An experimental investigation of the thermal performance of the He-Cooled Multi-Jet (HEMJ) modular divertor design developed by the Karlsruhe Research Center (FZK) was previously performed at Georgia Tech using air at Reynolds numbers (Re) spanning those at which the actual He-cooled divertor is to be operated. The electrically heated test section was constructed from a brass alloy with nearly the same thermal conductivity as the tungsten alloy from which the prototypical HEMJ is to be constructed. More recently, another experimental investigation was performed by the Georgia Tech group for a similar finger-type divertor module using both air and He as coolants. The results of these experiments suggest that, in addition to matching Re, dynamic similarity between the air and He experiments requires that a correction be made to account for the differences in the relative contributions of convection and conduction (through the divertor walls) to the overall heat removal rate by the module. This correction factor depends on the thermal conductivity ratio of the solid to the coolant.

This investigation is aimed at quantifying the correction factor necessary to achieve dynamic similarity for the HEMJ to account for changes in the coolant and solid thermal conductivities. Experiments have been conducted using two (one each brass and steel) HEMJ test modules cooled with air, Ar, or He. The test modules were directly heated using an oxy-acetylene torch achieving heat fluxes up to 3 MW/m². The results of these experiments, together with data obtained using the aforementioned air-cooled experiments at lower heat fluxes (<1 MW/m²) were used to develop a generalized correlation for the Nusselt number which accounts for changes in Re, as well as the solid to coolant thermal conductivity ratio. The correlation can be used to predict the performance of the actual He-cooled divertor at prototypical operating conditions.

I. INTRODUCTION

Several gas-cooled divertor designs have been investigated as part of the ARIES research program including the He-cooled multi-jet¹ (HEMJ), the He-cooled flat plate² (HCFP), and the He-cooled modular design with integrated fin array³ (HEMP). The purpose of these thermal-hydraulic and thermo-mechanical studies is to ensure that these designs can withstand the anticipated heat fluxes (~10 MW/m²) when operating in a magnetic fusion energy power plant without exceeding temperature and stress limits dictated by material properties.

The research group at Georgia Tech has utilized dynamic similarity to evaluate the thermal-hydraulic performance of the proposed divertor designs by performing a large number of low-temperature, low-pressure experiments. In general, test sections duplicating the geometry of the prototypical design are constructed out of brass (instead of tungsten and tungsten alloys) and cooled with air (instead of He). The experiments are conducted at non-dimensional mass flow rates (Reynolds number, Re) that span the expected operating Re value. The test modules are heated either electrically or, more recently, by an oxy-acetylene torch. Heat transfer coefficients (HTCs) are determined from calculated incident heat flux values and measured temperature readings from thermocouples embedded in the test section, and used to determine non-dimensional Nusselt numbers (Nu). The measured pressure drop across the test section is used to calculate a non-dimensional pressure loss coefficient (Kp). By correlating Nu and Kp to Re, parametric design curves are constructed that describe the thermal-hydraulic behavior of the design at prototypical operating conditions. For a given coolant inlet temperature and pressure, and a specified maximum allowable temperature for the pressure boundary, the maximum allowable incident heat flux and the corresponding pumping power fraction are determined based solely upon thermal-hydraulics (vs. thermal stress) considerations.

II. THE HEMJ AND HEMP DIVERTORS

Most recently, two modular finger-type divertors, the HEMJ and HEMP, designed by Karlsruhe Research
Center (FZK), have been studied at Georgia Tech. The HEMJ divertor cools the inner surface of the divertor tiles using an array of twenty-five round jets (Fig. 1). Hot He enters the inner tube at ~600°C and 10 MPa, passes through the array of round jets thus cooling the inner surface of the endcap (i.e. the pressure boundary), thereby removing the incident heat flux on the divertor tiles, and exits at ~700 °C. The HEMJ is designed to accommodate 10 MW/m² incident heat flux while keeping the tungsten alloy endcap peak temperature below 1300°C.

