University of California, San Diego

UCSD-ENG-108

Helium Cooled Tungsten Heat Sinks

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April 3, 2003

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Subcontracting Work Performed by UCSD for Plasma Processes Inc. as Part of DoE SBIR Phase I Grant "Helium Cooled Tungsten Heat Sinks"

Final Report

April 3, 2003

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I. Background

Fusion power plant studies have found helium to be an attractive coolant based on its safety advantages and compatibility with structural materials at high temperature. However, gas coolants in general tend to provide modest heat transfer performance due to their inherently low heat capacity and heat transfer coefficient. Innovative techniques have been proposed previously using porous metal heat transfer media infiltrated by the coolant [e.g. 1-5]. The general design strategy is to minimize the coolant flow path length in contact with the porous medium, and optimize the microstructure configuration to minimize the friction factor in that zone while simultaneously maximizing the heat transfer coefficient. Initial estimates indicate that a He-cooled porous medium configuration would be an attractive divertor candidate concept, being able to comfortably accommodate heat fluxes of ~ 5 MW/m^2 with the possibility of accommodating higher heat fluxes of up to 20-30 MW/m^2 albeit at the cost of higher pressure drops and higher system pressure, and/or of lower coolant inlet temperature and lower-quality heat extraction [6]

Our effort was focused on helping PPI optimize the design and microstructure characteristics of their proposed W porous medium for high heat flux application. The design optimization analyzes were carried out with our general porous flow model (MERLOT [6]) which was specifically updated for this task.

II. Proposed Configuration

Based on our past collaboration and initial optimization studies, PPI has evolved a helium cooled heat sink with plasma spray formed porous tungsten, as illustrated in Fig. 1. These initial studies showed that the ratio of effective heat transfer coefficient to pressure drop for a porous medium tends to increase with increasing porosity and with lower velocities, as shown in Fig. 2 [7]. When including as variables the channel width, the velocity and the porosity, further studies show that, for a given pressure drop, there is a definite advantage (based on the effective heat transfer coefficient) of going with a smaller channel dimension with a high porosity medium and a high fluid velocity [6]. PPI is trying to develop fabrication techniques that will allow for high porosities (about 70% or higher). Fig. 3 shows a cross-sectional view of an example porous medium structure that has been previously achieved (with porosities ranging from 40-55%).

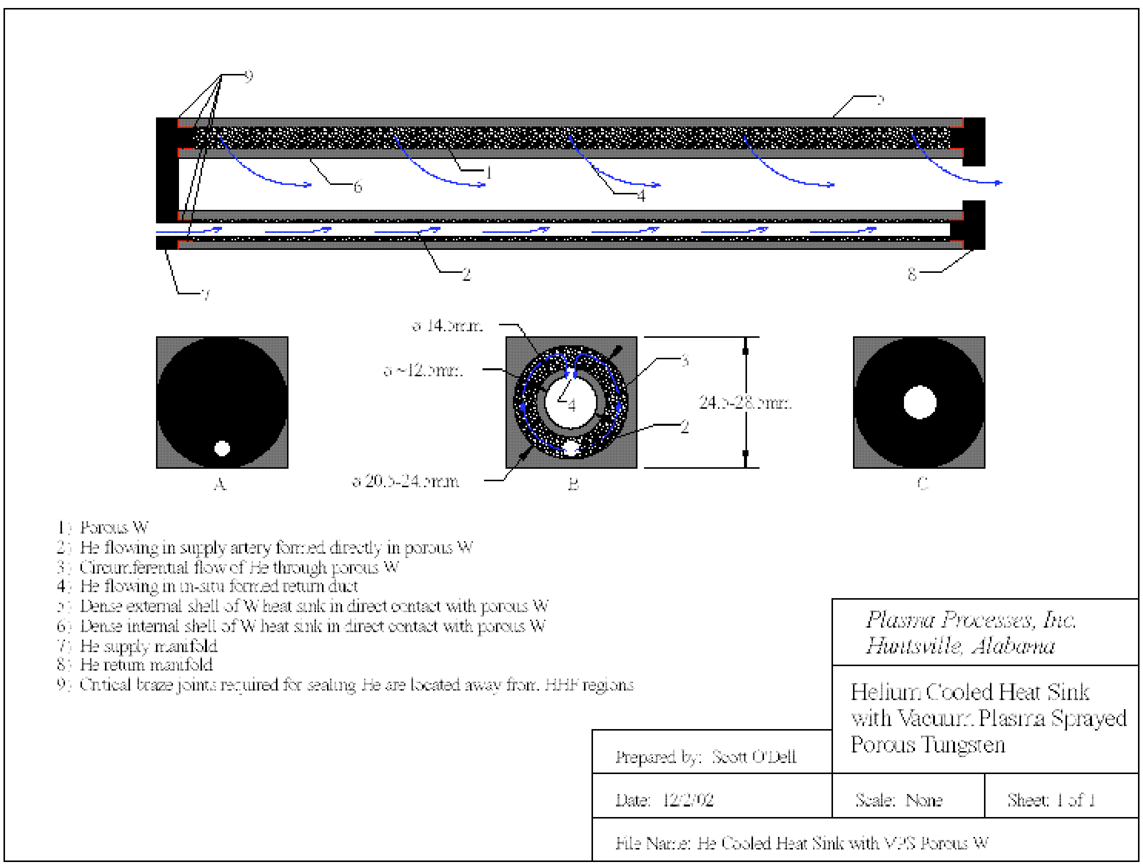

Figure 1 PPI's helium cooled heat sink with plasma spray formed porous tungsten.

