An Experimental Investigation of Structured Roughness Effect on Heat Transfer During Single-Phase Liquid Flow at Microscale

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Abstract

The effect of structured roughness on the heat transfer of water flowing through minichannels was experimentally investigated in this study. The test channels were formed by two 12.7 mm wide × 94.6 mm long stainless steel strips. Eight structured roughness elements were generated using a wire electrical discharge machining (EDM) process as lateral grooves of sinusoidal profile on the channel walls. The height of the roughness structures ranged from 18 μm to 96 μm, and the pitch was varied from 250 μm to 400 μm. The hydraulic diameter of the rectangular flow channels ranged from 0.71 mm to 1.87 mm, while the constricted hydraulic diameter (obtained by using the narrowest flow gap) ranged from 0.68 mm to 1.76 mm. After accounting for heat losses from the edges and end sections, the heat transfer coefficient for smooth channels was found to be in good agreement with the conventional correlations in the laminar entry region as well as in the laminar fully developed region. All roughness elements were found to enhance the heat transfer. In the ranges of parameters tested, the roughness element pitch was found to have almost no effect, while the heat transfer coefficient was significantly enhanced by increasing the roughness element height. An earlier transition from laminar to turbulent flow was observed with increasing relative roughness (ratio of roughness height to hydraulic diameter). For the roughness element designated as B-1 with a pitch of 250 μm, roughness height of 96 μm and a constricted hydraulic diameter of 690 μm, a maximum heat transfer enhancement of 377% was obtained, while the corresponding friction factor increase was 371% in the laminar fully developed region. Comparing different enhancement techniques reported in the literature, the highest roughness element tested in the present work resulted in the highest thermal performance factor, defined as the ratio of heat transfer enhancement factor (over smooth channels) and the corresponding friction enhancement factor to the power 1/3. [DOI: 10.1115/1.4006844]

Keywords: roughness, microscale heat transfer, structured roughness, minichannels

Introduction

Heat transfer in microscale passages is of great interest due to their application in micro heat exchangers, fuel cells, biomedical devices, etc. A number of earlier investigations reported a significant departure from conventional heat transfer predictions [1–6], while the more recent literature confirmed the validity of conventional theory to heat transfer during laminar liquid flow at microscale [7–10]. Errors in data reduction, large experimental uncertainty, and incorrect boundary conditions were found to be the major sources for discrepancies in earlier investigations [11–16]. Some of the experimental issues were discussed in the literature; e.g., Celata et al. [17] revealed the effect of inaccurate measurement of tube diameters on friction factors. Using a 40× microscope, the diameter was measured as 84.7 μm, and the resulting friction factor was significantly higher than the prediction. While using a 400× microscope, the diameter of the same tube was found to be 80.0 μm and the data were predicted very well by the conventional correlation. Li [18] indicated that Wu and Little [1] did not measure the wall roughness directly, but obtained the value of equivalent sand roughness from their f−Re−Re chart. Mala and Li [19] relied on the manufacturer data for the values of relative roughness and used the concept of roughness-viscosity to correlate their data. Herwig and Hausner [20] indicated that axial heat conduction was the major reason for discrepancy in the experimental data by Tso and Mahulikar [21].

Many experimental data reported in the literature indicate a lower value of fully developed the Nusselt number in the laminar flow, with a decreasing trend at lower Reynolds numbers [2,3,5,6,22–24]. The reason for lower heat transfer coefficients as compared to the conventional correlation could be attributed to the axial conduction effect. Lin and Kandlikar [25] proposed a model to quantitatively estimate the axial conduction effect. From their model, the axial conduction effect is not negligible for fluid flow in channels with the following characteristics: large wall thickness, small diameter, high wall thermal conductivity, low fluid conductivity, and low Re. On the other hand, some data exhibited a higher heat transfer coefficient, which can be mainly attributed to the roughness effect [1,10,26–30].

Heat transfer enhancement in both laminar and turbulent flows has been an area of great interest for a long time [31]. In macroscale laminar flow, some of the enhancement techniques receiving renewed interest are: twisted tubes [32,33], porous materials [34], and coiled wires [35]. In turbulent flow, enhancement structures on channel internal walls were investigated and found to enhance heat transfer. There is very limited data available analyzing...
systematic effect of enhancement techniques on heat transfer and pressure drop at microscale.

Friction factors in miniscale and microscale rough channels were systematically investigated by Kandlikar [36], Brackbill and Kandlikar [37,38], and Kandlikar et al. [39]. The effects of the roughness element height, pitch, and relative roughness on friction factor were studied experimentally. A model was proposed to predict the early transition from laminar to turbulent flow due to roughness effects [38,39]. However, there are very few studies in the literature that systematically investigate the effect of roughness on heat transfer at microscale [27].

