Experimental study of flow and heat transfer in rib-roughened rectangular channels

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Abstract

Secondary flow patterns, pressure drop and heat transfer in rib-roughened rectangular channels have been investigated. The aspect ratio of the channels is 1–8, and ribs are attached to the wide channel walls in order to set up swirling motions. The geometries tested consist of channels having cross ribs, parallel ribs, cross V-ribs, parallel V-ribs, and multiple V-ribs (Swirl Flow Tube). The flow patterns were investigated using smoke wire visualization and LDV measurements. The smoke wire experiments have been performed at Re = 1100 and the LDV measurements at Re = 3000 at periodic fully developed conditions. The heat transfer and pressure drop are described by $j$ and $f$ factors for Reynolds numbers from 500 to 15 000. The distributions of axial mean velocity and turbulent fluctuations are strongly influenced by the secondary flows. Large mean velocities and small fluctuations are found in regions where the secondary flow is directed towards a surface, while small mean velocities and large fluctuations are found in regions where the secondary flow is directed away from a surface. The Swirl Flow Tube provides a significant increase in the $j$ factor at Reynolds numbers from 1000 to 2000, but unfortunately also an increase in the $f$ factor. At higher Reynolds numbers, the $j$ and $f$ factors of the Swirl Flow Tube are of the same order of magnitude as for the other rib-roughened channels. It is found that the flow direction in a channel with parallel V-ribs has important influence on the $j/f$ ratio. At Reynolds numbers above 4000, this channel provides the highest $j/f$ ratio if the V-ribs are pointing upstream; while it provides the lowest $j/f$ ratio of all rib configurations, if the V-ribs are pointing downstream. © 1998 Elsevier Science Inc. All rights reserved.

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1. Introduction

Ribs are often used in order to enhance forced convective heat transfer between a wall and a fluid. However, depending on the application the design of the ribs will be different. In this investigation the interest is to study the flow fields and heat transfer in channels of compact heat exchangers, in which the ribs have a smooth shape due to the manufacturing process. There exists a variety of investigations concerning ribbed walls in square and rectangular channels. Han et al. [1] investigated the effects of rib shape, angle of attack and pitch to height ratio on friction and heat transfer in a parallel plate experiment for Re = 3000–30 000. It was found that a symmetrical rib arrangement gave the same results as a staggered rib arrangement, and that the shape of the ribs mainly influenced the friction factor and had only a very modest effect on heat transfer. The highest heat transfer for a given friction power was obtained with ribs having a 45° angle of attack. Metzger et al. [2] investigated the effects of rib angle and orientation on local heat transfer in a square channel. They expected different large scale secondary flows depending on the orientation of the ribs, i.e. a two-cell pattern when the ribs are placed in parallel planes, and a one-cell pattern when the ribs are placed in intersecting planes on the opposite walls. It was found that 60° angled ribs giving a two-cell pattern provided the best heat transfer performance. In a similar investigation, Han et al. [3] also included V-shaped ribs with 45° and 60° half-angle, where the ribs may point in the flow direction or opposite to it. The ribbed walls were in-line, and the Reynolds number was varied from 15 000 to 90 000. It was found that the
V-shaped ribs pointing upstream gave the highest heat transfer augmentation, while the V-shaped ribs pointing downstream generated the greatest pressure drop. It was believed that secondary flows are essential in the explanations to the observed differences between the channels, but no details of the flow fields were provided.

Taslim et al. [4] report local Nusselt numbers in a square channel roughened with angled, V-shaped, and discrete ribs on two opposite walls. Liquid crystals were used to monitor the wall temperature distribution. Only one wall was heated with a constant heat flux, while the other walls were adiabatic. In contrast to the results of Han et al. [3] and the results from the present investigation, Taslim et al. obtained the highest Nusselt numbers for the tube with V-shaped ribs pointing downstream. However, the difference in average Nusselt number between the tubes with V-shaped ribs pointing upstream or downstream was within the experimental uncertainty. The heat transfer and pressure drop in a square channel with wedge-shaped and delta-shaped turbulence promoters was investigated in [5], and it was found that the delta-shaped ribs should provide the best performance. Zhang et al. [6] studied the effects of ribbed-grooved walls in a rectangular channel with an aspect ratio 1–10 at $Re = 10,000–50,000$, and it is found that a ribbed-grooved wall generates a greater heat transfer enhancement than a ribbed wall but at essentially the same pressure drop penalty. In [7] it was shown that heat and momentum transport are uncoupled in the recovery region downstream a grooved section. Experiments were performed at $1500 < Re < 5000$ and it was found that the recovery lengths for shear stress and pressure gradients are significantly shorter than for temperature. Effects of rotation on the flow in a ribbed rectangular channel have been investigated in e.g. [8,9]. In these latter cases the flow field will be further complicated due to the Coriolis-induced flow. The influence of Prandtl number on heat transfer enhancement is turbulent flow in rib-roughened circular tubes has been investigated in e.g. [10,11].

While investigations of overall characteristics such as the friction factor and average heat transfer coefficient have been presented for many enhanced channels, investigations of the corresponding flow fields are rare. Such investigations may be very useful as the results are interpreted and as the physical mechanisms are of concern. Detailed LDV measurements of the flow field in a rectangular channel with transverse ribs at $Re = 33,000$ are presented in [12] and in [13] an investigation on the flow field and temperature field of a square channel with ribs only on one of walls at $Re = 65,000$ is reported. In both papers it was pointed out that secondary flow is an important mechanism for redistribution of momentum and temperature. Although a turbulent flow over a transverse rib may appear as two-dimensional, local permanent three-dimensional patterns exist, as presented in e.g. [14]. Hwang and Liou [15] have shown that using perforated ribs instead of solid ribs will give a superior heat transfer performance as well as reducing hot spots. Liou et al. [16] studied the periodic heat transfer in a channel with ribs detached from one wall, and an optimum clearance ratio was found.

It is known that turbulent flow in smooth square and rectangular channels exhibits a secondary flow which is important for the distribution of shear stress and heat transfer coefficient on the walls. Also, the mean velocity field is influenced due to the redistribution of momentum by the secondary flow, see e.g. [17]. The secondary flow is caused by small differences in forces exerted by on the one hand Reynolds stresses and on the other hand static-pressure gradients, see [18]. They also showed that the ratio of secondary flow velocities to primary flow velocities decreases with increasing Reynolds number. If the viscosity of the fluid varies in the flow field, the secondary flow pattern may be different from that with constant viscosity, see e.g. [19].

