DECAY OF SUPERCRITICAL-FLOW HEAT TRANSFER
ENHANCEMENT DOWNSTREAM FROM A GROOVED SURFACE

M. Greiner, R.-F. Chen, and R. A. Wirtz
Department of Mechanical Engineering
University of Nevada
Reno, Nevada

ABSTRACT

Recent experiments show that cutting transverse-grooves into a channel can enhance its heat transfer/pumping power performance by stimulating flow instabilities. In the current experiment, a fully-developed air flow in a rectangular channel encounters an isolated sawtooth section with thirteen, V-shaped grooves cut into one wall. Detailed local heat transfer and pressure gradient measurements are made in the grooved section and in the flat-walled recovery passage downstream to determine how far heat transfer enhancement extends downstream once the grooves are removed. These measurements are compared to the Nusselt numbers and friction factors in ungrooved and very long grooved channels.

The local heat transfer coefficients and pressure gradients drop off from the grooved channel values at different rates in the downstream recovery passage. In the Reynolds number range 1500 ≤ Re ≤ 3000, the pressure gradient relaxes to within 10% of the flat channel value in roughly ten hydraulic diameters, while the heat transfer decay length is approximately twice that distance. These results suggest that intermittently grooved channels represent an overall improvement in the heat transfer/pumping power performance of exchange passages.

NOMENCLATURE

α Groove length (Fig. 1).

b Groove depth (Fig. 1).

Dh Minimum channel hydraulic diameter, 2HW/(H + W).

f Fanning friction factor, −(dρ/dx)/(Dh/2ρV^2).

h(x) Local heat transfer coefficient, 1/υ(x).

H Minimum channel height (Fig. 1).

k Fluid thermal conductivity.

L Test section length.

L_e Entry section length.

L_g Grooved section length.

Nu Spatially averaged Nusselt number, q^−Dh/ΔT_k.

Δp Pressure drop.

Pr Fluid Prandtl number.

q^− Heat flux.

Q Fluid volume flow rate.

r(x) Local thermal resistance, ΔT/q^−.

R_f(x) Recovery coefficient for friction, Eqn. 2.

R_s(x) Recovery coefficient for dimensionless temperature, Eqn. 3.

Re Reynolds number, V Dh/υ.

R_e Critical Reynolds number.

To Fluid entrance temperature.

T_f(x) Local flat surface temperature.

ΔT(x) Local temperature difference between the flat surface and the inlet, T_e(x) − T_o.

ΔT̄ Temperature difference ΔT averaged over some distance.

V Mass average velocity based on minimum channel cross section, Q/WH.

ω Channel width.

x Axial coordinate.

Δx Distance downstream from the grooved section exit, x − L_g.

Δx_f Ninety percent decay length for friction factor.

Δx_e Ninety percent decay length for dimensionless temperature.

Greek

ν Kinematic viscosity.

ρ Fluid density.

θ(x) Local dimensionless upper wall temperature, r(x)k/Dh.

θ_f(x) Local dimensionless upper wall temperature in ungrooved duct.

θ_g(x) Local dimensionless upper wall temperature in fully-grooved duct.
INTRODUCTION

Recent numerical calculations and experiments have shown that cutting transverse grooves into the surfaces of parallel-walled passages causes unsteady flow to appear above a critical Reynolds number which is considerably below that normally associated with transition [Chadha et al., 1986; Karniadakis et al., 1988; Kozlu et al., 1988]. For certain groove configurations, this critical value is one-eighth the Reynolds number normally associated with the transition to turbulence [Greiner et al., 1988]. Visualizations of these supercritical flows show that unsteadiness appears within a few hydraulic diameters of the first groove [Greiner et al., 1990]. Measurements of the fully-developed heat transfer and pressure gradient characteristics indicate that heat transfer is enhanced by as much as a factor of 4.6 at equal Reynolds numbers and 3.5 at equal pumping powers relative to flat passages [Greiner et al., 1991a, 1991b].

In the current work, we experimentally study how far heat transfer enhancement extends downstream once the grooves are removed. The specific aim is to determine the distance for the pressure gradient and heat transfer to return to the flat channel values, and the average heat transfer/pumping power characteristics of different lengths of the downstream recovery section.

We consider the partially-grooved channel shown in Fig. 1. A fully-developed air flow from a rectangular channel enters a section with thirteen right-triangular grooves cut into the lower wall (length \(a\), depth \(b\)), and then passes into smooth-walled recovery section. The entrance channel is adiabatic. Starting at the grooved section inlet (\(x = 0\)), the lower wall temperature is held constant at the inlet value \(T\) (\(T_p\)), while the upper wall develops a thermal boundary layer by dissipating a uniform heat flux, \(q^*\). Experiments are performed for the Reynolds number range \(1500 \leq Re \leq 5000\), where \(Re = V D_a / \nu\) is based on the mass averaged velocity and hydraulic diameter of the minimum channel cross-section.

