HEAT TRANSFER AUGMENTATION THROUGH WALL SHAPE INDUCED FLOW DESTABILIZATION

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ABSTRACT

Experiments on heat transfer augmentation in a rectangular cross section water channel are reported. The channel geometry is designed to excite normally damped Tollmien-Schlichting modes in order to enhance mixing. In this experiment, a hydrodynamically fully developed flow encounters a test section where one channel boundary is a saw-tooth series of periodic, transverse grooves. Free shear layers span the groove openings, separating the main channel flow from the recirculating vortices contained within each cavity. The periodicity length of the grooves is equal to one-half of the expected wavelength of the most unstable mode. The remaining channel walls are flat, and the channel has an aspect ratio of 10:1. Experiments are performed over the Reynolds number range of 200 to 1500.

Streak line flow visualization shows that the flow is steady at the entrance, but becomes oscillatory downstream of an onset location. This location moves upstream with increasing Reynolds numbers. Initially formed traveling waves are two-dimensional with a wave length equal to the predicted most unstable Tollmien-Schlichting mode. Waves become three-dimensional with increasing Reynolds number and distance from onset. Some evidence of wave motion persists into the turbulent flow regime.

Heat Transfer measurements along the smooth channel boundary opposite the grooved wall show augmentation (65%) over the equivalent flat channel in the Reynolds number range 1200 to 4500. The degree of enhancement obtained is shown to depend on the channel Reynolds number, and increases with the distance from the onset location.

INTRODUCTION AND PROBLEM DEFINITION

In heat exchanger applications involving low density fluids (gases), it is well known that mechanical pumping power requirements may be comparable to heat transfer rates if flow velocities are not limited to small values. As a result, "compact" exchangers involving passages with small hydraulic diameter have been developed where gas side velocities are kept low, and surface area is maximized in order to make up for the relatively low heat transfer coefficients. Kays and London [1964] point out that another way to minimize friction power requirements is to select enhanced surfaces which "...plot 'high' on a heat transfer-friction power plot."

A recent review [Bergles and Webb, 1985] shows an exponential growth in the heat transfer augmentation literature during the past thirty years, underscoring the importance of the problem which, if current interest is any indication, has resulted a solution that is satisfactory for all applications. Enhancement schemes are classified as active (those requiring external energy input) or passive (which are inherently more reliable). Passive schemes generally involve an increase in the complexity of the heat transfer surface. For example, offset-strip fins, louvers and plates are added to plate fin exchangers in order to promote mixing in the plate-side stream, and to interrupt the thermal boundary layer along the plate-fin, giving rise to an increase in heat transfer coefficient [Mori and Nakayama, 1980; Joshi and Webb, 1987]. Since leading edges also cause very high surface shear stresses, these mechanisms also lead to significant increases in friction power requirements.

Due to the generally small passage hydraulic diameters and low face velocities employed in compact heat exchangers, gas side passage Reynolds numbers are relatively low (approximately 2000 or less), giving rise to laminar, or at best transitional flows. Therefore, a successful augmentation scheme must be operable in a flow with strong viscous damping characteristics.

The present investigation involves a passive scheme which enhances mixing in the fluid stream, not by turbulence, but by promotion of a secondary flow through careful design of the channel contour. The

NOTATION

a groove depth
b groove length
D_g hydraulic diameter
E enhancement ratio, Nu(ground)/Nu(flat)
G^2-1 inverse Graetz number, (x/D_g)/Re Pr
h heat transfer coefficient
H channel height
k fluid thermal conductivity
L test section length
N_u Nusselt number, hD_g/k
P r fluid Prandtl number
q** heat flux
Re, Reynolds number, V D_g/v
Re_c critical Reynolds number
T temperature

\( T_m \) mixed mean temperature
\( T_o \) lower wall temperature
\( T_s \) heated wall surface temperature
\( \nu \) average velocity
\( \nu \) channel span
x axial coordinate
\( \lambda \) flow periodicity length
\( \nu \) kinematic viscosity
Flow is inherently laminar, but is more complex than flat channel counterparts. The secondary flow is designed to increase convective mixing normal to the heat transfer surface, thus promoting transport augmentation.

