Comparison of 2D and 3D Heat Transfer Models around the Coolant Channel in the HTR-PM Side Reflector

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Abstract – In the HTR-PM pebble bed reactor heat is produced in a cylindrical core surrounded by a graphite reflector. The helium coolant flowing down through the core first flows up through 30 coolant channels in the reflector, cooling the reflector. Heat is also transferred through the reflector in radial direction to the pressure vessel and other surroundings. Usually heat transfer in the reflector region is modelled using a 2D axi-symmetric geometry, modelling the region containing the coolant channels as a homogenized mixture of coolant channel and graphite reflector using a porosity value, sometimes using very coarse meshes. In reality temperature gradients in angular direction will exist around the coolant channels, possibly effecting both heat transfer to the coolant in the channels and heat transfer through the graphite around the coolant channels to the outer boundary. This paper investigates the accuracy of the 2D model for calculating the temperature and heat transfer around the coolant channels in the side reflector by comparing calculations for a fine and a course 2D mesh with a 3D mesh in which the coolant channel geometry is explicitly modelled. Two cases were investigated, one representing full power operation, and the other a loss of forced coolant incident with no helium flow through the coolant channels. Calculations were performed with the pebFoam OpenFOAM solver. The course 2D mesh resulted in large errors in the reflector temperature field, overestimating the temperature drop across the coolant channel region. The 2D fine mesh compared reasonably well with the 3D mesh, although it resulted in both an overestimation of the effective heat transfer rate to the coolant channels and an underestimation of the effective resistance to heat transfer in the reflector around the coolant channel in the radial direction, both of which can lead to an underestimation of reflector and core temperatures.

I. INTRODUCTION

The pebble bed nuclear reactor, a high temperature gas cooled reactor design, is one of the main candidates for next generation nuclear power plants [1]. Key features include the passive safety of the reactor, higher thermal efficiency due to higher coolant outlet temperatures, and the possibility of on-line refuelling by extracting pebbles at the bottom of the bed and adding pebbles to the top. In these reactors, the fuel is contained within graphite spheres, which form a randomly packed bed in a cylindrical core cavity surrounded by a graphite reflector. The helium coolant first flows upwards through coolant channels in the graphite reflector, and then via a plenum downwards through the pebble bed, removing the fission heat.

Heat is also transferred from the pebble bed to the graphite reflector, which is cooled by the cold helium flowing upward through the coolant channels. These coolant channels are arranged in a ring in the reflector at equidistant positions. Finally a small portion of the heat is transferred through the reflector and via various core components to the outside.
In pebble bed calculations often 2D r-z models are used. In these models the ring of coolant channels is modelled as a homogenized zone in the reflector, using a porosity to represent the fractional volume occupied by the coolant channels [2]. Especially for system analysis purposes the meshes for these models are often quite coarse, using only a few or even one mesh cell for the coolant channel region in the reflector. This brings us to the question how accurate these models are in calculating the heat transfer through the reflector around and to the coolant channels.

To this end we compared results for heat transfer calculations using three different meshes for the reflector. The first two are 2D meshes using r-z coordinates and a homogenized coolant channel region, with one mesh (2D1) using only one mesh cell in the coolant channel region, and the other mesh (2DN) using a fine mesh in this region. The third mesh is a 3D model of a 12 degree wedge around one coolant channel, making use of the symmetry of the coolant channels in the reflector.

Heat transfer calculations were performed with pebFoam, which is described in the next section. A simplified geometric model of the Chinese High Temperature gas-cooled Reactor-Pebble-bed Module (HTR-PM) [3] was used, for which two separate cases were considered. In the first case, detailed in section III, the temperature distribution in the graphite reflector temperature is calculated during normal full power operation. In section IV the second case is presented, representing an incident scenario with a loss of forced cooling accident in which the flow through the coolant channels is zero.

II. REACTOR MODELLING

The reactor model under consideration is based on the Chinese HTR-PM design [3]. In this design, cold helium flows upwards through coolant channels in the radial reflector, cooling the reflector, before flowing into the cold Helium plenum on top of the pebble bed, and down through the bed. There are 30 coolant channels in the reflector, spaced evenly apart on a circle with a radius of 200 cm. The coolant channels are 18 cm in diameter.