The present work is aimed at systematically investigating the effects of structured roughness elements on heat transfer at microscale. The roughness parameters investigated include: roughness element pitch, \( \lambda \), height, \( H \), and the relative roughness \( H/D \).

**Experimental Setup**

Figure 1 shows a schematic of the test system. Degassed, distilled water from a reservoir is circulated through the system by a micropump as shown in Fig. 1(a). The water then flows through a bank of flow meters, which consists of two flow meters in parallel with working ranges of 10–100 ml/min and 60–1000 ml/min. The water flows through the test section and then enters a heat exchanger cooled by a chiller. The reservoir is maintained at atmospheric pressure, and the exit from the test section is slightly above the atmospheric pressure. Test section assembly is shown in Fig. 1(b). The two stainless steel test pieces with desired roughness profiles are assembled in the test section to form the flow channel. There is a tapered inlet section to allow for smooth entry into the channel. The channel exit is similar to the inlet and has the same header configuration. Further details of the test section assembly are given in an earlier publication by Kandlikar and Wagner [40]. Silicone film heaters with adhesive backs were applied on the back of the stainless steel test pieces for heating the channel walls. All interfaces between the microchannel assembly and the test fixture were sealed with an adhesive-backed silicon gasket.

The channel wall temperatures were measured by inserting the thermocouples at a fixed distance below the channel inner surface. The eleven thermocouples are located at distances of 16.2 mm, 24.4 mm, 32.7 mm, 40.9 mm, 49.1 mm, 57.3 mm, 65.6 mm, 73.8 mm, 82.0 mm, 90.2 mm, and 98.5 mm from the inlet section.

As described earlier, the flow channel was formed by two thin stainless steel sections. The advantages of using stainless steel walls are: (i) reasonably high thermal conductivity to conduct heat from the heaters to the inside channel wall, (ii) high strength to withstand high compression forces required to prevent leakage, and (iii) relatively low thermal conductivity to avoid severe axial heat conduction effects.

The structured roughness elements on the stainless steel walls were generated by using a wire EDM process. The workpiece traverse was controlled with a computer numerical control machine to generate the desired roughness profile. The pitch to height ratio of the structured roughness elements has been identified as a key parameter affecting fluid flow and heat transfer [40]. In this study, the sinusoidal roughness surfaces were employed with pitch to height ratios ranging from 2.6 to 13.9. The actual fabricated surface profiles were obtained from a confocal laser scanning Microscope. The surface profile of the roughness element and a cross-sectional view are shown in Fig. 2. Table 1 lists the details of the surface geometrical parameters investigated in this work—roughness element pitch \( \lambda \), height \( H \), \( \lambda/H \) ratio, gap \( b \), constricted gap \( \text{gap}_{b,ct} \), hydraulic diameter, \( D_h \), constricted hydraulic diameter \( D_{h,ct} \), and the ratios \( H/b \), \( H/b_{ct} \), \( H/D_h \), and \( H/D_{h,ct} \). The constricted parameters are listed as they have been identified by Kandlikar et al. [39] to be relevant in evaluating the roughness effects on friction factor at microscale.

The roughness surface profile has a sinusoidal shape that can be described by the following equation:

\[
y(x) = H \left[ \cos \left( \frac{\pi x}{\lambda} \right) \right]^P \tag{1}
\]

The power of the cosine function, \( P \), controls the slope of the profile near the peaks. The values of \( P \) for channels B, C, D, and E in Table 1 are 4, 12, 32, and 4, respectively.

**Effect of Axial Conduction.** The effect of axial heat conduction was investigated by Lin and Kandlikar [25], Maranzana et al. [41] and Guo and Li [12]. Figure 3 shows a schematic of a tube section showing the axial conduction effects on the temperature profile in the fully developed flow region. Under constant heat
Fig. 3 A schematic illustrating axial conduction effect on fluid temperature distribution

flux heating boundary condition, the wall temperature is higher in the flow downstream and hence heat is transferred from the downstream section to the upstream by conduction in the wall. Without considering axial conduction, the heat transfer rate to the fluid in the region up to the section under consideration is \( q_{\text{conv}} = q' A_h \), while with axial conduction the heat transfer rate is \( q_{\text{conv}} = q' A_h + q_{\text{cond}} \), where \( A_h \) and \( q_{\text{cond}} \) are the convective heat transfer area and the axial conduction heat transfer rate, respectively. Fluid temperature \( T_f \) at a given section increases further due to the axial conduction effect as the heat conducted upstream in the wall enters the fluid prior to the section.