One way to induce longitudinal vortices along a surface is to install half-delta wings on the surface. If the wing has an angle of attack to the flow, a wing tip vortex will be formed which can be traced as far downstream as 60 wing cords behind the wing. The vortex breaks up the boundary layer and entrains high momentum fluid close to the wall which increases the spanwise average skin friction and heat transfer. The flow field behind pairs and arrays of winglet-type vortex generators embedded in a turbulent boundary layer was investigated by Paulsey and Eaton [20], and another investigation of the flow field behind a single half-delta wing in a flat plate laminar boundary layer was reported in [21]. Numerical predictions of heat transfer and flow structures in laminar and turbulent flows in a rectangular channel with longitudinal vortices were presented in [22], and the flow and heat transfer in a rectangular channel with longitudinal vortex generators on one wall and rib-roughness elements on the other wall was studied numerically by Zhu et al. [23]. A review of recent progress in the use of longitudinal vortices for heat transfer enhancement is found in [24].

Rib-roughened and dimpled channels are frequently employed in radiators in order to improve the heat transfer performance on the liquid side. Investigations of the overall hydraulic and thermal performance of such channels are found in e.g. [25,26]. Typical properties of a radiator channel is that the width to height ratio is between 5 and 20 and that the roughness elements are rolled in the sheet-metal as the channels are formed. As the hydraulic diameter is in the range of 3 mm and the wall thickness usually 0.1–0.5 mm, it is not possible to have sharp edged roughness elements but they will rather appear as protuberances into the channel.

This work concerns a detailed investigation of the flow fields and the possibilities to create large scale secondary flows in a rectangular channel using ribbed walls. Pressure drop and heat transfer data are established for channels which all have equal hydraulic diameter, rib height, and rib pitch. Also, the considered rib configurations provide identical surface extension, which means that any variation in performance is due to rib configuration only.
2. Experimental arrangement

The investigation was carried out in four separate parts with different test rigs. In the smoke visualization and LDV experiments, a rectangular plexiglass channel of $14.5 \times 112.5$ mm (height $\times$ width) cross section has been used. To simulate the real rib-roughened surfaces, the rib configurations have been accomplished by attaching circular plastic tubes of 1.5 mm diameter to the walls (equal to rib height in this part). However, in order to facilitate the positioning of the ribs on the walls, a 0.10 mm thick plastic film with a print of the tested rib pattern is attached to each of the walls as templates. Double-sided adhesive tape of 0.10 mm thickness have been used to fix the templates to the plexiglass walls to fix the ribs to the templates. For an effective cross section of $14.1 \times 112.5$ mm, the hydraulic diameter is equal to 25.1 mm. The rib configurations tested are shown in Fig. 1. Angled ribs, V-shaped ribs, and multiple V-shaped ribs have been investigated. In the channels with parallel ribs on the two walls the ribs are staggered, and in the channels with crossed ribs the ribs are staggered in the channel centre line plane. The idea of the channel with multiple V-shaped ribs, also referred to as Swirl Flow Tube, is to establish eight longitudinal vortices which will be effective in exchanging fluid between the wall region and the core region. A similar channel with 32 vortices has also been tested, but due to the practical difficulties in manufacturing such a channel for a compact heat exchanger that channel will not be discussed in this paper. The rib pitch was 32 mm for the straight ribs and the V-shaped ribs, while a rib pitch of 56 and 70 mm was used for the multiple V-shaped ribs in the smoke wire visualization and the LDV measurements, respectively. The ribs in all the rib-roughened channels are oriented so that they make a $60^\circ$ angle away from the axial direction. Fig. 2 provides conjectured flow patterns for each channel. These patterns might be imagined if one considers the possible influence of the different rib configurations on the fluid flow. Similar ideas have been presented by Metzger et al. [2] for rib-roughened square channels.

2.1. Smoke visualization

The smoke wire technique, see e.g. [27], has been used to visualize the flow 20–24 hydraulic diameters downstream the channel inlet. The flow system consists of a fan, a rotameter to measure the flow rate (Krohne, flow range 3–30 m$^3$/h), and a by-pass arrangement to control the flow. Air is sucked into the channel from the quiescent room, and no contraction has been used at the inlet. The channels are rib-roughened all the way from the inlet. The smoke wire (NiCrome, 0.1 mm diameter) is placed in the vertical centre plane at 20 hydraulic diameters from the inlet, and a 5 mm wide light slot at 24 hydraulic diameters makes the smoke visible for a video camera (25 pictures per second), which is placed in front of the channel on the channel centre line. The smoke wire is wetted with paraffin oil and subsequently heated electrically up to approximately 300°C during 2 s. The video camera registers the smoke as it passes the illuminated cross section. In a later stage, the video pictures are digitized and stored on a computer disc.

2.2. LDV measurements

The fluid velocity was measured using a three-component LDV system with sidescatter arrangement, see Fig. 3. However, only one component was used at a time with a green colour of 514.5 nm wavelength. The system was composed of an argon ion laser (COHERENT INNOVA 90), the Dantec fibre-flow series of optical elements, including a transmitter box with a Bragg cell and colour separators, a one-component probe and a two-component probe with beam expanders of the factor 1.95, and a front lens of 310 mm focal length. The probes contained both transmitting and receiving optics, and the collected light was led to colour separators and further to photomultipliers, where it was transformed into electric signals. The signals were analyzed in a Dantec burst spectrum analyzer, BSA (57N10), and the positioning of the probes was controlled using a Dantec traverse (57G20 and 57B100). A computer equipped with Dantec BURSTware 3.1 was used to control the LDV hardware and to process the velocity data. The sidescatter arrangement gives an ellipsoidal measurement volume with approximately 0.05 mm width.

Fig. 1. Rib configurations: (a) cross rib-roughened; (b) parallel rib-roughened; (c) cross V-rib-roughened; (d) parallel V-rib-roughened; (e) Swirl Flow Tube. The solid arrows indicate the positions of the LDV measurements relative to the neighbouring ribs. (Rib angle $60^\circ$).
and 0.10 mm length. The output power of the laser was typically 1.0 W.

A TSI model 9306 six-jet atomizer was used as seeding unit with 7% sugar–water solution as seeding material. Vegetable oil was also tested as seeding material, but there was a problem with contamination of the plexiglass surface which lowered the data rate after approximately one hour operation. With the sugar–water solution, the system could be operated during more than 20 h without any effects of contamination.

The flow system for the LDV measurements consists of the same components as for the smoke visualization, but in addition the seeding unit and a plenum with two turbulence generating grids are placed upstream the test section, and another plenum including a particle filter is placed downstream the test section, see Fig. 3. No smooth contraction is used at the inlet of the rectangular channel. The test section consists of a 28 hydraulic diameters long smooth section followed by a rib-roughened section of equal length. The measurements were carried out at 50 hydraulic diameters from the inlet of the rectangular channel. At this position, an 8 mm slot was cut in the template in order to avoid disturbances of the laser beams. At the measurements in the rib-roughened channels 6000 samples were taken, and at the smooth channel measurements, which were performed mainly to check the quality of the flow in the test section, 20 000 samples were taken. Only validated bursts

Fig. 2. Rib configurations and conjectured secondary flow patterns: (a) cross rib-roughened; (b) parallel rib-roughened; (c) cross V-rib-roughened; (d) parallel V-rib-roughened; (e) parallel V-rib-roughened, (f) Swirl Flow Tube.