As we are only interested in comparing calculation methods with each other, we used a simplified model of the HTR-PM. The core is modelled as a cylinder 150 cm in radius, consisting of an 11 m high pebble bed, with an 80 cm high cold helium plenum on top, ignoring top and bottom reflectors and the cone shape at the bottom of the pebble bed. The core is surrounded by a 75 cm thick graphite reflector, with the coolant channels located at \( r = 200 \) cm, see Fig. 1. Other details in the reflector such as control rod holes are ignored, as are heat losses through the top and bottom.

![Fig. 1: Pebble bed model with reflector and coolant channels.](image)

Fig. 1: Pebble bed model with reflector and coolant channels. \( r \) and \( z \) coordinates are in meters.

II.A. The PebFoam Thermo Hydraulics Code

Thermo hydraulics calculations were performed with pebFoam, an OpenFOAM [4] application created to solve the heat and mass transfer in a pebble bed reactor, using finite volume discretization and supporting unstructured meshes of arbitrary shapes in 1D, 2D and 3D. It uses the porous medium approach to model the pebble bed, in which volume averaged values of the quantities of interest are calculated, using the porosity \( \varepsilon \) to represent the local packing structure. For a more detailed description see [5].

For the current work, pebFoam was extended to include the graphite reflector region. Heat and mass transfer in the pebble bed are modelled as described in [5]. Heat is transferred from the core to the graphite reflector, and through the reflector to the outside. Two ways of modelling the coolant channels were implemented in pebFoam, the first a 2D r-z
model using a porous region at the radial location of the coolant channel, and the second a detailed 3D model of the reflector and coolant channel geometry.

II.B. 2D Model and Meshes

For the 2D model, the region with the coolant channels, from \( r = 191 \) to \( r = 209 \) cm, is modelled as a porous region with constant porosity \( \varepsilon \). The mass and momentum balance for the helium in the coolant channel region use the following equations

\[
\nabla \cdot (\varepsilon \rho u) = 0 \quad \text{(eq. 1)}
\]

\[
\nabla \cdot (\varepsilon \rho u u) = \nabla \cdot (\varepsilon \mu_{eff} \nabla u - \varepsilon \rho - \varepsilon C_g u + \varepsilon \rho g) \quad \text{(eq. 2)}
\]

Where \( \rho \) is the helium density, \( u \) the velocity in the coolant channel, \( \mu_{eff} \) the effective viscosity, \( \rho \) the pressure and \( g \) the gravitational acceleration. The pressure drop over the coolant channel is modelled using a drag coefficient \( C_d \) which value was taken from THERMIX [6].

The heat transfer through the helium and the graphite reflector is modelled using

\[
\nabla \cdot \left( \varepsilon \rho c_p h \right) = \nabla \cdot \varepsilon c_{eff} \nabla h - q' \quad \text{(eq. 3)}
\]

\[
\nabla \cdot \left( (1 - \varepsilon) \lambda_{gr} \nabla T_{gr} \right) = Q' \quad \text{(eq. 4)}
\]

Where \( h \) is the helium enthalpy, \( c_{eff} \) is the effective thermal diffusivity, \( \lambda_{gr} \) is the graphite thermal conductivity, and \( T_{gr} \) the graphite temperature. The two equations are coupled through \( q' \), which is the volumetric heat transfer from the helium to the graphite in W/m² and is calculated using

\[
q' = \frac{4\varepsilon \theta_{ch} \mu_{ch} \text{Nu}}{D_{ch} D_{ch}} \quad \text{(eq. 5)}
\]

