The Effect of Multilayer Insulation on Thermal Loading in DEMO Systems

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ABSTRACT

This paper discusses the radiative heat exchange between major components of the DEMO tokamak. Since the additional multilayer insulation at the warm side of the thermal shields can substantially reduce the heat load to the magnets and thermal shields themselves, different shielding configurations have been investigated, including those with a passive multilayer insulation. Numerical analysis shows that excessively high thermal loads on the magnet system can only be avoided if the magnets’ thermal shielding is actively cooled, regardless of the additional insulation layers. The discussion is supported by a simplified theoretical model which is in good agreement with the numerical predictions.

1 INTRODUCTION

The realization of the fusion electricity by 2050 requires the development of a demonstration fusion power plant (DEMO) [1]. As the only step between ITER and a commercial fusion power plant, the DEMO will have to demonstrate the stable long term operation with the net electricity production of few hundred MWs. To support the development of a consistent design, a thermal radiation analysis of the global DEMO tokamak has been carried out at the “Jožef Stefan” Institute (JSI) within the “System level analysis - Global tokamak thermal analysis” project. The objective of the analysis is to evaluate the radiative heat exchange between the tokamak components due to their very different operational temperatures and, additionally, the thermal loading on the individual components. This is especially important for the systems with low operational temperatures, such as superconducting magnets, since high refrigerating power may significantly decrease the overall efficiency of the DEMO device.

The DEMO tokamak geometry and components are first presented in Section 2. The numerical aspects concerning the error of the radiation heat exchange simulation are given in Section 3. This section also presents a theoretical model of radiation heat exchange used to validate the numerical simulations. The operational temperatures of the components and the strategy employed to minimize the numerical errors are then given in Section 4.

In Section 5, the results of the simulations and theoretical analyses are presented. This section shows, in first place, the important reduction of the simulation errors with respect to previous attempts [2], which has been achieved through a mesh refinement and improved numerical setup. Also, in Section 5, this improved DEMO numerical model is used to investigate different shielding configurations, including those with passive Multi-Layer Insulation (MLI) placed next to completely passive cryostat thermal shield. The steady-state
numerical analysis is supported by the theoretical analysis. The conclusions are drawn in Section 6.

2 TOKAMAK GEOMETRY

The development of the DEMO tokamak model, shown in Fig. 1, started in 2014 at JSI [2]. The geometry of certain components such as vacuum vessel thermal shield (VVTS), cryostat thermal shield (CTS) and cryostat (CRY) are slightly simplified with respect to the real one to carry out the required analyses. Due to an axial (toroidal) symmetry of the tokamak geometry, only 1/18th (20 degrees) of the tokamak geometry is modelled. Further information on the geometry simplifications is available in Ref. [2]. The numerical model comprises the following components (see Fig. 1) listed from the outside-in:

- Cryostat (CRY): thickness of 60 mm, surface area of 307 m².
- Cryostat thermal shield (CTS): thickness of 60 mm, surface area of 231 m².
- Magnet system, including six poloidal field coils (PFC), toroidal field coils (TFC) and central solenoid (SOLENOID).
- Vacuum vessel thermal shield (VVTS): thickness of 40 mm, surface area of 412 m².
- Vacuum vessel with four ports (VV): thickness of 50 mm, surface area of 463 m².
- In-vessel components: Blanket (BLA) and divertor (DIV): surface area of 215 m².
- Multi-layer Insulation (MLI): thickness of 15 mm, surface area of 230 m².

![Figure 1: Global tokamak model.](image)

3 THERMAL RADIATION ANALYSIS

3.1 Numerical simulation

The numerical simulations have been performed with the Finite Element (FE) solver code ABAQUS [3]. In the ABAQUS code, the radiation heat exchange is modeled with the cavity thermal radiation model [3]. Here, cavities are represented as collection of surfaces, which are composed by elemental surfaces, named as facets. The exchange of radiation
between two or more surfaces depends strongly on the surface geometries and orientations, as well as on their radiative properties and temperatures. It is assumed that the energy exchange takes place in a perfect vacuum where no scattering, absorption and emission in the gas affect the transfer of the thermal radiation energy between the solid surfaces. For most gases (also for ionized plasma) this is a good approximation, and is also used in this analysis. It should be noted that in the current thermal model the effect of hot ionized plasma is taken into account indirectly by the temperature of in-vessel surfaces whereas the vacuum is present in the cryostat region.