Fig. 3. Schematic drawing of the LDV experimental set-up.
were stored and used in the evaluation of instantaneous velocities. No corrections for velocity bias have been employed. It should be noted that the LDV measurements were carried out at only one position for each channel. At similar locations upstream and downstream the time-averaged flow pattern should be identical due to the periodicity of the geometry and thus the flow. In between such locations the flow pattern will differ somewhat but the present study is a first comprehensive investigation of the flow pattern in rib-roughened rectangular channels with an aspect ratio of practical relevance and should provide some characteristic features of the flow fields. To obtain a complete flow picture complementary studies are required.

The smooth channel measurements consisted of 10 \times 14 points and covered one quadrant of the channel plus one row of measurement points on the other side of the lines of symmetry. The points closest to the walls were 0.5 mm from the short wall and 1.25 mm from the wide wall. In the rib-roughened channels, two different measurement grids were employed for each channel. A coarse grid, that was used for all these channels, consisted of 7 \times 11 points and covered the whole cross section with measurement points 2.0–2.9 mm away from the short and wide walls, respectively. A finer grid, that was used for the channels having straight or V-shaped ribs, consisted of 10 \times 11 points and covered approximately one quadrant of the cross section with measurement points 2.25 and 1.25 mm away from the short and the wide walls, respectively. For the channel with multiple V-shaped ribs, the finer grid consisted of 9 \times 15 points and covered a part of the cross section extending from 11.2 to 30.8 mm from one of the short walls and including points that were 1.45 mm away from each of the wide walls. With this grid, the flow in one swirl cell could be captured with reasonable resolution. The flow fields in the rib-roughened channels are not expected to be symmetrical in any respect. However, the more detailed measurements have been restricted to one quadrant of the cross section in order to limit the number of measurement points, and also due to the problem of having the \( W \) component laser beams close to the wide walls. Also, the presence of ribs makes it difficult to measure close to the walls at some locations. Despite these restrictions, it has been possible to obtain a picture and imagination of the flow fields in the channels in terms of secondary flows and distributions of axial mean velocity fluctuations at representative locations in the channels.

2.3. Pressure drop and heat transfer measurements

In the pressure drop and heat transfer test rigs, the channels are made of copper, and all of them have an internal cross section of 5 \times 40 \, mm\(^2\), which yields a hydraulic diameter equal to 8.89 mm. The channels are 500 mm long, and they are manufactured in two parts – a 10 \times 50 \, mm\(^2\) bar with a 5 mm deep and 40 mm wide milled slot and a 5 \times 50 \, mm\(^2\) planar rod. The two parts are screwed together and sealed with O-ring gaskets. The ribs are made of 0.5 mm copper wire and soldered on the wide walls in the channels. The streamwise spacing of the ribs is 25 mm.

The pressure drop and heat transfer investigations were performed separately. Air was used in both tests, and the pressure drop measurements were performed at isothermal conditions. The pressure drop rig consists of a fan, three rotameters to measure the flow rate (Krohne, flow ranges 0.12–1.2, 1.2–8, and 3–30 \, m\(^3\)/h, respectively), and a by-pass arrangement to control the flow. The air is sucked into the tested channel from the quiescent room. To enable pressure drop measurements, the channels are equipped with 10 pressure taps along the flow length with 0.5 mm hole diameter and 50 mm spacing. The pressure drops are measured with a micromanometer, FCO14 Furness Controls Ltd, and the signals are recorded by a Macintosh II using a MacADIOS card.

The heat transfer test rig consists of a fan and the same flow rate control system as in the pressure drop test rig. In addition, there is an electrical heater together with a small fan which provide heated air to a box upstream of the tested channel. In order to obtain constant temperature at the channel wall, the channel is mounted on a so-called water table. Water is passing the channel in cross flow with a speed of approximately 1 m/s, and a two-dimensional contraction is placed upstream the channel to make the water flow uniform. This provides the outer channel wall with a heat transfer coefficient that is much larger than the inside wall heat transfer coefficient. The thermal resistance on the outside is thus much less than that on the inside of the channel and is assumed negligible. Under the prevailing circumstances, also the thermal resistance in the channel wall is negligible. It can therefore be assumed that the inside wall temperature is the same as the temperature of the water. Since the water flow rate is high and the heat flux modest, the water temperature can be considered as constant and uniform. The temperature difference at the inlet is approximately 70°C and the temperature difference at the outlet is 2–20°C depending on the air flow rate.

The air temperatures are measured using 0.25 mm copper-constantan thermocouples, and the voltages are recorded by a Keithley 199 System DMM/Scanner. At the channel inlet and outlet, two thermocouples are used. In the determination of the heat transfer coefficient, the bulk mean temperatures at the inlet and outlet are used. At the end of the test section there is a large difference between the maximum and minimum temperatures. To obtain the outlet bulk temperature, the channel outlet is connected to an insulated mixing chamber. The air leaves the mixing chamber through two circular holes in which the thermocouples are placed.

An additional thermocouple is used to measure the temperature at the flow meters. The water temperature is taken by a mercury thermometer. No pressure drop measurements are carried out in the heat transfer rig. Only the pressures after the channel and at the flow meters are taken so that the densities can be calculated.
3. Data evaluation

The pressure drop data are evaluated using the Fanning friction factor, $f$, and the inlet loss coefficient, $K_c$. These parameters are determined from the nondimensional pressure drop equation, see e.g. [28,29],

$$\frac{\Delta p}{\rho \frac{U_m^3}{2}} = K_c + (f \text{Re})_{id} \frac{4x}{D_h \text{Re}}$$  \hspace{1cm} (1)

by performing a least squares fit of a line to the straight part of the plot of $\Delta p/(\rho \frac{U_m^3}{2})$ versus $x/(D_h \text{Re})$. The slope is equal to $4(f \text{Re})_{id}$, and the intercept is equal to $K_c$. The velocity $U_m$ is the mean velocity calculated from the mass flow rate divided by the density and cross-sectional area of the channel. The cross-sectional area is measured at the base of the roughness elements. The Reynolds number is determined from

$$\text{Re} = \frac{U_m D_h}{v}. \hspace{1cm} (2)$$

The pressure at the inlet, $x = 0$, is taken as the air pressure outside the channel. This means that the inlet loss coefficient includes acceleration of the fluid from rest. A term of value unity is thus added to $K_c$ as compared to the entrance pressure-loss coefficients presented in e.g. [30,31].