Where \( 4\varepsilon / D_{ch} \) is the coolant channel surface area per unit volume (in the annular coolant channel region between \( r = 191 \) and \( r = 209 \) cm), and \( \theta_{ch} \) is the helium thermal conductivity. Nu is the Nusselt number for the heat transfer from the coolant channel wall to the bulk of the helium, and is given by [6]

\[
\text{Nu} = (1 - f) \max \left[ 3.66 + 1.61 \left( \frac{\text{Re} \mu_{ch} D_{ch}}{L_{ch}} \right)^{1/3}, 0.644 \left( \frac{\text{Re} D_{ch}}{L_{ch}} \right)^{1/3} + f \max [3.667, 0.0214(\text{Re}^{0.8} - 100)\text{Pr}^{0.4} \left( 1.0 + \frac{D_{ch}}{L_{ch}} \right)^{2/3}] \right] \quad \text{(eq. 6)}
\]

Here Re and Pr are the helium Reynolds and Prandtl numbers, and \( L_{ch} \) is the channel length for which we used 11.8 m. \( f \) is a fraction depending on Re, with \( f = 0 \) for \( \text{Re} < 2300 \), \( f = 1 \) for \( \text{Re} > 10000 \), and \( f = (\text{Re} - 2300)/(10000 - 2300) \) in between.

The helium in the coolant channels can only exchange heat with the graphite in the coolant channel region. Heat exchange with the rest of the graphite reflector takes place by conduction through the graphite. Outside the coolant channel region, heat transfer in the reflector is modelled by eq. 4 with \( \varepsilon \) and \( \theta_{ch} \) both zero.

As sometimes very course meshes are used in pebble-bed calculations, we wanted to investigate possible effects of using course meshes on heat transfer in the coolant channel region of the reflector. To this end two different 2D r-z meshes for the reflector were compared. The first 2D mesh, the 2D1 mesh, uses only one mesh cell in the radial direction for the coolant channel region with a width \( \Delta r = 18 \) cm. The other mesh, the 2Dfine mesh, uses 18 cells in the coolant channel region with a width \( \Delta r = 1 \) cm per cell. Outside the coolant region both meshes used cell widths \( \Delta r = 1 \) and \( \Delta z = 2.5 \) cm, for a total number of mesh cells of 58x472 and 75x472 for the 2D1 and 2DN meshes respectively.

II.C. 3D Model and Mesh

In our 3D model, we modelled a pie-shaped section of the reflector around one coolant channel, spanning a 12 degree arc (360 divided by 30 coolant channels), using symmetry boundary conditions on the sides. A fine mesh is used for the graphite reflector, using 64 cells in the r-direction and 48 cells in the \( \theta \)-direction. The coolant channel itself used a 1D mesh with only one mesh cell for each axial coordinate, see Fig. 2. Again 472 cells are used in the z-direction. This geometry makes use of the flexible meshing capabilities of pebFoam, allowing complex cells bounded by an arbitrary number of faces. It also allows us to use the same heat transfer relations for the coolant channel as for the 2D case.

Heat transfer through the solid graphite reflector is modelled as before. For the heat transfer from the graphite coolant channel wall to the helium bulk inside the channel we use the same Nusselt number as in the 2D case (eq. 6). The transport equations for the helium are identical to those of the 2D model, again using \( C_d \) to model the drag force due to the coolant channel walls, with \( \varepsilon = 1 \).
II. Graphite and Helium Transport Properties

The reflector graphite thermal conductivity $\lambda_{gr}$ is calculated using the relationship for the HTR-10 graphite reflector without the effect of neutron irradiation as reported in [7], with $T_{gr}$ in °C.

$$\lambda_{gr} = 1.15 \left(1 - 1.084 \times 10^{-3} T_{gr} \right) + 0.743 \times 10^{-6} T_{gr}^2 - 0.213 \times 10^{-9} T_{gr}^3$$

(eq. 7)

Helium properties are calculated according to the KTA rules [8]. As the 2D1 and 3D cases use a 1D coolant channel geometry, the values for $\alpha_{eff}$ and $\mu_{eff}$ are unimportant. For the 2Dfine channel we have chosen $\alpha_{eff} = \text{Nu}_{efT}$ and $\mu_{eff} = \mu_{Tef}$. See section V for a more detailed discussion on these choices.

