

Heat transfer characteristics in forced convection through a rectangular channel with 60° tilted staggered ribs

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Abstract. In this paper we present new experimental results of investigation on average heat transfer characteristics of a forced air-flow through a rectangular channel with the lower and upper surfaces roughened by ribs; data for a rectangular channel with flat surfaces are presented for comparison as well. The channel cross-section is 120 mm wide and 12 mm high; the channel is operated with the lower and upper walls kept at fixed temperature whereas the sides are adiabatic. The ribs have a square cross section and are mounted 60° parallel-tilted (the angle is respect to main stream) in a staggered arrangement. The tested configurations differ each other for the rib side dimension, namely, 2 or 4 mm, and for their pitch-to-side ratio equal to 10, 20 and 40. Upstream the test channel, there is an entry-section consisting of a 800 mm long, rectangular duct with the same transverse dimensions as the test section but with flat and adiabatic walls. Air flow rates have been varied in order to have Reynolds numbers, based on the duct hydraulic diameter, ranging between 700 and 7500. The average Nusselt numbers are evaluated on the basis of the air-flow bulk-temperature at entrance and exit from the heated zone, as well as of the surface temperature measured by eight T-type thermocouples plugged into the heated walls. The test section is also equipped with static pressure taps placed at the heated zone ends. Results show an increase of the average Nusselt number, calculated as the ratio Nu/Nu_0 , for the all tested ribbed channels ranging between 1.0 and 5.0.

Introduction

Heat transfer in forced convection inside rectangular channels is a very interesting matter for industry as it is encountered in critical heat transfer applications like gas-turbine blade cooling [1], and in devices largely used such as plate-fin compact heat exchangers. In designing these devices, high values of heat transfer area per unit volume are searched for; however, if this parameter is increased over a given value, thermal performances start worsening. In fact, the higher the surface-to-volume ratio the narrower the passages, so air velocity has to be lowered to maintain acceptable the pressure drops; however, narrow passages and low air velocities lead to laminar flows laminarization that is of course characterized by a rather poor convective coefficient which eventually defeats completely the area increase benefits. To overcome this limit, heat transfer is enhanced by configuring surfaces with a large variety of fins and ribs, which are an efficient and cost-effective solution. The literature review shows that, starting from the '70s, a continuously growing number of studies have been dealing with heat transfer over rib-roughened surfaces, as this is a very attractive solution for both gas-turbine blade cooling and compact heat exchangers.

A first crucial feature for this kind of roughness is the rib arrangement with respect to the main stream. The number of possible configurations, indeed, is very large but four arrangements have been



particularly considered so far in literature, namely, transverse-, parallel-tilted-, cross-inclined- and V-shaped-ribs; in addition, ribs may be assembled on the opposite walls either staggered or in-line. In this paper, we focus particularly on the parallel-tilted and cross-tilted arrangements. A brief discussion of relevant works in literature is presented below making use of the following dimensionless geometric parameters: the channel aspect ratio AR , i.e., the base to height ratio (making reference to the channel cross section), the dimensionless pitch p/e , i.e., the pitch to rib height ratio, and the blockage factor e/D_h defined as the rib height to the channel hydraulic diameter ratio.

The first major studies were carried out starting from the first half of the '80s by Han [2-7], Park [8], and Kukreja [9]. It was observed that tilting the ribs with respect to the main flow direction results in higher heat transfer performance due to the presence of secondary flows. Actually, as a consequence of inclination, two counter-rotating vortices develop near the rib and slide along it. They interact with the main flow carrying part of the colder core towards the wall. Furthermore, the interaction between main and secondary flows affects the reattachment point, and hence the recirculation-zone between two successive ribs. Specifically, a great deal of work was devoted to the case with a tilt angle $\alpha = 45^\circ$; it was found that the optimum dimensionless pitch p/e strongly depends on both e/D_h and AR . The most studied configuration presents $AR=1$ with all the sides heated.

Among the recent studies, there are the papers of Liu et al. [10] [11], Choi et al. [12], and Smulsky et al. [13]. Liu et al. [10] [11] carried out experiments on heat transfer characteristics in steam-cooled rectangular channels with parallel-tilted ribs on the opposite walls, for two tilt angles, i.e., 45° and 60° , several aspect ratios within 0.5 and 2.0, and Reynolds number ranging from 10000 to 140000. They found that thermal performances for $\alpha=45^\circ$ are 15-25% larger than for $\alpha=60^\circ$. They also showed that heat transfer coefficients for steam are 12-25% higher than for air.