As the total pressure drop, from inlet to outlet, is considered it might be appropriate to introduce the so-called apparent friction factor, see [32]. Although, the apparent friction factors can be evaluated from the presented inlet loss coefficients and friction factors it is provided here. The pressure drop (excluding the effects of sudden contraction and sudden expansion at the inlet and outlet, respectively) is then written as

$$\frac{\Delta p_{\text{tot}}}{\rho \frac{U_m^3}{2}} = f_{\text{app}} \frac{4L}{D_h}. \hspace{1cm} (1b)$$

The apparent friction factor will also be used as the performance of the various ducts are compared.

The heat transfer data are presented as Colburn heat transfer factor, $j$, versus Reynolds number. The $j$ factor is defined as

$$j = \frac{\text{St \ Pr}^{2/3}}{\text{Re} \text{Pr}^{1/3}} = \frac{\text{Nu}}{\text{Re} \text{Pr}^{1/3}}. \hspace{1cm} (3)$$

The fluid properties are determined from the arithmetic mean values of temperature and pressure upstream and downstream of the channel. The heat transfer coefficient is obtained as

$$\alpha = \frac{\rho U_m A_e c_p}{A} \ln \left( \frac{T_{\text{in}} - T_{\text{wall}}}{T_{\text{out}} - T_{\text{wall}}} \right). \hspace{1cm} (4)$$

where $A_e$ is the cross-sectional area and $A$ the heat transfer area.

4. Estimation of uncertainty

4.1. LDV measurements

The uncertainty in the presented velocity data can be derived from the inaccuracy of the channel geometries and the uncertainties connected with the LDV technique. The dimensions of the plexiglass channel are accurate within ±0.1 mm and the rib height within ±0.05 mm. The accuracy of the rib pitch is depending on the care that is taken when attaching the ribs to the template, and it is estimated to be within ±0.5 mm. The major source of error is the orientation of the probes and the positioning of the measurement volume with respect to the channel. The adjustment of the probe orientation has been performed through measuring the $V$ and $W$ velocity components on the center line of a smooth channel and turning the probes until these components are less than 0.005 m/s, which is the estimated resolution in velocity due to the statistical uncertainty at this position. However, it has not been possible to check the orientation of the probes when the smooth channel is replaced by a rib-roughened channel, since there is no point in the cross section where the flow direction is known a priori. The error in the $U$ component, which is in the order of 1 m/s, is thus about ±0.5%, while in the $V$ and $W$ components, which are in the order of 0.1 m/s in the rib-roughened channels, the corresponding error is about ±5%. In the smooth channel, the $V$ and $W$ components are in the order of 0.05 m/s close to the corners while they are less than 0.01 m/s in the rest of the channel which gives an uncertainty greater than ±10% in magnitude. Evidently also the direction of the secondary flow will suffer a large uncertainty in this region.

The position of the measurement volume has been related to the channel coordinate system through finding a corner of the channel by analyzing the signals from the photomultipliers, and the uncertainty is estimated to be within ±0.05 mm in both $y$ and $z$ directions. Assuming a Gaussian distribution and taking the rms-value as an estimate of the true standard deviation, the statistical uncertainty in the velocity measurements is given as $\Delta U = \pm 1.96u'/\sqrt{N}$ with 95% confidence, where $N$ is the number of samples. In the worst case of a turbulent flow with $u' = 0.5$ m/s and $N = 3000$, then the uncertainty is about ±0.018 m/s, which in the wall regions would correspond to an uncertainty less than ±2% in the $U$ component. The error due to the signal processing in the BSA is negligible with the settings being used.

4.2. Pressure drop and heat transfer measurements

The following uncertainties have been estimated for the components in the pressure drop test rig. The micro-manometer has been calibrated to give a maximum error of less than ±1%. An important contribution to the pressure drop uncertainty is the error due to the scatter in the pressure drop data for each least squares fit. This scatter is due to uncontrolled parameters as the exact shape of the pressure tap holes, secondary flows, etc. The maximum errors in the slope $4(f \text{Re})_{id}$ have been estimated with the maximum likelihood method and 95% confidence to be within ±5%, see e.g. [33].

The thermocouples used in the heat transfer test rig are calibrated individually, and the maximum error is
estimated to be ±0.05°C. This estimate also holds for the mercury thermometer used for measuring the water temperature. The inlet air temperature controlled by the heater varies up to ±0.5°C.

The rotameters have been calibrated and the error is less than ±1%. Corrections are also made to allow for changes in temperature and pressure during the tests. The geometric quantities of the channels have been determined within ±0.05 mm and the fluid properties are estimated to be within ±1%.

An additional error in the $j$ factor is due to finite values of the thermal resistances of heat conduction in the channel wall and forced convection on the outer side of the channel. This error is estimated to be less than −1% for laminar flow. For the highest flow rates, this error is less than −5%. The negative sign means that the true values are greater than the measured and reported ones. There may also be an underestimation of the $j$ factor due to the sealings at the channel ends. The sealings cover less than 10 mm of each channel end, which for a 500 mm long channel may give an error up to 4%.

5. Flow distribution

5.1. Smooth channel measurements

The velocity field in a smooth channel has been measured in order to check the flow quality in the LDV experiment and also to verify that small spanwise velocities can be measured with the employed LDV system. Fig. 4 shows the flow in one quadrant of the smooth channel at a Reynolds number of 5800. The vector plot shows the secondary flow field, and it is seen that the secondary motion is concentrated to the region close to the short side and the corner. The motion is directed towards the corner along a bisector and away from the corner along the walls. On the vertical symmetry line $z/a = 0$ there is a point with zero secondary velocity at approximately $y/a = 7.3$, where the downward and upward motions meet. This means that the upper and lower corner vortices are separated along a line from the corner making approximately a 35° angle to the upper wall. It has not been possible to conclude if the lower corner vortex extends all the way down to the horizontal symmetry line or if it is split into several smaller counter-rotating vortices.

The largest magnitude of the secondary velocities is about 0.07 m/s, which corresponds to 1.7% of the maximum axial velocity, and it is found on the separation line between the corner vortices. The effect of the secondary flow on the axial velocity ($U/U_m$) and axial turbulence intensity ($u'/U_m$) is shown in the two contour plots of Fig. 4. The iso-lines are bulged towards the corner since the secondary velocities are transporting fluid with high momentum and low turbulence intensity towards the corner. It is also found that there is a region with strong fluctuations close to the upper wall. In this region the secondary flow transports fluid with low momentum and high turbulence intensity away from the wall. In general the flow field is symmetric with respect to the vertical and horizontal symmetry lines. On the lines of symmetry, no secondary velocities greater than the estimated uncertainty of 0.005 m/s are measured normal to these lines, expect close to the upper wall. There, it seems like the upper vortex is situated slightly more to the right than expected for a perfectly symmetric case. This lack of symmetry is believed to be due to imperfections in the channel geometry and not due to the measurement technique.