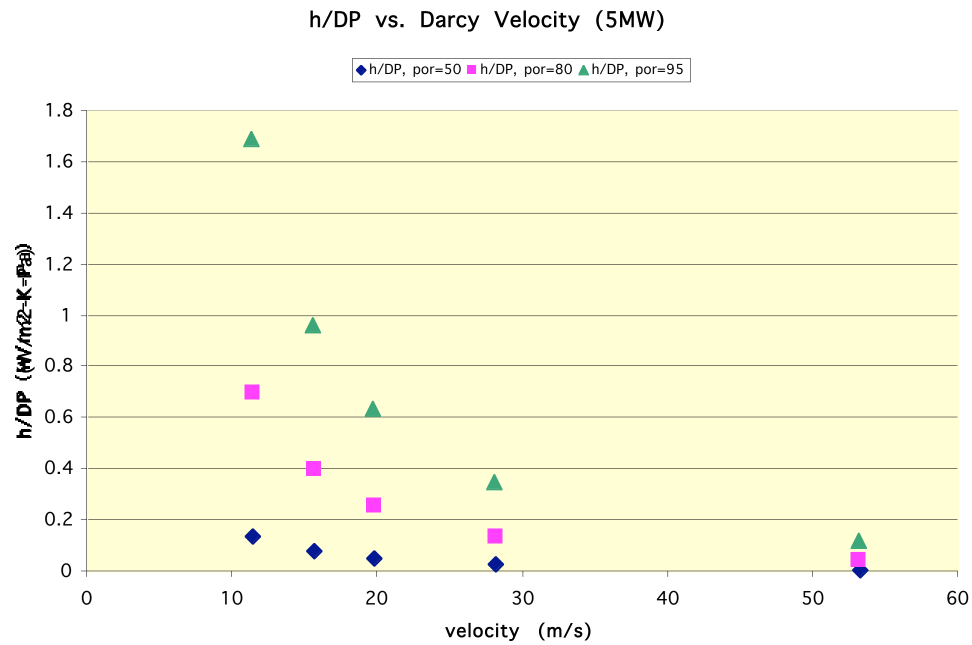

Figure 2 Ratio of effective heat transfer coefficient at outlet to pressure drop as a function of velocity for different porosities (q'' = 5 MW/m^2, solid microstructure characteristic dimension = 0.1 mm, k_s = 100 W/m-K, He inlet temperature = 823 K) [7]

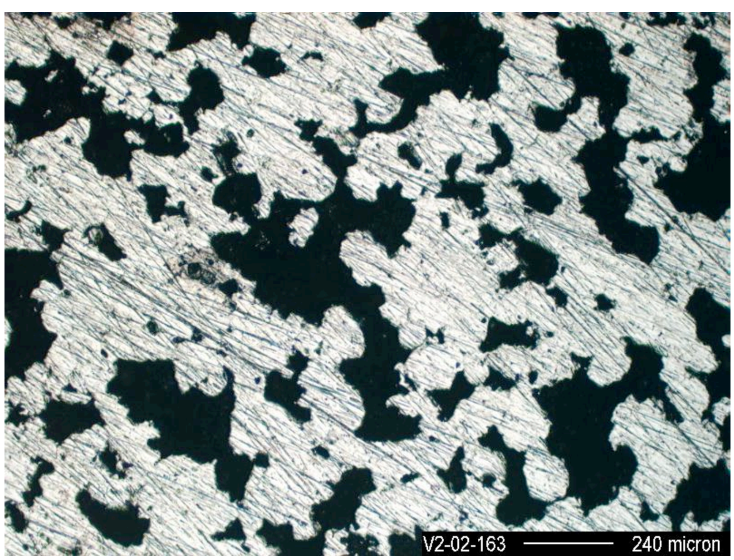

Fig. 3 Cross-sectional view of an example porous medium structure from PPI (with porosities ranging from 40-55%).

III. Optimization Studies

The trends identified from previous studies [6,7] have helped in guiding the evolution of the porous microstructure characteristics for high heat transfer performance. However, there are a number of further questions which arise when considering specifically the example mock-up shown in Fig. 1, such as:

1. Should the porous medium be included in the whole azimuthal direction?
2. Should the flow in the porous medium be divided in two shorter parallel sections rather than having the full flow rate routed through a longer section?
3. What is the effect of having an azimuthally varying heat flux on the porous medium channel as compared to a uniform average heat flux?
4. Should the flow inlet be in the high heat flux region or the low heat flux region?
5. Should the inlet or outlet be at a location with maximum heat flux or away from it?
6. For a given average porosity, is there an optimum radial porosity profile that would maximize the heat transfer performance?

These questions were addressed as part of the optimization studies carried out under this subcontract. They are discussed below from a thermo-fluid perspective with the intent of providing recommendations on how the design should be optimized but with the understanding that fabrication constraints might not allow complete implementation of all these recommendations.

III.1 Should the porous medium be included in the whole azimuthal direction?

From previous studies [6,7], it has been clearly shown that there is a major pressure drop penalty associated with flowing a fluid through a porous medium as opposed to a regular channel. Thus, it makes sense that such a penalty should only be accepted in areas where the high heat transfer performance of the porous medium is required. Typically, for a one-sided heat flux application (such as for a fusion divertor), only the upper region of the porous medium will see substantial heat flux and, if possible from the fabrication point of view, it is strongly recommended that the porous medium be installed only in the top half of the channel (i.e clockwise from 0° to 180° starting from "west" in cross-section B of Fig. 1). This convention of measuring the angle θ clockwise from the "west" side will be used throughout this report.

III.2 <u>Should the flow in the porous medium be divided in two shorter parallel sections rather than having the full flow rate routed through a longer section?</u>

For a given heat flux imposed on the upper side of the heat sink configuration shown in Fig. 1, the heat flux seen by the porous media will vary azimuthally, from a maximum at the top (θ=45°) to a low value at the sides (θ=0 and 180°) to even smaller values in the lower region (from θ=180° to 360°) which will not be considered in this study.