In the cavity radiation formulation [3], the radiation flux per unit area, $q$, into a cavity facet is defined as

$$ q_i = \sigma e_i \sum_{j=1}^{N} e_j \sum_{k=1}^{N} F_{ik} C^{-1}_{kj} \left( (T_j - T^Z)^4 - (T_i - T^Z)^4 \right), $$

where $N$ represents the number of facets forming the cavity, $e_i$ and $T_i$ are, respectively, the emissivity and temperature of facet $i$, $\sigma=5.67 \times 10^{-8}$ W/m$^2$K$^4$ is the Stefan-Boltzmann constant, $T^Z$ represents the absolute zero on the temperature scale used, $F_{ij}$ is the geometrical view factor matrix and $C_{ij} = \delta_{ij} - (1 - e_i) F_{ij}$ is the reflection matrix. Numerically, the radiative heat exchange between two arbitrary surfaces is defined by the concept of view factors [4], presented in Fig.2.

Figure 2: Elemental surfaces of area $dA_i$ and $dA_j$ separated by a distance $r$. Polar angles $\theta_i$ and $\theta_j$ with the surface normal $n_i$ and $n_j$ are used in the calculation of view factors $F_{ij}$.

Adopted from Ref. [4].

The view factor $F_{ij}$, defined as the fraction of the radiation leaving surface $i$ that is intercepted by surface $j$, is calculated as follows [4]:

$$ F_{ij} = \frac{1}{A_i} \int_{A_i} \int_{A_j} \frac{\cos \theta_i \cos \theta_j}{\pi R^2} dA_i dA_j. $$

For Eq. (2) applies reciprocity relation $A_i F_{ij} = A_j F_{ji}$, which is useful in determining one view factor from the other. For enclosed cavity problems, where all radiation leaving surface $i$ must be intercepted by the enclosure surfaces, it follows $\sum_{j=1}^{N} F_{ij} = 1$ [4]. Therefore, the total radiation power exchange between all systems in a steady-state closed cavity simulation should be equal to zero.
\[ \sum Q_i = 0 \] (3)

where \( Q_i \) denotes the net radiation power gained/lost for component \( i \). In practice, i.e. numerical simulations, this is not completely true due to numerical errors arising from the spuriously calculated view factors. For every facet with its view factor outside the pre-described tolerance limit (in our case the tolerance is \( \pm 0.05 \)), the heat exchange with an ambient is assumed. Therefore, one needs to specify the ambient temperature, \( T_{amb} \), which should be set to the approximate temperature of the surroundings. The relative error of the simulation is defined as a ratio between the net radiation power and the total power exchanged in a system:

\[ \text{Relative error} = \frac{\sum_i Q_i}{\sum_i |Q_i|}/2. \] (4)

### 3.2 Theoretical Model

A simplified theoretical analysis is conducted to validate the simulation results. This model is very convenient for investigations of the effects of multiple MLIs, since it is mathematically simple to solve. In the theoretical model, the tokamak components are approximated by infinitely large, parallel, gray surfaces. This is a reasonable assumption because the actual tokamak components exhibit relatively small curvatures, are closely aligned, and have similar topology.

The net heat flux between two infinite parallel plates at temperature \( T_1, T_2 \) with emissivities \( \varepsilon_1, \varepsilon_2 \), respectively, is given by [5]:

\[ Q / A = \sigma (T_1^4 - T_2^4) \left( \frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1 \right). \] (5)

Equation (5) takes into account the infinite number of reflections between the plates. For a completely passive component in a steady-state, the net thermal load loss is zero, meaning that emitted (\( Q_{out} \)) and absorbed (\( Q_{in} \)) heat fluxes are equal, which allows us to calculate the component’s temperature.

The schematic of theoretical model of a system including actively-cooled (AC) and passive (PA) components is presented in Fig. 3. In this study, four cases were analyzed: i) the case where all components are actively cooled, ii) the case where only the CTS is passive, iii) the case where passive MLI is placed next to a passive CTS, and iv) the case where two passive MLIs are placed next to a passive CTS. In the theoretical model the attachment joints between the shields, MLIs and tokamak components are neglected, therefore only thermal radiation heat transfer is taken into account.
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The refrigeration power, $P^\text{refrig.}_i$, required for keeping an actively-cooled component $i$ at its predetermined temperature $T_i$ can be estimated based on ideal Carnot Cycle as follows:

$$P^\text{refrig.}_i = \frac{Q_\text{in} - Q_\text{out}}{A_i} \left( \frac{T_{\text{amb}} - T_i}{T_i} \right),$$

where $A_i$ is the cross-sectional area of $i$-th component. The total refrigeration power of a system, $P_\text{tot}$, is defined as the sum of individual powers for VVTS, MAG and CTS (if it is actively-cooled), and is given by the following equation:

$$P_\text{tot} = \sum P_i.$$  

4 SIMULATION MODEL

Thermal radiation analysis is conducted numerically using the FE solver ABAQUS [3]. Since only 1/18th (20 degrees) of the entire geometry is modeled (see Fig. 1), a cyclic boundary condition is used to achieve a closed model. The temperature independent material properties and operational temperatures for individual components are summarized in Table 1. In a steady-state simulation with only actively-cooled components at predetermined operational temperature only the emissivity is required, while for a fully passive component where heat conduction takes place one needs to specify also the thermal conductivity, $\lambda$. A single passive MLI package consists of a 15 mm thick shell, with thermal conductivity equal to 0.1 W/m\cdot K and emissivity of 0.05 [6].

