THERMAL ANALYSIS OF A HIGH-TEMPERATURE FALLING BED FUSION REACTOR BLANKET

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Summary

A high temperature, falling bed blanket has been designed for a tokamak fusion reactor. The design centers on the use of a gravity flow of 0.5 to 1.5 cm diameter Al₂O₃ balls as the high temperature heat transfer media. This system has the advantage of being able to produce process heat at temperatures in excess of 1000°C while maintaining structural temperatures at ~ 400°C. Pumping powers for the system are low. A thermal hydraulic analysis of a representative blanket module has, however, identified several potential problem areas.

The most important of these is the fact that unless steps are taken to avoid it, a significant fraction of the total power deposited in the high temperature bed is extracted as low temperature heat through the active cooling of the structural shell of the duct channeling falling ceramic balls. The use of more insulating liner bricks was explored and rejected due to the excessively high temperatures that resulted in these bricks. Partitioning of the bed into regions of different velocity, together with the provision of an insulating gas-filled gap around the liner brick wall, appears to be the most promising method for reducing the loss of energy to the actively cooled structure.

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1. Introduction

This paper presents some interim results of the on-going design study of a high temperature blanket for a tokamak reactor. The study was motivated by the potential advantages of a fusion reactor (vis-a-vis other energy sources) in the generation of high-temperature process heat. Fusion power is produced from a non-exhaustible fuel, and in a way that allows the deposition of energy to take place outside the contaminated nuclear reaction zone. Within this context, the falling pebble bed was chosen as a system capable of meeting the following major goals:

1) The capability to produce process heat at a temperature in excess of 1000°C.
2) The development of a design that would allow the direct use of a non-activated process heat stream without recourse to the use of a high-temperature heat exchanger as a means of isolating the primary coolant loop due to concerns over transported radioactivity.
3) The elimination of ceramics as structural materials in the reactor.
4) The system should be nearly self-sustaining with respect to tritium breeding and this should be done in a way to ensure no contamination of the process heat stream.

The approach to a high-temperature fusion reactor blanket design suggested here attempts to avoid the problems of static designs through the use of a ceramic falling bed. As a high-temperature blanket concept, the falling bed is particularly attractive in that it allows the decoupling [1] of the structural materials from the high-temperature heat transfer medium. As shown in recent studies, by actively cooling the structural walls, modest temperatures (\(~400°C\)) can be maintained while producing a high (\(>1000°C\)) temperature in the falling ceramic pellets. From a feasibility point of view, this may prove much more reasonable than would the construction of an integral structure capable of withstanding 1000°C temperatures. Such a system also minimizes the pumping requirements to a few hundred kilowatts of power due to the fact that it can be operated at low pressures. Because of the high density of the falling particle bed compared to more conventional coolants, a relatively small volume throughput is sufficient to absorb the energy output of a fusion reactor. For example, if \(\text{Al}_2\text{O}_3\) is used as the coolant media at an entrance temperature of 300°C and an exit temperature of 1000°C, a throughput of less than 2 tons per second is required to absorb an output of 1000 MW of thermal energy.

The blanket consists of a series of bins surrounding the tritium-breeding blanket. These bins are filled with packed falling beds of \(\text{Al}_2\text{O}_3\) or MgO pebbles whose temperature rises as a result of fast neutron capture. The pebbles fall by gravity into heat exchangers where they give up their heat to a countercurrent of air or steam and are finally returned by a mechanical conveyor to the top of the blanket for another cycle through the system. Figure 1 gives a schematic representation of this process. A small upward flow of air (or steam) from the heat exchanger will be allowed to sweep through the falling bed blanket, in order to prevent any flow of gas from the bed into the heat exchanger that might contaminate the output from the latter.