MERLOT was run for two cases to help make the comparison between parallel and series flow. The basic geometry modeled by MERLOT is shown in Fig. 4 and the reference parameters (shown In Table 1) were assumed based on the geometry of Fig. 1 and typical helium conditions for a fusion divertor application.

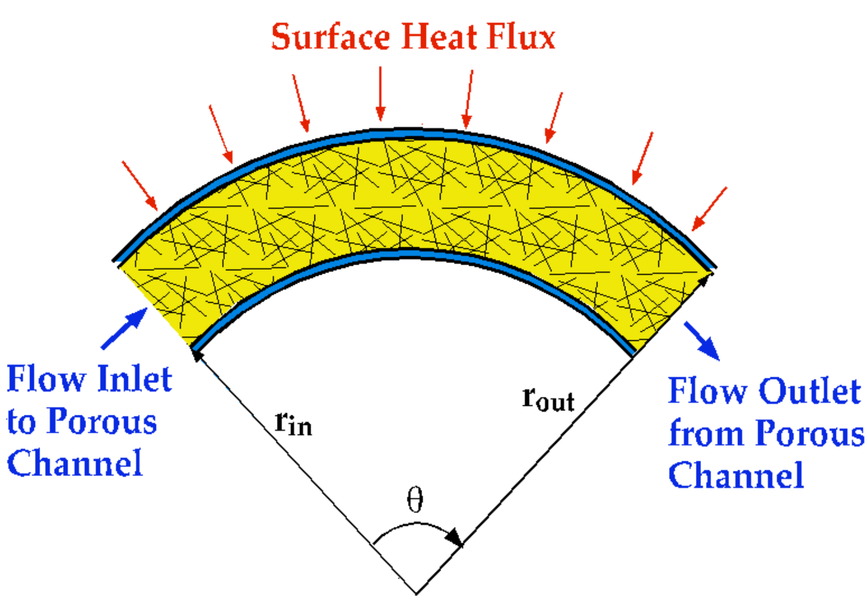

Figure 4 2-D cylindrical geometry porous flow configuration used in MERLOT

Table 1 Reference Parameters used for the Analysis (except if otherwise indicated)

Incident heat flux	10 MW/m^2
Helium inlet temperature	823 K
Helium outlet temperature	1223 K
Helium inlet pressure	4 MPa
Inner porous medium channel radius, r_{in}	7 mm
Outer porous medium channel radius, r_{out}	12 mm
Solid thermal conductivity, k_s	100 W/m-K
Porous structure characteristic dimension	50 μm
Porosity	50%

The first case assumes that the flow is divided in two streams and models half the flow from θ=0° to θ=90°. The heat flux seen by the channel is assumed to vary linearly from 0 at θ=0° to 20 MW/m^2 at θ =90°. The flow rate is based on the He inlet and outlet temperatures of 823 K and 1223K, respectively with a corresponding pressure drop of 0.68 MPa. The wall temperature and effective heat transfer coefficient results are summarized in Figures 5 and 6.

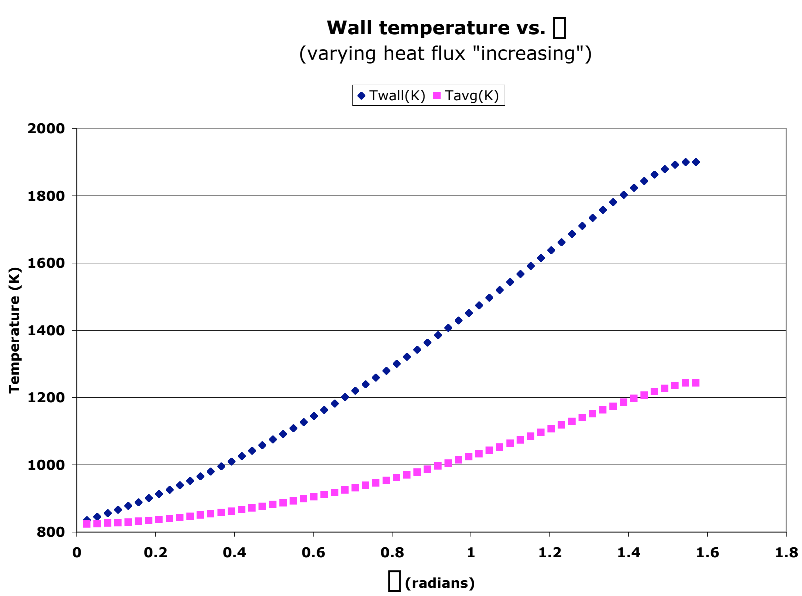

Figure 5 Wall and fluid temperatures along flow path for a case with the heat flux on the channel increasing linearly from 0 at θ=0° to 20 MW/m^2 at θ =90°(π/2 radians). The He inlet and outlet temperatures are 823 K and 1223K, respectively and the porosity is 50%.