5.2. Rib-roughened channels

The rib configurations tested are shown in Figs. 1 and 2 where arrows indicate the positions of the cross sections where the LDV measurements were performed relative to the ribs. For each geometry, measurements were only performed in one cross section at 50 hydraulic diameters from the inlet, while the smoke visualizations show the traces of the smoke at a position four hydraulic diameters downstream the smoke wire, i.e. at a position where the smoke has been convected more than one rib period in the axial direction. The pictures obtained from the LDV measurements and the smoke visualization only provide limited information on the flow in the rib-roughened channels. The smoke visualization is restricted to one Reynolds number equal to 1100, i.e. before transition to turbulent flow, and the LDV measurements
presented only contain measurement points from one cross section of the channel which might not be representative for all cross sections between two consecutive ribs. Despite these limitations, some results are obtained that are useful for the interpretation of the pressure drop and heat transfer results.

5.2.1. Cross rib-roughened channel.

Fig. 5 shows the results from the cross rib-roughened channel. It may be expected that a one cell secondary flow is set up, since the ribs turn the flow upwards along the right wall and downwards along the left wall. In the smoke visualization, the smoke wire is placed in the middle between the left and right walls, and close to the top and bottom of the channel the smoke visualization picture indicates a motion similar to the schematic pattern. However, further away from these walls the smoke visualization indicates motions in opposite direction across the mid-plane. It seems, at this specific instant, that the smoke traces have been influenced by several vortex like structures and not a stable one cell pattern.

The vector plot of the secondary mean velocity shows a flow pattern that is consistent with the schematic pattern. The motion is mainly directed along the walls according to the rib direction. A rib traverses the cross section at \( y/a = 4.2 \) on the left wall which causes the downward flow to turn away from the wall. Below the rib, the flow is turned back towards the wall, but some fluid also crosses the mid-plane and follows the upward flow along the right wall. The consequences of this motion are seen in the contour plots of axial velocity and velocity fluctuations. Clearly, the secondary flow over the rib pushes low momentum and high turbulent kinetic energy fluid from the wall towards the right hand side of the channel. Thus, the finite height of the ribs causes the real secondary flow to differ from a simple one cell pattern.

5.2.2. Parallel rib-roughened channel

Fig. 6 shows the results from the parallel rib-roughened channel. The schematic pattern is a two cell secondary flow with downward flow along the vertical walls and upward flow in the middle of the channel. In the smoke visualization picture it is seen that the smoke is compressed towards the middle at the bottom of the channel, while it is spread out towards the vertical walls at the top of the channel. This is in accordance with the schematic pattern, but as in the cross rib-roughened channel the smoke closer to the centre line has been convected horizontally, which shows that the instantaneous secondary flow is more complicated than the two cell pattern. The LDV measurements show that there is a strong upward flow in the middle of the channel which brings high momentum fluid to the top of the channel. Close to the vertical walls, fluid is moving downwards, which to some extent is detected with the finer measurement grid in the upper left quadrant. Due to mass conservation, the secondary flow velocity close to these walls have to be of larger magnitude than the velocity in the middle of the channel.

At the top of the channel, the secondary flow is from right to left instead of turning towards each of the ver-

Fig. 5. Cross rib-roughened channel. From left to right: Schematic pattern; smoke visualization at Re = 1100; secondary flow field; U velocity field; and u' velocity field at Re = 3000.
tical walls as in the schematic pattern. This is due to the location of the chosen cross section relative to the ribs. The top right corner of the cross section coincides with the end of a rib (see the arrow in Fig. 2), which will push fluid from right to left. It is expected that half a rib pitch downstream this cross section the flow will be from left to right instead, since the end of a rib will there appear at the top left corner.

5.2.3. Cross V-rib-roughened channel

The results from the cross V-rib-roughened channel are shown in Fig. 7. In contrast to the preceding channels, the visualization picture of this two cell secondary flow seems to be stable at Re = 1100, i.e. the smoke pattern is essentially the same at repeated tries. However, at Re = 1500, smoke visualization has indicated that transition to unstable flow has occurred. For this channel both the visualization and the LDV measurements show good agreement with the schematic secondary flow pattern. The axial flow is separated into the symmetric parts – one high momentum region on each side of the horizontal centre line separated by a region where the flow is from left to right. Regions of strong fluctuations coincide with regions where the secondary flow is forcing fluid away from a wall.

5.2.4. Parallel V-rib-roughened channel

Fig. 8 shows the results from the parallel V-rib-roughened channel. The schematic secondary flow is a four cell pattern where the ribs turn the flow towards the horizontal symmetry line along the vertical walls and thus cause a flow towards the top and bottom walls in the middle of the channel. In the smoke visualization, the smoke wire is placed at z/a = -0.5. As for the cross rib-roughened and parallel rib-roughened channels, the smoke visualization shows patterns that are varying significantly from one picture to another. However, the secondary flow pattern from the LDV measurements is in agreement with the schematic pattern, although somewhat affected by the position relative to the ribs. Regions of strong fluctuations are found close to the top and bottom walls where the secondary flow carries high momentum fluid towards the short walls and forces it to turn back along either the left or right vertical wall.

The main flow in Fig. 8 is in the same direction as the V-ribs (>). However, if the flow is in the direction opposite to the V-ribs (<), the secondary flow will be as in the schematic pattern of Fig. 8 with the arrows flipped 180°. Unfortunately, no smoke visualizations or LDV measurements were carried out with the flow in this direction. It may be estimated that the distribution of axial mean velocity would show one region of high momentum fluid in the centre of the channel and regions of low momentum fluid close to the short walls.

5.2.5. Swirl flow tube

Fig. 9 shows the results from the Swirl Flow Tube with multiple V-shaped ribs. In this channel, the ribs are arranged in order to create swirling motions in cells with an aspect ratio of unity. Thus, the secondary flow pattern will consist of eight longitudinal vortices according to the schematic pattern. A comparison between the
smoke traces in the visualization picture and the vector plot shows that the expected flow pattern is obtained. The smoke visualization picture is taken at $\text{Re} = 1100$.

It is obvious that small scale motions are influencing the smoke traces, but the large scale swirling motion is dominating.

Fig. 7. Cross V-rib-roughened channel. From left to right: Schematic pattern; smoke visualization at $\text{Re} = 1100$; secondary flow field; $U$ velocity field, and $u'$ velocity field at $\text{Re} = 3000$.