Earlier experimental and numerical studies on a HEMP-like divertor predicted the performance of the HEMP under prototypical conditions. However, as described by Mills et al., predicted prototypical divertor performance based on non-dimensional $N_u$ and $K_L$ values obtained from an air-cooled brass test module may not be accurate because these predictions do not account for the differences in the relative contributions of convection and conduction through the divertor walls.

The previous experiments performed on an electrically heated HEMJ module cooled with air were therefore re-examined by accounting for the difference in the relative contributions of convection and conduction, characterized by the ratio of the thermal conductivities of the solid and the coolant, to the overall heat removal capability of the divertor module. To this end, additional experiments have been performed using two HEMJ test sections made of brass and steel, respectively, heated with an oxy-acetylene torch at incident heat fluxes up to 3 MW/m² and cooled by air, Ar, or He. By varying both the coolant and divertor thermal conductivities, these experiments at different thermal conductivity ratios are designed to verify that these experiments can, by accounting for this effect, give accurate predictions of the thermal performance of the HEMJ design at prototypical conditions.

III. EXPERIMENT

The torch-heated test sections are composed of two primary components: a jet cartridge, which creates round jets, concentric with and inside a thimble that simulates the pressure boundary. Two thimbles were machined, one of C36000 brass alloy and one of AISI 1010 carbon steel. Two jet cartridges were also machined, both out of C36000 brass. The jet cartridges have 24 0.6 mm diameter jets arranged in groups of six placed on four concentric circles with radii of 2.22 mm, 3.52 mm, 4.77 mm, and 6.49 mm projected to a plane at the top of the jet cartridge centered about one 1 mm diameter jet. The
thimble and the jet cartridge are separated by a 0.9 mm gap. Schematic drawings of the jet cartridge and thimble are shown in Fig. 3. The volumetric flow rate of coolant into the test section is measured using a Brooks variable area flow meter (Model 1110) and used along with the density of the coolant calculated using temperature and pressure measurements from an OMEGA type E TC and OMEGA PX302-2KGV pressure transducer at the exit of the flow meter to determine the mass flow rate, $\dot{m}$. The temperature of the coolant was measured at the inlet, $T_i$, and outlet, $T_o$, of the test section by two OMEGA type E TCs and used along with the density of the coolant calculated using temperature and pressure measurements from an OMEGA type E TC and OMEGA PX302-2KGV pressure transducer at the exit of the flow meter to determine the mass flow rate, $\dot{m}$:

$$\frac{\dot{q}^c}{A_h} = \frac{\dot{m}c_p(T_o - T_i)}{A_h}$$

(1)

where $c_p$ is the constant pressure specific heat capacity of the coolant evaluated at the average of $T_i$ and $T_o$, and $A_h$ is the area of the heated surface, 227 mm².

The temperature distribution in the thimbles was measured using six 0.5 mm embedded type E OMEGA TCs (Fig. 3). The readings from the four TCs nearest the cooled surface spaced 90° apart and radially located 6.4, 4.3, 2.1, and 0 mm at axial locations 8.26, 6.88, 6.36, and 6.25 mm relative to the top of the thimble are used to determine the HTC. These TC readings are extrapolated through 0.5 mm of brass or steel to the cooled surface using $q^c$ assuming 1-D conduction. The extrapolated temperatures are then area averaged over the projected cooled surface area, $A_c = 132$ mm², resulting in an average cooled surface temperature, $T_c$.

The pressure drop, across the divertor $\Delta p$ was measured using an OMEGA PX26 differential pressure transducer. The inlet pressure was measured with an OMEGA PX302-300AV pressure transducer.

A total of seven experimental configurations were studied: the torch-heated brass thimble cooled with air, Ar, or He; the torch-heated steel thimble cooled with air, Ar, or He; and the electrically heated brass thimble cooled with air, from previous experiments5,6. Data from a total of 60 experiments are presented here.