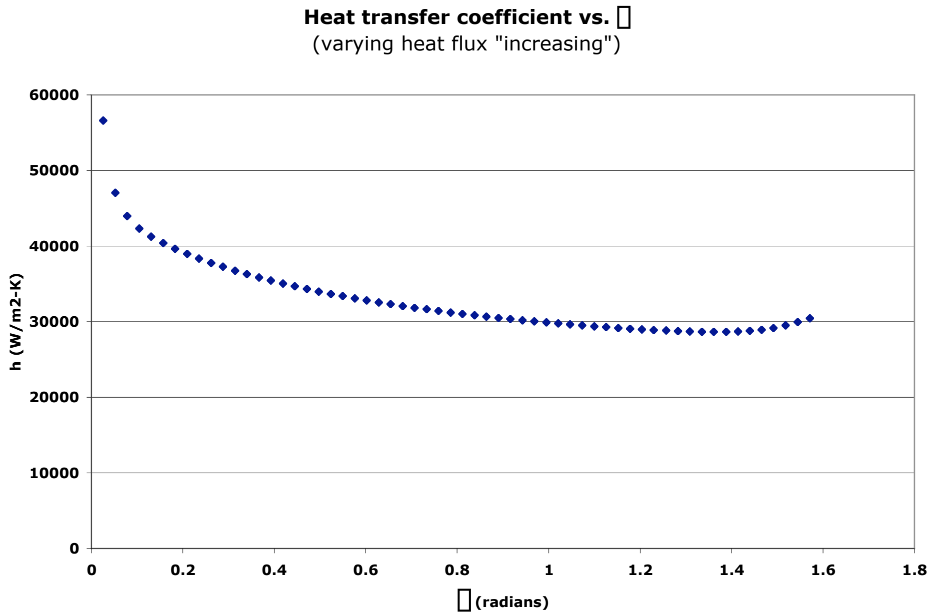

Figure 6 Effective heat transfer coefficient along flow path for a case with the heat flux seen by the channel increasing linearly from 0 at θ=0° to 20 MW/m^2 at θ =90° (π/2 radians). The He inlet and outlet temperatures are 823 K and 1223K, respectively and the porosity is 50%.

From Fig. 5, the maximum wall temperature is about 1900 K, and from Fig. 6, the effective heat transfer coefficient decreases from a maximum value of about 60,000 W/m^2-K close to the entrance to a value of about 30,000 W/m^2-K at the exit. A similar analysis was done for the case with the total flow rate routed through the porous medium from θ =0° to θ =180°. The heat flux to the channel was assumed to increase linearly from 0 at θ=0° to 20 MW/m^2 at θ =90°, and then to decrease linearly from 20 MW/m^2 at θ=90° to 0 at θ =180°. The flow rate was based on He inlet and outlet temperatures of 823 K and 1223K, respectively with a corresponding pressure drop of 1.8 MPa (about 3 times higher than for the previous case). The wall temperature and effective heat transfer coefficient results are summarized in Figures 7 and 8. From Fig. 7, the maximum wall temperature is about1700 K, and from Fig. 8 the effective heat transfer coefficient varies from about 75,000 W/m^2-K close to the inlet about 35,000 W/m^2-K at θ=90°. It then gradually decreases as θ approaches 180° where the heat flux is zero, and by definition the calculated heat transfer coefficient is also zero.

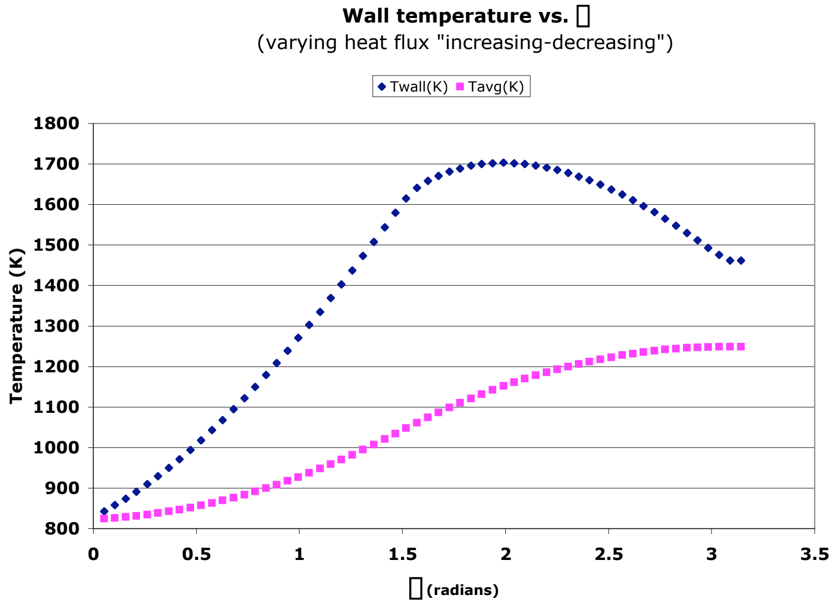

Figure 7 Wall and fluid temperatures along flow path for a case with the heat flux on the channel increasing linearly from 0 at θ=0° to 20 MW/m^2 at θ =90°(π/2 radians), and then decreasing linearly 0 at θ =180°(π radians). The He inlet and outlet temperatures are 823 K and 1223K, respectively and the porosity is 50%.

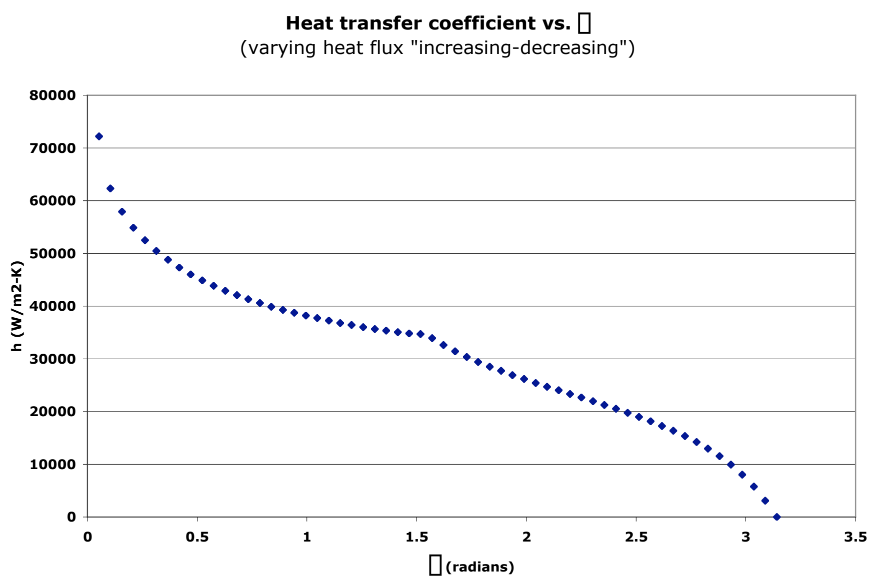

Figure 8 Effective heat transfer coefficient along flow path for a case with the heat flux on the channel increasing linearly from 0 at θ=0° to 20 MW/m^2 at θ =90°(π/2 radians), and then decreasing linearly 0 at θ =180°(π radians). The He inlet and outlet temperatures are 823 K and 1223K, respectively and the porosity is 50%.