Fig. 8. Parallel V-rib-roughened channel. From left to right: Schematic pattern; smoke visualization at $\text{Re} = 1100$; secondary flow field; $U$ velocity field, and $u'$ velocity field at $\text{Re} = 3000$. V-ribs pointing downstream (☞).
In the contour plot of the axial mean velocity in Fig. 9 it is shown that high momentum fluid is convected either to the right or to the left by the secondary flow. In the regions of low momentum fluid, the velocity fluctuations are approximately twice the velocity fluctuations in the high momentum regions. The coarse measurement grid, that covers the whole channel cross section, has too few measurement points to resolve the details of the flow. In the data from the finer measurement grid, that covers the second vortex from the top, the time averaged flow structure becomes clear. One quarter of a rib pitch upstream of the investigated cross section, the flow has passed over ribs on the right hand wall, while ribs will appear on the left hand wall one quarter of a rib pitch further downstream. This is the reason why the flow is not symmetric with respect to the vertical centre line. It can be noted that the secondary velocities are of smaller magnitude in this channel compared to the channels previously discussed. On the other hand, the length scale of the secondary flow is smaller for this channel, since the convective motions are mainly in the horizontal direction instead of in the vertical direction as for the other channels.

6. Pressure drop and heat transfer

Fig. 10 shows the $f$ factor versus the Reynolds number for the channels investigated. The analytical solution for fully developed laminar flow in a 1–8 rectangular channel is that $f \, \text{Re}$ equals 20.6, see [32]. The smooth channel data follow this line for Reynolds numbers below 2000. For turbulent flow, the data are compared with the Blasius relation [35]

$$f = 0.0791 \, \text{Re}^{-0.25},$$

In the region $2000 < \text{Re} < 6000$, Eq. (5) overpredicts the data; while for $\text{Re} > 6000$, the data and Blasius relation agree within the estimated uncertainty interval of ±6%.

The rib-roughened channels have somewhat greater friction factors than the smooth channel. The Swirl Flow tube data fall above the data from the other channels for $\text{Re} < 3000$. For higher Reynolds numbers, all
the rib-roughened channel data are almost on the same level, and the slope of the curves are close to zero indicating that the flow is in the fully rough regime according to the Moody chart for pipe friction, see e.g. [36]. At the highest Reynolds numbers, it is seen that the Swirl Flow Tube and the V-rib-roughened channels provide greater friction than the other rib-roughened channels. In the transition region, it can be noted that transition starts at lower Reynolds numbers for the channel with cross ribs than for the channel with parallel ribs. Similarly, transition starts earlier for the channel with cross V-ribs than for the channels with parallel V-ribs. If the data for the parallel V-rib-roughened channel are excluded, there is a good correlation between the level of the \( f \) factor in the transition region and the shape of the secondary flow pattern as presented in Figs. 5–9. A secondary flow pattern consisting of square cells corresponds to a high \( f \) factor, while a secondary flow pattern consisting of rectangular cells corresponds to a lower \( f \) factor, i.e. transition to turbulent flow occurs at higher Reynolds numbers as the width to height ratio of the secondary flow cells is increased.

Fig. 11 shows the inlet loss coefficient \( K_c \) versus the Reynolds number. The data for the different channels fall close together, which indicates that the inlet loss coefficient is not very much dependent on the rib geometry. At the lowest Reynolds numbers, \( K_c \) is close to 1.8, and for increasing Reynolds number, it drops to a minimum level in the transition region. For Reynolds numbers greater than 7000, all channels provide inlet loss coefficients in the interval 1.35–1.50. In [37], inlet loss coefficients for smooth rectangular channels with 1–9 aspect ratio were reported, and the present data for the smooth channel are in general 10% lower than the data in [37]. This difference may be due to that the edges of the copper walls at the channel inlet are slightly curved in the present investigation; while the edges were cut very sharp in the previous investigation.

Fig. 12 shows the measured \( j \) factors versus the Reynolds number. The analytical solution for fully developed laminar flow is that the Nusselt number equals 5.59, see [32]. However, the smooth channel data are average values obtained with a channel which is only 56.25 hydraulic diameters long, which is too short to avoid a significant influence of the development region. This is the reason why the smooth channel data are far above the analytical solution. The data for Reynolds numbers from 500 to 2000 could be correlated by the expression

\[
\text{Nu} = 3.83 \left( \frac{\text{Re} \cdot \text{Pr}}{L/D_h} \right)^{1/3},
\]

which is a modification of the correlation by Sieder and Tate [38] for circular tubes, where the original constant 1.86 is replaced by 3.83. At Re = 500, Pr = 0.72, and \( L/D_h = 56.25 \), Eq. (6) gives Nusselt numbers that are slightly lower than those for parallel plates, but higher than those for a 1–6 aspect ratio rectangular channel with simultaneously developing flow, see [32]. However, the Reynolds numbers above 650, Eq. (6) gives Nusselt numbers greater than those for parallel plates. In turbulent flow, the data are in good agreement with the Dittus–Boelter relation for cooling [35]

\[
\text{Nu} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3}.
\]

The \( j \) factors for the rib-roughened channels are expected to be representative for fully developed conditions. The influence of channel length on the heat transfer coefficient for a parallel rib-roughened channel with rib height/hydraulic diameter = 0.058 was investigated in [26]. It was found that a 64 \( D_h \) long channel provided the same average heat transfer coefficient as longer channels, while a 48 \( D_h \) long channel showed up to 15% higher average heat transfer coefficient at \( \text{Re} < 2000 \) and less than 5% higher for \( \text{Re} > 2000 \). It is therefore expected that the influence of the development region on the average heat transfer coefficients for the 56.25 \( D_h \) rib-roughened channels in the present investigation is small at both laminar and turbulent flow.

It can be noted that the smooth channel provides a greater \( j \) factor than some of the rib-roughened channels for the lowest Reynolds numbers due to the effect of the development region. The \( j \) factors in the transition
region do not show the same geometry dependence as the $f$ factors in Fig. 10. One explanation to this difference might be that the heater upstream of the channel, which is present only in the heat transfer test rig, induces fluctuations at the inlet which may cause the flow to become turbulent at lower Reynolds numbers and thus the influence of the rib geometry becomes weaker. Also, the cooling of the fluid may influence transition due to the change in dynamic viscosity with temperature. Since the dynamic viscosity of air increases with increasing temperature, the local Reynolds number will be lower at the inlet of the channel than at the outlet. Therefore, transition may occur at a higher average Reynolds number in the heat transfer test rig than in the pressure drop test rig, where the air temperature is constant.

