Augmented Heat Transfer in a Recovery Passage Downstream From a Grooved Section: An Example of Uncoupled Heat/Momentum Transport

Earlier experiments have shown that cutting transverse grooves into one surface of a rectangular cross-sectional passage stimulates flow instabilities that greatly enhance heat transfer/pumping power performance of air flows in the Reynolds number range 1000 < Re < 5000. In the current work, heat transfer, pressure, and velocity measurements in a flat passage downstream from a grooved region are used to study how the flow recovers once it is disturbed. The time-averaged and unsteady velocity profiles, as well as the heat transfer coefficient, are dramatically affected for up to 20 hydraulic diameters past the end of the grooved section. The recovery lengths for shear stress and pressure gradient are significantly shorter and decrease rapidly for Reynolds numbers greater than Re = 3000. As a result, a 5.4-hydraulic-diameter-long recovery region requires 44 percent less pumping power for a given heat transfer level than if grooving continued.

Introduction
Transport augmentation schemes are regularly used to reduce the size and operating costs of heat exchangers. Typical methods involve fins to extend the transport surface area, or offset strips that promote thin boundary layers (Kays and London, 1984). Nevertheless, heat transfer in these passages is generally limited by an absence of convective mixing since small geometric dimensions limit the Reynolds number.

Augmentation techniques have been developed that promote mixing in channels by intentionally stimulating normally damped Tollmien-Schlichting waves through the use of continuous arrays of spatially periodic disturbances such as transverse-cylinder eddy promoters (Greiner et al., 1988; Majumdar and Amon, 1992) or transverse grooves cut into the channel walls (Kozlu et al., 1988; Greiner et al., 1990, 1991a, 1991b). Deep grooves promote inflection points in the axial-velocity profile. Kelvin-Helmholtz instabilities of these points destabilize two-dimensional Tollmien-Schlichting waves in the channel mainstream for Reynolds numbers greater than Re > 350. Significant heat transfer augmentation coincides with the early onset of three-dimensional mixing at Re = 800, and this flow develops rapidly after the first groove. A significant result for air heat exchanger devices is that fully developed heat transfer is enhanced relative to laminar flow in a flat channel by as much as a factor of 4.6 at equal Reynolds numbers and 3.5 at equal pumping powers. Furthermore, significant augmentation extends into the transitional flow regime, 1000 < Re < 5000.

Investigations of heat transfer on flat surfaces downstream from disturbances have also shown favorable performance characteristics. For example, measurements made on a flat plate downstream from different turbulence-generating grids show that the ratio of heat transfer to friction factor increases linearly with increasing turbulence intensity (Blair, 1983). Maciejewski and Moffat (1992) measure heat transfer from a flat plate subject to very high levels of free-stream turbulence. Their measured heat transfer coefficients, along with that of Blair, increase linearly with the maximum turbulence intensity of the flow.

In the current work, a flat recovery region in a rectangular cross-sectional channel downstream from a grooved section is considered to determine what effect axially decaying unsteady flows and pressure gradients have on uncoupling heat and momentum transport. This paper reports measured axial variations of heat transfer, pressure gradient, and center-span velocity profiles for a range of Reynolds numbers, 1500 ≤ Re ≤ 5000. A method for determining axial-profiles of wall shear stress from pressure drop and velocity measurements is also developed. These profiles are compared directly to heat transfer measurements to determine whether the decay rates are significantly different, implying that heat/momentum transport mechanisms are uncoupled. As a practical measure of performance augmentation, the heat transfer/pumping power characteristics of the downstream recovery region are compared to the performance of a fully grooved channel.

Test Configuration
We consider the partially grooved channel shown in Fig. 1. A fully developed air flow from a 20:1 aspect ratio rectangular channel enters a section with thirteen right-triangular grooves cut into the lower wall (length a, depth b), and then passes into another flat-walled duct. Starting at the beginning of the grooved section, x = 0, the lower wall temperature is held constant at the inlet value T0, while the upper wall develops a thermal boundary layer by dissipating a uniform heat flux, q'.