Since the complexity of the flow is increased, it is expected that for a given Reynolds number, pumping power requirements for the present geometry will be greater than flow in a flat channel. However, increases in pumping power requirements come about because of organized mixing of the flow, because of the direct manipulation of the surface shear stress (as is done with a periodically interrupted surface), or through the introduction of chaotic motion (as in flow turbulence). Some evidence [Kozlu et al., 1988; Karniadakis et al., 1988] indicates that the pumping power cost of the current technique is favorably compared to more conventional methods. While this issue is not addressed in the present work, it is the subject of current investigation.

The focus of the present paper is to extend concepts which have been developed numerically [Ghaddar et al., 1986], using experimental techniques. In that work, rectangular shape grooves were studied under periodically fully developed flow conditions. A new cavity geometry which shows promise as an efficient heat transfer enhancement device is investigated. Transport measurements are made in a thermally developing flow along an adjacent flat surface in order that direct and meaningful comparisons may be made with the performance of a parallel channel flow.

The specific geometry considered in this work is indicated in Figure 1. A fully developed flow is discharged from a parallel-wall flow development channel whose upper wall is insulated and whose bottom wall is maintained at a uniform temperature, T_b. At the position x = 0, this flow enters a test section, of height h, whose upper surface dissipates a uniform heat flux, q^*, and whose lower surface is maintained at a constant temperature, T_p. Heat transfer measurements are performed at the upper wall under two sets of geometric conditions of the bottom surface: a flat surface, and a saw-toothed profile. The first condition provides a baseline against which enhancement, caused by a grooved lower surface, can be compared. It also serves to certify the validity of the measurement system.

The grooved surface is specifically designed to excite Tollmien-Schlichting waves in the channel flow which under flat wall conditions are damped at Reynolds numbers less than a critical value, Re_c = 15400 [Drazen and Reid, 1981]. The critical Reynolds number is reduced by introducing spatially periodic disturbances whose periodicity length is compatible with the most unstable Tollmien-Schlichting wave length. At supercritical conditions, traveling waves whose amplitude increase with Re = Re_c, promote transport perpendicular to the channel walls.

A groove aspect ratio of a/b = 2 is used so that grooves act as "open" cavities whose openings are completely spanned by free shear layers for the Reynolds number range considered [Yee, 1985]. These shear layers cause velocity profile inflection points to form in the channel. At Reynolds numbers above the critical value, Kelvin-Helmholz instabilities of these layers have been shown numerically [Ghaddar et al., 1986] and experimentally [Greiner, 1986] to destabilize the normally-damped Tollmien-Schlichting waves. The destabilized flow exhibits traveling waves which alternately pump core fluid to and away from the walls.

The channel periodicity length, a, is chosen to destabilize the most unstable Tollmien-Schlichting wavelength, \lambda. Since this wavelength is Reynolds number dependent, and the present investigation considers the range 200 < Re < 15000, a representative value of Re = 1400 is selected. At this Reynolds number \lambda = 2.4h [Ghaddar et al.], and the channel periodicity length, a, is chosen so that \lambda = \alpha/2.

![Figure 1. Geometric and thermal boundary conditions.](image)

![Figure 2. Closed loop test apparatus.](image)

The critical Reynolds number for the onset of natural oscillations has been shown to decrease with spacing between the grooves [Greiner, 1986]. In the current design, a "saw-tooth" profile is selected since it minimizes cavity separation, and the critical Reynolds number in fully developed flow is predicted to be roughly \( Re_c = 330 \) [Greiner et al., 1988]. Furthermore, this geometry is expected to produce an efficient cavity flow, and be easily manufactured.

Convective heat transfer measurements are performed for the Reynolds number range, 200 < Re < 15000, where Re = VD_b/\nu with V average velocity, D_b the channel hydraulic diameter, and \nu the fluid kinematic viscosity. The local Nusselt number, Nu = hD_b/\kappa, where \kappa is the local heat transfer coefficient and \kappa is the fluid thermal conductivity, is reported at five x-locations along the upper flat surface. The Nusselt number dependence on Reynolds number, channel location and channel geometry is determined experimentally.

