An experimental study of convective heat transfer from flat and ribbed surfaces†

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Heat transfer and friction characteristics of flow in a rectangular channel with a solid rib mounted on the bottom surface of the channel have been investigated experimentally. The experiments have been carried out at Reynolds number (based on hydraulic diameter of the channel) range of 12,800 to 29,400. The skin friction coefficient, Nusselt number, velocity and temperature profile results are presented. The surface temperature images obtained with the help of liquid crystal thermography supplement the understanding of flow physics with qualitative explanation. The average Nusselt number for the ribbed flow case is higher in comparison to that for the flat surface. The increase in skin friction coefficient for the ribbed flow case is comparatively less than the Nusselt number increase, indicating the effectiveness of the ribbed surface in heat transfer enhancement applications. The liquid crystal thermography images show the flow in the downstream region of the rib to be two dimensional in nature. The recirculation and reattachment region for the ribbed flow case and their role in heat transfer effectiveness is clearly visible in the liquid crystal images.

Flow interruption created in flow passages at periodic intervals is a popular means for heat transfer enhancement. The flow passages (which may be of various geometries, for example tubular or rectangular) or surfaces of devices such as heat exchangers, advanced gas-cooled reactor fuel elements, and electronic cooling devices are usually roughened or dimpled to improve the convective heat transfer. The roughness breaks up the viscous sublayer and promotes local wall turbulence. There is an increase in heat transfer from both the rough and smooth surfaces with simultaneous increase in the pressure drop. Thus, it is of interest to find out a roughness geometry that will yield high heat transfer augmentation with minimum pressure penalty. The rib turbulators have been investigated by many investigators as a possible means for heat transfer enhancement. These ribs increase the heat transfer in the channel flows by increasing the heat transfer area (fin effect) and by disturbing the laminar sub-layer (roughness effect).

Sundén† has presented an overview of heat transfer and fluid flow in rib-roughened rectangular ducts. Geometric parameters such as channel aspect ratio (AR), rib height-to-passage hydraulic diameter $e/D_h$ or blockage ratio, rib angle of attack $(\alpha)$, the manner in which the ribs are positioned relative to one another, rib pitch-to-height ratio $(s/e)$, rib shape and configuration have pronounced effects on both local and overall heat transfer coefficients. Some of these effects have been studied by many investigators.

Investigations in the past mainly concerned the overall pressure drop or friction factor and the average heat transfer coefficient or Nusselt number. More recent studies have used new techniques, i.e. laser Doppler velocimetry, laser holographic interferometry and liquid crystal thermography to study the details of the flow pattern and temperature field. Improvements on the understanding of the physical processes have been made.

The present investigations attempt to improve the physical understanding of the flow in a rib roughened channel by simultaneous quantitative representation of friction characteristics, heat transfer characteristics, velocity profiles and temperature profiles. Liquid crystal thermography has been used to visualize the temperature field of the bottom surface of the channel. Liquid crystal thermography uses liquid crystals, which exist between the solid and isotropic liquid phase of some organic compounds. Incident light is scattered selectively and the change in their colour is observed depending on the temperature. The change in color is monitored by image processing system and is used as a measure of the temperature field. One of the important advantages of this non-intrusive thermography technique is to illustrate the flow footprints and local values of temperature/heat transfer coefficient on the surface under investigation.

This simultaneous “flow visualization” enhances the understanding of underlying physics and helps the investigator in interpretation of the results.

Experimental Procedure

Fig. 1 shows the sketch of the experimental set up used in this work. The experimental facility comprises of a flow circuit, an image processing system, the traverse mechanism and the heating section. Velocity measurements have been carried out using a pitot static tube connected to a digital micromanometer (FC012 - Furness Controls Limited, England). Temperature in the thermal boundary layer is measured using a K-type thermocouple connected to a digital multimeter. Wall temperature distributions have been recorded with liquid crystal sheets, exposed to two 50 W tungsten-halogen lamps, providing an excellent colour rendering even in the long run, a reasonable high efficacy, high luminance and compact size as described by de Boer and Fischer².