In summary, it is preferable from a heat transfer performance standpoint to have the whole flow routed through a longer porous medium section than to have half the flow routed in parallel through half the section. The maximum wall temperature is 1700 K in the former case and1900 K in the latter case. However, this is achieved at the cost of a higher pressure drop (by about a factor of 3), 1.8 MPa in the former case compared to 0.68 MPa in the latter case.

III.3 What is the effect of having an azimuthally varying heat flux on the porous medium channel as compared to a uniform average heat flux?

A comparison of a case with varying heat flux to a case with a uniform heat flux can help shed some light on this question. The case with the heat flux on the channel increasing linearly from 0 at θ=0° to 20 MW/m^2 at θ =90°(π/2 radians), and then decreasing linearly 0 at θ =180°(π radians) is compared to a case with a uniform heat flux of 10 MW/m^2 over θ =0° to180°. The results for the former case are shown in Figs. 7 and 8 and have been discussed in the previous section. The results for the uniform case are shown in Figs. 9 and 10. The maximum wall temperature in the case of the uniform heat flux (Fig. 9) is about 1615 K and is lower than in the case of the varying heat flux (~1700 K from Fig. 7). The effective heat transfer coefficient in the case of the uniform heat flux (Fig. 10) ranges from about 85,000 W/m^2-K at the inlet to about 26,000 W/m^2-K at the outlet; overall, the hese values are higher than in the case of the varying heat flux (Fig. 8).

In summary, it seems that, for the same pressure drop and average heat flux to the coolant, the case with uniform heat flux would result in a lower wall temperature (better heat transfer performance) than the case with a peaked heat flux profile.

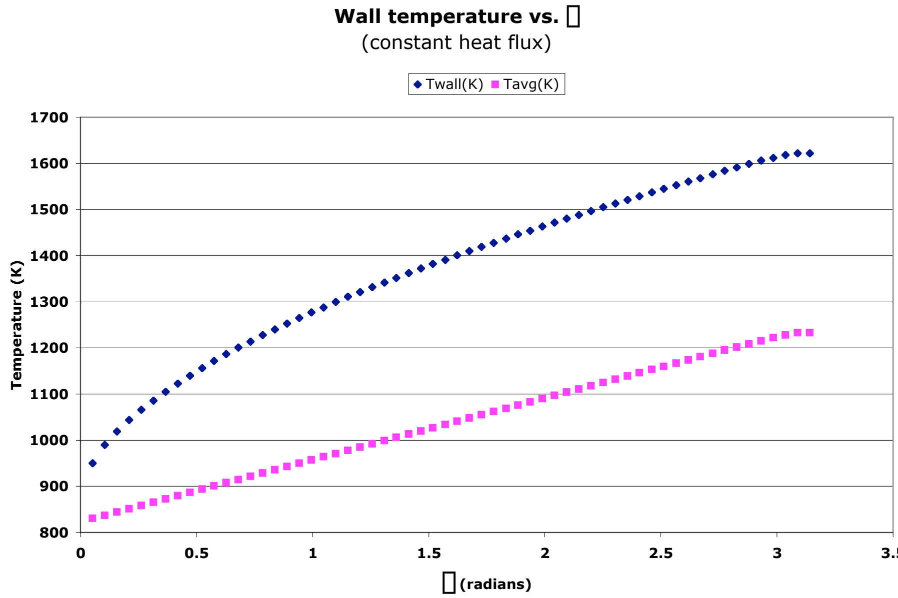

Figure 9 Wall and fluid temperatures along flow path for a case with a uniform heat flux of 10 MW/m^2. The He inlet and outlet temperatures are 823 K and 1223K, respectively and the porosity is 50%.

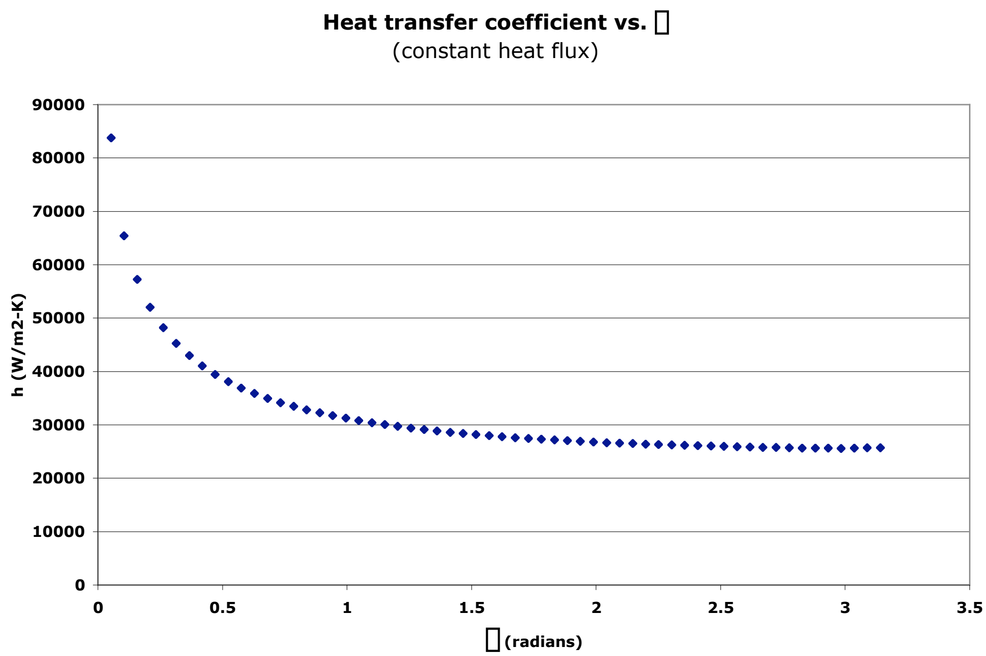

Figure 10 Effective heat transfer coefficient along flow path for a case with a uniform heat flux of 10 MW/m^2. The He inlet and outlet temperatures are 823 K and 1223K, respectively and the porosity is 50%.

