DESIGN AND FABRICATION OF A RECTANGULAR HE-COOLED REFRACTORY FOAM HX-CHANNEL FOR DIVERTOR APPLICATIONS

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A rectangular single channel low pressure drop helium-cooled refractory metal heat exchanger (HX) tube for divertor applications was designed and manufactured for testing in the SNL E-beam facility. A unique fabrication feature of the rectangular HX channel design is that all welds, brazes, and joints are located at or near the bottom of the rectangular channel, i.e., far from any heated surface. The HX tube concept uses a thin (~ 2mm) layer of open-cell refractory foam bonded underneath the heated surface to enhance heat transfer to the helium coolant.

The helium coolant flows through a 2-mm-wide slot and then through the thin foam layer (~2 mm × 12 mm × 127 mm; H/W/L) from the inlet to the outlet plenum. This design minimizes the path of helium flow through foam to about 11 mm and thus the pressure drop through the porous media is more or less constant along the length of the channel. The concept is scalable for cooling large flat surfaces, such as a flat-plate divertor, without substantially increasing the coolant pressure losses.

We present CFD analyses used to optimize the design for minimum pressure drop through the porous media and for highest uniformity of surface temperatures. A design-for-manufacturing concept for a single HX-channel was developed with the goal to minimize welds or joints near heated surfaces. Based on the advanced HX-channel design a number of HX-channels were manufactured using Mo as a surrogate material instead of tungsten.

I. INTRODUCTION

Divertor target plates in fusion power plants must perform under stringent requirements, such as high heat fluxes (> 10 MW/m^2), deliver heat at elevated temperatures for high efficiency, and use coolants that are compatible with the blanket power conversion systems. To satisfy these requirements, a helium-cooled refractory metal heat exchanger (HX) concept would be desirable. Furthermore, the coolant pressure drop should be kept to a minimum to help with the overall plant power balance and coolant exit temperatures should be as high as possible for efficient power conversion.

Malang and Hermsmeyer proposed a gas-cooled divertor in 2001 with helium flowing helium parallel to the heated surface through a thin slot (gap ~ 0.1 mm) at high velocity [1]. The design improved divertor plate cooling capacities from 5 MW/m^2 to about 10 MW/m^2 heat flux. A different class of divertor concepts, based on porous media for enhancing the heat transfer coefficient (HTC) was considered by Malang et al. in 2001, but not pursued due to uncertainties in predicting the heat transfer coefficient and due to geometric considerations [2]. Recently, we revisited the use of porous media for HTC enhancement for divertor applications [3]. Using axial helium flow through a tungsten tube containing tungsten foam, a maximum heat flux of 22.4 MW/m^2 was demonstrated [4].

Due to the inhibitive large pressure losses in flow through axial configurations of porous media, we re-designed the concept such that the coolant flows azimuthally through the porous media underneath the heated wall [5] as shown in Fig.1.

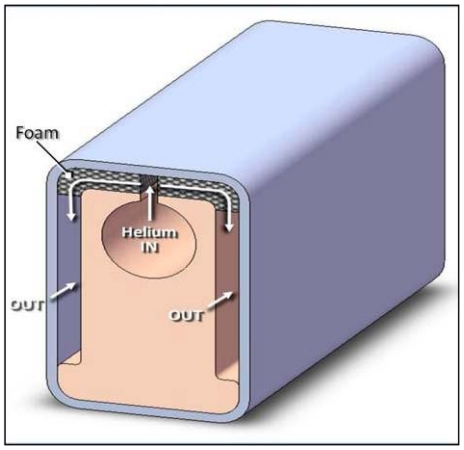

Fig. 1. CAD model of the HX-tube showing He-coolant flowing azimuthally through the porous media.