The obvious conclusion from Fig. 12 is that the Swirl Flow Tube provides a significantly higher $j$ factor for $Re < 2000$, and that the parallel V-rib-roughened channel provides a significantly higher $j$ factor for $Re > 5000$. At $Re = 3000$, the difference in $j$ factor for the rib-roughened tubes is small and, notably, the $j$ factor of the Swirl Flow Tube is not higher than for the other tubes. According to the LDV measurements in Fig. 9, the secondary flow exists also at turbulent flow and influences the flow distribution in the channel. However, the magnitude of the secondary flow is fairly weak as compared to the other channels, which may be the reason why the $j$ and $f$ factors of the Swirl Flow Tube do not differ more from the other rib-roughened channels.

6.1. Performance comparison

Fig. 13(a) shows the flow area goodness factor $j/f$. The greatest $j/f$ is attained by the smooth channel. In the laminar region, the curve is at a higher level than for fully developed conditions, since the $f$ factor is influenced by the development region while the $j$ factor is representative for fully developed conditions, as the inlet loss coefficient, $K_c$, takes into account all effects of the entrance and development region. However, at turbulent flow, the level is in accordance with standard correlations.

It is interesting to note that for $Re > 3000$, all curves for the rib-roughened channels are enclosed by the curves for the parallel V-rib-roughened channels, where it provides the highest $j/f$ with the flow in the direction opposite the V-ribs ($\ll$); while it provides the lowest $j/f$ ratio, if the flow is in the same direction as the V-ribs ($\gg$). The cross rib-roughened and parallel rib-roughened channels provide higher $j/f$ than the Swirl Flow Tube and the cross V-rib-roughened channel. The Swirl Flow Tube, which provide a higher $j$ factor than the other channels for $Re < 2000$, requires an increase in pumping power of approximately the same amount as the increase in heat transfer coefficient. It might therefore be possible to design a parallel rib-roughened channel by increasing the rib height and/or decreasing the rib pitch, and then a $j$ factor and $j/f$ ratio of the same order as for the Swirl Flow Tube are achieved. However, such a channel will have high $j$ and $f$ factors also for

\[ Re > 2000. \] Thus, for a heat exchanger channel operating at a Reynolds number interval which spans both laminar and turbulent flow, and where the heat load at the lower Reynolds numbers is critical to the design, the Swirl Flow Tube will be superior compared to the other rib-roughened channels.

In Fig. 13(b) the flow area goodness factor $j/f_{app}$ is presented. It is now found, as expected, that $j/f_{app}$ is less than 0.5, which would be highest obtainable value for boundary layer type flow over a flat plate. However, the conclusion concerning the ranking of the channels investigated is not affected if Fig. 13(b) is used for the evaluation.

The volume goodness factor is presented in Fig. 14(a) and (b). The diagrams show the heat transfer coefficient versus the pumping power per unit heat transfer area. Fig. 14(b) is based on the apparent friction factor and thus the overall pressure drop. Both figures indicate that the parallel V-rib-roughened $\ll$ channel provides the highest position and is thus the most efficient one. The Swirl Flow Tube and the cross and parallel rib-roughened channels fall in an intermediate level, while the parallel V-rib-roughened $\gg$ and cross V-rib-roughened channels take the lowest positions of the rib-roughened channels.
7. Swirling flow in rectangular channels

A secondary flow in the form of longitudinal vortices is an effective mechanism for heat transfer augmentation. It has been shown that helically rib-roughened circular tubes may provide heat transfer augmentation with $j/f$ ratios close to that of a smooth channel, see e.g. [39] and Webb [40]. In [26], it was found that channels with inclined ribs provided as high heat transfer augmentation as dimpled channels but at a significantly lower pressure drop penalty. In a rib-roughened channel, two mechanisms yielding the heat transfer augmentation can be identified namely: increase of turbulent fluctuations, and convective transport of fluid between the walls and the core region. Straight ribs and dimples mainly involve the former mechanism, while inclined ribs may involve both mechanisms. It is often desirable to keep the $j/f$ ratio as high as possible, which requires that the form drag is kept small and that stagnant and recirculation zones of low velocity are avoided.

It has been found that a secondary flow is set up in all the rib-roughened channels in this investigation. The parameter that differs between the channels is the aspect ratio of a swirl cell. In the parallel rib-roughened channel, two cells with aspect ratio 1–16 are obtained. In the cross rib-roughened channel and the parallel V-rib-roughened channel, cells with aspect ratio 1–8 are obtained, but the boundaries for each cell are different. In the cross rib-roughened channel, the cell boundaries are the channel walls, while in the parallel V-rib-roughened channel, each cell has two wall boundaries and two fluid-fluid boundaries. The swirl cells of the cross V-rib-roughened channel have aspect ratio 1–4, three wall boundaries and one fluid-fluid boundary. It has been observed that at a fixed Reynolds number, the smoke visualization pattern seems to be less perturbed for the 1–4 aspect ratio cells than for the 1–8 and 1–16 aspect ratio cells. It has also been noted, although not presented in this report, that increasing the Reynolds number has a larger influence on the instantaneous flow pattern for a wide swirl cell than for a square cell.

The main objective of creating a secondary flow is to establish or improve the exchange of fluid between the core region and the wall regions, and thus increasing the temperature gradient at the walls. In a rectangular channel, the largest gradients are found in the direction normal to the wide walls. Therefore, a convective motion in this direction will be more beneficial for heat transfer enhancement than a motion parallel to the wide walls. According to this, the secondary flow in the Swirl Flow Tube would be more effective than the secondary flows in the other rib-roughened channels.

The heat transfer measurements show that the Swirl Flow Tube does provide greater heat transfer enhancement than the other channels for Reynolds numbers below 2000, i.e. in the Reynolds number range where the smoke visualization indicated a fairly stable secondary flow. All the other rib configurations provide approximately the same heat transfer enhancement even though their secondary flow patterns are expected to be different. But, from the smoke visualizations it was found that most of the flow patterns were unstable at $Re = 1100$ and, consequently, the instantaneous flow patterns might be completely different from the schematic patterns. It should be mentioned that the high $j$ factor for the Swirl Flow Tube is accompanied by a high $f$ factor, thus, the $j/f$ ratio is of the same order as for the other channels.