Flow circuit

The experiments are performed in an open-loop airflow system. The air is sucked into the test section through a honey comb section, five anti-turbulence screens and a 3:1 contraction cone. The test section is followed by a flow straightener to minimize the influence of blower noise in the test section. Speed of the blower is controlled by a speed controller (Victor G1000) supplied by Kirloskar Electric Co. Ltd., India. The test channel is 3300 mm long with an aspect ratio of 1.8:1, (298x160 mm² in the vertical plane) and is made of perspex sheet of 12 mm thickness.

Heating section

A single aluminium plate (680x298x3 mm³), heated by stainless steel foil from underneath, acts as heat transfer surface. Six stainless stainless steel foils of dimension 690x47x0.045 mm³ connected in series are cemented onto the 25 mm thick bakelite sheet. A variac is used to control the power supply, i.e. heating level. The aluminium plate is highly polished in order to minimize emissivity and hence the radiative losses. In addition, to minimize the conductive heat losses, lower surface of the bakelite board is insulated using a 13 mm thick plate of bakelite with a 2 mm air gap in between.

The heat transfer surface is instrumented with thirteen calibrated thermocouples of the chromel-alumel type, along the centerline and spanwise direction of the heated plate in order to measure the plate temperature and check the spanwise temperature uniformity respectively. Conduction losses to the bottom surface of the heating section through the
The skin friction coefficient can be determined by measuring the velocity gradient in the near wall region of the channel from using:

\[ C_f = 0.0592 \text{Re}_{	ext{v}}^{\frac{1}{5}} \quad \text{for } 5 \times 10^3 < \text{Re}_v < 10^5 \]  

... (5)
\[ C_f = \frac{\mu (du/dy)|_{wall}}{1/2 \rho U^2} \] \hspace{1cm} \ldots (6)

where, the wall velocity gradient is determined from the quadratic curve fitting through the near-wall values of velocity.

**Thermal boundary-layer**

The non-dimensional temperature \( \Theta \) is defined as:

\[ \Theta = \frac{T_e(x)-T(x,y)}{T_e(x)-T_m} \] \hspace{1cm} \ldots (7)

where, \( T_e(x) \) is the local temperature of the plate, \( T_m \) is the free stream temperature and \( T(x,y) \) is the temperature inside the flow field. The theoretical non-dimensional temperature variation for laminar flow is obtained from Blasius solution\(^{10}\). The laws of the wall for turbulent flow temperature profile can be expressed as\(^{10}\):

\[ y^+ = 2.075 \ln y^+ + 3.9 \] \hspace{1cm} \ldots (8)

where,

\[ y^+ = \frac{y U_e \sqrt{C_f/2}}{\gamma} \]

Here the wall shear stress, \( \tau_w \) is obtained as earlier by Eq.(1). Therefore, basic unknown in the Eq.(8) is \( q_w \), which can be obtained by applying the boundary condition: @\( y^+ \delta_f \), \( T = T_m \) where, \( \delta_f \) and \( T_m \) are the thermal boundary layer thickness and free stream temperature respectively. Thus, using Eq.(8) it is possible to get the plot of \( \Theta(x,y) \) as a function of \( y^+ \delta_f \).

The variation of \( \delta_f / x \) versus Reynolds number comparison for laminar flow uses:

\[ \frac{\delta_f}{x} = 5.631 Re^{1/2} \] \hspace{1cm} \ldots (9)

The above relation is inferred from Eq.(2) by using \( \delta / \delta_f = Pr_f^{1/2} \). For turbulent flow, the thermal boundary layer thickness variation is expressed as:

\[ \frac{\delta_f}{x} = 0.4167 Re^{1/5} \] \hspace{1cm} \ldots (10)

The above relation is obtained from Eq.(3) and \( \delta / \delta_f = Pr_f^{1/2} \), where \( Pr_f \) is the turbulent Prandtl number\(^{10}\).

