ULTRA LOW PRESSURE-DROP HELIUM-COOLED POROUS-TUNGSTEN PFC

S. Sharafat[1], A. Mills[1], D. Youchison[2], R. Nygren[2], B. Williams[3], and N. Ghoniem[1]

[1]*University of California Los Angeles, Los Angeles, CA. 90095-1597* shahrams@ucla.edu
[2]*Sandia National Laboratory, Albuquerque, NM 87185-1129* dlyouch@sandia.gov
[3]*Ultramet Inc., Pacoima, CA. 91331* brian.williams@ultramet.com

A new class of helium-cooled high heat-flux plasma facing heat exchanger (HX) concept is presented. These unique "Foam-In-Tube" HX concepts are composed of a thin tungsten shell integrally bonded to an open-cell tungsten foam core. High heat flux tests show maximum heat loads of 22.4 MW/m^2 using 4 MPa helium at a flow rate of 27 g/s. Based on these impressive performance results, a unique and scalable heat exchanger channel with ultra-low pressure drop through the porous foam is presented. The primary advantage of the new concept is that pressure drop through the porous media and structure temperatures are nearly independent of HX tube length. The concept is modular in design and can be combined to meet divertor size requirements. From a manufacturing and reliability point of view, the advantage of the proposed concept is that it minimizes the need for joining to other functional materials.

I. INTRODUCTION

Plasma facing components, such as divertors and limiters in magnetically confined plasma devices will be subject to high thermal heat loads. The next step ITER Tokamak will expose divertors to quasi-steady state power densities of approximately 5 MW/m^2, to "slow" thermal transients of up to ~20 MW/m^2, and to short transients thermal pulses lasting (100 – 300) ms with energy densities of several tens of MJ/m^2 (~ 60 MJ/m^2). Candidate water-cooled ITER divertor concepts are under development based on, CfC or tungsten armor materials, which are joined to Cu-based heat sinks.[1] Research has also focused on developing more efficient cooling schemes, such as forced annular swirl flow.[2]

However, power requirements of future DEMO tokomaks will require helium cooled divertor concepts capable of handling up to ~15 MW/m^2 during normal operation for up to ~1000 thermal cycles.[3] Several helium-cooled divertor concepts are being developed without the use of Cu heat sinks. Candidate divertor concepts are impinging-jet cooled tungsten covered "finger-type" ODS-Ferritic tubes (HEMJ, HEMS)[4] and water cooled tungsten coated F82H screw-type concepts.[5]

A new class of helium-cooled HX concepts is under development, using porous materials, such as open-cell foam inside refractory tubes. High heat flux tests of such a Foam-In-Tube (FIT) HX tube were recently conducted by Youchison et al.[6] at Sandia National Laboratories and maximum heat loads of ~22.4 MW/m^2 were measured. In this work, we describe thermal-hydraulic analysis of these new Foam-In-Tube (FIT) HX concepts. Based on these initial impressive performance results, an advanced and ultra low-pressure drop short flow-path (SOFIT) concept was designed. Typical pressure drops through the foam section are estimated to be less than ~8 kPa at a helium flow rate of ~4 g/s and 150 kPa inlet pressure. At a heat load of 10 MW/m^2 the maximum surface temperature of a typical flat face SOFIT HX is estimated to be less than ~1100 ^{o}C . The primary advantage of the SOFIT HX concepts is that surface temperatures and the pressure drop through the porous media are nearly independent of HX tube length. The concept is modular in design and can be combined to meet divertor size requirements. The FIT HX concept has the added advantage that it minimizes the need for joining to other functional materials, which eases manufacturability and improves performance reliability.

