MEASUREMENTS OF FULLY-DEVELOPED AUGMENTED CONVECTION IN A SYMMERICALLY GROOVED CHANNEL

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ABSTRACT

Measurements of fully-developed augmented convection and pressure drop of air flow in an isothermal, symmetrically grooved channel are reported for channel Reynolds numbers ranging from 800 to 5,000. Grooves, oriented transverse to the flow, are of triangular shape with dimensions that are comparable to the hydraulic diameter of the channel. The grooved section is designed to excite instabilities in the flow leading to increased mixing at sub-transitional Reynolds numbers.

Local heat transfer measurements are made using a holographic interferometer. Interferograms, representative of the cross-span-average temperature of the air in the channel, are analyzed to produce data records of the air temperature distribution and the local heat flux along the grooved walls. Heat flux distributions are spatially averaged to produce a correlation of fully-developed Colburn j-factor for this surface configuration. A performance evaluation of the grooved surface applied to a simple heat exchanger shows that it provides thermal performance which is comparable to other surfaces commonly employed in compact heat exchangers.

NOMENCLATURE

- \( p \) pressure
- \( q \) heat duty
- \( R \) gas constant, also a performance ratio
- \( Re \) Reynolds number
- \( S_l \) longitudinal pitch
- \( S_d \) diagonal pitch
- \( S_t \) transverse pitch
- \( T \) temperature
- \( V \) fluid velocity
- \( W \) channel width
- \( \lambda \) wave length of He-Ne light
- \( \mu \) viscosity
- \( \Theta \) non-dimensional temperature
- \( \rho \) density

INTRODUCTION

Compact heat exchangers invariably incorporate heat transfer augmentation technology at the fluid-solid interface. The more popular and well documented passive techniques summarized by Webb [1994] include: roughened walls, extended surfaces, and displaced inserts. Displaced inserts (for example, twisted tapes) modify the core flow in such a way as to increase transport at the tube wall. Geometric modification of the heat transfer passage walls will also modify the core flow and give rise to an increase in surface conductance. Spatially periodic disturbance promoters, such as a series of transverse grooves cut into a flow channel wall, destabilize Tollmein-Schlichting waves, leading to a self-sustained two-dimensional oscillatory flow at a Reynolds number of \( O(350) \) [Greiner, 1987]. A subsequent transition to three-dimensional oscillatory flow at Reynolds numbers of \( O(700) \), accompanied by an increase in surface conductance, has been demonstrated in asymmetrically heated channels containing a single V-grooved wall [Greiner et al., 1990; Greiner et al., 1996].
The present investigation is concerned with symmetrically heated parallel-plate channels having a series of transverse V-grooves formed in both walls. Temperature field and groove surface local heat flux measurements, obtained with a holographic interferometer, are reported for fully-developed flow in a channel having an aspect ratio of 20:1 for the Reynolds number range, \( 800 \leq \text{Re} \leq 5000 \).

Figure 1 shows the groove channel geometry under consideration. Grooves are symmetrically placed in the channel walls and they span the width of the channel. Each groove is right-triangular with depth, \( a \), and opening, \( 2a \). The channel wall-to-wall spacing is \( H \), and the channel width (perpendicular to the plane of the figure) is \( W \). The channel walls are symmetrically heated to a uniform temperature, \( T_s \), and \( T_e \) is the temperature at the center line of the channel, opposite the vertices of two opposing grooves. This work is concerned with flow sufficiently far from the channel entrance or exit that the convection processes are fully-developed, in a groove-periodic sense.

**EXPERIMENTS**

Experiments are conducted in two open circuit flow channels. Each test channel consists of a grooved section followed by a flat walled section. (The flat-walled section is used to study fully-developed grooved channel-to-flat channel flow recovery. These measurements will be reported in a subsequent presentation.) The larger of the two channels (\( a = 12.0 \pm 0.1 \text{ mm}, H = 10.0 \pm 0.3 \text{ mm}, W/H = 20.4 \text{ mm} \)) has seven V-grooves located in each wall. This facility is used for heat transfer measurements. The small test channel, which is used for pressure drop measurements, is 14-th scale of the large unit. It has 13 groove pairs in the grooved section.