The azimuthal He-flow HX concept uses of refractory foam between two concentric channels. The advantages of this design are: (1) all-refractory metal HX

tube (no joining of dissimilar materials), (2) enhanced HTC, and (3) short azimuthal flow path through the porous medium. The pressure losses associated with flow through the porous media are limited to that of the azimuthal flow length (~ tube radius), regardless of the tube length. The HX tube pressure losses through the porous media are independent of tube length and thus, a low pressure drop gas-cooled HX concept using porous media becomes feasible.

In this work we report on the CFD analysis of the azimuthally cooled porous HX concept, which lead to a for-manufacturing-design concept that avoids welds and joints near heated surfaces. The single HX concept is undergoing testing at the SNL E-beam facility, following the procedure outlined in Ref. 4 and results will be published.

II. ADVANCED POROUS HX CONCEPT

A brief description of the advanced concept is given here; for more details the reader is referred to earlier publications [3-5]. The azimuthal flow HX concept is based on integrally CVD-bonded tungsten foam between two concentric tungsten tubes. The inner tube has a centrally located slot, which runs the length of the tube. The helium coolant initially flows radially through the slot then splits into two streams to flow azimuthally through the foam. Figure 1 shows a schematic CAD drawing of the advanced porous HX concept. The foam is also split into two segments, with an opening between them above the slot to allow the helium to impinge on the heated surface unimpeded before it flows azimuthally through the porous media. Because the coolant splits into two streams, pressure losses are equal to flow through a porous media with characteristic lengths of about ½ the tube diameter.

This advanced porous HX-concept lends itself readily to flat plate divertor designs with design modifications, such as tapered inner helium supply tubes to provide uniform flow rate along the top slot, thereby allowing for an overall thinner divertor plate.

III. POROUS HX-CONCEPT SOLID MODELING

As pointed out earlier uncertainties in predicting the heat transfer coefficient along a wall in contact with porous media constitutes a major drawback to design and to optimization of such concepts. In particular, applying correlations developed for packed beds and granular materials to low density open-cell foam structures is highly speculative. The reasons are that granular materials or packed beds have porosities in the range 0.4 – 0.6, while foam porosities range between 0.8 and 0.1. However, more limiting is the assumption of local thermal equilibrium between the two phases is invoked. The assumption that the solid and fluid phases have the same temperature field results in using a single homogeneous equation to describe energy transport between the two phases. Thus, an effective stagnant thermal conductivity, k_e, of a porous medium is used to account for the solid and fluid phase conductivities. The assumption of local thermal equilibrium does not hold if the stagnant conductivities of the solid and fluid are significantly different, which is the case when using helium as a fluid and tungsten as the low density foam structure.

Therefore, the use of computational fluid dynamics software, which directly solves the Navier–Stokes equations and the local energy equations in the fluid and solid phase, is preferred in simulating thermal-hydraulic performance of the advanced porous HX design.

However, prediction of flow and pressure distribution using a three-dimensional field solution of the Navier-Stokes equation does not come without its own challenges. First, one has to develop a realistic 3-D solid model of the porous structure, in this case the tungsten foam. The second challenge in using the direct CFD approach is that all geometric features and associated interstitial spaces have to be meshed into a large number of cells; this requires extensive computational resources even for geometrically small representative samples of porous structures.