IV. EXPERIMENTAL RESULTS

The area-averaged HTC, $\overline{h}$, was calculated from $\dot{q}^c$, $T_c$, $T_i$, and the area ratio $A_h/A_c$:

$$\overline{h} = \frac{\dot{q}^c}{T_c - T_i} \frac{A_h}{A_c}$$

(2)

The above equation assumes that all of the power incident on the thimble is removed by convection at the cooled surface, i.e. it ignores conduction through the annular walls of the thimble. The area averaged $\overline{Nu}$, $\overline{Nu}$, and the $Re$ of the central jet, $Re_j$, were calculated as follows:

$$\overline{Nu} = \frac{\overline{k}D_i}{k} \quad \text{and} \quad Re_j = \frac{mD_i}{A_j\mu_i}$$

(3a & 3b)

where $D_i$ is the diameter of the central jet (1 mm), $\overline{k}$ is the gas thermal conductivity based on the average of $T_i$ and $T_o$, $A_j$ is the total area of all 25 jets (7.57 mm²), and $\mu_i$ is the dynamic viscosity of the coolant at $T_i$.

Fig. 4 presents $\overline{Nu}$ as a function of $Re_j$, for all seven experimental conditions. Experimental uncertainty for both $\overline{Nu}$ and $Re_j$ was calculated using standard techniques14 and ranges from 2%-12% for $\overline{Nu}$ and 1% - 3% for $Re_j$. The largest uncertainties for $\overline{Nu}$ are seen for the smallest values of $T_o - T_i$.

As shown in Fig. 4, the seven experimental data sets each follow their own trend instead of conforming to a global $\overline{Nu}$ relation based solely on $Re_j$. This means it is incorrect to assume that $\overline{Nu}$ can be predicted by just $Re_j$—more specifically that all of the heat incident on the divertor is removed by convection at the cooled surface [Eq. (2)]. Since the HEMJ and HEMP divertor are both ‘finger-type’ and share many general features, it’s
reasonable to assume that their thermal performance can be generally characterized in the same manner. As shown in Figs. 1-3, the designs remove heat by both convection through the cooled surface and conduction down the walls of the divertor. Using a similar method presented in more detail in Ref. 13 regarding experimental and numerical studies on a HEMP-like divertor, an additional non-dimensional parameter $k_s/k_c$, or $\kappa$, was required to account for the change in conduction and convection heat removal rates at the cooled surface based on the different combinations of structure and coolant thermal conductivities.

As indicated on Fig. 4, the trends of $\bar{Nu}$ to $Re_j$ are layered according to $\kappa$. By relating both $Re_j$ and $\kappa$ to $\bar{Nu}$, and assuming a power law correlation, each of the seven experimental data sets is now shown to follow a global trend (Fig. 5). Nearly all of the data points fit within the range of $\pm10\%$ of the correlation as shown by the dashed lines. A dotted line indicates the prototypical operating point. The values of the fitting parameters are given by Eq. (4):

$$\bar{Nu} = 0.059 \cdot Re_j^{0.669} \kappa^{0.190}$$  \hspace{1cm} (4)

The approximate value of $\kappa$ for each experimental setup is shown in Table I and varies from $\sim370$ for the He-cooled steel thimble experiments to $\sim7000$ for the Ar-cooled brass thimble experiments. The prototypical value of $\kappa$ is expected to be near 340. By covering a large range of both $Re_j$ and $\kappa$, it is ensured that any extrapolation based on the experimental data is well characterized.

![Fig. 4. $\bar{Nu}$ versus $Re_j$ for all seven experimental configurations. Solid symbols for brass thimbles (air $\bullet$, Ar $\bullet$, He $\blacksquare$, and electrically heated air $\blacktriangle$) and open symbols for the steel thimble (air $\diamond$, Ar $\bigcirc$, and He $\square$). Prototypical $Re_j$ value of 21,600 shown with dotted line. Experimental uncertainty is indicated, though uncertainty in $Re_j$ is almost unnoticeable at this scale for most cases.]

![Fig. 5. $\bar{Nu}$ versus $Re_j$ and $\kappa$ correlation [Eq. (4)] for all experimental data shown with curve fit and $\pm10\%$ dashed lines. Solid symbols for brass thimbles (air $\bullet$, Ar $\bullet$, He $\blacksquare$, and electrically heated air $\blacktriangle$) and open symbols for steel thimble (air $\diamond$, Ar $\bigcirc$, and He $\square$). Prototypical value of 140 ($Re_j=21,600$, $\kappa=340$) shown with dotted line.]

