Characteristics of Heat Transfer in the Ribbed Rectangular Channel with Variable Heating Condition

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ABSTRACT: Surface heat transfer of a fully developed turbulent air flow in a 45° inclined ribbed square duct with two and four heating walls was experimentally investigated, at which the experimental works were performed for Reynolds numbers ranging from 7,600 to 24,900. The pitch-to-rib height ratio, p/e was kept at 8 and rib-height-to-channel hydraulic diameter ratio, e/D_h was kept at 0.0667. The channel length-to-hydraulic diameter ratio, L/D_h was 60. The heat transfer coefficient values were decreased with the increase in the number of heating walls. Results of this investigation could be used in various applications of internal channel turbulent flow involving roughened walls.

Nomenclature

A : heat transfer area [m^2]
D_h : hydraulic diameter [m]
e : roughness height [m]
h : heat transfer coefficient [W/m^2℃]
k : thermal conductivity [W/m^2℃]
Nu : Nusselt number, hD_h/k
p : roughness pitch [m]
$\dot{Q}$: heat transfer rate [W]
Re : Reynolds number, $\rho D_h u_b/\mu$
x : distance from entrance [m]

Subscripts

1 : inlet
2 : exit
ro : ribbed channel
sm : smooth channel
st : Sieder and Tate's correlation

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1. Introduction

Surface ribs can break up the viscous sublayer of a flow and promote local wall turbulence, which, in turn, enhances the heat transfer from the rib-roughened heated wall. In addition, a rib-roughened wall provides a greater surface area for heat transfer compared to a plain wall. When cooling gas turbine airfoils, rib turbulators are usually cast on two opposite sides of the cooling channel, since the heat transfer takes place from both the inner walls of the pressure and suction sides of the blade. In some applications for electronic equipment, heat exchangers, and nuclear reactors, rib turbulators are formed on one, two, three, or all four sides of the cooling channel. Han et al.[1-5] provided state-of-the-art reviews of turbine blade cooling and analyses of the heat transfer and friction characteristics in channels with two opposite ribbed walls. The effects of Reynolds number and rib geometry (rib height, rib spacing, rib angle-of-attack, and rib configuration) on the heat transfer and pressure drop in the fully developed region of uniformly heated square channels with two opposite ribbed walls have been investigated by Han et al.[1-2] Further studies of the combined effects of rib geometry and channel aspect ratio on the local values of the heat transfer and pressure drop have also been reported by Han et al.[3-5] These results show that angled ribs provide better heat transfer performance than transverse ribs, and that lower aspect ratio channels perform better than higher aspect ratio channels. Choi et al.[6] investigated convective heat/mass transfer characteristics and pressure drops in a square duct with ∧- and ∨-shaped rectangular ribs. They demonstrated that ∧-shaped ribs had higher heat/mass transfer coefficients than ∨-shaped ribs. Rhee et al.[7] examined the effect of the rib arrangement on an impingement/effusion cooling system with the initial cross-flow and showed that the average heat transfer coefficient with rib turbulators was approximately 10% greater than that measured without ribs. Higher values were obtained with smaller rib pitches. Zhang et al.[8] investigated the effect of the surface heating conditions on heat transfer enhancement in square channels with semicircular wavy tape, hemi-triangular wavy tape, and twisted tape inserts. Chandra et al.[9] did an experimental study of wall heat transfer and friction characteristics of fully developed turbulent air flow in a rectangular channel with transverse ribs on one, two, and four walls. Tests were performed for Reynolds numbers ranging from 10,000 to 80,000. The pitch-to-rib height ratio, p/e was kept at 8. They reported that the heat transfer coefficient and friction factor values were enhanced with the increase in the number of transverse ribbed walls. One objective of this study was to determine the effects of the number of heated walls on the wall friction and heat transfer coefficients in a ribbed square channel. In the past, the majority of investigations have considered turbine blade cooling in channels with transverse ribbed walls. However, there is a need to examine the effects of heated walls in a ribbed rectangular channels in heat exchanger systems.