<table>
<thead>
<tr>
<th>Component</th>
<th>T (K)</th>
<th>Material</th>
<th>Emissivity [6]</th>
<th>Thermal Conductivity (W/m\cdot K) [7]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Magnets</td>
<td>4</td>
<td>SS-316</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>CTS</td>
<td>80</td>
<td>SS-304</td>
<td>0.05</td>
<td>14.8</td>
</tr>
<tr>
<td>VVTS</td>
<td>80</td>
<td>SS-304</td>
<td>0.05</td>
<td>16.2</td>
</tr>
<tr>
<td>VV and LID</td>
<td>473</td>
<td>SS-316</td>
<td>0.25</td>
<td>-</td>
</tr>
<tr>
<td>CRY</td>
<td>293</td>
<td>SS-304</td>
<td>0.25</td>
<td>-</td>
</tr>
<tr>
<td>BLA</td>
<td>573</td>
<td>F82H-Eurofer</td>
<td>0.05</td>
<td>0.1</td>
</tr>
<tr>
<td>DIV</td>
<td>573</td>
<td>F82H-Eurofer</td>
<td>0.05</td>
<td>0.1</td>
</tr>
<tr>
<td>MLI</td>
<td>-</td>
<td>-</td>
<td>0.05</td>
<td>0.1</td>
</tr>
</tbody>
</table>
Most of the tokamak geometry is meshed with hexahedral elements. The mesh was systematically built from the inner side of the model towards outside, starting with the in-vessel components. Tetrahedral elements are used only in the vacuum vessel regions attached to the ports.

The analysis case where all components are actively cooled comprises three separate closed cavities within the tokamak geometry that do not interact, Figure 4. Therefore, the tokamak geometry could be decomposed into three independent sub-models shown in Figure 4, denoted as Model 1, 2 and 3, respectively, that form separate closed cavities:

- Model 1: Region between VVTS and CTS, including magnet systems.
- Model 2: In-vessel components (DIV and BLA) and inner surface of the VV.
- Model 3: Region between VV and VVTS and region between CTS and CRY.

Figure 4: Three independent sub-models (Model 1, Model 2, Model 3)

This is very convenient to perform grid refinement of the model meshes and setup of appropriate ambient temperature. Numerical errors are minimized through the refined model meshes, however, similar results are obtained with the analyses performed with the so-called “separate” and the “full” model approaches.

5 RESULTS

5.1 Mesh refinement study

Meshing process turned out to be the most important step in the model development as it has the strongest impact on the simulation errors. The number of facets with erroneous view factors can be substantially reduced if the mesh topology is adapted with the model geometry. Therefore, all sharp edges of the opposite facing components are projected to the surfaces which are being meshed. By doing so, the facet cannot interact with both accompanying surfaces next to the sharp edge, which has been identified as one of the reasons for the erroneous view factor. For example, all sharp edges of the magnet systems were projected to VVTS outer surface as shown in Fig. 5.
The results of the mesh refinement study for Model 2 in Fig. 4 are presented in Table 2. It can be seen that only 1.3% of total surface area with view factors outside the tolerance limits results in the total energy imbalance of 7% (Case 1). With edge projections and grid refinement, the error is reduced to 5.3% (Case 2). Additional mesh refinement in critical areas is required to reduce the error below 4% (Case 3). Moreover, the error is substantially decreased when the ambient temperature is set to somewhat more realistic value (Case 4). Operational temperatures of the vacuum vessel (VV) and in-vessel components (BLA and DIV) are, respectively, 473K and 573K. The ambient temperature is thus set to 523K (arithmetic average) reducing the error from previous 3.4 % to 0.6 %.

![Vacuum vessel thermal shield with detailed mesh (left). TFC (blue) in close proximity of the VVTS (red) (right).](image)

Figure 5: Vacuum vessel thermal shield with detailed mesh (left). TFC (blue) in close proximity of the VVTS (red) (right).