The ratio of axial conduction to convective heat transfer rates, \( q_{\text{cond}}/q_{\text{conv}} \), was defined as cond and \( M \) parameters by Guo et al. [12,41] and Maranzana et al. [12,41], respectively. The axial conduction effects are negligible when \( M \) is less than 1% [41]. Based on the test section material and configuration of the present study, for \( Re > 70 \), the \( M \) parameter is less than 1% and the axial conduction effects are negligible. In a recent work, Lin and Kandlikar [25] proposed a model to estimate the axial conduction effect given by

\[
\frac{Nu_{00}}{Nu_h} = \frac{1}{1 + \frac{k_s A_{hs}}{A_f} \frac{Nu_h}{Nu_00} (Re Pr)^{\frac{1}{2}}}
\]

where \( Nu_h \) is the theoretical \( Nu \), and \( Nu_{00} \) is without considering axial conduction, and \( k_s \) and \( k_f \) are the thermal conductivity of the channel wall and the fluid respectively. \( A_{hs} \) is the cross-sectional area of the channel wall, and \( A_f \) is the cross-sectional area for the fluid flow in the channel. A value of \( Nu_{00}/Nu_h \) equal to 1 indicates that the axial conduction has no effect. The lower the \( Nu_{00}/Nu_h \) value, the higher is the axial conduction effect. In the present study \( k_s = 14.9 \text{ W/m°C} \), \( k_f = 0.62 \text{ W/m°C} \), \( A_{hs} = 102 \text{ mm}^2 \), \( A_f = 6.35 \text{ mm}^2 \), \( Nu_h = 8 \), and \( Pr = 5.6 \). Using Eq. (2), \( Nu_{00}/Nu_h \) is estimated to be around 0.99 for \( Re \) higher than 300. The experimental values of \( Re \) employed in this study are higher than 300, and the effects of axial heat conduction are therefore negligible.

### Data Reduction

The test section was heated with electric heaters to provide a constant heat flux boundary condition. In order to calculate the local heat transfer coefficient, the wall temperature at 11 locations, and the inlet and outlet fluid temperatures were measured. Local fluid temperature was calculated from the inlet and outlet fluid temperatures given by

\[
T_{f,x} = \left( T_{i,o} - T_{i,x} \right) \frac{L_{h,s}}{L_{h,All}}
\]

where \( L_{h,s} \) is the distance from the inlet section and \( L_{h,All} \) is the total heated length. Local fluid temperatures are then used to calculate the local Prandtl number, viscosity, and thermal conductivity. The heat supplied to the fluid is then given by

\[
q_{\text{conv}} = \dot{m} C_p \left( T_{i,o} - T_{i,x} \right)
\]

where \( \dot{m} \), \( C_p \), \( T_{i,o} \), and \( T_{i,x} \) are mass flow rate, heat capacity, and outlet and inlet fluid temperatures, respectively. The total measured input power, \( q_{\text{in}} \), is obtained from the voltage and current from the power supply

\[
q_{\text{in}} = I \times V
\]

where \( I \) and \( V \) are current and voltage, respectively. A separate experiment was performed to calculate the heat loss in each individual location. The average heat flux is calculated from the following equation:

\[
q''_{\text{avg}} = \frac{q_{\text{in}} - q_{\text{loss}}}{A_h}
\]

where \( q_{\text{loss}} \) is the heat loss from the channel to the surrounding. Depending on the location, the maximum heat loss in any section was found to be 6% of the heat input in the experiments. \( A_h \) is the heat transfer area of the channel. The local heat transfer coefficient is then calculated as follows:

\[
h_x = \frac{q''_{\text{avg}}}{T_{w,x} - T_{i,x}}
\]

where \( T_{w,x} \) and \( T_{i,x} \) are the local wall and fluid temperatures. \( T_{w,x} \) is derived from one dimensional Fourier’s heat conduction equation knowing the heat flux, wall material thermal conductivity, the temperature measured by the thermocouple, and the distance of the thermocouple from the inside wall surface.

The local Nusselt number \( Nu_x \) is calculated as

\[
Nu_x = \frac{h_x D_h}{k_{i,x}}
\]

where the hydraulic diameter \( D_h \) is calculated as follows:

\[
D_h = \frac{4 a b}{2 (a + b)}
\]

where \( a \) is the width of the channel and \( b \) is the gap between the two opposing stainless steel test pieces. The constricted hydraulic diameter is calculated by using the constricted gap (the narrowest gap between the opposing roughness elements in a rough channel) \( b_{c} \) in place of \( b \) in Eq. (9).