For each of the two test channels, laboratory air enters the grooved section, passes through the flat-walled section, an exit plenum, and through ducting to a flow measurement section where the channel mass flow rate, \( \dot{m} \), and air temperature are measured. A calibrated ASME nozzle and differential pressure transducer having an accuracy of \( \pm 0.003^\circ \text{C} \) \( \text{H}_{2}\text{O} \) is used for flow rate measurements in the large channel. A bank of calibrated rotometers are used with the small channel. The flow rate in the channel is characterized in terms of channel Reynolds number,

\[
\text{Re} = \frac{\rho \nu D_h}{\mu} = \frac{2\dot{m}}{\mu (H + W)}
\]

where \( D_h = 2HW/(H+W) \) is the minimum hydraulic diameter of the grooved channel, \( V = \dot{m}/(\rho HW) \) and \( \rho, \mu \) are the fluid density and viscosity, respectively.

Twenty-four pressure taps are located along the centerline of one groove wall of the small test channel. Each tap is 0.5 mm in diameter. Pressure taps are located in the vertices of the thirteen grooves, one is located in a short flat section immediately upstream from the grooved section; and the remainder are located downstream from the grooved section. Pressure signals are routed through a Scanivalve to a differential pressure transducer having an accuracy of \( \pm 0.003^\circ \text{C} \) \( \text{H}_{2}\text{O} \). Differential pressure measurements are referenced to one of the measurement stations. A least-squares fit of the linear part of the pressure distribution in the grooved section gives the fully-developed pressure gradient, \( \nabla p \bigg|_{f,d} \), and the grooved channel friction factor is computed as

\[
f_g = \frac{-\nabla p \big|_{f,d} D_h}{2\rho \nu^2}
\]

A similar analysis of the (downstream) flat-channel data gives \( f \).

Six micro-foil heaters line the outer surface of each wall of the large channel. These are individually powered to maintain each wall temperature equal and isothermal to within \( \pm 0.4^\circ \text{C} \). Wall temperatures are monitored with copper-constantan thermocouples embedded in, and grounded to the surface of each of the triangular elements that form the grooved geometry. Measurements are made along the channel centerline at each of the vertices between adjacent grooves. Six additional detail measurements are made at the fourth triangular element of each wall to check for temperature asymmetry and spatial non-uniformity. The side walls of the large flow channel are of 4 mm thick plate glass.

Fluid temperature measurements are made with a holographic interferometer operated in real-time mode [Vest, 1979]. Measurements are made with the optical axis of the device aligned transverse to the mean flow. Interferograms are captured with a 600-line video camera and frame-grabber, and analyzed using an image processing system. The framing rate of the video system is 30 per second, so each frame represents a 33 msec. "snap-shot" of the temperature field. Since the light projected through the test section is collimated, each "snap-shot interferogram" represents a cross-span average of the temperature field in the test section. These supercritical groove-channel flows are known to be unsteady [Greiner et al., 1996, 1997], and randomly obtained interferograms of the temperature field at the same flow conditions are different from each other, in spite of the cross stream averaging that is implicit in the measurement technique. Therefore, for heat flux determination we randomly acquire several interferograms for a given flow condition and digitally average the images before analysis. We have found that averaging four "snap-shot" images allows us to maintain reasonable image contrast suitable for analysis.

Infinite-fringe interferograms are used to depict the basic features of the fluid temperature field in the grooved section. Wedge-mode interferograms are used to compute the heat flux distribution along segments of the grooved walls. In this case, fringe shifts are analyzed using an algorithm similar to one reported by McAuliffe and Wirtz [1991]. The locus of points of the bright and dark fringe
centerlines are identified. This determines the fringe displacement, \( d \), relative to a reference condition, and the temperature at each of these points is calculated. We use the groove wall temperature, \( T_s \), as our reference, so the temperature at a point near the wall is given as

\[
T = \left[ \frac{1}{T_s} + \frac{dR\lambda}{d \omega W K p} \right]^{-1}
\]

where \( K \) is the Gladstone-Dale constant, \( \lambda \) is the wavelength of He-Ne light, \( R \) is the gas constant for air, \( p \) is the pressure, and \( d \omega \) is the reference condition (no heat transfer) fringe spacing. A finite difference approximation is used to estimate the temperature gradient at the wall, \( \nabla T(0) \). We have found that in the grooved section, second- and third-order approximations give essentially the same result. The local Nusselt number based on wall-to-channel center temperature difference is then computed as

\[
Nu = \frac{-\nabla T(0)D_h}{T_s - T_c}
\]