III.4 Should the flow inlet be in the high heat flux region or the low heat flux region?

To help address this question, it would be useful to compare an increasing heat flux case to a decreasing heat flux case. For example, Figs. 5 and 6 show the results for an increasing heat flux case (from 0 at θ=0° to 20 MW/m^2 at θ =90°). This would represent a case where the inlet is at a low heat flux region and the outlet at the high heat flux region. A similar analysis but with a decreasing heat flux profile (from 20 MW/m^2 at θ=0° to 0 MW/m^2 at θ =90°) would simulate a case with the fluid entering in the high heat flux region and exiting from the low heat flux region. The results from such a case are summarized in Figs. 11 an 12.

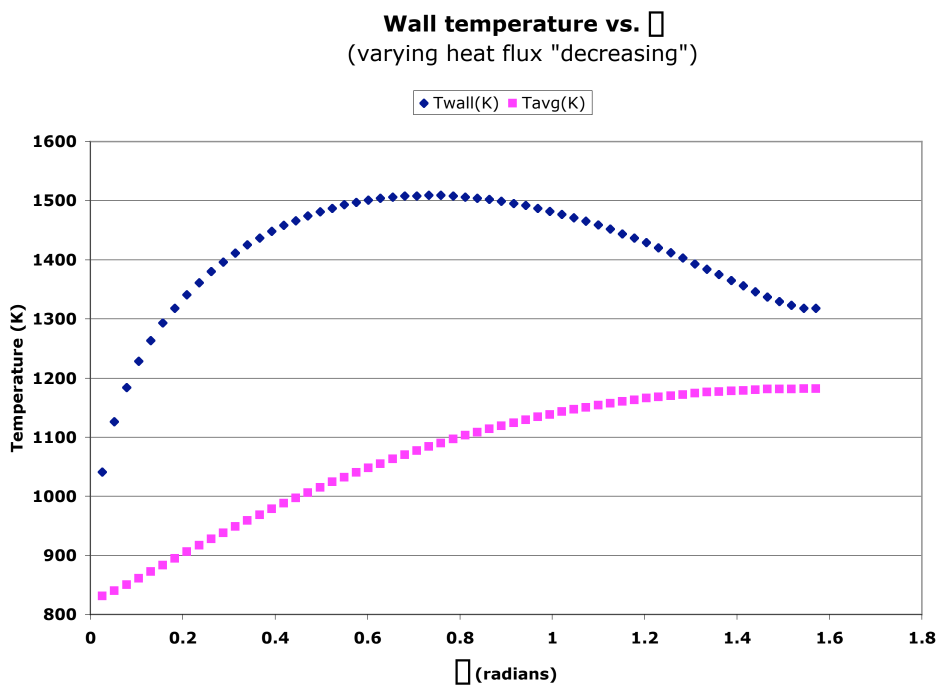

Figure 11 Wall and fluid temperatures along flow path for a case with the heat flux on the channel decreasing linearly from 20 MW/m^2 at θ=0° to 0 MW/m^2 at θ =90°(π/2 radians). The He inlet and outlet temperatures are 823 K and 1223K, respectively and the porosity is 50%.

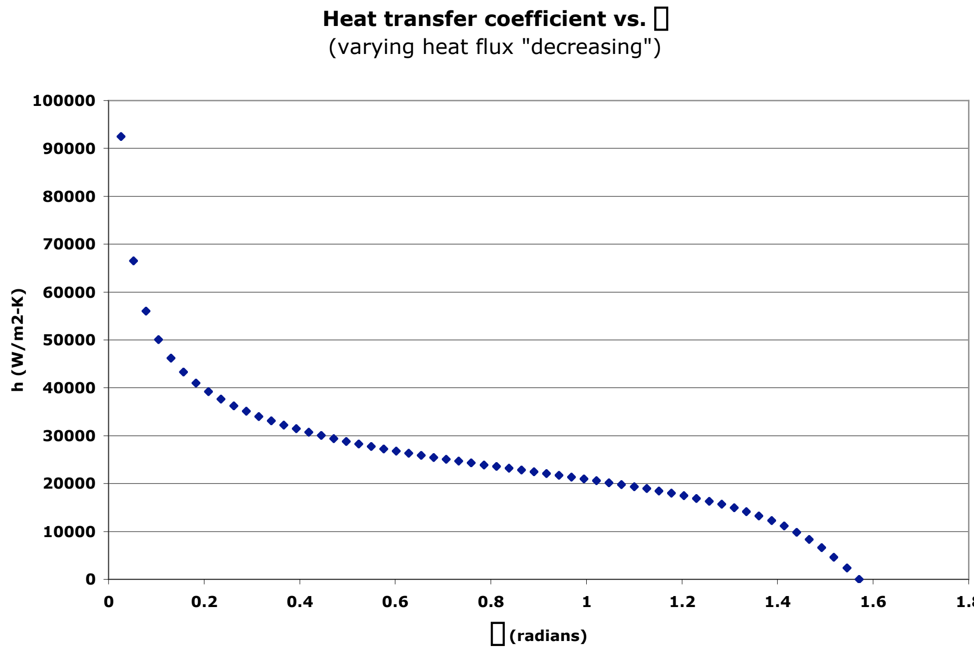

Figure 12 Effective heat transfer coefficient along flow path for a case with the heat flux seen by the channel increasing linearly from 20 MW/m^2 at θ=0° to 0 MW/m^2 at θ =90° (π/2 radians). The He inlet and outlet temperatures are 823 K and 1223K, respectively and the porosity is 50%.