The initial phase of the design here described is aimed at establishing the feasibility of the proposed concept and identifying potential problem areas. The work performed so far
includes an appraisal of the methodology available for predicting and controlling the movement of the pebbles, a determination of the criteria for pebble size selection, an approximate sizing of the heat exchangers, and a more detailed thermal analysis of the falling bed blanket based upon the engineering design and layout of reference [2]. According to this layout, the blanket will consist of 72 bins placed side-by-side. The bin walls will have an outer layer of actively cooled steel panels, and an inner layer of refractory brick. Figure 2 shows the cross section of a blanket element.

2. Gravity Flow of the Packed Pebble Bed

A study of this problem has established several important points:

1) The rate of particle flow can be controlled by a suitable design of the bin outlet, its value being determined by the shape of the funnel at the bottom of the bin, the diameter of the outlet orifice and the pebble size and material.

2) The pressure forces acting at the bottom of the bin increase with bed height only up to a certain point, reaching a saturation value when the bed height equals 2-1/2 times the bin diameter. This is expected to maintain the compression load on the pebbles within acceptable limits.

3) The velocity of the pebbles is constant throughout the bin volume, except for a small region around the sides of the bin, where it is reduced by 10%, and another region next to the outlet orifice, where the pebble trajectories are no longer vertical and where the velocity increases towards the center of the bin. The length of this zone of perturbed trajectories is a function of the material.

No major problems are anticipated that would preclude the use of gravity bulk solids flow in the system.

3. Pebble Diameter Selection and Heat Exchanger Sizing

The selection of pebble size is likely to be a trade-off between the conflicting requirements of reducing both the pressure drop through the pebble bed and the thermal stresses at the pebble surface. More work is being done in this area to determine the pebble diameter that will be satisfactory on both accounts. Some preliminary calculations, however, have shown that very small pebbles result in pressure drops greater than the weight of the bed, thus impeding the pebble flow, while very large particles may undergo fracture during the cooling process due to the generation of thermal stresses that are larger than the tensile fracture stress of alumina. The air velocity is, of course, a major factor determining the pressure drop in the heat exchanger. The mechanical design requirements preclude the use of high pressure gas, while the large amount of heat generated and the low specific heat of air or steam necessitate high volumetric flow rates; this can only be achieved with high gas velocities and/or heat exchanger flow areas. Figure 3 gives the results of parametric calculations of pressure drop (performed following the methodology of reference [3]) as a function of gas velocity (v), pebble diameter and the product of heat exchanger area and gas temperature rise (A × ΔT). The gas properties were taken to be those of air at 2 atm and 527°C; the total heat removal rate is 50 MW. The curves correspond to constant values of the pressure drop per foot of bed height. It is seen that, for the above postulated values of heat exchanger cross-sectional area and coolant temperature rise (A × ΔT = 28,000 m²°C), a pebble diameter of 1.1 cm results in a pressure gradient of 0.045 atm/m with an air
velocity of \(\approx 1.8\) m/s. This pressure drop is well below the value of \(\approx 0.223\) atm/m that would be required to support the weight of a bed of \(\text{Al}_2\text{O}_3\) packed with a 37% void fraction. Therefore, it is expected that this pebble size would result in adequate solid particle flow.

Calculations of the rate of heat transfer between the pebbles and the gaseous coolant have shown that the solid and gas reach thermal equilibrium within a very short distance from the gas inlet; this is due, primarily, to the large heat exchange surface per unit volume that results from the packed pebble bed geometry. Consequently, the outlet temperature of the gas will be essentially equal to the inlet temperature of the pebbles, and the heat exchanger height will be determined by the residence time required for sufficient decay of the pebble radioactivity. If the inlet air temperature is taken to be 100\(^\circ\)C and its outlet temperature 1000\(^\circ\)C, then for the above value of \(\Delta T\) the cross-sectional area of the heat exchanger should be 31 m\(^2\). A heat balance then gives a value of \(\approx 0.06\) m/min for the pebble velocity for a pebble temperature loss of 800\(^\circ\)C, and a heat exchanger height of 2.4 m would result in a residence time of 40 min, or 19 times the half-life of the activated pebbles.