This geometry is a modification of a "fully-grooved" passage studied in [Greiner et al., 1991a; 1991b]. In that work, a passage with forty-six cavities is examined, and the essential change is that, in the present passage, the last thirty-three grooves are replaced by smooth-walled recovery section. The results of the previous work show that the development of the velocity and temperature fields are essentially complete by the fifth groove for Reynolds numbers greater than \(Re = 1000\). This suggests that the conditions at the exit of the grooved section of the current experiment are fully-developed. Our general experimental protocol is to measure local heat transfer and pressure drop from the upper surface in the grooved and recovery sections of the channel for a range of Reynolds numbers. These results are compared to local data measured in fully-grooved and ungrooved passages.

The test section is 1220 mm long and 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 the thirteen cavities of the grooved surface (\(a = 24\) mm, \(b = 12\) mm, Fig. 1), and a flat aluminum plate downstream. The minimum channel spacing in both regions is \(H = 10\) mm making the hydraulic diameter \(D_h = 19\) mm, and the dimensionless grooved section length \(L_g/D_h = 16.4\). The test section is designed so that two different Plexiglas upper surfaces may be installed; one is for pressure drop measurements, the other is for heat transfer.

Pressure measurements are made to determine how the gradient, \(dp/dx\), depends on location and Reynolds number. The pressure surface has twenty-four 2 mm diameter taps along its centerline starting at \(x = 0\). The tap spacing is 24 mm in the grooved section and the region just downstream (0 \(\leq x < D_a \leq 21.4\)) and 102 mm apart further downstream. The pressure difference between the first and the other twenty-three taps is measured with an electronic pressure transducer (\(\pm 0.5\%\)) which is plumbed using a switching valve.

Temperature measurements are performed along the upper surface under conditions of uniform heat flux to determine the dependence of the coefficient of thermal resistance, \(r(\alpha)\), on channel location and Reynolds number. The resistance is defined as \(r(\alpha) = 1 / h(\alpha) = \Delta T(\alpha)/q^*\), where \(h(\alpha)\) is the local heat transfer coefficient, and \(\Delta T(\alpha) = T_s(\alpha) - T_a\) is the local difference between the surface and inlet temperatures. The heat transfer surface 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 sub-circuits are wired in parallel to a regulated DC power supply. The heat flux passing to the fluid from each plate is monitored by a 102 mm square thermopile heat flux gage (accuracy \(\pm 1\%) located at its center. During an experimental run, the trimming rheostats are adjusted so that the gage indicated heat flux is the same. The correction for radiation heat flux is negligibly small since the opposite wall is polished to a high gloss.

![Fig. 1. Partially-grooved channel geometry and thermal boundary conditions.](image)

**Fig. 1.** Partially-grooved channel geometry and thermal boundary conditions.

**Fig. 2.** Schematic of the experimental wind tunnel facility.

EXPERIMENTAL APPARATUS

A schematic of the open-loop wind tunnel used in this investigation is shown in Fig. 2. Laboratory air at approximately 26°C and 8.7 \times 10^4 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 \(b = 12\) mm (normal to the plane of the figure) is \(W = 203\) mm, and length is \(L_a = 1220\) mm. 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 plenum, flows through calibrated rotameters, and is drawn into a variable speed blower.
The heater plate assemblies contain copper-constantan thermocouple junctions located 0.28 mm beneath the wetted surface at eighteen points along the channel centerline. These thermocouples are referenced to a rake of three junctions located in the flow development section, at \( x = -150 \text{ mm} \). 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 difference between the thermocouple junction and the wetted surface.

**RESULTS**

Figures 3a and 3b are plots of the pressure drop between the inlet of the grooved section and twenty-three locations downstream for the highest and lowest Reynolds numbers studied, \( Re = 1500 \) and \( 5000 \), respectively. Included in these figures are lines indicating the location of the end of the grooved section, \( x/D_h = 16.4 \). We see that in both cases the slope increases abruptly from the flat channel value in the region near \( x = 0 \), reaches a constant gradient in the grooved section, drops off near the end of the last groove, and again reaches a constant but lower value in the recovery channel.

The local Fanning friction factor,

\[
f(x) = \frac{D_h}{2 \rho V^2} \frac{dP}{dx}(x)
\]

may be calculated by determining the slope of this data as a function of location, \( \frac{dP}{dx}(x) \). The gradients in the grooved and recovery sections are roughly constant in the domains, \( 7.6 \leq x/D_h \leq 13.8 \) and \( 26.8 \leq x/D_h \leq 55 \), respectively. Least-squares lines are fit to the data in these regimes to average out the effects of random errors. These lines are included in Figs. 3a and 3b. Comparing these figures, we see that the data for \( Re = 1500 \) follows the grooved section slope further downstream from that section's end (departing at \( x/D_h = 20 \)) than does the data for \( Re = 5000 \) (departing at \( x/D_h = 17 \)). It appears that the length of the downstream domain in which the grooves influence the pressure gradient decreases with Reynolds number. The dotted spline fit for the data in the region \( 13.8 \leq x/D_h \leq 32.2 \) show that the intermediate points do not follow a simple relation. To find the local gradient in this region we use a finite difference technique. While this method is strongly affected by random measurement errors, it appears to be the best technique available since no theoretically based relation is currently known to fit the data.