(4)

The average Nusselt number along the groove faces, \( \overline{Nu} \), is calculated by trapezoid rule integration. Then the Colburn j-factor for each groove, based on \((T_s - T_c)\) and the projected area of the groove opening \((2aW)\) is

\[
j_o = \frac{h_p}{c_p \rho V} Pr_2^{-1} = \frac{\overline{Nu}}{Re Pr_1^{1/3}}
\]

(5)

A Monte Carlo technique is used to assess experimental error. Estimates of the 99% confidence-level expected error (3\( \sigma \)-value) in measured quantities are used to generate sets of pseudo-data which are normally distributed, with standard deviation \( \sigma \), about nominal measured values. Typically, 5000 such sets are created for each measurement point. These are input to the data reduction algorithm and the standard deviation of the output about the nominal result (i.e., calculated quantities such as heat transfer coefficient, friction factor, Colburn j-factor, etc.) is then used to establish the 99% confidence-level expected error in the results.

We establish estimates of 3\( \sigma _{VT(0)} \) by experiment. Selected interferograms are repeatedly re-analyzed where fringe shifts are intentionally misread by one pixel location. The effect of these "misreads" on the computed value of \( \nabla T(0) \) are recorded, and the resulting values of the 3\( \sigma _{VT(0)} \) are calculated. This results in 3\( \sigma _{VT(0)} \) ranging from 0.3 \( ^{\circ} \)C/mm at high heat flux to approximately 1 \( ^{\circ} \)C/mm at the lowest heat flux measured. Other 3\( \sigma \)-values, for example error estimates of flow rate measurement, etc., are selected based on instrumentation specifications and our experience.

The above described analysis shows that the 99% confidence-level expected error in reported Reynolds numbers is less than 7.1% at the lowest flow rates, and these values decrease to less than 1% at \( Re > 2400 \). The error for the Colburn-j factor is less than 7% for \( Re > 2400 \).

**Figure 2** Grooved section axial pressure variation. \( Re = 2411 \).

> 1000; it equals 15% at \( Re = 800 \). The error in friction factor ranges from approximately 15% at the lowest flow rates to less than 5% at high Reynolds numbers.

**RESULTS**

Figure 2 shows the axial pressure distribution obtained in the small test channel when the channel Reynolds number, \( Re = 2411 \). The figure is typical of data obtained at \( Re \geq 1294 \). The second through 14-th measurement points from the channel entrance are in the grooved region. The figure shows that the flow is essentially fully-developed by the 4-th groove. Furthermore, the pressure in the 13-th groove is little affected by the transition to a smooth channel that follows. A linear, least-squares fit of measurement points 5 - 14 establishes the fully-developed grooved section pressure gradient \( \nabla P_{f.d.} \). Pressure-drop data for \( Re = 996 \) (the lowest \( Re \) considered) show that the flow is probably not fully developed until the 7-th groove. In this case, \( \nabla P_{f.d.} \) is based on measurement pts. 8 - 14, where the pressure profile is linear. A similar analysis of measurements downstream from the grooved section gives the fully-developed smooth wall pressure gradient.

Figure 3 plots measured values of \( f_f \) and \( f_p \) as a function of \( Re \). The measured results (open circles) are compared to the laminar result for flow between parallel plates, and a turbulent correlation for flow in a tube having the same hydraulic diameter (solid lines) [Kakac, et al., 1987].

\[
f_F = \frac{24}{Re}
\]

(6)

\[
f_F = 0.00128 + 0.1143 Re^{-0.311}
\]

(7)

For the purposes of comparison, these two correlations have each been extended to their intersection point, which occurs at \( Re = 1914 \).
Figure 3 Grooved and flat-channel friction factor.