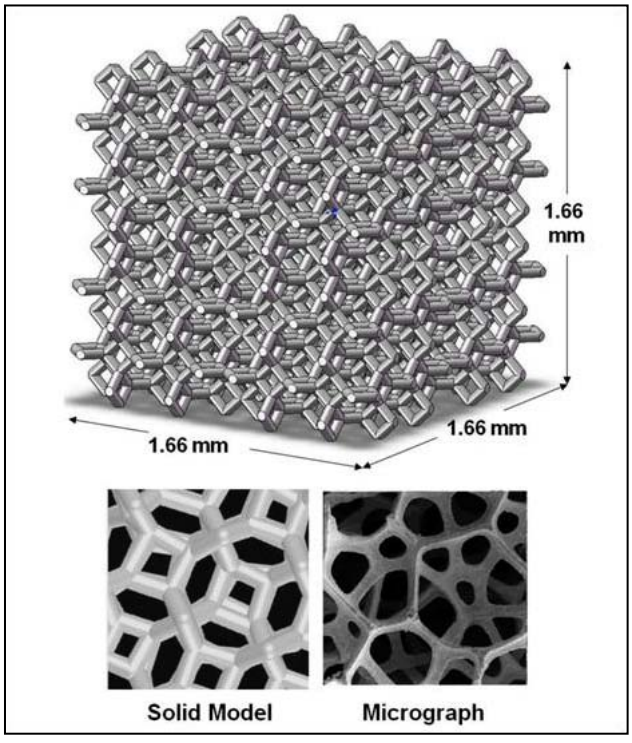

Fig.2. Solid model of a tetracaidecahedra 65 ppi foam with 19.7% relative density foam (1.66×1.66×1.66 mm^3) (top); (lower left) solid model (10 ppi, 19.7 % dense), (lower right) open-cell W-foam micrograph (50 X).

In this section we outline our approach to meeting these two challenges, by first describing our approach to modeling the porous structure.

III.A. Foam Model

A program was developed to calculate the coordinates of joined tetrakaidecahedra cells, based on required pore per inch (*ppi*) densities. The nodes are transferred to the 3-dimensional (3-D) solid modeling code and a script was programmed, which automatically connects the vertices with cylindrical geometries (ligaments). The radius of the ligaments determines the relative foam density, while the coordinates establish desired pore densities. This process creates a solid model of the foam structure having a true tetracaidecahedra cell structure with "tunable" pore densities and relative foam densities.

However, real foam structures are not symmetric and do not have perfect periodicity. In order to make the solid model more realistic, the cell vertices were displaced from their original coordinates randomly to within ±5% of original location. In Figure 2 we compare a 10 ppi 20% dense foam solid model with a typical tungsten foam micrograph, which shows a relatively good match-up between the two.

III.B. HX Section Model

The goal of the CFD modeling is to analyze a section of the advanced porous HX concept shown in Fig. 1. Performance of this concept is relatively independent of tube length; therefore we can limit the analysis to a small but representative section of the HX tube. Figure 3 shows the 10-mm wide porous HX tube segment model developed for CFD analysis. The foam has a relative density a 19.7% and a pore density of 10 ppi. The foam segment is about 11-mm long in the direction of helium flow and about 10-mm wide. The foam thickness is 2.5 mm, the entrance slot is ~1 mm wide, and the exit slot is about 2 mm wide. Other dimensions of the model are shown on the cross sectional view imbedded in Fig. 3.

The CFD model also has three thin (0.1 mm) side walls, which serve to confine the coolant (note the right-hand sidewall has been removed for illustration purposes).

IV. CFD ANALYSIS

The CFD analysis results are not meant to be deterministic, but serve only to better understand the impact of foam on cooling a heated wall. A commercial CFD code (SC-Tetra®) was used to determine velocity, temperature, pressure drop profiles of the coolant and structure. The SC-Tetra® code has several turbulence models with various low Reynolds number models for accurate simulation of near wall regions along with models for fluid solid conjugate heat transfer analysis. Turbulent heat transfer enhancement at the solid/fluid interface and phase change heat transfer are but some of the additional capabilities. The code uses varying fluid properties, which change with time, temperature, and flow conditions.