III. NORMAL OPERATION TESTCASE

The first test case represents the reactor under normal operating conditions. Only for the 2D1 mesh calculation was the pebble bed core region included, to calculate the heat transfer from the pebble bed to the reflector. To have a good comparison of the different meshes, for the other two investigated reflector meshes (2Dfine and 3D) only the reflector was modelled, and the calculated heat flux from the pebble bed to the reflector was used as the reflector inner boundary condition.

The pebble bed core calculations used the same power density distribution as in [5], with a total thermal power of 250 MW. The helium inflow velocity in the core was 2.0 m/s and the outlet pressure was fixed at 7.08 MPa. The core had a constant porosity of 0.39. As mentioned before, top and bottom boundary conditions for core and reflector were adiabatic. The coolant channels had a helium inflow velocity of 16.25 m/s with an inflow temperature of 528.5 K. The outlet pressure for the coolant channels was 7.16 MPa. At the reflector outer boundary ($r = 225$ cm) a constant flux boundary condition with a flux of 2997 W/m² was applied, resulting in a total heat loss in the radial direction of 500 kW. For the 2D calculations, the coolant channel region between $r = 191$ and $r = 209$ cm had a constant porosity of $\varepsilon = 0.3375$, corresponding to 30 coolant channels of 18 cm diameter.

III.A. Results

For each of the three meshes, temperature distributions were calculated. The heat flux from the pebble bed to the reflector as calculated by the 2D1 problem was used as input for the 2Dfine and 3D cases, so boundary conditions were identical for all three cases. As heat fluxes were fixed, conservation of energy fixed the helium outflow temperature of the coolant channel. A comparison of the outflow temperatures for the three meshes confirmed this was indeed the case, with almost identical helium temperature profiles throughout the length of the reflector for the three grids.

For each mesh the average reflector temperature as a function of $z$ was calculated by averaging over the $r$- and (in the 3D case) $\theta$-directions. As can be seen in Fig. 3, the reflector heats up from the top of the pebble bed (starting at $z = 0$) and reaches its maximum temperature at the bottom of the bed ($z = 11$ m), where helium temperatures in the core are highest, and thus the heat flux from the core to the reflector is highest. This is also the place where the largest temperature differences are between the three meshes, with the 2D1 mesh resulting in an average temperature >25 K higher than the other two cases.
Fig. 3: Reflector temperature profile as a function of z-coordinate, averaged over r- and θ-directions. Top of pebble bed is at z = 0.

As the previous results were for the average reflector temperature, we expect local temperature differences to be higher. To this end we have plotted the reflector temperature at the bottom of the core (z = 11 m) as a function of the radial position inside the reflector in Fig. 4. For the 3D mesh these values are averaged over the angular direction.

For a more detailed look at the temperature profiles around the coolant channel for the 3D mesh, we show in Fig. 5 the reflector temperature profile as a function of the angular position θ at three different radial positions: at r = 1.91 m, touching the coolant channel on the core side of the reflector, at r = 2 m, through the middle of the coolant channel, and at r = 2.09 m, touching the coolant channel at the outside of the reflector. Again the axial position was at the bottom of the reflector, at z = 11 m, where temperature differences are large.

For the 2D1 mesh the reflector temperature near the inside edge is >50 K higher than for the other two meshes, mostly due to a much higher temperature drop in the coolant channel region, indicating that this mesh is far too course to properly capture the heat transfer in the coolant channel region. The difference between the 2Dfine and 3D meshes is much smaller, and only in the coolant channel region more than 10 K. However, the total temperature drop for the 3D mesh is 14 K smaller than for the 2Dfine mesh (309 K versus 323 K). This differences has two causes. First, the temperature drop for the 3D mesh is lower than for the 2Dfine mesh in the part of the reflector on the core side of the coolant channel (r < 191 cm). This is due to a slightly lower average reflector temperature resulting in higher graphite conductivity, and due to the non-uniform temperature profile in the angular direction in this region, as we will see later. Second, the 3D mesh results in a lower temperature drop in the coolant channel region, resulting in significantly higher temperatures in this region. This indicates a higher resistance to heat transfer from the reflector to the coolant channel.
At the core side of the coolant channel \((r = 1.91\,\text{m})\) the angular temperature profile shows a strong drop in temperature near the coolant channel (at \(\theta = 0\)), due to the large local heat flux to the coolant channel. In most of the reflector region on the core side of the coolant channel \((r < 1.91\,\text{m})\), the angular temperature profile looks similar, with a lower temperature around \(\theta = 0\), and higher temperatures near \(\theta = \pm 6\). This temperature profile is the main cause of the lower temperature drop over the reflector region on the core of the coolant channel for the 3D mesh compared to the 2Dfine mesh. Although the average temperature is almost identical near the coolant channel, the lower temperature around \(\theta = 0\) results in higher graphite conductivity in this region, leading to higher effective conductivity averaged over \(\theta\).