At Reynolds numbers above 3000, the Swirl Flow Tube provides the same heat transfer enhancement as the parallel rib-roughened channel despite the large difference between the expected secondary flow patterns. This fact may be interpreted as that the mixing due to the secondary flow is of minor importance for the heat transfer enhancement. However, the secondary flow has a strong influence on the axial flow distribution in the channel cross section as seen in Figs. 5–9. The parallel V-rib roughened channels provide significantly different $f$ factors depending on the flow direction over the ribs. If the V-ribs are pointing upstream ($\ll$), the $f$ factor will be greater than if the V-ribs are pointing downstream ($\gg$). At the highest Reynolds numbers, the difference is about 25%. However, no similar behaviour is seen in the $f$ factor data of Fig. 10. The secondary flow for the case with V-ribs pointing downstream will
concentrate high momentum fluid into two regions, one close to each short wall. At the center of the channel, there is a region of low momentum fluid with intense fluctuations. It may therefore be assumed that the local heat transfer coefficient will be larger close to the short walls and smaller at the centre of the wide walls, see [4]. For the case with V-ribs pointing upstream, the secondary flow will be reversed and thus create a region with high momentum fluid associated with large local heat transfer coefficients at the centre of the channel, while regions with low momentum and intense fluctuations will occur close to the short walls. It seems likely that, for a given Reynolds number, the high momentum region in the latter case will involve higher velocities than the high momentum regions of the former case, since all four vortices will act together transporting high momentum fluid towards the centre line in the latter case, while in the former case, two vortices will transport high momentum fluid towards one of the short walls and the other two vortices will transport high momentum fluid towards the other short wall. According to the measurements, the average isothermal shear stress is approximately the same for both cases, while the average heat transfer coefficient is higher for the latter case. This is in agreement with the data of Han et al. [3] at $Re = 14200$.

To get a further understanding of the differences in flow characteristics between the channel with V-ribs pointing upstream (a) and the channel with V-ribs pointing downstream (b), it is useful to study a vortex line close to a wall with V-ribs, see Fig. 15. The vortex line is bent to a V-shape similar to the ribs if the vortex line is close to the wall. Due to the existing velocity gradients close to the rib, the vortex line will be stretched and vorticity amplified. Consequently, the vortex line now has one axial and one spanwise vorticity component. The axial component is associated with the secondary flow discussed previously, while the spanwise components have similar behavior as the original vortex line. For the case of Fig. 15(a), the axial vorticity components will act as an inflow pair of vortices resulting in thinning of the boundary layer and as an effect the heat transfer is enhanced. Self-propagation of the axial vorticity components is directed towards the wall which tends to keep the strong vorticity close to the wall. It seems likely that this vortex interaction increases the local heat transfer and shear stress. In Fig. 15(b), also vortex stretching may occur, but as the vortex line is bent, the axial vorticity components act differently compared to Fig. 15(a). Now, an outflow vortex pair occurs and the boundary layer becomes thicker. Self-propagation of the axial vorticity components is, for this case, directed away from the wall which tends to transport vorticity away from the wall and, thereby, the local heat transfer and shear stress are decreased. This simplified way of considering the flow is not valid close to the short walls. However, in a channel with large width to height ratio, the influence of the short walls may be of importance only in small fractions of the cross section.

8. Practical significance

In many heat exchanger applications, e.g. radiators and charge air coolers, there is a great need to improve the performance of the heat exchangers in order to meet demands of larger heat load, smaller size etc. However, more knowledge is needed concerning the mechanisms of the fluid motion in roughened channels to enable significant improvements. This paper brings about information of the flow structure, pressure drop, and heat transfer in some rectangular rib-roughened channels which may be useful for design engineers and researchers who are interested in heat transfer augmentation and its application.

9. Conclusions

Smoke wire visualizations at $Re = 1100$ and LDV measurements at $Re = 3000$ have been presented for rib-roughened channels with parallel, crossed, V-shaped, and multiple V-shaped ribs (Swirl Flow Tube) together with thermal and hydraulic performance for Reynolds numbers from 500 to 15 000. The following conclusions have been found:

1. The Swirl Flow Tube provides a significant increase in the $j$ and $f$ factors for Reynolds numbers from 1000 and 2000, while at higher Reynolds numbers these are at the same level as those of most of the other rib-roughened channels. The high heat transfer enhance-
ment in the low Reynolds number range is caused by a stable secondary flow.

2. The flow direction in a channel with parallel V-ribs is important. At Reynolds numbers above 4000, the highest \( j/f \) ratio of all channels tested is provided with the V-ribs pointing upstream, while the lowest \( j/f \) ratio is provided with the V-ribs pointing downstream. Suggestions to explain this difference are provided based on the measured flow distributions and consideration of the vorticity mechanisms in these channels.

The flow pictures presented are of a qualitative nature. The smoke visualizations of unstable flow fields do not provide general information, but merely indications of the stability of the flow. The LDV measurements provide data of the average flow situation in one cross section of each channel, while measurements in the whole region between two consecutive ribs need to be carried out in order to give a complete description of the flow. It is concluded that, among the geometries investigated, further work should be concentrated on the Swirl Flow Tube and on the channel with V-ribs pointing upstream.

Nomenclature

- **heat transfer area, \( A \)**: \( m^2 \)
- **half the distance between the wide channel walls, \( a \)**: \( m (0.0145/2 \text{ and } 0.005/2) \)
- **channel cross sectional area, \( A_c \)**: \( m^2 \)
- **specific heat, \( c_p \)**: \( J/Kg K \)
- **hydraulic diameter, \( D_h \)**: \( m \)
- **pumping power per unit heat transfer area, \( E \)**: \( W/m^2 \)
- **Fanning friction factor, dimensionless \( f \)**
- **Colburn heat transfer factor, \( j \)**
- **inlet loss coefficient, \( K_c \)**: \( \text{dimensionless} \)
- **channel length, \( L \)**: \( m \)
- **number of samples, dimensionless \( N \)**
- **average Nusselt number \( (\lambda A/f D_h) \)**: \( \text{dimensionless} \)
- **Prandtl number \( (U_m D_h/c_p) \)**: \( \text{dimensionless} \)
- **Reynolds number \( (U_m D_h/\nu) \)**: \( \text{dimensionless} \)
- **Stanton number \( (Nu/Re Pr) \)**: \( \text{dimensionless} \)
- **temperature, \( T \)**: \( K \)
- **bulk mean velocity, \( U_m \)**: \( m/s \)
- **local axial velocity, \( U \)**: \( m/s \)
- **root-mean-square value of axial velocity fluctuations, \( U' \)**: \( m/s \)
- **vertical velocity component, \( V \)**: \( m/s \)
- **horizontal velocity component, \( W \)**: \( m/s \)
- **coordinate in axial flow direction, \( x \)**: \( m \)
- **vertical coordinate, \( y \)**: \( m \)
- **horizontal coordinate, \( z \)**: \( m \)

Greek symbols

- **heat transfer coefficient, \( \lambda \)**: \( W/m^2 K \)
- **kinematic viscosity, \( \nu \)**: \( m^2/s \)
- **density, \( \rho \)**: \( kg/m^3 \)

Subscripts

- **fd** refers to fully developed conditions
- **in** at the channel inlet
- **out** at the channel outlet
- **wall** at the channel wall

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References