I.A. High-Heat Flux Divertor Concepts

The main ITER divertor concepts under development consist of geometrically different CfC and tungsten armor configurations joined to water-cooled Cu-based heat sinks. CfC flat tiles were metal cast to CuCrZr heat sink blocks, while CfC-monoblock sheets were drilled and metal cast to copper tubes. Tungsten macrobrushes were e-beamed to CuCrZr heat sink blocks with a cooling tube running through them. Various tungsten monoblocks were bonded directly to OFHC-Cu cooling tubes (HIP'ed) and plasma-sprayed tungsten was directly applied onto CuCrZr heat sink blocks. These concepts were tested to heat loads of between 18 to 19 MW/m^2 and fatigue tested up to 1000 cycles. The CfC monoblocks were tested up to 25 MW/m^2 and the plasma-sprayed tungsten was thermal fatigue tested at 5.5 MW/m2 for 1000 cycles. These armored divertor concepts have shown satisfactory performance for normal ITER operating conditions and the enhanced cooling schemes show promise for further raising surface heat load capabilities. Water–cooled 32 × 400 mm divertor target mockups with W-rod armor and dual cooling channels were tested in Sandia's electron beam facility (EB1200), which sustained heat loads of 22 – 25 MW/m^2.[7]

The DEMO-relevant divertor concepts under development are the helium-jet impingement cooled[4] and the water cooled screw-type concepts.[5] The water-cooled screw-type divertor test section was made of a ~570 mm long 10 mm ID ferritic steel (F82H) tube with an internally machined M10 thread geometry. Tests with pressurized water (< 2 MPa), volumetric flow rates of < 10 ℓ/s, and axial velocities of < 7 m/s reached critical heat flux limits with surface heat loads of ~13 MW/m^2. The multiple helium-jet cooled HEMJ concept consists of 18 mm wide hexagonal tungsten tiles brazed to 15 mm diameter and 1.03 mm thick W-1% La_2O_3 thimbles. The tungsten armor was brazed to an ODS-Eurofer ferritic steel tube, creating a ~115 mm long structure. Maximum performance of the HEMJ concept was 12.5 MW/m^2 at a nominal helium flow rate of 6.8 g/s with a pressure drop of about 0.1 MPa. The HEMJ concepts withstood 100 thermal cycles without raising the W/W braze temperature above 1300 ^{o}C.

Helium-cooled divertor concepts using porous materials have been under investigation for over a decade.[8] Youchison et. al.[9] demonstrated a maximum heat flux capability of 29.5 MW/m^2 for helium-cooled copper tubes, which contained an annulus of sintered 1.02 mm copper (OFC) spheres. The copper-based divertor mockup consisted of two parallel tubes cooling a flat-faced copper block (Glidcop™ Al-15) with a volume of (38 × 63 × 32) mm^3 (L × W × H). Pressure drops across the module with a helium flow rate of 20 g/s and an inlet pressures of 4 MPa were about 20 kPa. Maximum heat transfer coefficients of ~26 000 W/m^2-K were determined.

ITER-relevant divertor concepts use water-cooled Cu alloys in combination with tungsten or CfC armor, while helium-cooled DEMO concepts are based on a complex structures bonding tungsten armor with ferritic steel structures. This work reports on a recent efforts to develop a new class of helium-cooled high heat flux plasma facing components, made by integrally bonding open cell refractory foam structures to refractory tubes. The advantage of the proposed concept is that it minimizes the need for joining to other functional materials. Sample tungsten Foam-In-Tube (FIT) heat exchanger (HX) tubes were fabricated and tested at the Sandia E-beam facility (EB1200). Impressive maximum heat flux levels of ~22 MW/m^2 were measured. We report here details of the thermal hydraulic and thermo-mechanical modeling efforts. Based on these extraordinary performance results, an advanced FIT concept is being developed. We present here the new concept, which maximizes heat transfer, reduces temperatures, minimizes stress, while maintaining an ultra-low pressure drop through the porous media.

II. FOAM-IN-TUBE (FIT) CONCEPT

The Foam-In-Tube (FIT) heat exchanger concept consists of an integrally bonded porous material to the inside of a tube made of the same refractory material. Ultramet Inc.[10] used chemical vapor deposition (CVD) to produce a number of FIT HX tubes. Figure 1 shows such a FIT component, made by CVD-bonding a 20% dense tungsten foam, with a pore density of 10 PPI (pore per inch) to the inside wall of a CVD-tungsten tube. The OD and ID of the tube were 16.2 mm and 12.7 mm, respectively. The foam section was about 38 mm long.