The temperature profile through the centre of the coolant channel, at \(r = 2\,\text{m}\), has a steep gradient in the angular direction. The temperature difference in angular direction is almost \(15\,\text{K}\) over a length of \(11\,\text{cm}\), resulting in an angular temperature gradient \(\partial T/\partial \theta = 1.3\,\text{K/cm}\), still significantly lower than the gradient in radial direction \(\partial T/\partial r = 2.7\,\text{K/cm}\).

In Fig. 4 we saw the radial temperature profile for the 2Dfine mesh drops much faster in the coolant channel region than for the 3D mesh. In Fig. 5 we see the cause. All three figures show a sharp temperature gradient in the graphite towards the cooler coolant channel. At the core side of the coolant channel \((r = 1.91\,\text{m})\) this results in a lower graphite temperature at the coolant channel wall for the 3D mesh, and thus lower heat flux towards the helium. As a result, halfway the coolant channel \((r = 2\,\text{m})\), graphite temperatures are higher for the 3D mesh. This results here in a higher heat flux to the helium in the coolant channel, resulting in only a slightly higher temperature for the 3D mesh than for the 2Dfine mesh at the outer side of the coolant channel region \((r = 2.09\,\text{m})\), even though the 2D cases overestimate the heat flux from the graphite to the coolant channel. The 2D models assume there is no resistance in the graphite for the heat transfer towards the coolant channel, but in reality there is. This results for the 2D meshes in a much faster temperature drop in the reflector in the coolant channel region, and as in this case the heat fluxes were fixed, an overall much lower temperature difference between the reflector graphite and coolant channel. Although the differences between the 2Dfine and 3D mesh are not very large, the fact that the 2Dfine case overestimates heat transfer from the graphite to the helium in the coolant channel might lead to non-conservative results for the core temperature.

IV. NO COOLANT FLOW TESTCASE

The second case considers a situation where there is no flow through the coolant channels, like in a loss of forced coolant incident. Where our first case focused more on the heat transfer from the graphite to the helium in the coolant channel, the second case investigates the heat transfer through the graphite around the coolant channel to the outside of the reflector, as the thermal conductivity of stagnant helium is small.

As we are still only interested in the effect of modelling choices on the heat transfer, and not so much in the actual temperature profiles in real cases, we assumed the heat flux from the core to the reflector would remain the same as in the full power case, and used the same fixed flux boundary condition on the inside of the reflector. We used adiabatic top and bottom boundary conditions for the coolant channel, assumed no flow through the coolant channels, and did not solve the mass and momentum balances for the helium. The outer boundary condition for the reflector was changed to a fixed temperature boundary condition with \(T_{\text{gr,r=2.25}} = 540\,\text{K}\).
IVA. Results

As with the full power case, we first look at the reflector temperature as a function of axial position, averaged over \( r \)- and \( \theta \)-directions, see Fig. 6.

This time the difference between the 2D1 and 2Dfine mesh is marginal, but the 3D case results in a significantly higher reflector temperature (>20 K) at the bottom of the core. As the flux through the reflector is fixed, the higher temperature for the 3D mesh indicates the 2D meshes underestimate the resistance to heat flow through the reflector in the radial direction.

![Graph showing reflector temperature profile as a function of axial position \( z \), averaged over \( r \)- and \( \theta \)-directions.](image1)

Fig. 6: Reflector temperature profile as a function of axial position \( z \), averaged over \( r \)- and \( \theta \)-directions.