This study extends the existing knowledge base by providing experimental data for square channels with one ribbed wall. The square channel was heated on either two or four walls, and one wall was roughened with ribs inclined at 45°. Tests were performed for Reynolds numbers (Re) ranging from 7,600 to 24,900, a pitch-to-rib-height ratio (p/e) of 8.0, and a rib-height-to-channel hydraulic diameter ratio (e/D_h) of 0.0667. The results of this investigation can be used in various applications, such as turbine blade internal cooling systems, electronic equipment, heat exchangers, and nuclear reactors.

2. Experimental apparatus and test procedure

Figure 1 shows a schematic diagram of the test apparatus. A 195 W blower delivered laboratory air through a 8.16 cm diameter flexible

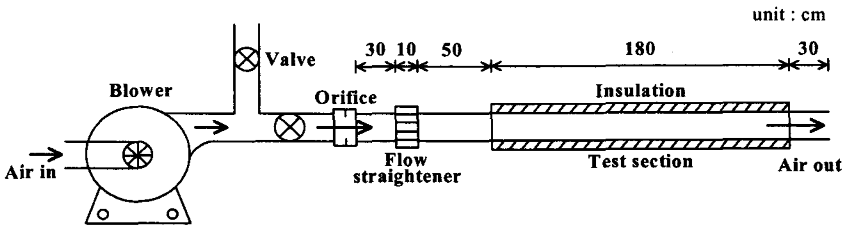
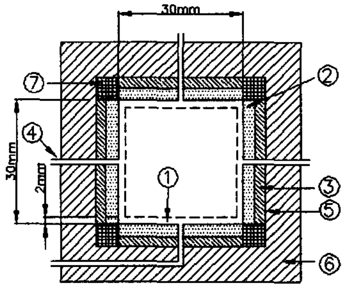


Fig. 1 Schematic of experimental setup.



① Rib type roughness (height: 2 mm)
② Al plate (thickness: 5 mm)
③ Woven heater (Omego, USA)
④ Pressure tap
⑤ Fiber gasket (thickness: 0.8 mm)
⑥ Pine wood
⑦ Insulation (gypsun)

Fig. 2 Details of test section.

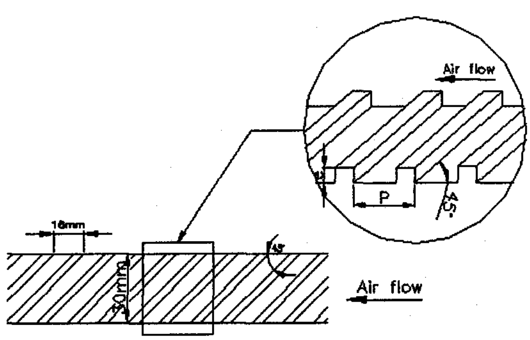


Fig. 3 Rib arrangement on bottom wall.

tube at room temperature. The air then passed through an orifice to measure flow rates, through a long straight square channel with a hydraulic diameter of D_h=3.0 cm and length of L_h=90 cm, and finally through the heated square test channel with D_h=3.0 cm and length L=180 cm. The entrance section was long enough to ensure a hydrodynamically fully developed flow just before the heated test channel. The air was exhausted into the atmosphere at the end of the test channel. The test channel is shown in Fig. 2. It was constructed of four 0.5-cm-thick aluminum plates, and it had a cross-section of 3.0×3.0 cm. The aluminum plates were separated by 0.08-cm-thick fiber to reduce the conduction heat loss and to obtain an average heat transfer coefficient. Square sharp-edged aluminum ribs with a height of e=0.2 cm (e/D_h =0.067) and an equal spacing of p=1.6 cm (p/e =8) were glued with silicone adhesive onto the channel walls. As shown in Fig. 3, the ribs were positioned with an inclination angle of 45°. The entire test section was covered tightly with 5-cm-thick wood insulation. The channel walls were heated individually with electric woven heaters that were embedded and placed flat between the aluminum and wood plates to ensure good thermal contact. A transformer controlled the heaters independently so that a constant heat flux thermal boundary condition was created on the channel walls. Thermocouples embedded on the walls were used to measure the rib spanwise surface temperature at one downstream location in the test section to examine the temperature uniformity along the rib surface. The thermocouple beads were carefully embedded into the wall and then ground flat to ensure that they were flush with the surface. Wall temperatures along the axial direction of the four channel walls were measured using 40 copper constantan thermocouples. The temperature of the air entering and leaving the test channel was also recorded. Ten temperature positions, which were transversed vertically, were used to measure bulk mean air temperature entering the test channel and leaving the test channel, respectively. All of thermocouples used in the experiment were carefully calibrated to an accuracy of 0.15℃. The Reynolds numbers based on the channel hydraulic diameter (D_h) ranged from 7,600 to 24,900. A 8.16 cm dia-pipe, equipped with an orifice plate, was used to measure mass flow rate. The airflow was varied to obtain the desired Reynolds number.