EXPERIMENTAL APPARATUS

Measurements were made using the temperature-controlled recirculating water channel shown in Figure 2. A centrifugal pump delivers distilled water through a bank of rotameters and control valves to the right side of a partitioned reservoir which contains a cooling coil for temperature control. The left side of the tank is fed via a weir at the top of the partition and can supply up to a 1.2 meter head to the test channel. The flow passes two honeycomb sections and enters the channel flow development section through a "soda straw" flow straightener. The flow development section has height, H = 20 mm, width, W = 263 mm giving a hydraulic diameter, D_h = 36.4 mm. Velocimetry measurements show that the flow development length of 67 D_h is sufficient to assure fully developed conditions at the test section inlet (x = 0 in Figure 1). The bottom and side walls of the flow development section are aluminum, and the top surface is Plexiglas. The temperature at x = -30 mm is monitored by a digital thermometer and maintained at \( T_b = 29.4 \pm 0.6 \) \^C, which is within 3 \^C of the laboratory room temperature.

The test section is 30.3 hydraulic diameters long, and includes forty-six V-shaped grooves which span the lower surface. These are constructed by mounting right-triangular aluminum ribs, 12 mm high and 24 mm at their base, to a 12 mm thick aluminum base plate, which is in turn backed by a water jacket for temperature control. The bottom wall is maintained isothermal to within \pm 0.2 \^C while the overall driving temperature difference for experiments ranges from 2 to 10 \^C. A 12 mm thick aluminum plate is substituted in place of the triangular elements for the base-line flat channel experiments.

Two different upper surfaces are employed in the test section: one for flow visualization experiments, the other for heat transfer measurements. Both are fabricated from 25 mm thick Plexiglas. Flow visualizations are performed by injecting colored tracer into the flow field and recording the resulting patterns on video tape. A variable volume flow rate syringe pump is used to inject the pigment at channel center-height and mid-span through an L-shaped tube (1.0 mm O.D.) inserted through the channel ceiling and bent downstream. For each Reynolds number, the tracer flow rate is adjusted so that its velocity at the injector tip is the same as the average fluid velocity, V, thus minimizing disturbances to the channel flow. For the applicable channel
Reynolds number range, the injector Reynolds number is less than 40, thus causing negligible flow disturbance. The pigment is a solution of red food coloring, diluted with roughly seventeen parts water. While the dye density is slightly greater than that of the working fluid, its settling velocity is much less than that of the center channel speed, even at the lowest Reynolds numbers considered.

The heat transfer surface has six custom heater/thermocouple/heat flux gage assemblies bonded to its surface. The heat flux passing to the fluid from each assembly is monitored by a 102 mm by 102 mm thermopile-type heat flux gage (accuracy ± 1%) located at its center. The assemblies contain copper-constantan thermocouple junctions located 0.28 mm beneath the wetted surface at eighteen equally spaced points along the channel center-line. These thermocouples are referenced to a junction located at the center of the inlet channel at x = -50 mm. After corrections are made for the conduction temperature drop between the wall thermocouples and the wetted wall surface, and the local fluid mean temperature is computed using an energy balance, the local temperature difference between the wall and the mixed mean fluid temperature, AT = T* - Tm, is used with the local heat flux to determine the local heat transfer coefficient, h.

Each of the six combination gage's heaters are wired in series with a trimming rheostat, and these subsystems are wired in parallel to a regulated DC power supply. During an experimental run, the trimming rheostats are adjusted so that the indicated heat flux through each combination gage is the same, resulting in a heat flux uniformity of ± 1% of the average heat flux input.

Following the test section the flow passes another "soda straw" flow straightener and enters a small plenum chamber with a return line to the pump. Edge walls of the test section are 6 mm thick Plexiglas, and all metallic surfaces are coated with black epoxy paint. The apparatus is covered with 50 mm thick expanded foam insulation during data collection.