This geometry is a modification of a "fully grooved" passage studied by Greiner et al. (1991a, 1991b). In that work, a passage with forty-six cavities is examined. Flow visualizations and pressure measurement from the previous work show that the flow is essentially fully developed by the fifth groove for Reynolds numbers greater than Re = 1000, indicating that the flow conditions at the exit of the grooved section of the current experiment are fully developed in a groove-periodic sense. Dimensionless temperature difference measurements are not fully developed, however. Our general experimental protocol is to measure local heat transfer and pressure drop from the upper, flat surface along with profiles of time-averaged and root-mean-squared (rms) axial velocity at the center span in the grooved and flat sections of the
channel for a range of Reynolds numbers. These results are compared to local measurements in the same passage with fully grooved and fully flat lower surfaces.

An open-loop wind tunnel is used in this investigation. Laboratory air at approximately 26°C and 8.7 × 10^3 Pa (elevation 1300 m) is drawn through a filter/screen box, an entrance nozzle, and then into a flat Plexiglas flow development section. The channel height is H = 10 mm, width (normal to the plane of the Fig. 1) is W = 203 mm, and length is l = 1.34 m. For the Reynolds numbers considered in this work, the flow is fully developed by the time it exits this section. The air then flows through the test section described below. Upon leaving that section, the fluid enters a large, baffled exit-plenum, flows through calibrated rotameters (whose three standard deviation flow rate uncertainty is always less than 4 percent), and is drawn into a variable speed blower.

The test section is 1.10 m long. Its chassis consists of a 12-mm-thick aluminum lower surface and two Plexiglas side walls. The lower aluminum plate is backed by a water jacket for temperature control, and has provisions for mounting a set of right triangular ribs, to form a thirteen-cavity grooved surface (a = 24 mm, b = 12 mm, Fig. 1), followed by a flat 12-mm-thick aluminum plate downstream. The minimum channel spacing is H = 10 mm, making the dimensionless grooved-section length Lg/Ls = 16.4, and flat recovery section length Ls/Ls = 40.3. The chassis is designed so that two different Plexiglas upper surfaces may be installed; one is for pressure drop and velocity profile measurements; the other is for heat transfer experiments.

The upper plate used for heat transfer measurements has six custom heater/thermocouple/heat-flux-gage plates bonded to its surface. The heaters are standard electrical resistive foil elements. Each of the six heaters is wired in series with a trimming rheostat, and these subcircuits are wired in parallel to a regulated DC power supply. The heat flux passing to the fluid from each plate is monitored with an accuracy of ±1 percent (99 percent confidence level) by a 102 mm square thermopile heat-flux gage located at its center. During an experimental run, trimming rheostats are adjusted until the gage-indicated heat flux is the same for each heater plate. The axial variation of the centerspan heat flux is therefore less than 2 percent.

The six plate assemblies contain copper-constantan thermocouple junctions located 0.28 mm beneath the wetted surface at 18 points along the channel centerline. These thermocouples are referenced to a rake of three locations installed in the flow development section, at x = -150 mm. The local difference between the surface and inlet temperatures is determined by measuring the thermocouple voltage difference. A small correction is made for the conduction temperature drop between the local thermocouple and the wetted surface. Temperature measurements are made for a range of heat flux and indicate that natural convection effects are unimportant (Greiner et al., 1991a, 1991b). The uncertainty (one standard deviation) in dimensionless temperature difference (inverse Nusselt number) β(x) is always less than 7 percent (Greiner et al., 1991a). The correction for radiation heat flux is found to be very small (less than 1 percent) since the aluminum grooved wall is maintained at a high gloss (emissivity 0.09).

Pressure measurements are made to determine how the pressure and its gradient, dp/dx, depend on location and Reynolds number. In this experiment, the upper plate used to measure pressure has 24.2-mm-diam taps along its center span (x = 0) starting at x = 0. The tap spacing is 24 mm in the grooved section and the region just downstream (0 ≤ x/Dh = 21.4) and 102 mm apart further downstream. The pressure difference between the first and the other 23 taps is measured with an electronic pressure transducer with a ±2.5 Pa accuracy (99 percent confidence) and plumbed using a switching valve.