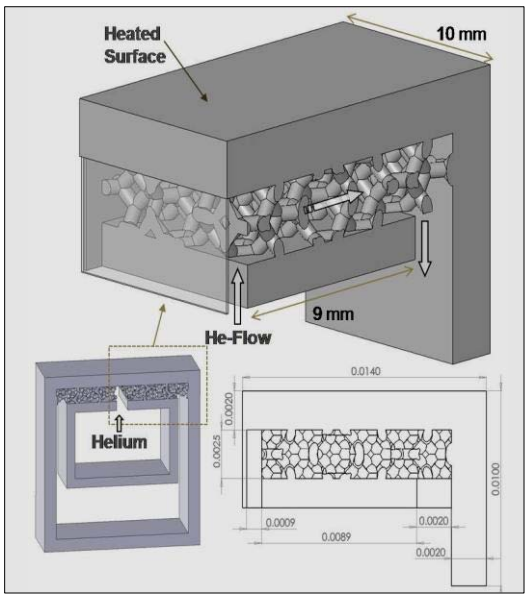

Fig. 3. Solid model of the HX section used for CFD (upper); cross section (lower right) dimensions in mm.

Flow parameters, coolant, and structure properties used in the CFD analysis are listed in Table 1. A no-slip boundary condition and compressible flow is used with the RNG k-E model. The model shown in Fig. 3 was meshed and a total of 15 million elements were generated. The computation time was of the order of 5 hours on a dual core 64-bit machine.

TABLE 1: Parameters used in the CFD Analysis.

Parameter/Property	Value/Correlation
Helium	
Inlet Pressure (MPa)	4
Inlet Temperature (K)	573
Flow Rate (kg/s)	4.84×10^{-3}
Inlet Flow Velocity (m/s)	150
Viscosity (Pa-s); *T(K)*	$3.2\times10^{-7}\cdot T^{0.72}$
Density (kg/m^3); *P(atm), T(K)*	$48.75\cdot P/T$
Thermal Cond. (W/m-K); *T(K)*	$2.5\times10^{-3}\cdot T^{0.72}$
Gas Constant (J/kg-K)	2078.5
Specific Heat (J/kg-K)	5156.3
Tungsten	
Specific Heat (J/kg-K)	f(T)[(a)]
Thermal Cond. (W/m-K)	f(T)[(b)]

[(a)] $C_p=128.308+3.2797\times10^{-2}T-3.4097\times10^{-6}T^2$

[(b)] $\lambda=174.9274 - 0.1067T + 5.0067\times10^{-5}T^2 - 7.8349\times10^{-9}T^3$

IV.A. Pressure Losses

At 4 MPa He inlet pressure, a flow rate of 4.84×10^{-3} kg/s, an inlet velocity of 150 m/s, and inlet temperature of 573 K (300 °C) the average pressure drop across the 11-mm long foam section was estimated to be about 182 kPa. At inlet velocities of 32 m/s and 100 m/s, the pressure drops were 52 kPa and 112 kPa, respectively. In comparison, with a helium inlet velocity of 32 m/s, 28×10^{-3} kg/s flow rate at room temperature, SNL measured a pressure drop of 98 kPa for a 0.0381 m long foam section, which had a relative density of 20% and 10 ppi pore density. It is of interest to note that the pressure through the thickness of the foam (top to bottom) shows very small variations, indicating a fairly uniform flow rate across the foam.

IV.B. Coolant Temperature and Velocity

The mean temperature rise in the helium was calculated to be about 100 K, while in the corner region above the outlet the helium is ~200 K higher, because of slower flow velocities (about 70 m/s). As expected at the entrance to the foam, vertical intermixing of the coolant is minimal, but increases towards the exit of the foam. The most pronounce intermixing occurs near the exit of the foam. Fig. 4 shows the velocity contours of the helium coolant. The inlet velocity is about 150 m/s but locally along the foam section local velocities reach about 250 m/s. These high coolant velocities are very effective in increasing the HTC especially when they occur near the surface.