The local convective heat transfer coefficient of the heated surface is presented in terms of the local Nusselt number \( Nu_x \) and is defined as:

\[ Nu_x = -\int_0^x \frac{dT}{T_e(x)-T_m} dy \] \hspace{1cm} \ldots (11)

Here the air temperature gradient \( dt/dy|_{wall} \) is determined by fitting quadratic function through the near wall values of temperature. The correlation for the local Nusselt number for laminar flow is given by\(^{10}\):

\[ Nu_x = 0.332 Re^{1/2} Pr_f^{1/3} \text{ for } 0.6 < Pr < 50 \] \hspace{1cm} \ldots (12)

The correlation for the local Nusselt number for turbulent flows is:\(^{11}\):

\[ Nu_x = 0.0296 Re^{1/2} Pr_f^{1/3} \text{ for } Re_x < 10^8 \] \hspace{1cm} \ldots (13)

The properties of air are calculated at the film temperature, \( T_f = (T_e(x)+T_m)/2 \).

**Results and Discussion**

The flow and heat transfer test results are presented here along with the respective laminar and turbulent profiles in order to study the influence of two dimensional rib of square cross section on the wall parameters. In addition, liquid crystal images are used to assist in interpreting the influence of rib in flow modulation and heat transfer enhancement.

**Velocity field**

Figs 2 and 3 compare the experimental dimensionless transverse coordinate \( (y/\delta) \) versus the dimensionless local velocity \( (u/U_e) \) profile
without and with rib respectively with flat plate laminar and turbulent profile. The velocity profile in Fig. 2 indicates the flow on the flat surface to be laminar. The velocity profiles in Fig. 3 are seen to be turbulent in nature due to the presence of the rib.

Figs. 4 and 5 show the variation of the dimensionless boundary-layer thickness \( \delta / \delta \) as a function of local Reynolds number \( R_e \). Fig. 4 supports the earlier observation that the flat surface velocity profile is laminar. The boundary layer thickness for the rib roughened surface is larger than the laminar and turbulent boundary layer (Fig. 5).

The Reynolds number dependence of skin friction coefficient, \( C_f \) is shown in Figs 6 and 7. At the inflow plane for smooth surfaces, there is about 8-10% deviation from the laminar correlation which diminishes well as the Reynolds number increases. For ribbed surfaces, the skin friction coefficient does not have a systematic variation. This indicates the complexity of the rib roughened surfaces in comparison to flat surface.

**Thermal field**

Figs 8 and 9 compare the temperature profile of the flat surface and ribbed surface respectively with the laminar and turbulent profile of a flat plate. The measured flat surface temperature profile in Fig. 8 is similar to the flat plate turbulent profile indicating the temperature profile to be turbulent in nature. In comparison to the flat surface (Fig. 8), Fig. 9 shows the temperature profile to be turbulent in nature for the rib roughened surface. Figs 10 and 11 show the variation between the dimensionless thermal boundary-layer thickness and the local Reynolds number. From Fig. 10, the temperature profile is observed to be close to laminar in nature. The trend shown for the ribbed surface in Fig. 11 is different from that of the flat surfaces shown in Fig. 10.

The local Nusselt number versus the local Reynolds number plot is shown in Figs 12 and 13 for smooth
Fig. 6—Skin friction coefficient as a function of Reynolds number for a flat surface

Fig. 7—Skin friction coefficient as a function of Reynolds number for a flat surface with a rib

Fig. 8—Temperature profiles in flow over a flat surface

Fig. 9—Temperature profiles in flow over a flat surface with a rib

Fig. 10—Variation of thermal boundary layer thickness for flow over a flat surface

Fig. 11—Variation of thermal boundary layer thickness for flow over a flat surface with a rib
and ribbed surfaces, respectively. For smooth surfaces there exists a good agreement between the measured data and turbulent correlation. This is understandable because the Nusselt number follows closely the local temperature profile. The figures show that at low Reynolds number, the local Nusselt number of the ribbed channel is significantly higher than that of smooth channel; but at a higher Reynolds number, this difference is less. Also the local Nusselt number of the ribbed channel increases monotonically with an increase in the local Reynolds number.