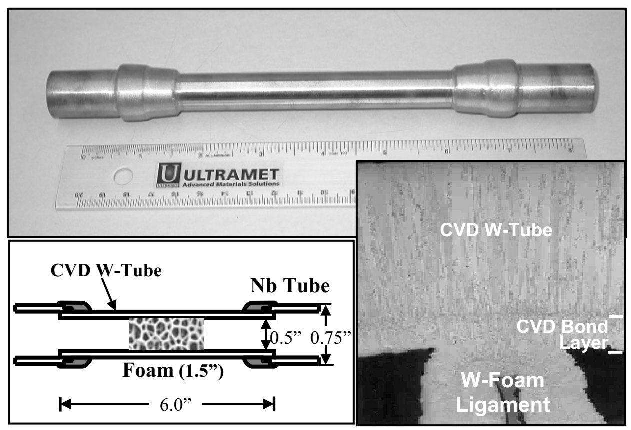

Fig. 1: The axial flow FIT heat exchanger tube concept; as fabricated by Ultramet Inc. and tested at the SNL (top); nominal dimensions (bottom left); optical micrograph showing a W-foam ligament CVD-bonded to the inner wall of a 0.05” thick W-tube (25X) (bottom right).[10]

Figure 2 shows a micrograph of the cross section of a CVD bond between a W-foam ligament and the W-tube. During the CVD-bonding process, a dense layer of CVD tungsten is deposited on the wall, which engulfs parts of the ligament producing a high quality bond. These bonds are critical for conducting heat from the wall along the ligaments deep into the foam structure. The conduction along ligaments results in spreading the heat over large surface areas inside the foam. The exposed surface area of a 12% dense 80–100 PPI open cell foam varies from ~1.2×10^4 to ~1.8×10^4 $m^2/m(solid)^3$.[11] The open-cell foam structure not only provides very large heat transfer areas, but also promotes coolant mixing and turbulence.

The axial-flow FIT HX tube depicted in Fig. 1 was tested at EB-1200 high heat flux Plasma Materials Test Facility (PMTF) at the Sandia National Laboratories (SNL).[9] The FIT HX tube experienced a maximum heat load of 22.4 MW/m^2 along the axial centerline of the top surface with an associated average heat flux of ~14 MW/m^2. The 4-MPa helium flowing at 27 g/s produced a pressure drop of 92 kPa and removed 7.2 kW at steady state. The HX tube eventually shattered after several cycles at these severe heat flux levels.[6]

II.A. Thermal Hydraulic Model

Detailed heat transfer and pressure drop modeling was performed based on experimentally-derived pressure drop data established at Ultramet (using air) and Sandia (using He), coupled with an extended Darcy's law (Eq. 1). The model is able to incorporate the effects of foam density, number of pores per inch (ppi), and ligament size for various coolant flow rates. Flow measurements were readily reproduced by the model and pressure drop and heat transfer coefficients as functions of foam parameters were established. The model predicts a pressure drop through 15% dense, 20 ppi foam over a length of 5 cm of about 39 kPa, which agrees well with the SNL-measured pressure drop of ~35 kPa for 150 kPa inlet helium pressure and a 3.6 g/s flow rate. Fig. 3 shows pressure drop estimates for various foam morphologies and good agreement between model and measured pressure drop data is demonstrated. Permeability as a function of foam density and ppi was determined (see Table I). As expected, the data shows that permeability can be increased by using lower ppi foam at lower density. Based of these benchmarked model results, the model provides a good level of confidence for estimating pressure drop as a function of W-foam parameters, such as density, ppi, ligament size, coolant flow rate and inlet pressure.