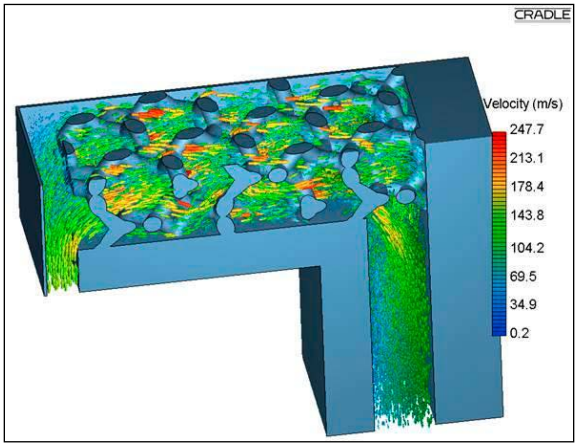

Fig. 4: Section view of the helium velocity field (q''=10 MW/m^2; V_{in} = 150 m/s; T_{in}=300 °C) [note: print copy is b&w, but color online].

IV.C. Structure Temperature

The maximum surface temperature on the 2-mm thick W-top is 1198 K (925 °C) with an applied surface heat load of 10 MW/m^2. Of particular interest was the uniform surface temperature along the top surface and across the wall, thus indicating the effectiveness of cooling the corner of the square HX-tube. This can be explained due to (1) the effective conduction of heat along the 2-mm thick sidewall, (2) the conduction along foam ligaments located at the corner, and (3) the effective mixing of helium near the corner, where the coolant has to make a 90° bend and simultaneously traverse foam ligaments. The ligaments are conducting heat from the surface to the interior of the channel, thus providing a larger surface area for cooling. It is also important to note that the temperature of the ligaments near the top was about 200 K higher than the coolant temperature in the same region. This is one of the reasons why it is critical to use CFD analysis for modeling high porosity foams, instead of using analytical approaches developed for low porosity packed beds. The solid and fluid temperatures are clearly not in equilibrium and thus using a single homogenous equation for temperature fields may not accurate.

V. DESIGN AND FABRICATION OF A SINGLE CHANNEL HX

Figure 5 shows the CAD solid model of the single HX-channel. This design uses a square outer tube, which eliminates the need for any welds or joints near the heated surface. All other internal components are slid into the outer channel and then e-beam welded near the inlet and outlet of the helium coolant supply pipes.

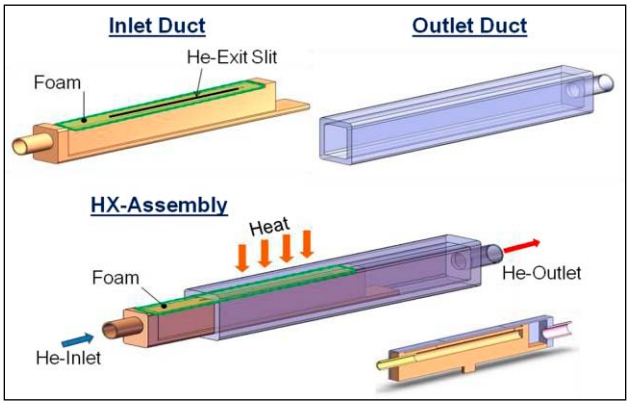

Fig. 5: Square channel HX-design based on a single outer tube to house internal components – "Slide-in Design".

The "Slide-in" design concept was realized by Ultramet Inc. [6]. Molybdenum was used as a surrogate material instead of tungsten development of this concept. Fig. 6 shows the finished components prior to assembly and a partially assembled HX-channel. Prior to assembly the 2-mm thick Mo-foam is placed on the inner tube above the 2-mm wide He-exit slit. Fig. 7 shows assembled HX-channels along with details of the e-beam welded inlet and outlet caps.

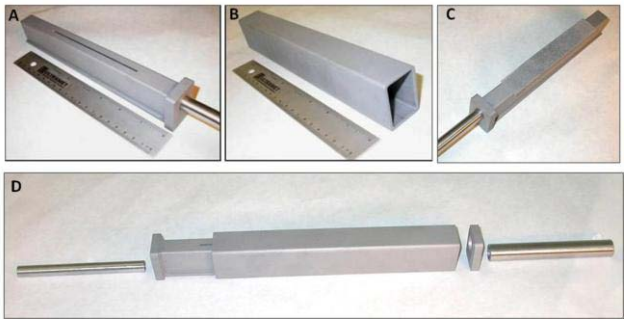

Fig. 6: HX-channel components made of Mo prior to assembly: (A) internal He-supply tube, (B) external tube, (C) internal He-supply tube with Mo-foam covering the He-exit slot, (D) partially assembled HX-channel.