The same approach is used in the grid refined of sub-Models 1 and 3 in Fig 4. Finally, the ambient temperature for Model 1 is set to 80K and for Model 2 to 293K. The corresponding simulation errors are 4.3% and 2.8%, respectively. The total energy imbalance for the so-called “full” model approach is 4.8%.

<table>
<thead>
<tr>
<th>Model 2</th>
<th>Error %</th>
<th>Error surfaces %</th>
<th>Improvements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1</td>
<td>7.0</td>
<td>1.3</td>
<td>$T_{amb} = 0$ K. Initial mesh.</td>
</tr>
<tr>
<td>Case 2</td>
<td>5.3</td>
<td>0.7</td>
<td>$T_{amb} = 0$ K. Edges projection from VV on BLA and DIV, mesh refinement in problematic regions.</td>
</tr>
<tr>
<td>Case 3</td>
<td>3.4</td>
<td>0.3</td>
<td>$T_{amb} = 0$ K. Additional mesh refinement in problematic regions.</td>
</tr>
<tr>
<td>Case 4</td>
<td>0.6</td>
<td>0.3</td>
<td>$T_{amb} = 523$ K. Edge projections and full mesh refinements.</td>
</tr>
</tbody>
</table>

### 5.2 Base case - Active CTS

Further analysis is based on the “full” model since the “separate” model approach cannot be used in the simulation cases with passive components. Simulation with the “full” model still allows us to define individual interactions for independent model regions and their corresponding ambient temperatures, which substantially improves the accuracy of the numerical simulations.

The numerical results for the simulation case where all components are actively-cooled are presented in Table 3. This table includes the total net (absorbed or emitted) powers for...
individual system and corresponding net heat fluxes. Note that positive net power and heat flux indicate that the component absorbs radiation from its surrounding, while the negative values indicate that the component is an emitter. The results show that the warmer (internal) components are the emitters and the cooler (external) components are the receivers, as expected. The thermal load on VVTS is the highest as it encloses the entire VV. The cryostat is also subjected to substantial thermal load since it interacts with VV ports. On the other side, the net heat flux on magnet system is rather low since these components are being shielded by thermal shields at 80 K. Based on the simulation results, the total refrigeration power, Eq. 6, of the total system (VVTS, CTS and magnets) is approximated to 430 W/m². The theoretically predicted total refrigeration power is equal to 392.5 W/m². It can be noted that this rather simple theoretical model and the numerical simulation of the tokamak geometry are in quite good agreement.

Table 3: Results for case with active CTS.

<table>
<thead>
<tr>
<th>System</th>
<th>Total Net Power (kW)</th>
<th>Net HF (W/m²)</th>
<th>( P_{\text{sim}}^{\text{refrig.}} ) (W/m²)</th>
<th>( P_{\text{theor}}^{\text{refrig.}} ) (W/m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>BLA and DIV</td>
<td>-57.5</td>
<td>-268</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>VV with LIDS</td>
<td>-27.6</td>
<td>-58</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>VVTS</td>
<td>49.6</td>
<td>121</td>
<td>322.27</td>
<td>327.96</td>
</tr>
<tr>
<td>Magnets system</td>
<td>0.1</td>
<td>0.2</td>
<td>13.01</td>
<td>16.78</td>
</tr>
<tr>
<td>CTS</td>
<td>8.2</td>
<td>36</td>
<td>93.91</td>
<td>47.80</td>
</tr>
<tr>
<td>CRY</td>
<td>23.1</td>
<td>76</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>TOTAL</td>
<td>-4.0 (imbalance)</td>
<td>-</td>
<td>429.18</td>
<td>392.54</td>
</tr>
</tbody>
</table>

5.3 Passive CTS

This case is similar to the previous one, but unlike other components which are actively cooled, the CTS here is assumed to be a 3mm thick shell that is completely passive (without cooling). For such component in steady state balance, the net thermal load loss is zero, meaning the absorbed power from cryostat equals the emitted power towards the magnets and VVTS. The average temperature of the CTS obtained in the simulation is 289 K, while the theoretically predicted temperature CTS is 242 K. Due to the significantly higher CTS temperature with respect to actively-cooled CTS case (80 K), the thermal load on the magnet system is now substantially higher (3.5 kW in Table 4 versus 0.1 kW in Table 3). This results in significantly higher refrigeration power. For the simulation, this value equals to 1015 W/m², while the theoretical model predicts the refrigeration power of 1038 W/m². Again, very good agreement between simulation results and theoretical prediction is observed. The thermal loads on other components, such as DIV, BLA, CRY practically remain unchanged, and are therefore not shown in Table 4.

Table 4: Results for case with passive CTS.

