW–grid plate corner joint. The stresses for the 6 × 1 configuration are within the required limits.

The results show that stresses in 7 × 2 and 6 × 2 configurations of the breeder unit module are within limits for INRAFMS during a category IV accident, with sufficient margins. From analytical analysis it was found out that by using higher grid plate thickness of 35 mm instead of 20 mm, we can remove the vertical grid plate and change from 6 × 2 to 6 × 1 configuration. The volume of structural material in the 6 × 2 configuration is 0.098 m³, and 0.102 m³ in the 6 × 1 configuration; these are very similar. Hence, it is recommended to use the 6 × 1 configuration with increased grid plate thickness instead of the 6 × 2 configuration since this reduces the number of breeder units and simplifies the flow.
mechanism because of the removal of vertical grid plates, which require active cooling. The breeder unit module will be designed according to the new optimized dimensions.

3.2. Structural analysis of the $6 \times 1$ breeder module configuration with actual FW and grid plate channels

Here we analyze the $6 \times 1$ breeder module configuration with actual FW and grid plate channels instead of using an equivalent FW thickness for accurate stress evaluation. The von Mises stress plot is shown in figure 13, and the total deformation is shown in figure 14. The maximum von Mises stress is 495 MPa, and the maximum deformation is 0.7 mm.

Stress linearization is performed to estimate the membrane and membrane plus bending stress at critical and high stress locations, as shown in figure 15. All paths are located at the radial center of the box structure.

The stress linearization results along with safety margins for the chosen paths are shown in table 7. All paths show stress within the required limits.

**Table 4. Stress linearization at the middle of the grid plate.**

<table>
<thead>
<tr>
<th>Configuration</th>
<th>4 × 2</th>
<th>5 × 2</th>
<th>6 × 2</th>
<th>7 × 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress classification</td>
<td>$P_m$</td>
<td>$P_m + P_b$</td>
<td>$P_m$</td>
<td>$P_m + P_b$</td>
</tr>
<tr>
<td>Value (MPa)</td>
<td>25</td>
<td>555</td>
<td>27</td>
<td>418</td>
</tr>
<tr>
<td>Safety margins</td>
<td>10.5</td>
<td>0.7</td>
<td>9.7</td>
<td>0.94</td>
</tr>
</tbody>
</table>

**Table 5. Stress linearization at the FW–grid plate corner.**

<table>
<thead>
<tr>
<th>Configuration</th>
<th>4 × 2</th>
<th>5 × 2</th>
<th>6 × 2</th>
<th>7 × 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress classification</td>
<td>$P_l$</td>
<td>$P_l + P_b$</td>
<td>$P_l$</td>
<td>$P_l + P_b$</td>
</tr>
<tr>
<td>Value (MPa)</td>
<td>130</td>
<td>580</td>
<td>70</td>
<td>403</td>
</tr>
<tr>
<td>Safety margins</td>
<td>3.03</td>
<td>0.68</td>
<td>5.62</td>
<td>0.97</td>
</tr>
</tbody>
</table>

**Table 6. Stress linearization for the $6 \times 1$ configuration.**

<table>
<thead>
<tr>
<th>Location</th>
<th>Middle of grid plate</th>
<th>FW–grid plate corner</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress classification</td>
<td>$P_m$</td>
<td>$P_m + P_b$</td>
</tr>
<tr>
<td>Value (MPa)</td>
<td>22</td>
<td>241</td>
</tr>
<tr>
<td>Safety margins</td>
<td>11.9</td>
<td>1.63</td>
</tr>
</tbody>
</table>

**Figure 12.** Von Mises stress plot for the $6 \times 1$ configuration.

**Figure 13.** Von Mises stress plot.
4. Summary and conclusions

An analytical calculation using various mass flow rates and different FW channel configurations is performed. The HTCs of FW channels are calculated by varying the mass flow and number of circuits. With these HTCs, the maximum temperature of the FW are calculated using ANSYS. Similarly, the pressure drop is also determined analytically for the different flow rates and number of circuits. The thermal efficiency of the blanket module can be increased by lowering the mass flow from the present 1.6 kg s\(^{-1}\) to 1 kg s\(^{-1}\). Acceptable thermal performance with this flow can be achieved by using a square or circular 15 mm channel. The number of circuits should be 16, with five channels in each circuit. The advantage of having a circular channel cross section over a square cross section is mainly due to its ease of fabrication. The circular channel can be easily fabricated through gun drilling or deep drilling. However, the circular channel gives slightly higher temperatures (~10 °C) compared to the square channel of the same parameters. The circular channel also gives a higher velocity and higher HTC, which is another advantage of this configuration. Both configuration have the FW temperature below the limits. Furthermore, the structural performance of both configurations is similar, and stresses are within the acceptable limits.

The optimization study of breeder module size is performed to find the suitable configuration for the breeder box structure. The structural analysis shows that the 6 × 1 configuration with the required grid plate thickness is sufficient to withstand an in-box LOCA with respect to category IV events as per RCC-MR design criteria. The optimized dimensions of a single breeder unit module are 237.5 mm poloidally, 402 mm toroidally, and 480 mm radially, with a 35 mm grid plate thickness. This reduces the number of breeder units to 6(6 × 1) compared to 10(5 × 2) in the present design [2], and also simplifies the flow mechanism because of the removal of vertical grid plates, which require active cooling. The effect of such a configuration on tritium breeding and other loads, including electromagnetic and seismic loads, will be studied in future work.

Acknowledgments

This work has been performed under the Institute for Plasma Research (IPR), University of California, Los Angeles (UCLA)
collaboration. We acknowledge Dr Alice Ying, UCLA, for her valuable comments and suggestions during this simulation work and preparation of the manuscript. We also acknowledge E Rajendrakumar, IPR, for his continuous support in this work.

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