### Uncertainty Analysis

To assess the accuracy of the measurement, an uncertainty analysis was performed. The uncertainty in the calculated parameters...
such as $\text{Nu}$, $\text{Re}$, $G$, and $f$ is generally denoted as $\delta y$. It is a function of variables $x_1, x_2, \ldots, x_n$. The uncertainty in a parameter is determined as described as follows:

$$y = f(x_1, x_2, \ldots, x_n),$$

$$\delta y = \left[ \sum_{i=1}^{n} \left( \frac{\partial y}{\partial x_i} \delta x_i \right)^2 \right]^{1/2}$$

(10a)

where $\delta x_1, \delta x_2, \ldots, \delta x_n$ are the uncertainties in the independent variables.

The uncertainties of experimental measurement devises and parameters are listed in Table 2. The resulting uncertainty in $\text{Nu}$ is 7%–27% in the present study.

### Results

#### Experimental Data Validation With Smooth Channel

In order to accurately estimate the heat loss, a separate controlled experiment was conducted. The channel plate was heated by a power supply without any fluid flow. The heat supplied was lost by conduction through both the assembly parts and the insulation by natural convection to the surroundings. It was noted that the heat losses were occurring mainly from the end regions (inlet and outlet sections), while the central region was well insulated and contributed very little to heat losses. It was therefore decided to use the central one third region of the test section for data analysis after accounting for the heat losses.

Figure 4 shows the calculated $\text{Nu}$ in smooth channel A-1 plotted as a function of $1/\text{Gz}$ for the data taken only in the central region. The reciprocal Graetz number $1/\text{Gz}$ is defined as $1/\text{Gz} = (L_{ch}/D_h)/(\text{Re Pr})$. The following correlation validated by Harms et al. [42] for thermally developing flow in a smooth rectangular channel flow for different aspect ratio $z$ is also plotted to compare with the present data.

$$\text{Nu} = 8.24 - 16.8z + 25.4z^2 - 20.4z^3 + 8.7z^4, \quad 1/\text{Gz} \geq 0.1$$

(10b)

$$\text{Nu} = 3.35(1/\text{Gz})^{-0.130} z^{-0.120} \text{Pr}^{-0.038}, \quad 0.013 \leq 1/\text{Gz} < 0.1$$

(10c)

\[
\text{Nu} = 1.87(1/\text{Gz})^{-0.300} z^{-0.056} \text{Pr}^{-0.036}, \quad 0.005 \leq 1/\text{Gz} < 0.013
\]

(10d)

For the laminar fully developed flow in a channel of aspect ratio $z$, the Nusselt number is given by

$$\text{Nu} = 8.235(1 - 2.0421z + 3.0853z^2 - 2.4765z^3 + 1.0578z^4 - 0.1861z^5)$$

(11)

It is seen from Fig. 4 that the conventional correlation for smooth channel are able to predict the data well. However, in the developing region, the discrepancies are higher than the uncertainty limits. One of the factors causing contributing to the larger error may be due to the fact that the conventional correlations for laminar developing flow in rectangular channels with high aspect ratios are not as reliable as that for circular tubes, and the implementation of correct boundary condition at the walls poses additional uncertainties. Further study on the high aspect ratio channels in the entrance region is warranted.

#### Roughness Effects on Laminar Flow Heat Transfer

Water flow in rough channels was tested to investigate the effects of roughness on heat transfer. The effects on pressure drop were reported in a separate publication by Wagner and Kandlikar [40].

The surface roughness element structure details are given in Table 1. Channels B-1, B-2, C-1, C-2, E-1, and E-2 have the same roughness element pitch $\lambda = 250 \mu m$, and channels C-1, C-2, D-1, and D-2 have similar roughness element height $H$ of about 35 $\mu m$. Each roughness element is tested with two different gaps (channel heights) resulting in two different hydraulic diameters.