The radial temperature profiles at \( z = 11 \) m are shown in Fig. 7, where we see that the cause of the temperature difference between the 3D and 2D meshes is inside the coolant channel region. The temperature drop over the coolant channel region is >20 K larger for the 3D mesh, indicating a higher resistance to heat transfer in this region than for the 2D meshes.

![Graph showing reflector temperature for the no flow case at \( z = 11 \) m as a function of \( r \), averaged over \( \theta \)-direction for the 3D mesh.](image2)

Fig. 7: Reflector temperature for the no flow case at \( z = 11 \) m as a function of \( r \), averaged over \( \theta \)-direction for the 3D mesh.

To find the cause of this difference in temperature drop in the coolant channel region we look again at the reflector temperature versus \( \theta \) at \( z = 11 \) m and \( r = 1.91 \) m, touching the coolant channel on the core side of the reflector, \( r = 2 \) m, through the middle of the coolant channel, and \( r = 2.09 \) m, touching the coolant channel on the outside of the reflector, see Fig. 8. Here we see on the core reflector side of the coolant channel (Fig. 8 top) a higher reflector temperature just in front of the coolant channel (\( \theta = 0 \)), and on the outer reflector side of the coolant channel (Fig. 8 bottom) a lower temperature at \( \theta = 0 \). This is because the coolant channel acts as a hole in the heat conduction through the reflector, due to the low thermal conductivity of helium. Thus on the core side of the reflector just in front of the coolant channel, the channel causes the cooler outside of the reflector to be further away, resulting in a locally higher temperature, as heat has to ‘travel’ a longer path through the graphite to the outside. On the outside of the reflector the situation is reversed. The result is an effectively higher resistance to heat transfer in the radial direction around the coolant hole than for the ‘uniformly mixed’ geometries of the porous models in the 2D meshes, resulting in a higher temperature drop over the coolant hole region. An important implication is that in incident scenarios where heat removal by the coolant is impaired and heat has to be removed through the reflectors, the temperature drop over the reflector for the same heat flux to the outside will be higher than estimated by the 2D porous model, resulting in higher core temperatures than estimated.
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Fig. 8: Reflector temperature profiles at \( z = 11 \) m for the three meshes as a function of \( \theta \) and at \( r = 1.91 \) m (top), \( r = 2.09 \) m (middle) and \( r = 2.09 \) m (bottom).

V. DISCUSSION

Here we will first discuss our investigation in three possible causes of errors in the calculations, followed by two possible improvements to the 2D porous model of the coolant channel region.

VA. Investigation of Uncertainty

The aim of the calculations was to compare different calculation strategies and investigate the accuracy of the 2D porous model compared to a 3D model. Accurately calculating a realistic temperature profile was of lesser importance. To that end we strived less to very realistic cases and more to cases in which the 2D and 3D meshes could be easily compared, for example by using identical boundary conditions for each mesh. Here we discuss three possible sources of errors which could compromise the validity of our comparison.

First we investigated if the meshes were fine enough, except of course in the coolant channel region for the 2D1 mesh. Calculations were performed for all three meshes with a mesh twice as coarse in each direction, thus reducing the total mesh size by a factor 8. Resulting temperature profiles were compared with the finer meshes, and only minor temperature differences were observed.

Second, as the outer reflector surface is only 16 cm away from the coolant channel and we imposed here a uniform flux or temperature boundary condition, this boundary condition could possibly affect the temperature profiles around the coolant channel in for 3D mesh. For comparison calculations were performed with the outer reflector boundary extended to \( r = 2.75 \) m, thus moving the uniform boundary condition much farther away from the coolant channel. No significant differences in temperature profiles were observed.

Finally, the magnitude of \( \alpha_{\text{eff}} \) could possibly affect heat transfer for the 2Dfine mesh, as it affects the heat transfer in radial direction through the helium. Calculations were performed with two times higher and much lower (\( \alpha_{\text{eff}} = \alpha_{\text{He}} \)) effective thermal conductivities. Again, both calculations showed no significant changes in temperature profiles. The sensitivity of the temperature profiles to changes in \( \mu_{\text{eff}} \) was not investigated, as the helium velocity stayed almost uniform throughout the coolant channel domain.