Choi et al. [12] studied the local heat transfer characteristics on two kinds of surfaces, namely, smooth or dimpled, with parallel-tilted ribs ($\alpha = 60^\circ$). They found that ribs are more effective than dimples in enhancing heat transfer. Finally, Smulsky et al. [13] investigated the effect of different tilt angles of a single rib on a flat surface. The local heat transfer maximum for 50° is approximately 40% higher than that for 90° , while its position depends on rib side dimension.

Finally, in the last decade, rectangular channels with only one heated, ribbed surface have aroused great interest for applications in renewable energy technologies as solar air heaters. Reviews on this subject are reported in Bhushan and Singh [15], and Hans et al. [16] [17].

A closer look reveals that most of the work done up to now focused on fully turbulent flow regimes. Moreover, two aspects are very little investigated, namely, the impact of high values of the blockage factor, and the laminar-to-turbulent flow regime transition. Both these aspects are very relevant in designing compact heat exchangers, as well shown by the specific literature [18]. For this reason we started an experimental campaign aimed at investigating heat transfer coefficients and pressure drops in rectangular ducts with variously arranged square ribs, in a range of Reynolds numbers between 600 and 8000. In these conditions, it is possible to explore both the turbulence onset and the transition region. In this paper we present a selection of the available results while experimental runs are under completion.

Experimental setup

As schematically shown in figure 1(a), the experimental setup consists of two independent circuits, namely, the air-circuit containing the test section, and the heating-water circuit used to control the temperature of the channel walls.



Figure 1. Schematics of the experimental setup: 1. fine mesh screen cover; 2. convergent air inlet; 3. entry-section; 4. test-section; 5. exit-section; 6. rotameters; 7. by-pass; 8. blower; 9. heat bath; 10. water circuit piping (a), test section with the duct on the wall backside (b)

Through a convergent, room air flows into the circuit; a fine mesh screen covers the convergent inlet whereas at its outlet there is a flow straightener consisting in a matrix of staggered 4-mm-diameter, 40-mm long thin polystyrene tubes. Inside the convergent a thermo-resistance, i.e., a resistance thermometer, is mounted to measure the air temperature. From the straightener air flows into an entry-section that is a rectangular duct with the same cross dimensions as the tested channel; entry-section is 0.8-m long, i.e., near 40 times the channel hydraulic diameter, and its walls are made with 10-mm thick plexiglas plates and are not heated. At the end of this section, air enters the test-section that is a 120-mm wide, 12-mm height, 880-mm long duct; its lower and upper walls are two aluminium plates of 10-mm minimum thickness with a rib configured face.

As shown in figure 1(b), the backside of each plate is covered by a cap strongly tightened to the plate, so that they form a jacket where the heating water flows. Inside and outside the water jacket there are ribs which prevent the plate buckling. To check if temperature is uniform over the heated walls, four thermocouples are embedded in the lower wall and three in the upper one; each thermocouple is cemented into a 1.8-mm-wide, 0.5-mm-depth groove cut in the flat face; thermocouple locations are displayed in figure 2 whereas table 1 lists the exact position of their junctions. Eventually, the test-section sides are closed by 4-mm-thick glass plates to allow optical access inside the channel.

Table 1. Probe positions on ribbed channel wall.

Upper wall		Lower wall	
x (mm)	y (mm)	x (mm)	y (mm)
120	60	70	60
440	60	120	60
760	60	440	60
		760	60

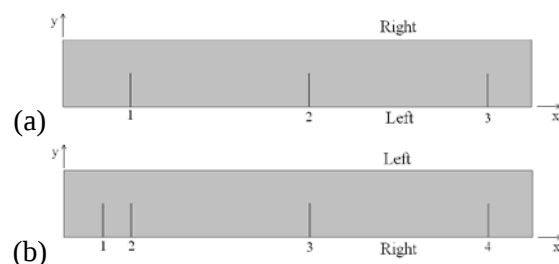


Figure 2. Thermocouples locations on upper (a) and lower (b) test channel walls.

At the test-section outlet, there is a short exit-section equipped with two turbolizer rows followed by a convergent that conveys air through a 20-mm-wide, 6-mm-high channel partially filled by a fine-meshed plastic net; after the screen, a PTR and a thermocouple are positioned on the centreline of this channel to measure the air bulk-temperature. Finally, inside the couplings between the entry-section and the test-section, and between this one and the exit-section there are two pressure taps, each consisting in a 1.5-mm-diameter hole (great care was devoted in eliminating any burr). Downstream the exit-section there are three rotameters, i.e., float-type flow-meters, connected in parallel with full

scale of 6, 23.5 and 40 m³/h respectively, a metering valve, and a 7-stage, 30-kPa-head, 5.5-kW-power blower operating in suction mode. The exhausted air is discharged outside the laboratory. Air temperature is also measured upstream the flow-meters by means of a thermocouple plugged in the pipe.