The heat transfer performance in the case of decreasing heat flux is superior to the case of increasing heat flux, with corresponding maximum wall temperatures of about 1500 K (From Fig. 11) and about 1900 K (from Fig. 5), respectively. The effective heat transfer coefficient values also reflect this superiority, with a maximum value of over 90,000 W/m^2-K at the entrance for the decreasing heat flux case (Fig. 12) compared to less than 60,000 W/m^2-K at the entrance for the increasing flux case (Fig. 6). Since the flow rate and porous medium characteristics are unchanged, the pressure drop is also the same for both cases.

Thus, it seems clear that there are advantages of routing the flow in the direction of decreasing heat flux.

III.5 Should the inlet or outlet be at a location with maximum heat flux or away from it?

MERLOT solves the flow conditions in 1-D (radially) and the energy equation in 2-D (radially and azimuthally). This is reasonable for most porous flow cases since the flow development lengths tend to be short in such cases. However, for a case such as shown in Fig. 1 B where the flow enters or exits radially in the high heat flux region, there are concern that there might be a stagnation region at this location as the flow turns from the azimuthal direction to the radial direction (or vice versa). Even if considered only as a fluid entry point, any benefit associated with radial jet cooling would be dampened by the presence of the porous medium. This cannot be modeled by MERLOT and needs to be confirmed experimentally. However, until such experimental confirmation, it seems prudent that such an entry or exit point be placed in a region away from the high heat flux to avoid the possibility of the wall overheating in the stagnation region.

III.6 For a given average porosity, is there an optimum radial porosity profile that would maximize the heat transfer performance?

This question relates directly to the desirability of adjusting the local porosity characteristics to optimize the thermo-fluid performance. This was assessed by considering a simple porosity variation example: a linear radial variation. A number of cases were run with different linear porosity profiles for a given average porosity. The results are summarized in Fig. 13 in terms of the effective heat transfer coefficient at the outlet for a uniform heat flux case of 10 MW/m^2 and inlet and outlet helium temperatures of 823 and 1223 K, respectively. This effective heat transfer coefficient is directly representative of the maximum wall temperature at the flow outlet in this case. From the figure, it is very interesting to see that the heat transfer performance is maximized for an outer wall porosity of about 70% for a case with a 50% average porosity (the corresponding porosity at the inner wall would be about 30%). It seems that as the outer wall porosity is increased two conflicting mechanisms occur: on one hand, the lower porosity there results in a higher local velocity and better convective heat transfer to the flow; on the other hand, the increasing porosity at the outer wall reduces the conducting solid area and its fin-like effect. The combination of these effects would result in an optimum linear porosity variation profile. From Fig. 13, this optimum tends to shift slightly to higher outer wall porosity as the average porosity is increased (from about 70% for an average porosity of 50 % to about 80% for an average porosity of 65% and about 85% for an average porosity of 80% as shown in the figure). The corresponding pressure drop values for these different average porosity cases are1.82 MPa, 0.49 MPa and 0.13 MPa for average porosities of 50%, 65% and 80%, respectively.

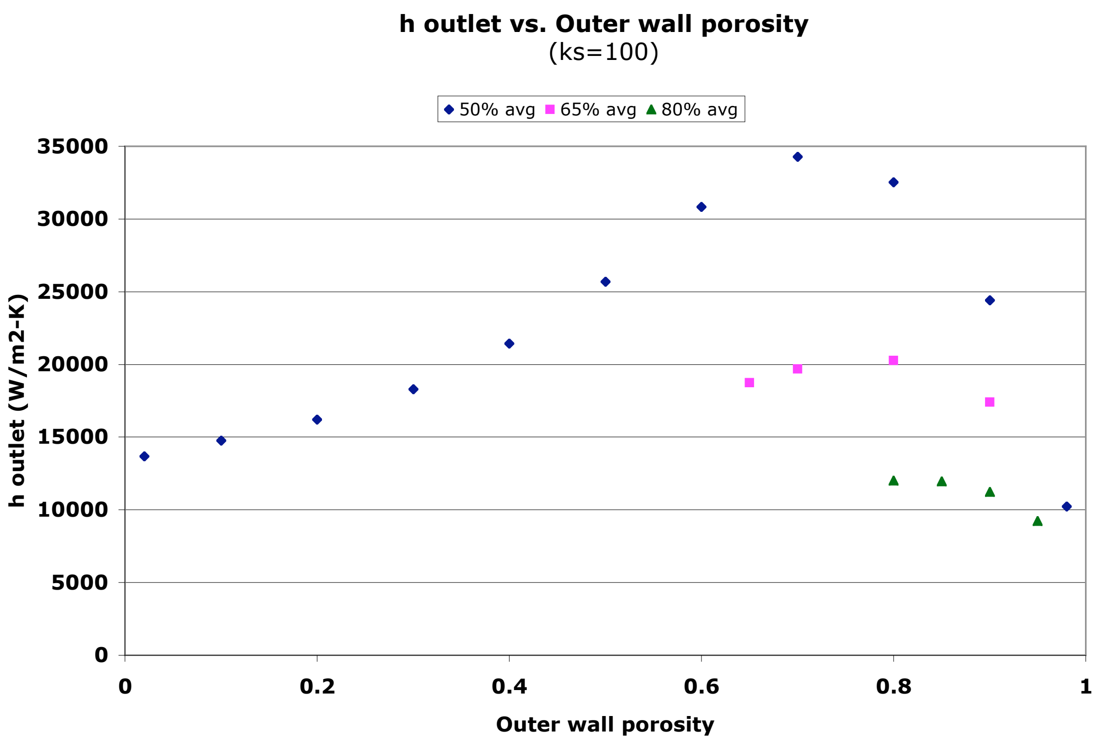

Figure 13 Effective heat transfer coefficient at outlet as a function of outer wall porosity for a linear radial porosity variation from the inner to the outer wall. Cases with different average porosities are shown.