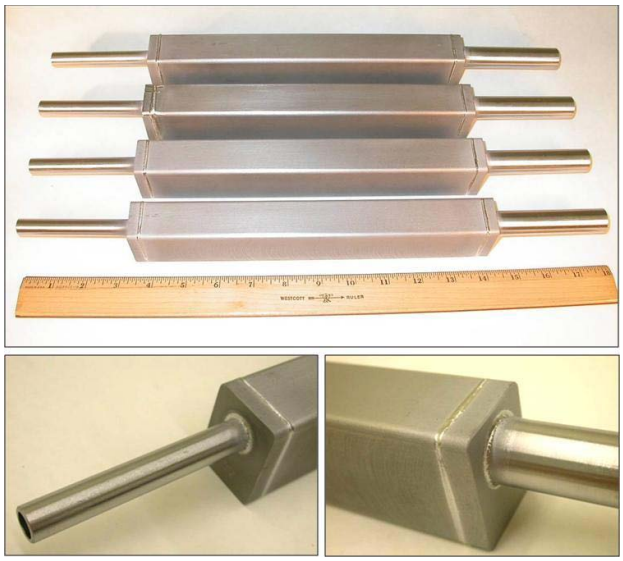

Fig. 7: Assembled HX-channels (top) and close-up view of e-beam welds of inlet and outlet caps (bottom).

VI. SUMMARY

An advanced porous helium cooled heat exchanger concept was designed and fabricated of Mo. The concept uses a thin layer of refractory foam attached between two concentric tubes. The foam serves to improve heat transfer to helium coolant flowing azimuthally through the foam underneath the heated surface (see Fig. 1). The foam is nominally between 1 and 2 mm thick and the coolant path length through the foam is about the length of the radius of the tube (~11 mm for the current design). The short flow path through the porous media, the foam keeps pressure losses at a minimum while the enhancing heat transfer coefficient beneath the heated surface.

At 4 MPa He inlet pressure, a flow rate of 4.84×10^{-3} kg/s, an inlet velocity of 150 m/s, and inlet temperature of 573 K (300 ^{o}C) the average pressure drop across the 11-mm long foam section was estimated to be about 182 kPa. Because the pressure losses through the foam (azimuthal direction) do not scale with HX-tube length, the pressure drop is relatively independent of HX-tube length.

CFD analysis also showed that with a surface heat load of 10 MW/m^2, helium inlet temperature of 300 ^{o}C, inlet velocity of 150 m/s, at 4 MPa pressure, the upper surface temperature of the HX would reach about 950 ^{o}C, which was about 360 ^{o}C lower than the same HX without foam. The heat transfer coefficient along the heated wall was extracted from CFD analysis and ranged between 12,500 W/m^2-K and 25,000 W/m^2-K for the modeled flow conditions.

To minimize welds or joints near heated surfaces, a concept based on a single external square tube with one internal helium supply tube was designed. This design provides a "slide-in" mechanism to push the internal supply pipe into the external tube and apply e-beam welds only to the both end-caps, which are far from the heated surface. A 2-mm thick foam covering the upper slit of the internal helium supply pipe. Helium enters the channel from one end, is guided upwards through the 2-mm wide slit, flows azimuthally through the foam between the heated surface and the internal supply pipe, before exiting on the opposite end of the HX-channel.

Several HX-channels were made of Mo, which was used as a surrogate material for tungsten. The HX tubes were sealed and tested for leaks. The HX-channels are undergoing flow and heat tests at the Sandia National Laboratory's EB1200 High Heat (E-beam) Facility.

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