The heating circuit is mainly composed by a heat bath which provides high mass flow-rate of water at constant temperature, and by the channels built into the upper and lower test-section walls (water and air stream in counterflow); two thermocouples are placed inside each water channel, near the inlet and outlet ports, respectively. The heat bath is the ThermoHaake B12 with a tank of 12-dm³, a 3-kW heater and a high precision controller; water temperature inside the tank is kept constant within 0.01 K.

Measurements, data processing and error analysis

The thermocouples used are T-type with 0.5-mm-diameter wires, whereas the PTR are 4-wires, 100-ohm, Platinum type, i.e., PT100, with dimensions of 2 mm x 4 mm. Both thermocouples and thermo-resistances were all preliminary calibrated over five points within the temperature range from 22 to 60 °C, by means of the ThermoHaake heat bath. The probes were immersed all together into the bath while devoting great care in their positioning; for each calibration point, 160 readings per probe were collected; the resulting standard deviation is of 0.02 K for the thermocouples, and of 0.01 K for the thermo-resistances. All temperature measurements are performed by means of an Agilent 34970A data logger equipped with a relay multiplexer and a 6½ digit multimeter. The channels are sequentially read, by waiting a 0.5-s settling time after each channel-locking, with an integration time of 400 ms which ensures a standard deviation of 0.01 K that is less than or equal to the probe uncertainties; consequently, reading cycles are performed every 20 s. Pressure drops are measured by connecting the two pressure taps, placed at the inlet and outlet of the test channel, to a differential micromanometer with a full scale of 250 Pa and a 0.125-Pa sensitivity. Air volume-flow-rate is measured by means of the aforementioned rotameters which have a 2% nominal accuracy; however, by means of a calibration performed by measuring pressure drops of laminar air-flows through a smooth circular tube, we found that their accuracy is better of 1%. Finally, in order to guarantee repeatability to measurements, we adopted a precise test procedures described in the following. First, we power the bath heater and water starts to circulate through the entire circuit included the ducts on the test-section wall backside. After 30 m, that is the time to stabilize the channel wall temperature to a prefixed value within a 0.01-K band, we power the blower and air starts to flow through the test section. Then we need to wait for other 30 m in order to attain regime conditions with all temperature time-fluctuations within 0.02-K band; in this conditions, channel wall-temperature is uniform within 0.1 K over the entire heated length. Eventually, we start to collect 20 reading-cycles which take 400 s; during this time, measurements of the air-volume-flow-rate and of reading-cycles was chosen first by collecting data for N=200 for some of the most representative regime conditions, and then by calculating their average standard deviation as a function of N; as a result, we found that at N=20 the average standard deviation becomes less than 0.01 K, and thus we assumed that 20 is the minimum reading-cycle number to be collected.

The average Nusselt over the test section and the apparent Darcy friction factor are calculated as follows

$$Nu = \frac{D_h}{k} \frac{\rho \dot{V} c_p}{A_s} \ln \frac{\theta_i}{\theta_o} \quad (1)$$

$$f = \frac{2 \Delta p A^2 D_h}{\rho \dot{V}^2 \ell_{taps}} \quad (2)$$

Where $\dot{V}$ the air volume-flow-rate, ρ the density calculated at the temperature where $\dot{V}$ is measured, c_p the specific heat at constant pressure, k the thermal conductivity, A_s the total heated area, θ_i and θ_o the wall-to-air-bulk-temperature difference at the test section inlet and outlet respectively, D_h the hydraulic diameter, Δp the pressure drop, and ℓ_{taps} the distance between the pressure taps.

The error analysis performed according to Moffat [19] gives an uncertainty less than 3% on the convective coefficient, and less than 7% on the apparent Darcy friction factor

Results

Experimental results here presented refer to a rectangular channel with parallel-tilted ribs on the lower and upper surfaces in a staggered arrangement, as schematically shown in figure 3. Ribs have a square-cross section and are made of linden wood, i.e., a material with a very low thermal conductivity, to minimize their fin effect on heat transfer. Tests have been carried out on many configurations differing for the rib side dimension and the pitch-to-side ratio as listed in table 2, whereas channel main dimensions are listed in table 3.

All data are for fixed and uniform wall-channel-temperature at the nominal value of 40°C, and for air-flows entering into the test-section at room temperature of near 22°C (elaborations, indeed, account for their actual values).

Figure 4(a) displays the average Nusselt number plotted versus the Reynolds number for all the tested configurations. Values obtained for the rectangular channel with flat surfaces, operated at the same conditions, are also plotted for comparison.