Figure 5 shows the experimental $\text{Nu}$ of rough channel D-2 normalized by laminar developing flow correlation $\text{Nu}_{\text{th, plain}}$ as a function of $\text{Re}$. $\text{Nu}_{\text{th, plain}}$ is the theoretical $\text{Nu}$ derived from plain channel equation, Eq. (10). Roughness element D-2 has $\lambda = 400 \mu m$, $H = 37.7 \mu m$, and a hydraulic diameter of 1.76 mm ratios as are in Table 1. The $\text{Nu}$ data for D-2 are significantly above the prediction from the conventional developing flow correlation for a smooth channel. The higher performance of D-2 surface is clearly attributable to the enhancement resulting from the roughness structures over the plain surface. In developing flow, $\text{Nu}$ is a function of $\text{Re}$ and increases as $\text{Re}$ increases. In Fig. 5, $\text{Nu}/\text{Nu}_{\text{th, plain}}$ is independent of $\text{Re}$ which implies that the slopes of the experimental data and the theoretical values are identical. As mentioned in Experimental Data Validation With Smooth Channel Section, the data were taken in the central region of the channel. The heating length from the flow entrance location $L_{ch}$ is about 50 mm, while the thermal entry region length is about 330 mm at $\text{Re} = 1000$. The experimental data and the theoretical

\[
\text{Nu} = 1.87(1/\text{Gz})^{-0.300} z^{-0.056} \text{Pr}^{-0.036}, \quad 0.005 \leq 1/\text{Gz} < 0.013
\]
values both depend on Re, while their ratio Nu/Nuth,plain is seen to be independent of Re. Since there are no correlations available for heat transfer in the entry region of rough channels, more detailed conclusions cannot be drawn from Fig. 5 at this time.

Amon and Mikic [43] numerically investigated flow patterns and heat transfer in slotted channels. Their numerical work indicated that recirculating vortices are generated behind the slot. These vortices are detrimental to heat transfer, but under certain conditions, an oscillatory separated flow was formed resulting in heat transfer enhancement. Dhariiya and Kandlikar [44] numerically investigated the effect of roughness on heat transfer for water flow in rough channels with the same roughness element structure and channel geometries used in the present study. The roughness elements can be considered as a series of slots in the flow direction. They found that there are no recirculating vortices generated over the pitch of the roughness elements because of the smooth profile.

Effect of Roughness Element Pitch on Heat Transfer. To study the effect of pitch λ, heat transfer data of C-2 and D-2 are plotted in Fig. 6 for comparison. Both surfaces have similar roughness elements with H ~ 35 μm and Dk ~ 1.7 mm, but with different pitches of λ = 250 μm for C-2 and λ = 400 μm for D-2. The λ/H ratios for C-2 and D-2 are 7.8 and 10.6, respectively. Since the aspect ratios of the two channels are the same, Nu/Nuth,plain is the same for the two data sets. As shown in the Fig. 6, Nu/Nuth,plain is significantly higher than 1 for both data sets. By comparing the two data sets, it is found that there is no significant effect of pitch on heat transfer for the two roughness elements.

The effect of roughness on heat transfer under fully developed flow conditions is shown in Fig. 7. All the test channels have similar roughness element height (H = 32.0 μm for Channel C-1, C-2 and H = 37.7 μm for Channel D-1, D-2). The experimental Nusselt number values are normalized by the theoretical Nusselt number for fully developed laminar flow in a smooth channel. The ratio Nu/Nuth,plain is plotted as a function of λ. Data sets with the same roughness structure but different hydraulic diameters (smaller diameter C-1 and larger diameter D-1 and D-2) are shown. The rough channels show enhanced performance over a smooth channel, but the effect of λ is seen to be insignificant in the fully developed region as well.

Effect of Roughness Element Height on Heat Transfer. Figure 8 shows the effect of roughness element height on heat transfer coefficient. The experimental Nu normalized by the theoretical laminar fully developed flow in a plain channel, Nu/Nuth,plain is plotted as a function of roughness element height H. The data sets are for a roughness element pitch λ = 250 μm with two different hydraulic diameters. Channel B-1, C-1, and E-1 are attributed as small diameter channels while channels B-2, C-2, and E-2 are attributed as large diameter channels. Both data sets exhibit an increase in heat transfer with the roughness element height. However, for H = 96.3 μm it is observed that the enhancement for the large diameter channels was significantly lower than that for small diameter channels. The heat transfer coefficient is thus seen to be dependent on the relative roughness H/Dk and increases with it. Hence with the same H = 96.3 μm, the enhancement for large diameter channels is lower than that for small diameter channels. From Table 1 it is seen that H/Dk values are 5.1% and 11.1% for the large channel B-2 and the small channel B-1, respectively. The relative roughness of B-2 (H/Dk = 5.1%) is similar to that for channel C-1 (H/Dk = 4.3%). Comparing Nu/Nuth,plain for these two geometries as shown in Fig. 9, it is seen that the heat transfer enhancement of C-1 and B-2 are quite similar. In general, heat transfer enhancement increases with the increasing of H/Dk for both channel diameters.