**Transient heat transfer experiment**

The transient liquid crystal thermography (LCT) technique offers significant advantage of yielding local heat transfer coefficients over complete test surfaces in a single experiment. In this case, the surface is heated to a constant temperature and suddenly cooled by an ambient mainstream. The images are captured at regular intervals and then temperature contours are drawn. The colour images of the transient test experiment conducted at $Re(D) = 20900$ for a ribbed surface are shown in Figs 14a-d. In the coloured images, black indicates the coldest regions, followed by red, green, yellow and blue respectively for increase in the temperatures.

Initially, the plate is heated up to $41^\circ C$ and correspondingly the LC sheets appears to be pure blue, then on forced cooling the gradual changes in temperature with time occurs which is clearly depicted in the images. Fig. 14a shows the constant
plate temperature at the start of the experiment by uniform colour distribution. Figs 14b, 14c and 14d show the colour/temperature distribution at later time, i.e. after 240, 480 and 720 s respectively. The green/red region in Figs 14b, 14c and 14d indicate lower temperature. The existence of low temperature at a distance from the downstream edge of the rib indicates that the heat transfer is maximum near the reattachment region. Comparatively, high temperature near the rib points to the low heat transfer zone due to the stagnant flow in the recirculation zone. Overall, the visualization image shows that the flow is two-dimensional as was expected from the 2-D nature of the rib-channel geometry. The existence of definite two-dimensionality in the downstream region of the flow behind rib is supported by the uniform colour distribution in the span wise direction. Further, after ten minutes of cooling, the plate cools to a temperature less than 35°C, and the thermal field is outside the range of liquid crystal and it appears black.

Conclusions

Fluid flow and heat transfer characteristics in the $Re(D_0)$ range of 12,800 to 29,400 have been carried out experimentally. The present investigation shows that the velocity profile for the smooth surface case is very similar to the Blasius profile indicating the flow to be laminar in nature. The skin friction coefficient versus Reynolds number plot supports the observation regarding the laminar nature of the flow. Contrary to the above finding, the temperature profile for the smooth wall case is observed to be similar to the law of the wall profile indicating the flow to be turbulent in nature. The Nusselt number versus Reynolds number plots support the observation about the turbulent nature of the flow. The local Nusselt number, average Nusselt number and skin friction coefficient for the ribbed surface are higher than that of the smooth surface. Both the Nusselt number and the skin friction coefficient are functions of the Reynolds number, the former being a stronger function of the Reynolds number than the latter one. The above observation supports the square rib to be an effective turbulator for heat transfer enhancement point of view.

The transient liquid crystal thermography shows the flow behind the rib to be two dimensional in nature. The ineffectiveness of the recirculation region and the effectiveness of the reattachment region for heat transfer enhancement are clearly visible from the liquid crystal images.

Nomenclature

- $C_f$ = Skin friction coefficient
- $C_p$ = Specific heat
- $D_h$ = Hydraulic diameter of the duct
- $Nu_l$ = Local Nusselt number
- $Pr$ = Prandtl number
- $Pr_l$ = Turbulent Prandtl number
- $q_h$ = Heat flux supplied by the heating foil
- $q_w$ = Wall heat flux
- $Re(D_0)$ = Reynolds number based on hydraulic diameter
- $Re_{xy}$ = Local Reynolds number ($U_{xy}/
u$)
- $T(x,y)$ = Temperature of the air
- $T_{r(x,y)}$ = Local temperature of the plate
- $T_f$ = Free stream temperature
- $T_w$ = Temperature in wall coordinate
- $u'$ = Velocity in wall coordinate
- $u, v$ = Velocities components in cartesian coordinates ($x,y$)
- $V_{x,y}$ = Streamwise (axial) and transverse coordinates respectively (Fig. 1)
- $y'$ = Transverse wall coordinates
- $\delta$ = Velocity boundary layer thickness
- $\delta_l$ = Thermal boundary layer thickness
- $\Theta$ = Non-dimensional temperature
- $\gamma$ = Kinematic viscosity
- $\mu$ = Dynamic viscosity
- $\rho$ = Density of the air
- $\tau_w$ = Wall shear stress

References