The friction factor for \( Re = 3000 \) is shown as a function of dimensionless location in Fig. 4. Solid lines indicate the friction factors based on the least-squared lines in the regions in which they are fit to the data, and triangles denote the values from the finite differencing technique. Dashed horizontal lines indicate the friction factors presented in [Greiner et al. 1991a] for ungrooved and fully-grooved channels. For the Reynolds number range investigated, the friction factors measured in the recovery section of the partially-grooved channel are scattered within 9% of the data for ungrooved ducts. The friction factors in the grooved section, however, are consistently lower than those reported for the fully-grooved passage, by roughly twenty percent. This difference may be due to the elliptic nature of the problem, but is not fully understood at this time.

The local finite difference values increase at the grooved channel inlet \( (0 \leq x/D_h \leq 7) \), vary randomly about the value determined by the least-squared fit of the present data, and rather quickly drop off in the recovery section. Error bars indicate the uncertainty caused by the pressure drop measurement. At this Reynolds number, we see that the pressure gradient has essentially returned to the ungrooved channel value within ten hydraulic diameters of the start of the recovery section.
We now turn to measurement of the local heat transfer along the upper surface. A plot of dimensionless temperature \( \theta(x) = r(x)/k/D_h \), where \( k \) is the fluid thermal conductivity, versus location for \( Re = 3000 \) is shown in Fig. 5. Included in this figure is the theoretical line for the development of the wall temperature for flow between two parallel plates with the same thermal boundary conditions [Lundberg et al. 1963], and symbols for measurements in the channels with flat, partially-grooved and fully-grooved bottom walls. The agreement between the theoretical and measured ungrooved passage results confirm the accuracy of the measurement technique.

All three of the measured profiles are the same near the channel inlet. The dimensionless temperature of the ungrooved channel increases downstream. The grooved channel temperatures, however, depart from those in the ungrooved duct as the influence of the bottom wall cavities reaches the upper surface. The grooved channel temperature profiles become fully-developed (roughly constant) in less than seven hydraulic diameters (five grooves) from the first groove. At the end of the grooved section, the partially-grooved channel temperature profile slowly departs from that of the grooved duct and eventually rejoins the ungrooved passage curve. The upper wall temperature is seen to be lower than the ungrooved channel profile for roughly twenty hydraulic diameters beyond the end of the groove section.

As quantitative indicators of the relaxation of the friction factor and heat transfer coefficient to the ungrooved channel values, we define the local recovery coefficients for friction and dimensionless temperature, respectively, as

\[
R_f(x) = \frac{f_f(x) - f_f}{f_f - f_f} \quad (2)
\]

and

\[
R_\theta(x) = \frac{\theta_f(x) - \theta_f}{\theta_f(x) - \theta_f(x)} \quad (3)
\]

In these expressions, \( f_f \) and \( f_f \) are the friction factors measured in the recovery and grooved portions of the partially-grooved channel, respectively, and \( \theta_f(x) \) and \( \theta_f(x) \) are the local dimensionless temperatures measured in the ungrooved and fully-grooved channels, respectively. These coefficients are unity where the partially-grooved values are the same as the grooved values, and zero where they equal the values in the ungrooved channel.

The friction factor and dimensionless temperature recovery factors are both plotted against location for \( Re = 3000 \) in Fig. 6. We see that the friction factor drops off to the flat channel value far more rapidly than does heat transfer. As a measure of the friction factor and heat transfer decay lengths, we define \( \Delta x_f \) as the distance between the end of the grooved portion \( (x/D_h = 16.4) \) and the location where the friction recovery factor is \( R_f = 0.1 \), and \( \Delta x_\theta \) as the distance for \( R_\theta = 0.1 \). The intersection of the interpolated lines in Fig. 6 with the horizontal line at \( R_f = R_\theta = 0.1 \) shows that for \( Re = 3000, \Delta x_f/D_h = 8.6 \) while \( \Delta x_\theta/D_h = 19.5 \).

Figure 7 shows the dependence of these decay lengths on Reynolds numbers. The figure shows that the heat transfer enhancement length is more than twice that for the friction factor for the full Reynolds number range investigated. While these lengths are roughly constant below \( Re = 3000 \), they both decrease for higher Reynolds numbers above that value.

Earlier studies show that fully-developed grooved channels enhance heat transfer for a given pumping power.
Fig. 8. Nusselt number versus dimensionless pumping power for a fully-developed flat channel, fully-developed grooved channel, and different recovery section lengths in an intermittently grooved channel.

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References


Conclusions

The heat transfer and pressure gradient in a flat channel downstream from a grooved duct drop off from the grooved channel values at different rates. In the Reynolds number range 1500 \( \leq Re \leq 3000 \), the pressure gradient returns to within ten percent of the flat channel value in roughly ten hydraulic diameters, while the heat transfer decay length is approximately twice that distance.

We can conclude that intermittent grooving improves the heat transfer/pumping power performance of grooved channels. However, further information on the development of supercritical flow in short grooved channels is necessary to determine the performance and optimal design of these passages.