Profiles of the streamwise velocity component are measured using a single component laser velocimeter in conjunction with a counter processor and frequency shifter. Transmitting optics

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**Nomenclature**

- a = groove length (Fig. 1)
- A = channel cross-sectional area = HW
- b = groove depth (Fig. 1)
- Du = minimum channel hydraulic-diameter = 2HW/(H + W)
- f = Fanning friction factor = τ/((ρ/2)Y^2)
- Jg, f = Fanning friction factor, in fully developed flat and grooved channels, respectively
- H = minimum channel height (Fig. 1)
- k = fluid thermal conductivity
- Lg = recovery-section length, Fig. 1
- Ls = recovery-section length, Fig. 1
- M = dimensionless momentum-flux gradient, Eq. (3b)
- Nu = zone-average Nusselt number, Eq. (9)
- p = pressure
- Pr = fluid Prandtl number
- q = heat flux
- Re = Reynolds number = VD/ν
- Rg = wall shear stress recovery coefficient = (fLg/fLs - fLs/fLs)
- S = velocity profile shape factor, Eq. (4a)
- T0 = fluid entrance temperature
- T0 = local flat-surface temperature
- u = axial velocity
- Ue = average center-span axial velocity = (1/1H) ∫0H u0 dx
- V = mass-averaged velocity based on minimum channel cross-sectional area
- W = channel width, Fig. 1
- x, y, z = coordinates, Fig. 1
- θ = local, dimensionless, upper-wall temperature = (T0 - T0)/k/q^*Dg
- β = local dimensionless temperature in fully flat and fully grooved ducts, respectively
- ν = fluid kinematic viscosity
- ρ = fluid density
- τ = effective wall shear stress
- Φ = zone-average dimensionless pumping power, Eq. (10)

**Subscripts and Superscripts**

- ( ) = time-averaged steady component
- (') = unsteady component
- (0) = center-span value, z = 0
- (rms) = root-mean-square deviation
with a beam crossing angle of 8.9 ± 0.2 deg (99 percent confidence) produce an ellipsoidal measurement volume with a 0.5-mm minor diameter and a 1.0-mm major diameter. The resulting velocity calibration uncertainty is less than 3 percent. Receiving optics are arranged in back-scatter mode. Particle seeding (propylene glycol) is introduced into the flow at the entrance nozzle of the wind tunnel. Variations in particle seeding rate show no effect on reported values.

The LDV is traverse mounted with a positioning accuracy of ±0.1 mm and measurements are made across the channel height, 0 < y/H < 1, at center span, z = 0. At each location 1024 samples of the axial velocity are taken. Increasing the sample size shows that the time-averaged \( \bar{u} \) and root-mean-square fluctuation \( u_{rms} \) components do not change. Sampling rates are typically 4–30 Hz with a validity rate of 15–20 percent, so that it takes approximately 0.5 to 4 minutes to acquire each data point. For each Reynolds number, profiles are measured at nine axial locations in the entrance, groove, and recovery regions \( 0.8 \leq \chi/D_0 \leq 45.07 \), well before end of the channel at \( \chi/D_0 = 56.7 \). Further information on the experimental apparatus and procedures is described in Chen (1993).

**Wall Shear Stress.**

A method to determine the wall shear stress in the recovery region from pressure and velocity measurements is now developed. A force/moment-flux balance on a recovery-region control volume of \( \Delta x \) with cross-sectional area \( A \) gives

\[
(p_r - p_{wz} \Delta x) A = 2\tau(H + W) \Delta x
\]

\[
= - \int_x \rho u^2 dA + \int_{x+\Delta x} \rho u'^2 dA. \tag{1}
\]

The pressure, wall shear stress, and velocity are now broken into time-averaged and unsteady components, for example, \( u = \bar{u} + u' \). The resulting equation is then averaged over time and the limit is taken as \( \Delta x \to 0 \), giving

\[
f = P + M, \tag{2}
\]

where \( f \) is the Fanning friction factor (dimensionless wall shear stress). In Eq. (2), the local, dimensionless pressure gradient, \( P \), and the local, dimensionless momentum-flux gradient, \( M \), are defined as

\[
P = \frac{D_0}{2pV^2} \left( - \frac{dp}{dx} \right)
\]

\[
M = \frac{D_0}{2} \left( \frac{d}{dx} (s^2 + T^2) \right), \tag{3b}
\]

respectively. In Eq. (3b), the velocity-profile shape factor, \( S \), and the spatial-average of the streamwise turbulence intensity, \( T \), are:

\[
S = \sqrt{\frac{1}{A} \int_A \left( \frac{u}{V} \right)^2 dA}, \tag{4a}
\]

\[
T = \sqrt{\frac{1}{A} \int_A \left( \frac{u_{rms}}{V} \right)^2 dA}. \tag{4b}
\]

The squares of these terms characterize the momentum-flux per unit mass of the steady and unsteady components of the axial velocity.