RESULTS

Flow Visualization

Figures 3 and 4 show a series of streakline patterns which document the onset of natural flow oscillations and the subsequent breakdown to turbulence. These visualizations are produced by injecting colored tracer at the leading edge of the thirty-eighth groove (x/Do = 24.4). The streakline pattern shown in Figure 3a is typical of subcritical flows. The flow is steady and the grooves contain slowly turning vortices. The outer channel streaklines move parallel to the flat wall, much like the flow in ungrooved (flat) channels. At Re = 600, small amplitude laminar waves are intermittently observed between long periods of steady flow. At a Reynolds number of 700 (Figure 3b), the flow is almost continuously oscillatory, with occasional steady periods. A traveling wave structure develops with a regular wavelength, which Figure 3b shows to be roughly equal to two groove lengths. The channel geometry is designed so that the wavelength of the most unstable Tollmien-Schlichting mode is equal to two groove lengths, and it is not surprising that this mode is the first to be excited. These streaklines are "smooth" and views from the top of the tank indicate that they are mostly two-dimensional. As the Reynolds number increases, however, the patterns become more irregular and three-dimensional. Small scale structures are superimposed on the long wavelength Tollmien-Schlichting waves in Figure 4a for Re = 1000. As the Reynolds number increases, the length scales of the smaller structures decrease and the three-dimensionality of the flow increases, as seen in Figure 4b.

The dye injection visualization technique loses its effectiveness at demonstrating the smallest scale motion at higher Reynolds numbers where the tracer rapidly dissipates. The dye fan envelope does appear to experience periodic large scale oscillatory motion when viewed in real time, suggesting that the groove spanning free shear layer may be capable of affecting fully turbulent flows in channels. This visualization technique, however, is severely limited at this turbulence level.

A graph showing the fraction of time the flow exhibits an oscillatory behavior, as a function of Reynolds number, is presented in Figure 5. These data are determined by viewing a flow pattern for a given time period and measuring the fraction of this time the flow is "oscillatory." These measurements are somewhat qualitative because the oscillatory amplitude varies continuously, and does not exhibit an "on/off" behavior. Multiple data points are reported for Reynolds numbers at which more than one observation is made, indicating the imprecision of this technique. Figure 5 shows that the oscillatory flow...
Figure 5. Oscillatory flow time fraction versus Reynolds number. Two-dimensional transition occurs in the range $600 < Re < 660$.

Time-fraction increases sharply in the Reynolds number range 600 to 660. As the Reynolds number increases beyond 700, the flow is observed to be continuously oscillatory.

If the observed critical Reynolds number, $Re_c$, is defined (arbitrarily) as the value at which the flow is oscillatory fifty percent of the time, then for this channel location, $Re_c = 630 \pm 20$. While oscillatory modes are known to be susceptible to forced flow rate modulation [Greiner et al., 1986], it is not known what effect any flow rate oscillations inherent in the apparatus has on the value of $Re_c$. The dependence of the observed value of $Re_c$ on location is discussed in the following section in connection with the onset of heat transfer enhancement. Reference to Figure 7 (triangular symbols) shows that $Re_c$ decreases with distance downstream. However, for the present apparatus an asymptotic value is not achieved. Work in progress with a test section which is $60 \, \text{D}_h$ long shows $Re_c$ approaches a value of 350 after about $35 \, \text{D}_h$ in good agreement with previous predictions [Greiner et al., 1988].

**Heat Transfer**

For each Reynolds number and axial location, local temperature measurements are made for a range of wall heat flux in order to detect any effect of natural convection, and to eliminate the effects of systematic temperature offset errors. A straight line is fit to each $q^*$ versus $\Delta T$ data set, and the slope of this line is used to determine the local heat transfer coefficient, $h = -q^* / \Delta T$. The relation is found to be highly linear, indicating the absence of significant buoyancy effects. In the following heat transfer plots, error bars representing the 99.7% confidence level due to random errors are indicated where the bar size exceeds data point symbol size. In many cases the error bar size is smaller than the symbol size, and is not shown.

Figure 6 shows the local Nusselt number as a function of inverse Graetz number for a flat channel flow. Data for $Re < 3500$ collapse to a single line, while that for $Re > 4000$ show the onset of significant turbulent transport. This data is used as the base-line for comparison with grooved channel measurements.

Figure 7 shows the grooved channel Nusselt number as a function of inverse Graetz number for $300 < Re < 15000$. The data is now seen to depart from a single curve at $Re = 1200$. At $Re = 1200$, the heat transfer coefficient is seen to increase at $Gz^{-1} = 0.003$, and the departure point has moved upstream to $Gz^{-1} = 0.0008$ at $Re = 2000$, with the trend continuing with increasing Reynolds number. The heat transfer coefficient is greater than corresponding flat channel values for $Re > 1200$, and it is actually less than the flat channel values for $Re < 1000$, as explained below.