Early Transition From Laminar to Turbulent Flow. Figure 10 shows the Nu data sets of B-1, B-2, and C-1 plotted as a function of Re. The channels B-1 and B-2 have the same roughness elements but with different Dk, while the channels B-1 and C-1 have the similar Dk and roughness element pitch λ, but different roughness element heights H. At lower Re, the effect of Re on the three data sets is negligible. However, for Re > 900, the Nu for B-1 is significantly enhanced. Some scatter is seen around Re = 700–900 for B-1. This is believed to be due to a flow regime transition from laminar to turbulent flow. In laminar flow the boundary layer is thick and Nu is small. While in turbulent flow, the heat transfer coefficient is high.
due to the presence of turbulent eddies and flow mixing. Also in
turbulent flow the boundary layer thickness decreases with increasing
Re resulting in an increase in Nu with Re. Transition from lami-
nar to turbulent was observed from the slope and magnitude of heat
transfer coefficient. At Re less than 900 the Nu is about 20 and at a
nearly constant value while for Re higher than 900 the Nu dramati-
cally increases to 80 and the values increases with Re. Re = 900
was therefore taken as the transition Re for this surface. The data of
B-2 and C-1 revealed that the Nu is independent of Re, indicating
that no early transition occurred for B-2 and C-1; the transition is
seen to occur for these two channels around Re = 2000.

The heat transfer coefficients for B-1, B-2, and C-1 are shown in
Fig. 11. It is observed that h for C-1 is higher than that for B-2,
while Nu values for the two channels are similar. This is due to the
fact that the hydraulic diameter of C-1 ($D_h = 0.74 \text{ mm}$,
$D_{h,ct} = 0.68 \text{ mm}$) is lower than that of B-2 ($D_h = 1.87 \text{ mm}$,
$D_{h,ct} = 1.71 \text{ mm}$). Earlier transition was observed only for B-1.
Comparing Figs. 10 and 11, it is concluded that the earlier transi-
tion is due to the high value of relative roughness $H/D_h$. The corre-
csponding $H/D_h$ values for these surfaces, it is seen that
for B-1 surface, $H/D_h = 11.1\%$, while it is $5.1\%$ and $4.3\%$ for B-2
and C-1 surfaces. In Fig. 12 it is seen from earlier discussion, that the surface B-1 per-
formed the best among the different surfaces tested. The friction
factor increase and the heat transfer enhancement for the rough-
ess element B-1 are displayed in Fig. 13 in terms of the $f$ and
$j$-factor, which are commonly used in reporting performance data
for compact heat exchanger surfaces. The dimensionless heat
transfer coefficient $j$ is defined as

$$0 < \frac{H}{D_h} \leq 0.08, \quad Re_t = Re_o - \frac{Re_o - 800}{0.08} \frac{H}{D_h}$$

(12a)

$$0.08 < \frac{H}{D_h} \leq 0.25, \quad Re_t = 800 - 3,270\left(\frac{H}{D_h} - 0.08\right)$$

(12b)

where $Re_t$ is the transition Reynolds number from laminar to tur-
bulent flow and $Re_o$ is the transition Reynolds number for a
smooth channel with the same geometry and aspect ratio. The cal-
culated transition Re based on $H/D_h = 0.14$ is $Re_t = 604$. From
Figs. 10 and 11 the transition Reynolds number is seen to be
around $Re = 900$. The difference in $Re_t$ for these two studies may
be attributed to the differences in the surface profiles used in the
two studies; the smoother profile in the current study yields a
higher value of $Re_t$.

The roughness effect on friction factor for the geometries shown in
Table 1 is reported from a separate publication by Wagner
and Kandlikar [40]. Figure 12 shows friction factors for three
surfaces B-1, B-2, and C-1. It is seen that the transition from lami-
nar to turbulent flow occurs between Re = 600 and Re = 1100 for
these surfaces. The transition Re reported from the friction factor
studies is similar to that observed from the heat transfer data in
the present study for B-1. Similar observations are made by com-
paring the friction factor data for B-2 and C-1 with the heat trans-
fer data shown in Figs. 10 and 11.