The term \( P \) can be determined in a straightforward way from pressure measurements. The momentum-flux gradient \( M \) is zero in fully developed flow, but must be evaluated in axially varying situations, such as the recovery region of the present test configuration. In order to evaluate \( M \), we must calculate \( S \) and \( T \) at each axial location and then determine the axial gradient of the sum of their squares, Eq. (3b).

Evaluation of \( S \) at a given axial location requires that the variation of the time-averaged velocity in the \( y,z \)-plane be known. To approximate \( \bar{u}(y,z) \), we make convenient and qualitatively plausible assumptions that (1) the dependence of \( \bar{u} \) on the spatial variables can be separated by factoring, and (2) the spanwise \( z \) variation caused by boundary layers that develop along the channel side walls may be represented by a smooth, power-law profile, i.e.,

\[
\bar{u}(y,z) = \bar{u}(y) \left( \frac{1 - |z|}{W/2} \right)^m. \tag{5}
\]

In this expression, \( \bar{u}(y) \) is the time-averaged velocity profile measured at the center span (\( z = 0 \)). The power \( m \) characterizes the side-wall boundary layer thickness. Its value is found by substituting the assumed velocity profile into the continuity equation, yielding the result, \( m = \frac{U_s}{V} \frac{1}{1 - \frac{1}{n} \int_0^h \left( \frac{\bar{u}(y)}{V} \right)^2 dy} \). \tag{6}

To approximate \( T \), we assume the spanwise variation of \( u_{rms}(y,z) \) decreases as the side walls are approached in the same way that the time-mean velocity decreases. Similar to Eq. (5), we write,

\[
u_{rms}(y,z) = \nu_{rms}(y) \left( 1 - \frac{|z|}{W/2} \right)^m, \tag{7}
\]

where \( \nu_{rms}(y) \) is measured at center span, and the power \( m \) has the same value as that determined from the time-averaged velocity. The resulting approximation for \( T \) is

\[
T = \sqrt{\frac{1}{V} \frac{1}{2} \int_0^h \left( \frac{\nu_{rms}(y)}{V} \right)^2 dy}. \tag{8}
\]

In summary, to evaluate the dimensionless momentum-flux gradient \( M \), first center-span velocity profiles of \( \bar{u}(y) \) and \( \nu_{rms}(y) \) are measured at several axial locations. Equations (6) and (8) are evaluated at each axial location using numerical integration. Finite differencing is then used to evaluate the derivative in Eq. (3b). It is noted that a similar analysis using a trap- eozoid-shaped spanwise velocity profile (instead of the power-law contours of Eqs. (5) and (7)) produces essentially the same results as described in the following section (Chen, 1993).

**Results.**

**Velocity.** A series of center-span axial-velocity profiles for \( Re = 2000 \) and 4000 are shown in Figs. 2(a) and 2(b), respectively. Profiles are presented for six axial locations and are nondimensionalized by the channel mass-averaged velocity, \( V \). Lines are used to connect the average values to show time-mean profiles, \( \bar{u}/V \). Horizontal "error bars" show the magnitude of the local streamwise turbulence intensity, \( u_{rms}/V \), on either side of the time-mean value. The two-standard-deviation uncertainty level (95 percent confidence) of the time-averaged velocity is six percent of \( u_{rms} \).

We first consider the steady component of the velocity profiles. Upstream from the grooved section (\( x/D_0 = -5.33 \)), the mean profiles for both Reynolds numbers are smooth and symmetric about \( y/H = 0.5 \), and their shapes are representative of fully developed laminar and well-mixed transitional flat-channel flows. At \( x/D_0 = 13.85 \), which is above the leading-edge peak of the last groove, the time-averaged velocity profiles are skewed away from the bottom grooved wall since enhanced mixing and
momentum transport from the grooves retards the flow at the bottom of the channel. It is interesting to note that the velocity profiles in the grooved region are very similar for both Reynolds numbers. The profiles in the recovery region (\(x/D_H > 16.4\)) regain symmetry further downstream. For \(Re = 2000\), the profile at the last measurement location exhibits center-channel flatness compared to \(x/D_H = -5.33\), indicating it is still not fully recovered, while that for \(Re = 4000\) appears to be fully recovered. This is the first of several indicators that show that the recovery lengths for this flow decreases with increasing Reynolds number.