4. Thermal Analysis of the Falling Bed Blanket

The thermal analysis of the falling bed blanket is based upon a two-dimensional finite difference solution of the energy equation. Correction factors are then applied to the results yielded by this model to take into account the heat losses through the small sides of each bin. It is anticipated that only a very small gas flow will sweep the pebble bed inside the blanket. As a result, the heat capacity of this gaseous current will be much smaller than that of the solid stream, and the gaseous heat convection inside the blanket can be neglected. Since, as stated above, the pebble trajectories are straight vertical lines, all the transverse heat transfer inside the bed can be described in terms of an equivalent bed thermal conductivity. The heat conduction through the ceramic walls and pebble bed in the longitudinal direction has been neglected. This is reasonable, since the longitudinal temperature gradients in the wall are found to be much smaller than the transversal ones, and since, inside the bed, convection is the dominant heat transfer process. With these assumptions, the energy balance in the bed reduces to:

\[
G_c \frac{\partial T}{\partial y} = q^{\cdot\cdot\cdot} + k_b \frac{\partial^2 T}{\partial x^2} \tag{1}
\]

under steady-state conditions, and where the equivalent bed thermal conductivity \(k_b\) will be discussed below. \(G\) is the solid mass velocity, \(c\) is the solid specific heat, and \(T\) is the temperature in eq. (1). For the ceramic walls, the energy balance gives:

\[
q^{\cdot\cdot\cdot} + k_w \frac{\partial^2 T}{\partial x^2} = 0 \tag{2}
\]

where \(q^{\cdot\cdot\cdot}\) is the volumetric heat generation and \(k_w\) is the wall thermal conductivity. The temperature of the steel panels was assumed to be constant, as some pressure drop calculations showed that it is possible to establish coolant flows inside the panels that are large enough to make the coolant temperature rise negligible.

Figure 4 gives a schematic representation of the system in its finite difference description. Equation (1) can now be written for each of the \(n\) nodes at a given axial position.
The volumetric heat generation $q''''$ decreased exponentially with distance and its value was taken from reference [4]. Thus: $q'''' = q_0'''' e^{-\alpha x}$. Equation (2) can be solved analytically using the appropriate boundary conditions. The outer wall surfaces are assumed to exchange heat with the steel panels through a gas-filled gap of width $\delta$. Then, if $T_s$ is the panel temperature and $k_w$ the thermal conductivity of the gas in the gap, the continuity of the heat flux at the outer brick wall surface required that
\[
\frac{\Delta T_0}{\Delta y} = q'''' + k_b \frac{T_2 - T_1}{\Delta x} - h \frac{T_1 - T_{w1}}{\Delta x}
\]
\[
\frac{\Delta T_2}{\Delta y} = q'''' + k_b \frac{T_3 - T_2}{\Delta x} - k_b \frac{T_2 - T_1}{\Delta x}
\]
\[
\vdots
\]
\[
\frac{\Delta T_n}{\Delta y} = q'''' + h \frac{T_{w1} - T_n}{\Delta x} - k_b \frac{T_n - T_{n-1}}{\Delta x}
\]

whereas the continuity of the heat flux at the inner wall surface provides the second boundary condition:
\[
at x = 0: \quad T = T_s + k_w \frac{dT}{dx} \frac{\delta}{k_g}
\]

where $T_b$ is the bed temperature. The resulting temperature distribution in the wall is:
\[
T = -\frac{q''''}{k_w a} e^{-\alpha a} + (B - \frac{h}{k_w} T_a) a + D - \frac{\delta h}{k_g} T_a
\]

where $h$ is the heat transfer coefficient at the wall-bed interface, with
\[
B = \frac{h}{k_w} T_b - \frac{q''''}{k_w a} e^{-\alpha a}
\]
\[
D = T_s + \frac{q''''}{k_w a^2} + \frac{q'''' \delta}{a k_g} + \delta \frac{k_w}{k_g} B
\]