Also shown in the figure are recent computational results for the same grooved channel geometry [Greiner et al., 1997, open diamond symbols]. The flat-channel experimental results are in reasonable agreement with Eqs. (6) and (7). These measurements lay approximately 10% above the correlations. This may be due to experimental offset error, or it may be due to residual mixing in the flow from the (upstream) grooved section. The grooved section measurements are about 15% higher than the direct numerical simulation results. This may be due to the above mentioned potential offset error. On the other hand, the numerical simulation is observed to produce a curious shift in the slope and magnitude of $f_0$ at $Re \approx 600$. Figure 3 shows that the measured grooved channel friction factor increases to a maximum value at $Re \approx 3800$, and then slowly decreases as $Re \to 10^4$. Measured values of $f_0$ are tabulated in Table 1.

Figure 4 Infinite-fringe interferogram at 6-th groove. $Re \approx 2400$.

Figure 4 shows an infinite fringe interferogram of the fully-developed temperature field at the sixth groove location. The channel Reynolds number is 2400. The image is rotated 45° (anti-clockwise) to facilitate analysis, so the flow appears to be from upper right to lower left in the figure. Each face of each groove is $\sqrt{2a} = 17$ mm long. Each fringe represents an isotherm, and the temperature difference between each fringe is approximately 4.6 °C. The figure shows an essentially symmetric temperature distribution with the core of the flow nearly isothermal. The flow re-circulates within each groove cavity, and thermal boundary layers form near each groove face. Since the re-circulatory flow first impinges on the windward groove face, thin thermal boundary layers are evident. The boundary layer thickens near the vertex of the groove, and then rapidly thins along the leeward face due to a second impingement of the flow. As the flow turns again to rejoin the core fluid motion, the boundary layer quickly thickens. This pattern, which is representative of images obtained over the entire Reynolds number range investigated.

Figure 5 Dimensionless temperature iso-contour maps at $Re = 800$, 2400 and 4800.
Figure 6 Local Nusselt number in grooved surface.

suggests that the heat flux distribution along the groove faces is very non-uniform.

Figure 5 shows iso-contour maps of the non-dimensional temperature, $\Theta = (T-T_0)/(T_r-T_0)$ where $T_0$ is the inlet temperature. These maps are created by analysis of interferograms such as Figure 4. The figure shows the temperature field adjacent to the sixth groove for three channel Reynolds numbers, Re = 800, 2400 and 4800, respectively. It displays trends in temperature field development as the coolant flow rate increases. An increase in flow rate results in a reduction in the axial temperature variation of the core flow, generally thinner thermal boundary layers along the groove walls, and a more uniform temperature in the recirculating flow regions.

Figure 6 shows the local groove surface Nusselt number for the conditions of Fig. 5. Nu is plotted versus non-dimensional position measured from the vertex of the groove, $x/\sqrt{2a}$. Also shown in the figure is a computational result for Re = 850 [Greiner, et al., 1997, dashed line]. The local Nusselt number on the windward groove face ($x/\sqrt{2a} < 0$) is seen to be minimum near the vertex of the groove and increase towards the groove opening. The heat flux along the leeward groove face is considerably smaller than that observed along the windward face. It is minimum near the groove vertex and increases to a local maximum at $x/\sqrt{2a} \approx 0.3$ (due to the second impingement of flow discussed in connection with Fig. 4). The heat flux then passes through a local minimum and increases again toward $x/\sqrt{2a} = +1$.

The measurements at Re = 800 are in reasonably good agreement with the numerical simulation (Re = 850) except that the data do not increase without bound as $x/\sqrt{2a} \rightarrow \pm 1$. Figure 6 shows the data generally leveling off as $x/\sqrt{2a} \rightarrow \pm 1$. On the other hand the numerical simulation shows $Nu(\pm 1) \rightarrow \infty$ at both groove tips. As a consequence, experimentally determined estimates of overall heat transfer coefficient will be lower than those predicted by the numerical code.