For the smooth rectangular-channel, the average Nusselt number remains nearly a constant for Reynolds numbers up to 2100, and then it starts to increase exhibiting a power-law dependence on Re with an exponent of about 0.8. The figures also reports as solid lines the values calculated for a flat rectangular-channel by the Shah and London correlation with Wibulsas correction for $Re < 2300$, and by the Gnielinski correlation for $Re \geq 2300$ [20]; all predictions take into account thermal effects of the entry-region. As it can be seen, present data agree very well with predictions.

Table 2. Configurations parameters.

e [mm]	p [mm]	e/D_h	p/e	e/h
2	20	0.0917	10	0.1667
2	40	0.0917	20	0.1667
2	80	0.0917	40	0.1667
4	40	0.1833	10	0.3333
4	80	0.1833	20	0.3333
4	160	0.1833	40	0.3333

Table 3. Main channel parameters.

Channel height, h	12.00 mm
Channel length, L	880.00 mm
Channel width, w	120.00 mm
Hydraulic diameter, D_h	21.82 mm
Height to width ratio	0.10

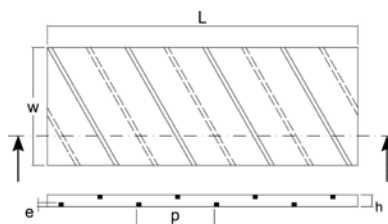


Figure 3. Tilted parallel staggered configuration

For all the tested ribbed configurations, however, the average Nusselt number displays an increasing trend even from the minimum value of the investigated Reynolds numbers, revealing a power-law dependence on Re yet with an exponent ranging between 0.80 and 0.95. The lack of a constant Nu region induces to conjecture that flow becomes turbulent at $Re < 600$ in these channels; however, no comparison with other results in the open literature is possible because seemingly there are not data for this kind of geometries at $Re < 2000$.

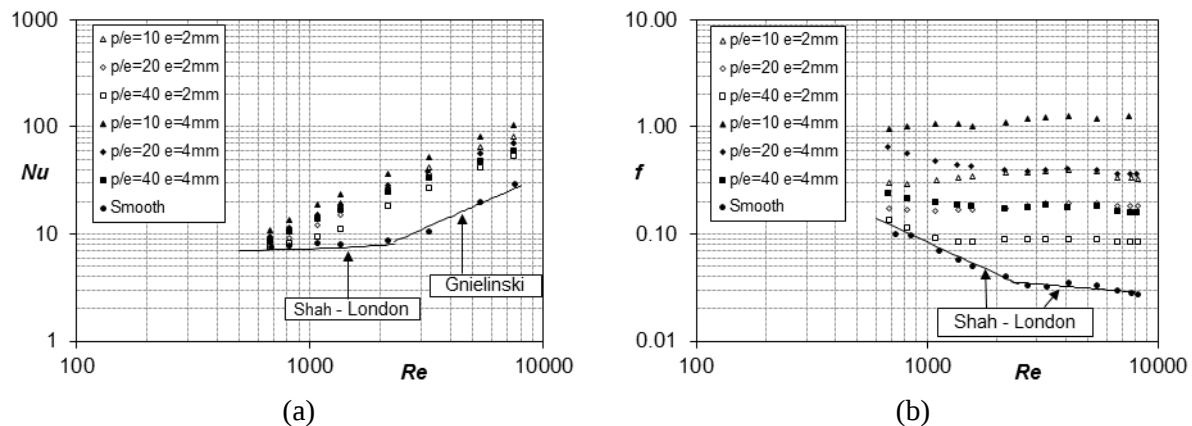


Figure 4. Average Nusselt number (a) and Darcy friction factor (b) versus Reynolds number

Among the tested configurations here reported, the highest heat transfer increase is for the $p/e=10$ and $e=4\text{mm}$, since it displays a mean value of the ratio Nu/Nu_0 equal to 3.2 (Nu_0 refers to the flat channel at the same operating conditions) with a maximum of 5.0 at $Re=3250$; whereas the configuration with $p/e=40$ and $e=2\text{mm}$ is the less performing one being characterized by a mean Nu/Nu_0 ratio equal to 1.6. The Nusselt number ratio and the friction factor ratio decrease by increasing the pitch-to-side ratio, whereas they increase by increasing the rib side (or blockage values).