and the temperature of the wall surface facing the blanket is:
\[
T_a = -\frac{q''''}{k_w a} e^{-\alpha a} + Ba + D
\]
\[
= \frac{1 + \frac{h a}{k_w} + \frac{h \delta}{k_g}}{1 + \frac{h a}{k_w} + \frac{h \delta}{k_g}}
\]

where $q''''$ is the maximum volumetric heat generation rate and $a$ is the wall thickness. Both ceramic walls are treated similarly, using appropriate values for $T_o$, $h$, $a$ and $T_b$ in each case.
Since the gas velocities inside the blanket are expected to be very small, the equivalent thermal conductivity of a pebble bed with stagnant gas, as given by references [5] and [6] is used:

\[
\frac{k_b}{k_f} = \varepsilon \left(1 + \frac{h_{rv} D_p}{k_f} \right) + \frac{1 - \varepsilon}{1 + \frac{h_{rs} D_p}{k_f} + \frac{2 k_f}{3 k_g}}
\]

(3)

where \( \phi \) is a function of \( \varepsilon \), the bed void fraction, and \( k_g/k_f \), and where

\[
h_{rs} = 0.1952 \frac{D_p}{2 - \varepsilon} (T_0 K) \]

\[
h_{rv} = 0.1952 \frac{(T_0 K)^3}{1 + \frac{\varepsilon}{1 - \varepsilon}} \frac{1 - \varepsilon}{2 - \varepsilon}
\]

\( k_f \) = gas thermal conductivity
\( \rho \) = emissivity of the solid surface.

The thermal conductivity of the bed in the immediate vicinity of the wall \( (K_d^a) \) differs from that inside the wall given by eq. (3) and is given by an expression similar to eq. (3), only using a corrected value for \( \varepsilon \) and \( \phi \). It should be noted that the equivalent thermal conductivity thus obtained takes into account the effect of thermal radiation, which is apt to be very significant at the high temperatures anticipated for the blanket. The heat transfer coefficient between a packed bed of flowing particles and a surface is given by reference [7] as:

\[
h = \frac{1}{R_a} \left[ 1 - \frac{R_w}{2R_a} \ln \left(1 + \frac{2R_a}{R_w}\right)\right]
\]

with

\[
R_a = \frac{4k_b}{\varepsilon \rho} \quad \text{and} \quad R_w = \frac{D_p}{2k_b}.
\]

(4)

\( R_a \) represents a thermal resistance of the bed in the region near the surface. \( R_w \) is an additional contact resistance at the bed-wall interface. \( D_p \) is the pebble diameter and \( k_b^a \) is the equivalent bed thermal conductivity near the wall. These expressions are based on an analysis of heat transfer at low temperatures where the thermal radiation effects are not significant and are neglected. In the present study, the radiative heat transfer is taken into account by assuming it to be independent of, and additive to, the convective mode [8]. A radiative thermal resistance (the inverse of an equivalent heat transfer coefficient for radiation) is therefore combined with the value of \( R_w \) given by eq. (4).

The above equations allow the determination of a complete temperature distribution throughout the blanket provided the inlet temperature and the velocity of the solid stream are known. A computer code was prepared to perform these calculations. Although this program was written specifically for the design under consideration, it was made general enough so that it can handle some changes in geometry, materials and nodal layout. The following results are based on an inlet temperature of 260°C. The pebble velocities used were obtained
by iteration, such that the outlet bulk temperature was always 1090 ± 11°C. The resulting pebble velocities are of the order of 0.5 m/min. The heat load on the steel panel cooling system could be computed from the temperature distribution if each blanket element was of a length much greater than its thickness. In actual fact, the ratio of these two dimensions is expected to be only ∼ 1.5; therefore, a correction must be applied to the losses obtained from the present two-dimensional model. Since the heat generation decreases exponentially with increasing distance from the plasma region, it seems reasonable to also expect an exponential decrease in the heat losses from the blanket to the wall (and, of course, in the heat generation in the radial walls themselves). With these assumptions, the heat loss per square foot of radial wall was estimated to be 43% of that for the inner wall.