The time-averaged velocity profile shape factor, \(S\) (Eq. (4a)), is presented in Fig. 3(a) as a function of location and Reynolds numbers. Since this parameter is based on the square of the velocity, it is a measure of profile peaking in that it is larger for peaked contours, such as laminar, parabolic profiles, than for flatter shapes characteristic of well-mixed flows (we note that \(S = 1\) for slug flow). This is shown in Fig. 3(a) where the entrance-region values are seen to decrease toward unity with increasing Reynolds number. In the grooved region, values of \(S\) tend to collapse to the same low value, indicating the shape of the time-averaged velocity profiles are essentially independent of Reynolds number. In the recovery region downstream from the grooves, the values of \(S\) eventually climb back to their flat channel levels. It is noted that this recovery is very rapid for \(Re > 3000\).

We now return to Fig. 2 so we may consider the local streamwise-turbulence intensity, \(\text{u}_{rms} / \text{V}\). Upstream from the grooves, the level of unsteadiness increases with Reynolds numbers, as expected. Instability in the grooved-section causes the level of mixing to be much larger than that at the entrance. In the downstream recovery region, unsteadiness diminishes rapidly at the channel center height, where the time-averaged velocity profiles are fairly flat. The fluctuations also decay quickly near the walls. In the intermediate, shear layer region, the decay of unsteadiness is much less rapid. A surprising observation may be made regarding unsteadiness in the grooved region. The dimensionless center-span turbulence intensity is actually larger at \(Re = 2000\) than it is at \(Re = 4000\), especially close to the lower wall.

The spatially averaged turbulence intensity, \(T\) (Eq. (4b)), indicates the mean level of unsteadiness in the flow, and its variation with location and Reynolds number is shown in Fig. 3(b). As already mentioned, the entrance flow exhibits mixing, which increases with Reynolds number, with \(T\) ranging from approximately 3 to 15 percent. The grooves serve to increase this level sharply and it appears that the Reynolds number dependence in

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**Fig. 3** Steady- and unsteady-velocity profiles parameters versus location for \(Re = 1500\) to 5000. Vertical lines show the extent of the grooved section. (a) Time-averaged velocity profile shape factor. (b) Spatially averaged turbulence intensity.

**Fig. 4** Dimensionless pressure gradient \(P\) (solid circles) and dimensionless momentum-flux gradient \(M\) (open symbols) versus location at \(Re = 3000\). A vertical line shows the end of the grooved section. Horizontal lines show the value of \(P\) measured in fully grooved and fully flat channels. Error bars show the uncertainty (95 percent confidence level) in \(P\) at representative locations.
Uncoupled Heat/Momentum Transport. Comparing Figs. 4 and 5, we see that the shear stress (and pressure gradient) in the recovery region merges back to its flat channel value much more rapidly than the dimensionless temperature. Recovery coefficients are now defined to compare their recovery behaviors quantitatively. The recovery coefficient for shear stress at a given location, $R_\tau$, is defined as the difference between the Fanning friction factor at that location and the fully developed value in the flatly flat channel $f_\infty$, divided by the maximum difference, $R_\tau = (f(x) - f_\infty)/(f_\infty - f_\infty)$. The recovery coefficient for dimensionless temperature is defined similarly, $R_\theta = (\theta_\infty(x) - \theta(x))/(\theta_\infty(x) - \theta_\infty)$.