Figure 6 shows that we cannot measure $\nabla T(0)$ in the last 2–3 mm near the tip of the windward groove face (5% - 10% of the groove surface). For the purposes of determining the overall heat transfer coefficient, we linearly extrapolate the data to $x/\sqrt{2a} = -1$.

Figure 7 shows measured fully-developed grooved and flat-channel Colburn j-factors (open circles). The flat-channel measurements are obtained 55 Ds downstream from the grooved section where it is thought that groove-induced mixing has dissipated. Also shown in the figure are computational results of Greiner et al. [1997], and the laminar result for flow between parallel plates,

$$j_R = \frac{5.86}{Re Pr^{1/3}}$$  \hspace{1cm} (8)

Note in Eq. (8), the coefficient is 5.86 instead of the expected value, 7.54, since j is based on the temperature difference, $T_r - T_c$.

The flat-channel measurements are in almost identical agreement with Eq. (8) at Re = 800, and they reside increasingly above Eq. (8) as the Reynolds number increases toward the transition point. This may be due to residual grooved-induced mixing in the flow. At Re > 2400, there is a transition in $j_R$ which is suggestive of transition toward turbulence.

The groove section measurements at Re = 800 are in good agreement with the simulation results of Greiner et al. This may be somewhat coincidental since our pressure profile measurements indicate that at this Reynolds number, the flow is probably not fully-
Figure 8 Corrugated-fin (grooved channel) and pin-fin heat exchange cores.

An extrapolation of the present experimental results suggests that \(j_o\) is greater than \(j_e\) for \(Re \geq 450\), in agreement with the simulation. There is a continuous decrease in \(j_o\) as \(Re\) increases, and it appears that \(j_o\) will approach \(j_e\) (resulting in no heat transfer augmentation) as \(Re\) increases beyond a value of about 10,000. Table 1 summarizes measured values of \(j_o\) and \(j_e\). Note, the second reported \(j\)-factor value at \(Re = 2400\) is measured in the 5-th groove. The close agreement with the 6-th groove value and pressure profile measurements at \(Re = 1294\), tend to confirm that the flow at \(Re \geq 1600\) is fully-developed.

Table 1 Summary of \(j_o\) and \(j_e\) at sixth groove-pair.

<table>
<thead>
<tr>
<th>Reynolds No.</th>
<th>Friction fac. (\mu)</th>
<th>Reynolds No.</th>
<th>Colburn (j)-factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>996</td>
<td>0.060</td>
<td>800</td>
<td>0.0130</td>
</tr>
<tr>
<td>1294</td>
<td>0.062</td>
<td>1600</td>
<td>0.0091</td>
</tr>
<tr>
<td>1625</td>
<td>0.065</td>
<td>2400</td>
<td>0.0074</td>
</tr>
<tr>
<td>2411</td>
<td>0.074</td>
<td>2400</td>
<td>0.00715th</td>
</tr>
<tr>
<td>3134</td>
<td>0.083</td>
<td>3200</td>
<td>0.0064</td>
</tr>
<tr>
<td>3814</td>
<td>0.089</td>
<td>4000</td>
<td>0.0054</td>
</tr>
<tr>
<td>4627</td>
<td>0.087</td>
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</tr>
<tr>
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<td>0.086</td>
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</tr>
</tbody>
</table>

PERFORMANCE EVALUATION

The potential of the grooved surface as a high performance heat exchanger core can be assessed by comparing its performance with other high performance surfaces. Consider a plate-fin exchanger having face area, \(A_f\) and flow length, \(L\). The coolant (air, \(Pr = 0.71\)) approaches the exchanger at temperature \(T_s\) and face velocity is \(V_o\). The plates of the exchanger are at temperature \(T_s\). In the following we compare the heat duty, \(q\), and pressure drop across the exchanger, \(\Delta P\), assuming the overall size of the exchanger \((A_fL)\), temperature difference, \((T_s - T_o)\), and the coolant flow rate are fixed (an FG-1a comparison, Webb, 1994).