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factor increase and the heat transfer enhancement for the rough-
ess element B-1 are displayed in Fig. 13 in terms of the $f$ and
$j$-factor, which are commonly used in reporting performance data
for compact heat exchanger surfaces. The dimensionless heat
transfer coefficient $j$ is defined as

$$0 < \frac{H}{D_h} \leq 0.08, \quad Re_t = Re_o - \frac{Re_o - 800}{0.08} \frac{H}{D_h}$$

(12a)

$$0.08 < \frac{H}{D_h} \leq 0.25, \quad Re_t = 800 - 3,270\left(\frac{H}{D_h} - 0.08\right)$$

(12b)
It is found that the enhancement in $f$ and $j$ follow a somewhat similar trend. Both of the data revealed a slope change between $Re = 700$ and $Re = 900$, which indicated an earlier transition to turbulent flow.

The heat transfer enhancement is generally compared with friction factor increase while considering their applicability in heat exchangers. The enhancement is measured in terms of a thermal performance index $\eta$ that compares the heat transfer enhancement to the pumping power requirement and is defined as

$$\eta = \frac{\text{Nu}_{\text{th,plain}}}{f / f_{\text{th,plain}}}$$

(14)

The thermal performance index for surface B-1 is compared with some of the recent enhancement studies reported in the literature [32–34] with tape and other inserts in the flow channel. Figure 14 shows the results of this comparison, and it can be seen that the performance factor for the surface B-1 is better than the other surfaces. The heat transfer enhancement $\text{Nu}/\text{Nu}_{\text{th,plain}}$ reported by Wongcharee and Eiamsa-ard [33] using alternate clockwise and counterclockwise twisted tapes is about 6–13, but the friction factor increase is about 8–15 fold. The heat transfer enhancement $\text{Nu}/\text{Nu}_{\text{th,plain}}$ reported by Huang et al. [34] with a porous medium is very high, about 50–60 fold. In the laminar flow, heat transfer performance was significantly enhanced by inserting twisted tapes, porous material, and coiled wires. Similar observations are made by Krishna et al. with twisted tape inserts in a plain tube with a full twist, and Akhavan et al. [35] for a coiled insert. From the data in Fig. 14, it is seen that by generating internal roughness structure on microchannels the heat transfer coefficient is efficiently enhanced.

**Heat Transfer Enhancement.** From the experimental data reported in this investigation, it was found that structured roughness enhanced the heat transfer. The Nu data from fully developed laminar flow in eight rough channels, and the friction factor data in [40], are summarized in Table 3. For all rough channels, heat transfer was enhanced and $\text{Nu}/\text{Nu}_{\text{th,plain}}$ are all found to be greater than 1. The enhancement was normalized with respect to the plain channel results.

Although heat transfer enhancement of B-1 is as high as 377%, which is significantly higher than the other channels, it should be noted that the enhancement of the friction factor is 371%, which is also significantly higher than the other channels. From the summary of $\text{Nu}$ and $f$ data in the present study and [40], it was found that heat transfer enhancement due to roughness is higher than the friction factor in the parametric ranges investigated. The simulation work of Dharaiya and Kandlikar [44] indicated that the laminar fully developed Nu for the same roughness structure is 30.7, which is very close to the experimental Nu = 28.9 obtained in the present study; there is only a 6% difference between the simulated results and the experimental data.

Coleman et al. [45] experimentally and numerically assessed the effects of transverse rib roughness pitch-to-height ratios, $\lambda/H$, on heat transfer in turbulent flow. The maximum drag was reported to be at $\lambda/H = 5$. The authors reported a smooth transition from skimming flow to interactive flow at a rib spacing of $\lambda/H = 5$. In the extremes where $\lambda/H$ is significantly greater than 5, or less than 5, the roughness effect is expected to diminish. Although these publications discussing roughness ratios focus on turbulent flows, the concept may still hold true for microscale laminar flows. Webb et al. [46] revealed flow patterns over transverse-rib roughness as a function of rib spacing. The flow separates at the rib and reattaches six-to-eight rib heights downstream from the rib. The reattachment does not occur for $\lambda/H$ less than 8. The heat transfer coefficient attains the maximum values near the attachment point; the highest enhancement occurs for $\lambda/H$ between 10 and 15.

To study the effect of pitch to height ratio, the heat transfer and friction factor enhancement are plotted as a function of $\lambda/H$ in Fig. 15. It is found that the enhancement decreases with the increasing $\lambda/H$. Pethkool et al. [47] investigated the heat transfer enhancement in a helically corrugated tube. Their data also showed that the effect of relative roughness was higher than the effect of pitch. Figure 16 shows the effect of relative roughness on the enhancements. Both heat transfer and friction factor enhancements increase with increasing of $H/D_h$. Further experiments are needed to cover the wider ranges of pitch to height ratio, although the trends observed in the present work is in agreement with the effects seen for other similar geometries by earlier investigators.
mentally investigated in the present study. The following
nnels and channels with structured roughness surfaces was experi-

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Fig. 15 Friction factor $f$ and Nu enhancement ratio of as a function of $j/H$ for all rough channels tested

Fig. 16 Friction factor $f$ and Nu enhancement ratio of as a function of $H/D_h$ for all rough channels tested

Conclusions

The heat transfer performance of water flowing in smooth chan-
nels and channels with structured roughness surfaces was experi-
mentally investigated in the present study. The following
conclusions are drawn based on the experimental results:

1. The experimental data of smooth channels from the experi-
mental setup developed in this study indicated that the Nu
agrees well with conventional correlation predictions in
both developing and fully developed flows.