The modified Darcy equation[12] (Eq. 1) was used to determine permeability (K) and form coefficient (C) using helium and air flow test data for W-foam:

$$\frac{dP(x)}{dx} = \frac{\mu(x)}{K} V(x) + \rho(x) C\, V(x)^2 \qquad (1)$$

where P is the pressure, V is the velocity, and μ and ρ are dynamic viscosity and density of the fluid, respectively. The permeability and form coefficient are determined using:

$$K = \frac{D_f^2 \Phi}{a(1-\Phi)^2}; \quad C = \frac{b(1-\Phi)}{D_f^2(1-\Phi)^3} \qquad (2)$$

where D_f is the foam ligament diameter, Φ is the porosity, and (a) and (b) are constants determined from pressure drop data. Table 1 shows estimated permeability and form coefficient for different W-foam morphologies.

TABLE 1: Calculated permeability (K) and form coefficient (C) for various W-foam configurations

Pores Per Inch/Foam Density	Ligament Diameter D_f (m)	Permeability K (m^2)	Form Coefficient C (1/m)
10 PPI:			
10%	6.8 E-04	3.8 E-08	85.0
15%	8.3 E-04	2.1 E-08	122.7
20%	1.0 E-03	1.4 E-08	163.5
20 PPI:			
10%	4.6 E-04	1.7 E-08	125.0
15%	5.7 E-04	1.0 E-08	173.0
20%	6.8 E-04	6.7 E-09	242.2

Helium velocity and pressure profiles along the W-foam section were also determined. The helium gas velocity was found to increase from ~120 m/s at the entrance to about 170 m/s at the exit of the foam section.

Correspondingly, the helium gas density dropped from an inlet value of 0.24 kg/m^3 to about 0.17 kg/m^3 at the foam exit (helium inlet pressure is 150 kPa and flow rate is 3.8 g/s; tube ID is 0.5 inch). Convective heat transfer coefficients (h) were based on the Nusselt number

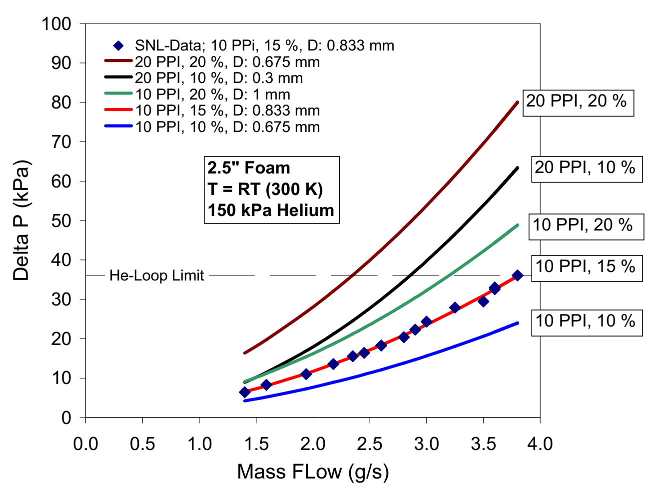

Figure 2: Helium pressure drop for different W-foam configurations (D: average ligament diameter, W-foam length 2.5" (6.35 cm); W-tube ID 0.5" (1.27 cm); 150 kPa helium pressure at RT; ◆ SNL data; lines: modeling).

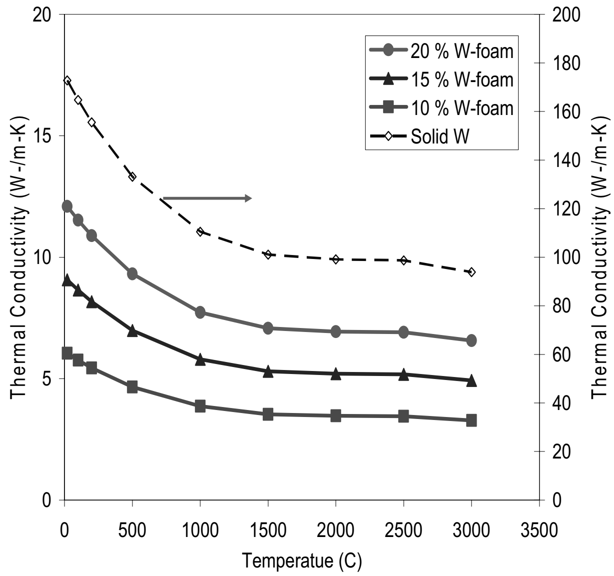

Fig. 3: Thermal conductivity of various W-foam morphologies as a function of temperature.