The effect of changing the solid thermal conductivity was also investigated and the results summarized in Fig. 4 for cases with conductivities of 10, 100 and 1000 W/m-K, respectively and an average porosity of 50%. The heat transfer performance increases markedly with the higher conductivity case. However, the effective heat transfer coefficient at the outlet is maximized at about the same outer wall porosity value (~70%) for all three conductivity cases.

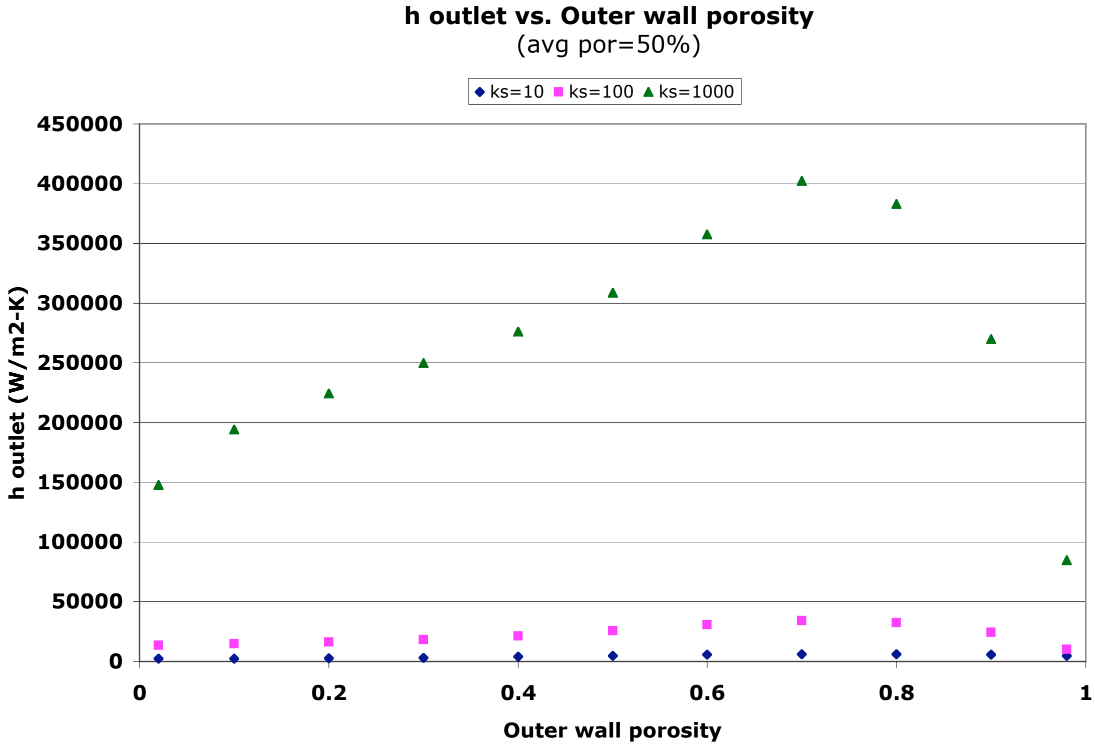

Figure 14 Effective heat transfer coefficient at outlet as a function of outer wall porosity for an average porosity of 50% and a linear radial porosity variation from the inner to the outer wall. Cases with different solid thermal conductivities are shown.

Summary and Conclusions

The following observations can be made from the results of the optimization studies.

- It seems sensible to accept the pressure drop penalty associated with porous flow only in areas where the high heat transfer performance of the porous medium is required. Typically, for a one-sided heat flux application (such as for a fusion divertor), only the upper region of the porous medium will see substantial heat flux and, if possible from the fabrication point of view, it is strongly recommended that the porous medium be installed only in the top half of the channel.

- It is preferable from a heat transfer performance standpoint to have the whole flow routed through a longer porous medium section than to have half the flow routed in parallel through half the section. For the example case considered, the maximum wall temperature is 1700 K in the former case and1900 K in the latter case. However, this is achieved at the cost of a higher pressure drop (by about a factor of 3), 1.8 MPa in the former case compared to 0.68 MPa in the latter case.

- For the example cases analyzed, for the same pressure drop and average heat flux to the coolant, the case with uniform heat flux would result in a lower wall temperature (better heat transfer performance) than the case with a peaked heat flux profile.

- For a case with an azimuthally varying heat flux profile on the porous tube (as for the case of a typical fusion divertor configuration), better heat transfer performance is achieved by routing the flow in the direction of decreasing heat flux.

- Unless and until experimental results indicate otherwise, it seems prudent that flow entry or exit point be placed in regions away from the high heat flux to avoid the possibility of a stagnation region leading to wall overheating.

- For a case with a varying radial porosity profile, it seems that an optimum point exists for a given average porosity. For example, for a linear porosity radial profile, the analysis results indicate that the heat transfer performance is maximized for an outer wall porosity of about 70% for a case with a 50% average porosity (the corresponding porosity at the inner wall would be about 30%). It seems that as the outer wall porosity is increased two conflicting mechanisms occur: on one hand, the lower porosity there results in a higher local velocity and better convective heat transfer to the flow; on the other hand, the increasing porosity at the outer wall reduces the conducting solid area and its fin-like effect.

The combination of these effects would result in an optimum linear porosity variation profile. This optimum tends to shift slightly to higher outer wall porosity as the average porosity is increased but does not seem to shift as the solid thermal conductivity is changed.

- It must be noted that the above observations are based on analytical results and would need to be confirmed experimentally.

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