Consider the two exchanger cores shown in Figure 8. Case I shows an edge view of thin corrugated fins that span the gap between the exchanger plates. The space between each corrugated fin forms a symmetrical V-grooved passage having groove depth, \(a\). The minimum spacing between corrugated fins is \(H\), so the transverse fin pitch is \(S_t = H+a\). The pin array has \(H/a = 10/12\). The heat transfer surface area per unit volume of the corrugated-fin array is \(2\sqrt{2}/S_t\). The approach flow is quantified in terms of face Reynolds number

\[
Re = \frac{\rho V_o 2S_t}{\mu} = Re
\]

Case II shows an edge view of a staggered pin-fin array that spans the gap between the exchanger plates. The transverse pitch of the array is equal to that of the corrugated fin array, \((H+a)\), and the pin fins have diameter, \(a\), equal to the corrugated fin array groove depth. The longitudinal pitch of the array, \(S_p\), is adjusted so that the pin-fin array has the same heat transfer surface area-to-volume ratio as the corrugated fin array. This requirement gives \(S_t = \pi a/2\sqrt{2}\). The pin-fin array friction factor, \(f_r\), and \(j\)-factor, \(j_r\), are given by correlations for staggered tube banks attributed to Zhukauskas [Kakac et al., 1987].

Assume each heat exchanger is isothermal at temperature, \(T_s\), and that fins in each are perfect conductors. We also assume that the mean fluid temperature in the corrugated-fin array, \(T_m\), is equal to \(T_s\) (the center temperature defined in Fig. 1). Greiner et al. [1997] show that this condition is satisfied in grooved channel flows when \(Re \geq 850\). Then the ratio of heat duties of the two exchangers at the same coolant flow rate is equal to the ratio of their respective effectiveness'.
\[ R_q (Re_o, L) \equiv \frac{q_o}{q_T} = \frac{1 - e^{-n u_o}}{1 - e^{-n u_r}} \quad (11) \]

where \( n u \) is the number of transfer units of each exchanger. For the present example

\[ n u_o = \frac{2L}{H} j_o \text{ Pr}^{\frac{2}{3}} \quad (12) \]

and,

\[ n u_r = \frac{2\sqrt{2L}}{S_x - \alpha} j_r \text{ Pr}^{\frac{2}{3}} \quad (13) \]

where \( S_x = \sqrt{S_t^2 + S_i^2 / 4} \) is the diagonal pitch of the pin-fin array (\( V_{max} \) occurs along the diagonal pitch in the present example).

In addition, the ratio of pressure drops is given by

\[ R_p (Re_o) = \frac{\Delta P_o}{\Delta P_T} = \frac{2S_x (S_x - \alpha)^2}{H^2} \frac{f_o}{f_T} \quad (12) \]

Since the coolant flow is assumed to be fully-developed in both cases, \( R_p \) is independent of exchanger length, \( L \).

Figure 9 plots \( R_q \) (dash lines) and \( R_p \) (solid line) as a function of face Reynolds number, \( Re_o \), for exchangers having \( L/S_t = 50 \) and 100. The calculation shows that the heat duty of the corrugated fin design is lower than that of the pin-fin design (by as much as 28% for the shorter exchanger). However, at equal flow rates, the pressure drop across the corrugated fin array is less than 15% of that for the pin-fin array for \( Re < 5000 \).

CONCLUSIONS

Air flows at \( Re \geq 1296 \) in symmetrically v-grooved channels become fully-developed, in a groove-periodic sense, after roughly four groove-pairs, and the fully-developed character of the flow maintains to the last groove-pair in the series. Thin thermal boundary layers form adjacent to groove faces, and the heat flux distribution is non-uniform with a minimum at the vertex of each groove. The Nusselt number of the windward face is approximately twice that of the leeward face.

The fully-developed j-factor, \( j_o \), becomes greater than the equivalent flat channel values when \( Re > \Omega(450) \), and it appears that heat transfer augmentation is maintained to \( Re \approx 10^4 \).

The comparison of the performance of a simple plate-fin heat exchanger which employs the v-groove technology to the same exchanger having a pin-fin array core is arbitrary in the selection of pin-fin diameter and pitch. However, it shows that the v-groove technology can produce enhanced performance that is comparable to other augmented surfaces used in compact heat exchange design.

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