for developing boundary layer with uniform heating and were found to be highly dependent on the length of the W-foam section:[13]

$$\mathrm{Nu}(x)=\frac{h(x)x}{k_m(x)}=\frac{q''}{T_w(x)-T_\infty}\frac{x}{k_m(x)}=0.886\,\mathrm{Pe}_x^{1/2} \qquad (3)$$

where x is the axial position along the foam, q'' is the surface heat load, T_w and T_∞ are coolant temperatures near and away from the wall, respectively, Pe is the Peclet number, and (k_m) is the effective thermal conductivity of the porous media, which is estimated using:[11]

$$k_m(x)=\Phi k_{He}(x)+k_{foam};\ \frac{k_{foam}}{k_s}\approx 0.35\left(\frac{\rho^*}{\rho_s}\right) \qquad (4)$$

where k_{He} and k_s are the thermal conductivity of helium and bulk W, and ρ^* and ρ_s are densities of W-foam and bulk W, respectively. Thermal conductivity of W-foam with relative densities of 20, 15, and 10 % were calculated and are shown in Fig. 4. The contribution to the thermal conductivity of helium (0.149 W/m-K) is neglected. Using Fig. 44 an average thermal conductivity value of 4.5 W/m-K is taken for evaluating the effective heat transfer coefficient. Using the Peclet (Pe) number:

$$Pe(x)=\frac{\rho(x)V(x)c_p\,x}{k_m} \qquad (5)$$

the convective heat transfer coefficient is estimated using equation (21):

$$h(x)=\frac{0.886\,Pe(x)^{1/2}k_m}{x} \qquad (6)$$

The effective heat transfer coefficients near the heated wall of a FIT HX tube was estimated for a number of W-foam morphologies and at different flow rates and results are shown in Fig. 5. As expected, the heat transfer coefficient decreases with lower coolant flow rates and drops off sharply along the axial flow path of the foam. Within a distance of only 10 mm the h drops from a high value of > 20 000 W/m^2-K at the entrance to < 8 000 W/m^2-K.

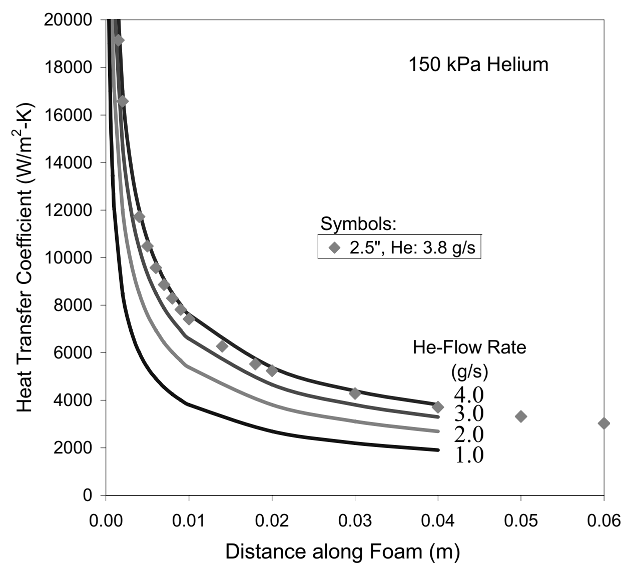

Fig. 4: Effective heat transfer coefficients along the W-foam (solid lines: 1.5 inch long, 20 PPI, 15% dense; P_{in} = 1.5 atm, T_{in} = 300 K, W-Tube ID/OD = 1.27/1.52 cm; ◆ symbols: 2.5 inch long 10 PPI, 15% dense foam).

II.B. Shortcomings of the FIT Concept

The FIT concept is based on axial flow of coolant through a porous media inside a tube, which has several drawbacks. The high pressure drop through the porous media makes it impractical to cool large areas. For example, the helium flowing through a 10 PPI – 15% dense W-foam at ~ 4 g/s and 150 kPa of a 12/15 mm ID/OD tube will be completely depressurized after ~ 18 cm. Assuming that the entire 18 cm length is used to cool an external heat load, the pressure drop per square meter would be of the order of 55 MP/m^2.