Figure 4b shows the experimental values of the apparent Darcy friction factor f plotted versus Reynolds number for all the tested configurations. For comparison, the figure also reports as solid lines the values f_0 calculated for a laminar and turbulent flow through a flat rectangular channel by means of Shah and London correlation [20]. From this figure, it is evident that f is substantially higher than f_0 for all configurations; more precisely, the worst configuration is for $p/e=10$ and $e=4\text{mm}$, as it displays a mean value of f 52 times larger than f_0 , but f/f_0 is reduced to 2.2 for $p/e=40$ and $e=2\text{mm}$. These data display the typical trend of flows through a duct with walls characterized by an equivalent sand roughness, namely, the friction factor becomes independent of Re for sufficiently high values. It is noteworthy that at the highest of the two blockage values, corresponding to a rib height of 4 mm, the flow appears to be turbulent at any Re ; at the lowest blockage, this behavior occurs for $p/e=10$ and 20, while the turbulence onset shifting to higher Re for $p/e=40$.

As a concluding remark, from an engineering point of view, it is interesting to compare performance of different ribbed channels, and indeed there are currently several criteria to carry out this comparison. The Criterion we used compares performances by taking into account at the same time both the heat transfer enhancement (Nu/Nu_0) and the pressure-drop penalization (f/f_0) at constant pumping power

$$\frac{Nu/Nu_0}{(f/f_0)^{1/3}} \quad (3)$$

Figure 11 shows a graph where the values calculated by this Criterion is plotted versus the Reynolds number for all the tested configurations. According to this Criterion, the highest value is 1.9 obtained at $Re=3250$ for the configuration with $p/e=20$ and $e=2\text{mm}$. Generally speaking, the most efficient configuration is for $p/e=40$ and $e=4\text{mm}$ at low Reynolds number, and for $p/e=20$ and $e=2\text{mm}$ at high one.

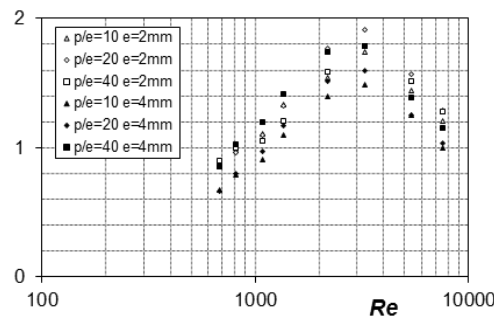


Figure 11. Tilted staggered ribs comparison at constant pumping power.

Conclusions

This paper reports about an experimental investigation on heat transfer characteristics of a forced air-flow through a rectangular channel with the lower and upper surfaces roughened by ribs with different configurations, at Reynolds numbers ranging between 700 and 7500. The enhancement effect is essentially due to the periodic streamline deflection induced by the ribs. Configurations differ each other for rib side, namely, 2 and 4 mm, and for the pitch-to-rib-side ratio equal to 10, 20 and 40. The Nusselt number averaged over the entire channel, and the Darcy friction factor are evaluated for each configuration and they have been plotted as a function of the Reynolds number; experimental results for a plate channel are also reported as reference values.

Heat transfer measurements show for all configurations a power-law dependence on Reynolds number with an exponent ranging between 0.80 and 0.95. As there is not a region at constant Nusselt number, we conjecture that flow is turbulent even from the lowest investigated values of the Reynolds number. Among the tested configurations, the one with $p/e=10$ and $e=4\text{mm}$ shows the maximum thermal improvement with respect to the flat channel, and in fact it is characterized by a mean enhancement factor of 3.2, with a maximum of 5.0 at $Re=3250$, but in spite to this increment the apparent Darcy friction factor worsens of nearly 27.5 times. In order to compare performances by taking into account at the same time both heat transfer enhancement and pressure-drop penalization, we used a comparison criterion based on a fixed pumping power. The configuration with $p/e=20$ and $e=2\text{mm}$ displays the better performances with a value of 1.9 at $Re=3250$.

Nomenclature

Symbol	Quantity	Unit
A_S	Total heated area	m^2
AR	Channel aspect ratio	-
D_h	Hydraulic diameter	m
e	Rib side dimension	mm
$\dot{V}$	Volume flow rate	$\text{m}^3 \text{s}^{-1}$
c_P	Specific heat	$\text{J kg}^{-1} \text{K}^{-1}$
f	Darcy-Weisbach friction factor	-
h	Channel height	mm
k	Thermal conductivity	$\text{W K}^{-1} \text{m}^{-1}$
ℓ_{taps}	Distance between pressure taps	m
p	Ribs distance	mm
Δp	Pressure drop	Pa
Nu	Nusselt number	-
w	Channel width	mm

Greek symbols

α	Rib angle respect to main stream	deg
θ	Wall-to-air-bulk-temperature difference	°C
ρ	Density	kg m ⁻³

Subscripts

i	Inlet
o	Outlet

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