<table>
<thead>
<tr>
<th>System</th>
<th>Total Net Power (kW)</th>
<th>Net HF (W/m²)</th>
<th>( P_{\text{sim}}^{\text{refrig.}} ) (W/m²)</th>
<th>( P_{\text{theor}}^{\text{refrig.}} ) (W/m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>VVTS</td>
<td>50.2</td>
<td>121.8</td>
<td>324.35</td>
<td>327.96</td>
</tr>
<tr>
<td>Magnets system</td>
<td>3.5</td>
<td>9.6</td>
<td>690.71</td>
<td>710.53</td>
</tr>
<tr>
<td>CTS</td>
<td>-7.3E-05</td>
<td>0.00</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>TOTAL</td>
<td>-4.3 (imbalance)</td>
<td>-</td>
<td>1015</td>
<td>1038</td>
</tr>
</tbody>
</table>
5.4 Passive CTS with MLI

By adding a single MLI at the warm side of the CTS, i.e. between CTS and CRY, the heat load on CTS shown in Table 5 is reduced to 1.8 kW which is, as could be expected, between the active and passive results. The average CTS temperature obtained in the simulation is, in this case, 244 K. The MLI temperature is somewhat higher with an average value of 314 K. The theoretically predicted CTS and MLI temperatures are 206 K and 270 K, respectively. Comparing to passive case, use of MLI shielding therefore reduces the heat load on the magnet system, which is, however, still well-above the active case. The total refrigeration power, based on simulation results and theoretical model, is estimated to 680.95 W/m² and 705 W/m², respectively, which is again above the active case.

Table 5: Results for case with passive CTS with one MLI.

<table>
<thead>
<tr>
<th>System</th>
<th>Total Net Power (kW)</th>
<th>Net HF (W/m²)</th>
<th>P&lt;sub&gt;refrig. sim.&lt;/sub&gt; (W/m&lt;sup&gt;2&lt;/sup&gt;)</th>
<th>P&lt;sub&gt;refrig. theor.&lt;/sub&gt; (W/m&lt;sup&gt;2&lt;/sup&gt;)</th>
</tr>
</thead>
<tbody>
<tr>
<td>VVTS</td>
<td>49.88</td>
<td>121.1</td>
<td>322.59</td>
<td>327.96</td>
</tr>
<tr>
<td>Magnets system</td>
<td>1.8</td>
<td>4.96</td>
<td>358.36</td>
<td>376.58</td>
</tr>
<tr>
<td>CTS</td>
<td>-4.4E-03</td>
<td>-0.0</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>MLI</td>
<td>-4.4E-03</td>
<td>-0.0</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>TOTAL</td>
<td>-4.3 (imbalance)</td>
<td>-</td>
<td>680.95</td>
<td>704.55</td>
</tr>
</tbody>
</table>

The refrigeration power for the case with two MLIs is estimated only by the theoretical model. The results show that additional MLI reduces the refrigeration power of the magnet system from previous 376.58 to 257.9 W/m². The total refrigeration power is estimated to 576.58 W/m² which is still approximately 50% higher than the base case. This analysis can be easily expanded to further investigate the effect of multiple MLIs.

6 CONCLUSIONS

Thermal radiation exchange between different tokamak systems in a steady-state is analyzed numerically with the FE code ABAQUS. In order to reduce the energy imbalance in the simulation (numerical error), an extensive grid refinement study has been performed. The final energy imbalance for the full tokamak model is reduced below 6%. This model is then used to investigate the effect of the passive CTS and the addition of MLI insulation on the CTS. The base (active CTS) case, involving only actively cooled components, is characterized by the smallest thermal loading on magnet system, and the lowest refrigeration power required for component cooling. In parallel, the simpler theoretical analysis, where the components are approximated by infinite parallel plates, is conducted.

By use of passive CTS, the heat load on magnet system is significantly increased. The simulation shows that 130 % more refrigeration power is required for cooling of VVTS and magnet systems. A similar result is obtained with the theoretical model which predicts approximately 160% increase of refrigeration power in this case. By adding the MLI on the warm side of the CTS (between the CTS and CRY), both, the temperature of CTS and thermal loading on magnets reduce. Nevertheless, the simulation and theory still predict 60% and 80% higher total refrigeration power than in the base case with active CTS. By acknowledging relatively good agreement between the simulation results and theoretical model, only theoretical approach is used to investigate the effect of two MLIs. The results show that even in this case, a 50% higher total refrigeration power than in the base case is required. Therefore, from the perspective of cooling efficiency, the most efficient design includes
actively cooled CTS. The perspective for future work is to investigate the effect of MLI in combination with actively cooled thermal shields.

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REFERENCES