The other concern regarding the FIT concept is the possibility of flow instabilities for a parallel flow configuration. If FIT HX-tubes were placed in parallel, a flow perturbation caused by localized heating or porous media irregularities in one tube, would result in decrease flow in adjacent tubes.

The third main drawback of the FIT concept is the sharp drop in heat transfer coefficient over small length. For example, at ~4 g/s and 150 kPa the heat transfer coefficient drops from ~20,000 W/mK to less than ~ 4000 W/mK at about 4 cm from the entrance (see Fig. 3).. Such a sharp drop in heat transfer coefficient, not only minimizes the effective cooling length of the FIT concept, but also results in sharp temperature variations along the HX-tubes. Hence, the FIT concept does not lend itself to uniform temperature distributions along heated surfaces.

III.ULTRA LOW PRESSURE DROP FIT CONCEPT

A new class of porous high heat-flux components with a short flow path Foam-In-Tube (**SOFIT**) concept is under development. The general concept is shown in Figs. 5 and 6. It consists of two concentric W-tubes with W-foam sandwiched between them. An individual coolant channel is shown, in which the W-foam heat exchanger is selectively located to minimize the flow-path length through the porous media, maximize heat transfer and minimize stress while maintaining an ultra-low pressure drop. The concept is modular in design and can be combined to meet the size requirements of divertors without increasing the pressure drop through the porous media.

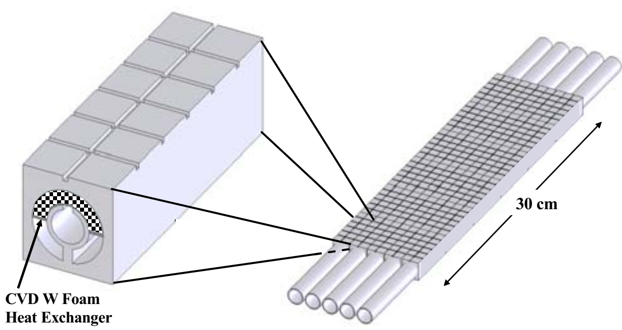

Fig. 5: CAD model of an individual W-foam/W-tube SOFIT coolant channel (left) and modular divertor panel composed of multiple channels (right).

The helium flow path of the SOFIT concept is shown schematically in Fig. 2. The inner tube is closed off at one end, thus forcing the pressurized coolant to flow radially through the upper slit and through the foam located in the annulus between the two tubes. Compared with a simple foam filled tube (FIT), which forces the coolant to flow axially through the porous medium (see Fig. 3), the proposed configuration minimizes the path length of helium flow through the porous media. Because pressure drop is proportional to the flow length through porous media, this new concept will incur very low pressure drops. Typical helium pressure drops through the foam section of an 11 mm ID tube are calculated to be ~ 7 kPa at flow rates of 4 g/s (Section II.B) . The main advantage of this concept is that the length of the HX tube no longer plays the dominant role in pressure losses and thus relatively long HX tubes with porous material structures can be developed.

III.A. Pressure Drop in Short Flow-Path FIT Concepts

The pressure drop in the FIT concept was estimated using parameters and helium properties at T = 600 K and *P = 1.5* atm as listed in Table 2.

TABLE 2: Parameters used for ΔP Calculations

Slot Length	25 mm	Inlet Temp.	600 K
Slot Width	4 mm	Viscosity	31.7×10^{-6} kg/ms
Permeability	6.8×10^{-9} m^2	Density600K	0.243 kg/m^3
Flow Rate	4 g/s (½ ea)	Pressure	150 kPa

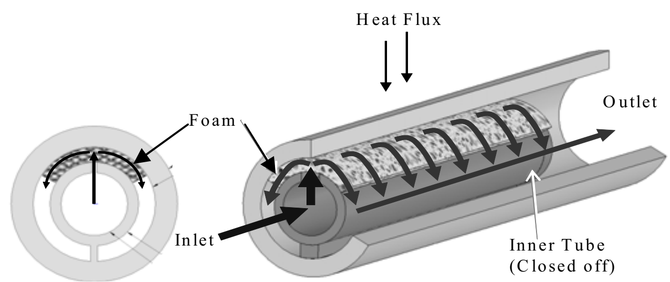

Fig. 6: CAD model of an individual W-foam/W-tube coolant channel showing the helium flow path (note the inner tube is closed off).