5. Discussion of Results

A series of parametric calculations was performed using tentative base values (and ranges) for the following design parameters:

- pebble diameter: 1.27 cm
- wall loading: 2 MW/m² (range: 1 to 3 MW/m²)
- inlet pebble temperature: 260°C
- outlet pebble temperature (bulk): 1090°C
- steel panel temperature: 93°C
- ceramic wall density: 75%
- ceramic wall material: Al₂O₃ (and SiO₂)

Unless otherwise stated, the steel panels were assumed to be kept at 93°C by their active cooling system. The volumetric heat generation was taken from reference [4]. The gap between ceramic wall and steel panel was initially taken to be 0.13 mm, as expected for a normal contact between the brick and the steel surface. A later set of calculations, presented at the end of this paper, shows the effect of variations in the gap thickness.

Furthermore, in an attempt to obviate some of the possible problems identified through these calculations, the overall thickness of the blanket was varied within the range of 20 to 40 cm. Figure 5 presents the pebble bed and wall temperatures for the base case dimensions of Figure 2 and a wall loading of 2 MW/m² at three axial positions corresponding to 30, 304 and 638 cm from the inlet (approximately, inlet, midpoint and outlet of the blanket). Both the wall and the pebbles exhibit very large transversal temperature gradients. This has two unfavorable effects: potentially large thermal stresses in the wall and high (near melting point) temperatures in wall and pebbles. Since the pebble temperature rises are largely determined by the local heat generation and since the bulk outlet pebble temperature is fixed by the temperature requirements in the gas stream out of the heat exchanger, it follows that the maximum pebble temperature can only be reduced by either reducing the thickness of the pebble bed (and, thereby, the spread of the values of the volumetric heat generation) or by adjusting the pebble velocity to the local heat generation rate. The latter effect can be obtained by dividing the blanket into several regions and designing the outlet orifice of each region so as to produce the required mass flow rate. It is also interesting to note that at the top of the bed, the wall surface is hotter than the bed, and the heat generated in the wall contributes to raise the temperature of the pebbles. Since the brick wall in the upper blanket region loses heat to both the steel panel and the pebbles, its temperature
must peak at some point inside the brick. The highest brick temperatures must occur, however, at the bottom of the bed, since the wall heat generation is assumed to be independent of height and the wall boundaries are hottest at the bottom. It follows then that the maximum wall temperature is reached at the bottom of the inside wall surface. This is the value given in the following curves and tables.

Figure 6 shows the heat output (for the entire reactor) from the pebble stream and from the steel panels as a function of wall loading. The relative proportion of heat carried by the panel coolant is larger for the lower wall loadings. This is due to the following: the inlet pebble temperature is taken to be constant (independent of wall loading); on the other hand, the wall heat generation (and wall temperature) increases with wall loading; as a result the transfer of heat from the wall to the pebbles near the top of the blanket, mentioned in the previous paragraph, is larger for large wall loadings. Consequently, the net heat transfer from the pebbles to the wall (also shown in the figure) decreases with increasing wall loading. A heat balance confirms the validity of this reasoning. This effect may be significant through its influence on the thermal efficiency of the system, particularly if no use is made of the panel-coolant heat.