The axial variations of the recovery coefficients for $Re = 2000$ and 4000 are shown in Figs. 6(a) and 6(b), respectively. Open diamonds represent $R_\tau$ while the filled circles indicate $R_\theta$. The shear stress drops back to its flat channel value more rapidly than the heat transfer for both Reynolds numbers. Furthermore, the decay lengths for pressure gradient and temperature difference both decrease as the Reynolds number increases. However, the decrease in the pressure gradient recovery length is much more dramatic than that for heat transfer. These results suggest that the heat/momentum transport mechanisms are not coupled in the recovery region. Moreover, over short recovery regions (i.e., five hydraulic diameters), the heat transfer is slightly less than that if the grooves continued, but the pressure drop is significantly lower, especially for $Re > 3000$. This suggests that short flat regions, intermittently placed in a grooved channel, may be useful in reducing the overall pumping power requirement for a given heat transfer level.

Momentum and Heat Transfer. Figure 4 shows the local dimensionless pressure-gradient, $P$, as a function of location for $Re = 3000$ (solid circles). These gradients are the slope of least-squares lines fit to consecutive sets of four pressure measurements. Vertical error bars indicate the two-standard-deviation uncertainty (95 percent confidence level) in $P$ at representative locations. A vertical dashed line indicates the end of the grooved section, and two horizontal lines show the fully developed gradients measured in fully grooved and flat test sections (Greiner et al., 1991a, 1991b). We see that the pressure gradient in the partially grooved test section is high in the grooved region. In the recovery section, its initial dropoff is very rapid, returning most of the way to the flat passage gradient within four hydraulic diameters, and then trailing off more slowly downstream. Examination of the other Reynolds numbers investigated shows a marked decrease in the initial rapid recovery length for Reynolds numbers above $Re = 3000$.

The local dimensionless momentum-flux gradient $M$, calculated from velocity measurements described in the preceding section (Eq. (3b)), is also shown in Fig. 4 (open circles). We see that the momentum-flux gradient is much smaller than the pressure gradient and thus may be effectively neglected in Eq. (2). This observation is true for all Reynolds numbers investigated (Chen, 1993). We therefore conclude that, for the present flow, the pressure gradient is an accurate reflection of the total shear stress, $f \approx P$.

Figure 5 shows a plot of dimensionless temperature difference, $\theta(x)$, versus location for $Re = 3000$ in fully flat, partially grooved, and fully grooved channels (Greiner et al., 1991c). The theoretical line for developing laminar-flow heat transfer between parallel plates with the same thermal boundary conditions (Lundberg et al., 1963) is included in this figure and confirms the measurement technique. The vertical line at $x/D_\infty = 16.4$ indicates the end of the grooved region of the partially grooved channel.

The temperature differences in the fully and partially grooved channels are measurably less than those in the flat passage, indicating enhancement. At the end of the grooved region of the partially grooved channel, the temperature profile departs from that of the fully grooved duct and eventually rejoins the flat passage value after roughly 20 hydraulic diameters. This recovery length is typical for the Reynolds numbers below $Re = 3000$, but it decreases mildly at higher Reynolds numbers.
augmentation of 8 percent compared to the same zone location in developing grooved channel at equal pumping power. This corresponds to a 30 percent pumping power reduction for a given Nusselt number (Chen, 1993).

Conclusions

Earlier work shows that transverse grooves enhance the heat transfer/pumping power performance of exchange passages in the Reynolds number range 1000 < Re < 5000 by stimulating three-dimensional flow instabilities (Greiner et al., 1991a, 1991b). The current work demonstrates that the resulting unsteady flow persists for a considerable distance downstream from the grooved region in that the time-averaged and unsteady velocity profiles, as well as the heat transfer coefficient, are dramatically effected for up to 20 hydraulic diameters past the end of the grooved section. The recovery lengths for shear stress and pressure gradient drop off rapidly as the Reynolds number increases beyond Re = 3000, while the decrease in the heat transfer decay length is much less rapid, leading to a net heat transfer/ pumping power performance advantage relative to flow in a fully grooved channel. The difference between the recovery lengths for shear stress and heat transfer indicate that heat/momentum transport mechanisms in this region are uncoupled.

The augmented performance observed in the recovery region of the present study suggests that intermittently grooved passages, in which unsteady flow alternately develops in grooved sections and decays in flat regions, would yield even higher heat transfer/pumping power performance than continuously grooved passages. However, further information on the development of destabilized flow in short grooved channels is necessary to determine an optimal design of these passages.

Acknowledgments

This work was sponsored by National Science Foundation under grant number CBT-8708802 and Gas Research Institute under contract number 5087-260-1562.

References