2. The heat transfer coefficient was found to be significantly
enhanced by the roughness structures. Surfaces with higher
$H/D_h$ had large enhancement in both heat transfer and pres-
sure drop.

3. The roughness element pitch did not affect heat transfer
significantly in the range investigated. Relative roughness
$H/D_h$ has a more significant effect on heat transfer than the
height $H$ alone. Further testing is needed before specific
recommendations can be made on the roughness profile.
However, it is noted that sharp edges and abrupt area
changes should be avoided.

4. The thermal performance index, defined as Nu enhance-
ment over a plain surface divided by the friction factor
enhancement to one-third power as given by Eq. (14), for
roughness structures with high relative roughness was
found to be at least comparable to the tape and coil inserts
examined by earlier investigators in the laminar region, and
superior to those structures in the turbulent region beyond

Re = 900. Structured roughness surfaces are therefore rec-
ommended as heat transfer enhancement devices at
microscale.

5. An earlier transition from laminar to turbulent flow was
observed in high $H/D_h$ rough channels, and the transition
Reynolds number obtained from heat transfer studies corre-
sponded with the transition noted from the friction factor
data by Wagner and Kandlikar [40]. The transition Reyn-
olds numbers were in reasonable agreement with the transi-
tion models based on relative roughness proposed by
Brackbill and Kandlikar [38], although the smoother profile
in the present study seems to have delayed the transition.

Nomenclature

$a =$ channel height, m
$A_h =$ flow heat transfer area, m$^2$
$A_{ch} =$ cross section area of channel wall, m$^2$
$A_f =$ cross section area of fluid flow in channel, m$^2$
$b =$ channel height, m
$h_{ct} =$ constricted channel height considering roughness
height, m
$c_p =$ heat capacity, J/kg °C
$D_h =$ hydraulic diameter, m
$D_{ch} =$ constricted hydraulic diameter (using constricted
channel gap), m
$f =$ friction factor, dimensionless
$f_{th,plain} =$ theoretical friction factor for smooth channels,
dimensionless
$Gz =$ Graetz number, dimensionless
$h =$ heat transfer coefficient, W/m$^2$-°C
$k_e =$ local heat transfer coefficient at location $x$, W/m$^2$-°C
$H =$ roughness element height, m
$l =$ current, A
$k_w =$ thermal conductivity of wall material, W/m °C
$k_f =$ thermal conductivity of fluid, W/m °C
$k_{wx} =$ local thermal conductivity of fluid, W/m °C
$L_{ch} =$ distance at location $x$ in the heated section from the
entrance, m
$L_{h,All} =$ total heating length of the channel, m
$m =$ mass flow rate, kg/s
$Nu =$ Nusselt number, dimensionless
$Nu_{th,plain} =$ Nusselt number without accounting for axial conduc-
tion, dimensionless
$Nu_{h,plain} =$ Nusselt number for plain channel in developing flow,
dimensionless
$Nu_e =$ local Nusselt number, dimensionless
$Pr =$ Prandtl number, dimensionless
$q_{conv} =$ input power to the fluid, W
$q_{mea} =$ measured input power by power supply, W
$q_{loss} =$ heat loss, W
$q_{avg} =$ average heat flux, W/m$^2$
$Re =$ Reynolds number, dimensionless
$Re_e =$ transition Reynolds number, dimensionless
$Re_{th} =$ transition Reynolds number for smooth channel,
dimensionless
$T_{li} =$ fluid temperature at channel inlet, °C
$T_{lo} =$ Fluid temperature at channel outlet, °C
$T_{wx} =$ local fluid temperature, °C
$V =$ voltage, V

Greek Symbols

$\alpha =$ aspect ratio $a/b$, dimensionless
$\eta =$ performance factor, dimensionless
$\lambda =$ roughness elements pitch, m

References

istics of Gas Flow in Fine Channel Heat Exchangers Used for Microminiature