V.B. Investigated Improvements to the 2D Model

To improve the results of the 2Dfine mesh, two possible improvements to the porous model of the coolant channel region were investigated. First, as the coolant channel is circular, the porosity in the coolant channel region from \( r = 1.91 \) to \( r = 2.09 \) m is not uniform but changes with \( r \). Also, the area of the coolant channel surface per unit volume changes with \( r \). Thus two separate adjustments to the 2Dfine model were investigated. First, the porosity \( \varepsilon \) was not taken constant in the coolant channel region, but was evaluated separately for each mesh cell. Second, the coolant channel surface area per unit volume, the term \( 4\varepsilon / D_{\text{ch}} \) in eq. 5, was not taken constant, but also evaluated separately for each mesh cell. Calculations were performed using only an adjusted porosity field, and using both an adjusted porosity field and coolant channel surface area. However, both adjustments resulted in only minor changes (<1 K) in the calculated temperature profiles.

VI. CONCLUSIONS

This paper investigated the accuracy of the use of 2D meshes for calculating the temperature and heat transfer around the coolant channels in the side reflector of HTRs, by comparing calculations using a fine and a coarse 2D r-z mesh with those using a 3D mesh. In the coolant channel region the 2D meshes used a homogenized model with a porosity to
represent the coolant channel volume, while the coolant channel geometry was explicitly modelled in the 3D mesh. The course 2D1 mesh used only one cell in the r-direction in the coolant channel region, where the 2Dfine case used 18 cells ($\Delta r = 1$ cm). Outside the coolant channel region the two meshes were identical.

The first investigated case represents full power normal operation of the reactor, with a high heat transfer rate from the graphite reflector to the helium flowing through the coolant channels. In this case the course 2D1 mesh overestimated the temperature drop over the coolant channel region by >50 K, resulting in a large overestimation of the reflector temperature on the core side of the coolant channel region. The difference between the 2Dfine and 3D meshes was much smaller, but the 2Dfine mesh also overestimated the heat transfer rate to the coolant channel compared to the 3D mesh, resulting in smaller temperature differences between the reflector and the helium in the coolant channel region and a slightly higher temperature drop in this region. The lower resistance to heat transfer to the coolant channel for the 2Dfine mesh is because the 2D model ignores the resistance in the graphite. This resistance is because part of the graphite is further away from the coolant channel than other parts, and leads to the observed large reflector temperature gradients in angular direction near the coolant channel. Thus, for the same average reflector temperature the heat transfer to the coolant channel is higher for the 2D mesh than for the 3D mesh, resulting in a lower average reflector temperature.

In the second case there was no flow in the coolant channels, representing for example a loss of forced coolant accident. As the conductivity of the stagnant helium is very low, this case mostly focused on the heat transfer in radial direction through the graphite reflector around the coolant channel. Resulting temperature profiles showed no significant difference between the 2D1 and 2Dfine meshes. However, the 3D mesh resulted in a significantly larger temperature drop over the coolant channel region of >20 K. This is caused by a higher effective resistance to heat transfer in the radial direction through the graphite around the coolant channel in the 3D mesh than for the 2D model due to the specific geometry around the coolant channel.

The comparison of the 2D and 3D meshes show a 2D mesh can result in reasonably accurate temperature profiles in the coolant channel region of the reflector, as long as the mesh in the coolant channel region is fine enough. Still, one should be careful with the 2D model, as it resulted in both an overestimation of the effective heat transfer rate to the coolant channels and an underestimation of the effective resistance to heat transfer in the reflector around the coolant channel in the radial direction. Both effects can lead to an underestimation of reflector and core temperatures. To amend these problems, a more detailed model for the effective Nusselt number for the 2D mesh should be developed, including the resistance in the graphite for heat transfer towards the coolant channel as a function of specific reflector geometry and graphite thermal conductivity. Also, the conductivity of the graphite in the coolant channel region should be adjusted in the 2D porous model to compensate for the added effective resistance to heat transfer in radial direction due to the geometry.

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