<table>
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The pressure loss coefficient, $K_L$, was also calculated for each experiment and is plotted versus $Re_j$ in Fig. 6. The density, $\rho_L$, and velocity, $V_L$, used in calculating $K_L$ are determined using the outlet pressure and inlet temperature:

$$K_L = \frac{\Delta p}{\frac{1}{2} \rho_L V_L^2}$$  \hspace{1cm} (5)
A curve fit assuming a power law plus a constant trend for $K_L$ is given by Eq. (6):

$$K_L = 1.39 \left( Re_j / 10^4 \right)^{-0.50} + 1.32$$

(6)

Since $K_L$ represents the “hydraulic” characteristics of the module, it is appropriately correlated by only matching $Re_j$ (and geometry). The loss coefficients from the electrically heated experiments are not included as those experiments did not use an accurate differential pressure measurement technique.

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Lines of constant $T_s$ are constructed by calculating $\overline{q''}$ from $\overline{T_s}$ and $Re_j$, by way of Eqs. (1), (2-4). First a guess is made for $T_s$ and $T_c$ allowing for the calculation of $K_L$ and thus $Nu$ by Eq. (4) and $\overline{h}$ by Eq. (3a). Then, a thermal resistance analogy for the conduction through the $\Delta z = 1$ mm pressure boundary allows for the calculation of $\overline{q''}$:

$$\overline{q''} = \frac{\overline{T_s} - T_i}{A_k \frac{1}{h_k} + \frac{\Delta z}{k_s \left( \frac{T_s + T_c}{2} \right)}}$$

(8)

where $k_s$ is calculated based on the average of $T_s$ and $T_c$. This new value of $\overline{q''}$ gives new guesses for both $T_s$ by Eq. (1) and $T_c$ by Eq. (2). The process repeats until the values of $T_s$, $T_c$, and $\overline{q''}$ converge.

Lines of constant $\beta$ are calculated using Eqs. (5-7) as follows: as $\rho_c$ and $V_c$ depend on $\Delta p_s$ an iterative process is applied to Eq. (5) using $K_L$ calculated with Eq. (6) from an assumed $Re_j$. With this value of $\Delta p$, $\overline{q''}$ is calculated based on a fixed value of $\beta$ using Eq. (7). This also requires iteration since $\rho_c$ depends on $\overline{q''}$.

As shown in Fig. 7, at the prototypical $Re_j$ of 21,600, the HEMJ is predicted to withstand $\overline{q''}$ of ~14.1 MW/m² with $\overline{T_s}$ of 1200 °C while having a $\beta$ value of ~8% based on a heated surface area of 227 mm². This corresponds to a heat flux of ~11.4 MW/m² for a hexagonal tile area of 280.6 mm² corresponding to a tile flat-to-flat distance of 18 mm (Fig. 1).
VI. CONCLUSIONS

As the HEMJ divertor design has been shown to withstand over 10 MW/m² incident heat flux at its nominal mass flow rate, an experimental investigation was undertaken to characterize the divertor’s behavior under a wide range of operating conditions. These experiments used air, Ar, and He over a range of $Re_j$ ($8 \times 10^3$-$6 \times 10^4$) and system pressure (0.6-1.4 MPa) to cool divertor test sections made of brass or steel and heated with an oxy-acetylene torch to heat fluxes ranging from 0.5-3 MW/m². Using nondimensional analysis and dynamic similarity, the results of these experiments were extrapolated to prototypical conditions (He cooling with inlet temperature of 600°C and pressure of 10 MPa) in order to produce performance curves. These curves allow magnetic fusion reactor designers to predict the HEMJ’s behavior at normal and off-normal conditions with any variety of materials selection. Specifically, the maximum allowable incident heat flux can be determined for a given coolant flow rate (i.e. Reynolds number) and maximum allowable pressure boundary temperature. Values of the corresponding pumping power fraction are also given.

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