Using the modified Darcy's law[12] (Eq. 1) the pressure drop of helium with a 0.15 MPa inlet pressure is 8.3×10^5 Pa/m. The velocity (V) is calculated using $V = \dot{m}/\rho A$, where $\dot{m}$ is the flow rate, ρ is coolant density, and A is the flow area (slot width times slot length).

The length of the flow path through the W-foam for a tube with an ID of 14 mm is ~10 mm each side (see Fig. 2). This assumes a 90^{o} coverage on both sides of the slot. For such a non-optimized configuration the pressure drop through the foam is estimated to be ΔP = 6.3 kPa. The pressure drop through the slot has to be added, which can be estimated using the expression for the pressure drop through a nozzle:

$$\Delta P = \rho V^2 \quad (7)$$

The pressure drop through the a 4 mm wide slot is ΔP = 3.3 kPa, where the density of helium is taken at RT (ρ= 0.486 kg/m^3). Hence, the total pressure drop is estimated to be ~10 kPa with a flow rate of 4 g/s and a helium inlet pressure of 150 kPa. Considering that the SOFIT pressure drop is relatively independent to HX-tube length, these are extraordinary small pressure drops when compared with the axial flow FIT concept. If the inlet pressure of helium is increased to 1 MPa, then the total pressure drop (foam plus slot) will be even further reduced to an ultra low value of ΔP = 3.15 kPa.

III.B. Convective Heat Transfer Model

To estimate the heat transfer coefficient (h) a thermal boundary layer is assumed to commence at the stagnation line and the flat plate Nusselt number (Nu) is used:[13]

$$\mathrm{Nu} = \frac{\bar{h}_c}{k_m} = \frac{1.77}{L}\mathrm{Pe}_L^{1/2} \quad (8)$$

where L is the flow length through the porous media, Pe is the Peclet number, and (k_m) is the effective thermal conductivity given by Eq. (4).

The effective heat transfer coefficient for the short flow-path FIT HX tube can be estimated using Eq. (8):

$$\begin{aligned}\bar{h}_c &= 1.77\frac{k_m}{L}\mathrm{Pe}_L^{1/2} \\ &= 1.77\left(\frac{k_m}{L}\right)^{1/2}\left(\frac{\dot{m}}{A_c}c_p\right)^{1/2}\end{aligned} \quad (9a)$$

Using the parameters listed in Table 2, an effective foam thermal conductivity of k_m = 4.5 W/mK, the helium specific heat capacity c_p = 5200 J/kgK the effective heat transfer coefficient is estimated to be :

$$\bar{h}_c = 17\,000 \quad W/mK \quad (9b)$$

This value compares reasonably well with the detailed convective heat transfer model results at the entrance of the axially cooled FIT concept shown in Fig. 5. However, because the thermal boundary layer will have a finite thickness at the stagnation line, the above value of $\bar{h}_c$ will be an upper bound.

It is of interest to compare the heat transfer coefficient of the short flow path FIT concept with a foam-free unobstructed jet-impinging concept (similar the HEMJ). The heat transfer coefficient for the case of no foam between the two concentric tubes, can be estimated using the Reynolds number for a jet through a slot of width (t) is:

$$\mathrm{Re}_t = \frac{\dot{m}}{A_c}\frac{t}{\mu} = 5050 \quad (10)$$

where A_c = 3.17x10^{-5} m^2, $\dot{m}$ = 4x10^{-3} kg/s; μ_{He} = 31.7 kg/ms, ρ_{He} = 0.486 kg/m^3, k_{He} = 0.229 W/mK (He is assumed to be at 600 K and 1.5 atm) and where we have chosen a 1.27 mm wide and 25 mm long nozzle (these dimension match available experimental data).[14] From Ref. 14, the Nusselt number is $Nu_t \approx 45$ and the heat transfer coefficient is then estimated to be:

$$h_c = \frac{k_{He}}{t}Nu = 8110 \quad (11)$$

which is about half of the heat transfer coefficient of the foam–filled FIT concept. To estimate the pressure drop through the nozzle we calculate the velocity through the nozzle using $V = \dot{m}/\rho A$=259 m/s:

$$\Delta P = \rho V^2 = 33 \;\; kPa \quad (12)$$

which is about 4 times higher than the pressure drop of short flow-path FIT concept, because the delivery slot width is 1.27 mm wide instead of 4 mm. If the inlet pressure of helium is increased to 10 MPa, then the velocity would drop to 38.8 m/s and the pressure drop through the slot would decrease to about 5 kPa, which is only about twice the FIT pressure drop ΔP of 3.3 kPa.

The uninhibited (no-foam) jet-impinging flow has higher pressure drops along with lower heat transfer capabilities compared with the SOFIT concept. Hence, the SOFIT concept with a higher effective h and lower ΔP clearly outperforms the simple uninhibited jet FIT concept.

IV. DISCUSSION

The main drawbacks of the axial flow through a porous media in a tube (FIT concept) are (1) a relatively short tube-length with high heat transfer coefficient, (2) a high ΔP, (3) a non-uniform structural temperature distributions, and (4) the potential for flow instabilities. These shortcomings can be addressed by locating the porous media directly above a slit of an internal tube, which forces a jet of coolant to flow radially through a short section of the porous media (SOFIT concept).

The advantage of the SOFIT flow configuration is that ΔP is relatively independent of tube length, and therefore it becomes feasible to cool large areas. For example, the pressure drop through a 10 PPI – 15% dense W-foam at ~ 4 g/s and 150 kPa of a 12/15 mm ID/OD tube is about 10 kPa, for a 4 mm by 630 mm long slit. The pressure drop through the porous media is independent of axial length of the porous media, provided adequate flow rates can be supplied to the pressurized internal tube (see Fig. 6). The pressure drop through the porous media of the SOFIT concept would be < 1 MPa/m^2 compared with about 55 MPA/m^2 for the FIT concept, for a 12/15 mm ID/OD tube. Of course, the total pressure drop would include inlet and outlet pressure drops associated with the SOFIT HX tubes.

Hence, the SOFIT concept offers several advantages over the FIT concept. These include (1) a short flow path and thus low pressure drops through the porous media, (2) utilizing the high heat transfer coefficients associated with the entrance flow through porous media, (3) a low pressure drop, which is relatively independent from HX-tube length, and (4) a fairly uniform temperature distribution along cooled surfaces.

V. CONCLUSIONS

Recent high heat flux tests of helium-cooled porous tungsten foam inside a CVD-tungsten tube (FIT concept) have demonstrated maximum heat loads of ~22.4 MW/m^2 before failure.[6] However, three main drawbacks are associated with the FIT concept: (1) large pressure drops, (2) potential flow instabilities for parallel flow configurations of FIT HX tubes, and (3) highly non-uniform temperature distribution along HX tubes due to sharp drops in heat transfer coefficient along the porous media.

However, based on the impressive heat load performance results of the FIT concept, a modified concept was developed, called SOFIT. The SOFIT concept has a radial and very short flow path through the porous media and as such provides an ultra low-pressure drop alternative to the FIT concept. Typical pressure drops through the foam section of SOFIT HX-tubes are estimated to be less than ~10 kPa with a helium flow rate of ~ 4 g/s and a 150 kPa inlet pressure. At a heat load of 10 MW/m^2 the maximum surface temperature of a SOFIT HX tube concept is estimated to be less than ~1100 ^{o}C. The temperatures along the surface of a flat SOFIT HX-tube concept are relatively uniform compare with a FIT concept. Furthermore, from a manufacturing and reliability point of view, the advantage of the proposed all tungsten SOFIT concept is that it minimizes the need for joining to other functional materials.

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