Figure 7 shows the maximum pebble temperature at the outlet. The melting point of alumina is reached for wall loadings of \( \approx 1.5 \text{ MW/m}^2 \). The inside surface temperature of inner and outer ceramic walls at the top and bottom of the bed are also given. Again, the melting point of alumina is reached at \( \approx 1.5 \text{ MW/m}^2 \). Calculations performed with thinner ceramic walls (keeping the overall blanket thickness and the gap between brick and panel constant) showed a slight decrease of the total heat carried by the pebble stream. Within the accuracy of the present two-dimensional calculations, it appears that the increase in bed volume is offset by the larger amount of heat loss from the pebbles through the thinner and cooler ceramic wall. As stated above, the heat losses through the radial walls were only estimated in the present study. A subsequent three-dimensional analysis of the blanket will allow a more accurate evaluation of this phenomenon. It seems, however, that a reduction of the wall thickness alone is not likely to result in a significant increase of the pebble stream output power unless it is accompanied by an increase in the gap thickness (more on this later).

Figure 8 shows the reduction in the maximum temperatures and temperature differences between walls that can be achieved by a reduction in the overall blanket thickness while keeping the bulk outlet pebble temperature constant. Figure 9 shows the penalty paid in terms of a lower power output. As mentioned above, an alternative solution to the problem posed by high wall and pebble temperatures would involve the use of baffles to divide the bed into regions of different pebble velocities. This situation is presently being analyzed.

An increase in the steel panel temperature results, of course, in smaller heat losses. Table 1 gives the values of power transported by the pebble stream and panel coolant together with the maximum wall and pebble temperatures for the hotter panels. The percent heat losses decrease from 35 to 31 when the panel temperature increases from 93 to 538°C. It should be noted that this large increase in the panel temperature results in a much smaller relative decrease of the pebble-bed to panel temperature difference, due to the very high temperatures reached by the pebbles, hence the modest gain achieved in the thermal efficiency. The pebble temperatures are seen to increase even more slowly, so that the penalty to be paid in terms
of increased brick and pebble temperature is small; qualitatively, it can be seen that to hotter panels there correspond lower heat losses, which require smaller driving temperature differentials; thus, when the panel temperature is raised by a certain amount, the pebble temperatures should increase by a smaller amount, so that the overall pebbles-to-panel temperature difference decreases. This effect is enhanced by the fact that the pebbles-to-wall heat transfer coefficient increases with increasing temperature.

Calculations were also performed for ceramic walls of silica bricks. Figure 10 shows the power distribution as a function of wall loading. The power yield of the pebble stream is seen to be considerably improved over the base case with alumina walls. This is due to the very low thermal conductivity of the silica bricks, and is attained at the expense of excessively high wall temperatures (well above the melting point). These results indicate, however, that a good thermal efficiency for the blanket can be achieved with the use of a material of low thermal conductivity, but such a material would have to have a very high melting point, or a very low heat generation when submitted to a neutron flux. A logical way of meeting these two requirements of low conductivity and heat generation would be to increase the gap thickness at the expense of the wall thickness. In all the above calculations it was assumed that the ceramic bricks rest against the panel surface. If a frame of high-temperature alloy (such at TZM) can be used to support the wall, then the steel panels, now serving only the function of vacuum boundary, need not touch the brick and the resulting brick-panel gap can be increased to provide additional thermal resistance while decreasing the wall heat generation. Table 2 shows the thermal efficiency gain associated with an increase in gap thickness. The heat losses are reduced from 35 to 23% when the gap increases from 0.13 to 12.7 mm. It is seen that the relative increase in thermal efficiency that is obtained with a given increase in gap thickness is smaller for the larger gaps. This is due to the enhanced radiative component of the wall-to-panel heat transfer that occurs with these large gaps and their correspondingly high wall temperatures. The overall wall-plus-gap thickness is constant for the four cases shown in the table so that the brick wall is thinner, and generates less heat for the thicker gaps; this explains the decrease in the maximum wall temperatures observed for gap thicknesses beyond 2.54 mm. In order to reduce the amount of radiant heat lost, a second wall arrangement was considered in which a thin layer of high temperature alloy is inserted in the gap. In addition, it was assumed that the blanket elements could be gathered in clusters of three without a panel coil between the central element and the two adjacent ones. This would reduce the amount of heat loss to side-walls. The inserted shielding layer was taken to have the same emissivity as the panels. Table 3 shows the resulting reduction in heat losses. (Actually, the first two cases in the table show only the gains achieved by clustering the blanket elements, without radiation shield, which can only be inserted in the wider gaps). Although the shielding layer increases considerably the resistance to radiative heat transfer, this effect is partly offset by the increase in wall and pebble temperatures, so that the thermal efficiency shows only a moderate gain. As mentioned above, for a given bulk outlet temperature of the pebble stream, the maximum pebble temperatures can be reduced by adjusting the local pebble velocity to the local heat generation rate. This is achieved by increasing the velocity of the pebbles adjacent to the hot wall, and will result in lower temperatures for the hot wall and heat losses. This possibility is now being investigated. Thus, while the
above results have identified the design features and parameters most likely to affect the thermal behavior of the blanket, they do not represent the best performance that can be achieved with the falling bed concept. The work currently in progress is expected to result in a significantly more favorable power and temperature distribution.