The degree of heat transfer enhancement obtained is more easily seen in Figure 8, which is for a super-critical Reynolds number of 3000. Part (a) of the figure compares the local Nusselt number for the grooved channel flow with that obtained with the flat configuration, and part (b) shows the enhancement factor, $E = Nu_{\text{grooved}} / Nu_{\text{flat}}$, both plotted against dimensionless axial location. As expected, flat channel values show a steady decrease in the stream-wise direction, consistent with a thermally developing flow. On the other hand, the grooved channel values show enhancement at $x/D_h = 6$, with an augmentation of approximately 65% farther down stream.

Figure 9 shows $E$ as a function of $x/D_h$ for subcritical (Re = 300), supercritical (3000), and turbulent (5000) flows. For a subcritical Reynolds number of 300, the groove geometry is seen to actually decrease the heat transfer coefficient along the upper wall by about 10% relative to the ungrooved geometry. Groove spacing free shear layers relax the no slip condition along the lower wall, causing the velocity maximum to shift downward, resulting in reduced transport along the upper wall. As a consequence the enhancement ratio is less than unity over the entire length of the channel. For $Re = 12000$ (not shown in Figure 9), traveling waves are visually observed for $x/D_h > 10$. The grooved channel heat transfer is seen to exceed the ungrooved values for $x/D_h > 15$, resulting in enhancement ratios of approximately 1.1.

As the Reynolds number is further increased, the break even point ($E = 1$) rapidly moves upstream, and the magnitude of the local enhancement ratio increases. For $Re = 3000$, visual observations of the flow show remnants of a traveling wave structure blurred by turbulent mixing. The break even point shifts to $x/D_h = 7$, and $E > 1.5$ for $x/D_h > 16$ (Figures 8b and 9).

As Reynolds number is increased beyond 3000, the grooved channel local heat transfer coefficient continues to increase.
ungrooved heat transfer coefficient increases at a faster rate due to increased turbulent mixing. As a consequence, the local enhancement ratio is reduced as shown in the figure for Re = 3000. This result is effectively the same for all 5000 < Re < 15000.

Figure 10 includes a Re versus x/Dh map of groove-induced heat transfer enhancement. The figure shows the locus of points where enhancement exceeds 10% (E = 1.1). Also shown are wave onset locations determined from analysis of video records of flow visualization experiments. The figure indicates that the occurrence of groove-induced oscillations is closely correlated with effective heat transfer augmentation. Points to the right of the data band of the figure experience heat transfer augmentation via this mechanism. Points to the left of the band experience no enhancement, or a degradation in performance. Enhancement is seen to move upstream with increasing Reynolds number until Re = 4000, and move downstream for higher Reynolds numbers. It is currently thought that for Re > 4000, turbulent mixing mechanisms become sufficiently strong that they dominate the large scale wavy structures, reducing the difference between grooved and ungrooved channel flows.

Enhancement data as a function of Reynolds number for five measurement stations of the current apparatus is condensed in Figure 11. For the current channel configuration, the figure shows that significant, and spatially constant (i.e. fully developed) enhancement, is obtained for x/Dh > 16. Maximum heat transfer enhancement of approximately 65% is obtained over the range, 2000 < Re < 4000. At higher Reynolds numbers turbulence appears to overwhelm the natural oscillations, leading to a reduction in E.

Figure 11. Local grooved channel enhancement versus Reynolds number for 9.8 < x/Dh < 27.7. Enhancement factor is fully developed for x/Dh > 16.
CONCLUSIONS

Groove induced flow oscillations occur at super-critical Reynolds numbers. The onset location for such oscillations is Reynolds number dependent. Oscillation intensity and onset location rapidly move upstream with increases in Reynolds number. For the current channel configuration, oscillatory flows are observed for Re ≥ 630, and persists to Re = 4800 (approx.).

The oscillatory flow mechanism is responsible for augmentation of heat transfer. For the current channel configuration, enhancement in excess of 10% extends over the range, 1200 < Re < 4800, and x/Dh ≥ 16. Maximum enhancement of 65% occurs at Re = 3000 ± 1000. Turbulent mixing at higher Reynolds numbers degrades the effectiveness of the mechanism.

ACKNOWLEDGEMENT

This work is supported by the Gas Research Institute under contract number 5087-260-1562.

REFERENCES