6. Conclusions

1) The concept of a falling bed blanket appears feasible as regards the control of the bulk solids flow.

2) From pressure drop and thermal stress considerations, the pebble size is anticipated to be within the 0.5 to 1.5 cm range. The heat exchanger height, dictated by pebble residence time requirements, will be in the 3 to 5 m range. The total cross-sectional area of the heat exchangers, for a 500-MW output in the pebble stream will be \( \approx 300 \, \text{m}^2 \), distributed in several heat exchangers placed around the periphery of the reactor.

3) The heat load on the cooling system is significantly dependent on the wall loading for the reactor, lower wall loadings resulting in higher fractional heat losses.

4) The maximum ceramic wall and pebble temperatures can be reduced by decreasing the blanket thickness or dividing the blanket into regions of different pebble velocity.

5) Due to several features of the falling bed blanket concept (such as its high maximum temperatures and flat geometry), the heat lost through the system boundaries can be a critical issue in the design. It is expected, however, that a suitable configuration of the containing walls and pebble velocity distribution will result in acceptably low heat losses.

References


Table 1. Effect of Panel Temperature on Heat Losses

<table>
<thead>
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<th>Panel Temp. (°C)</th>
<th>Power (MW)</th>
<th>Temperature (°C)</th>
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<tbody>
<tr>
<td>93</td>
<td>266</td>
<td>144 (35)</td>
</tr>
<tr>
<td>260</td>
<td>272</td>
<td>138 (34)</td>
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<tr>
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<td>128 (31)</td>
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Table 2. Effect of Wall-to-Panel Gap on Thermal Efficiency

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<tr>
<td>2.54</td>
<td>296</td>
<td>114 (28)</td>
</tr>
<tr>
<td>7.62</td>
<td>307</td>
<td>102 (25)</td>
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<td>12.7</td>
<td>312</td>
<td>95 (23)</td>
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Table 3. Effect of Radiation Shield and Clustering of Blanket Elements on Thermal Efficiency

<table>
<thead>
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<th>Gap (mm)</th>
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<th>Temperature (°C)</th>
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</thead>
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<tr>
<td>12.7</td>
<td>330</td>
<td>76 (19)</td>
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Fig. 1. Scaled vertical section of reactor and heat exchanger.

Fig. 2. Detail of a blanket element.
Fig. 3. Effect of system parameters on pressure drop.

Fig. 4. Schematic of node structure used in thermal hydraulic analysis.
Fig. 5. Typical temperature map (base case).

Fig. 6. Effect of wall loading on power distribution (base case).
Fig. 7. Effect of wall loading on temperature (base case).

Fig. 8. Effect of blanket thickness on temperatures.
Fig. 9. Effect of blanket thickness on power distribution.

Fig. 10. Effect of wall loading on power distribution (silica walls).