The air bulk velocity in the test section was obtained from the measurements in the smooth tube before the test section. The power supplied to the heaters was adjusted such that the wall temperatures at a particular cross-sectional

Table 1 Experimental work

Class	Details of test section	Cross-sectional area of test section	30 mm× 30 mm
		Aspect ratio (W/H)	1
		Length of test section	1,800 mm
		Roughness pitch to height ratio (p/e)	8
		Attack angle	45°
		Roughness height to hydraulic dia. ratio (e/D_h)	0.0667
		Material of duct	Al
	Conditions	Working fluid	Air
		Ambient temperature	24~28℃
		Reynolds number range	7,600~ 24,900

location were the same by controlling the supplying voltage and checking out 4 different wall temperatures in the channel, and the difference between the wall temperature at the end of the channel and the exit air temperature was around 15℃. Over the range of test conditions of Table 1, the room temperature was kept between 24 and 28℃. When a thermal steady-state condition was reached in the test channel, the inlet and exit air temperatures, atmospheric temperature, and temperatures along the channel walls were recorded. The upstream pressure, pressure drop across the orifice, atmospheric pressure, and pressure drop over a specified channel length were also recorded.

The heat transfer coefficient was calculated from the net heat transfer rate per unit surface area exposed to the cooling air ($\dot{Q}$), the wall temperature (T_w), and the bulk mean air temperature (T_b).

$$h = \frac{\dot{Q}}{A(T_W - T_b)} \tag{1}$$

where A is the total heat transfer area of the smooth walls (i.e., excluding the surface area of the ribs). The heat transfer rate ($\dot{Q}$) was defined as

$$\dot{Q} = m' C_p (T_{b2} - T_{b1}) \tag{2}$$

where, C_p is specific heat of air, T_{b1} and T_{b2} are bulk temperatures at the inlet and exit of test section. Exit bulk temperatures were obtained by averaging local velocities and temperatures measured at 25 cm downstream from exit of test section. Specific heat of 1,005.7 J/kg℃ was used at 35.6℃. The m' is the mass flow rate at test section.

After confirming that the present experimental results in a smooth channel were coincident with Sieder and Tate's correlation,[10] Sieder and Tate's[10] correlation was chosen as reference. The following Nusselt numbers normalized by Sieder and Tate's correlation were used to investigate the effect of roughness.

$$\frac{\mathrm{Nu}_{sm}}{\mathrm{Nu}_{ss}} = \frac{(h_{sm} D_h / k)}{\left\{0.027\mathrm{Re}^{0.8}\mathrm{Pr}^{1/3}(\mu/\mu_w)^{0.14}\right\}} \tag{3}$$

$$\frac{\mathrm{Nu}_{ro}}{\mathrm{Nu}_{ss}} = \frac{(h_{ro} D_h / k)}{\left\{0.027\mathrm{Re}^{0.8}\mathrm{Pr}^{1/3}(\mu/\mu_w)^{0.14}\right\}} \tag{4}$$

$$\mathrm{Nu}_{ss} = 0.027\mathrm{Re}^{0.8}\mathrm{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{5}$$

where, subscripts, ro, sm and ss mean ribbed channel, smooth channel, and Sieder and Tate's[10] correlation. The thermal conductivity of air was 0.0268 W/m℃ at 35.6℃. Nu is defined as:

$$\mathrm{Nu} = \frac{hD_h}{k} \tag{6}$$

Maximum uncertainty of Nusselt number (U_{Nu}) became 9.5% by utilizing the following Kline and McClintock.[11]

$$U_{\mathrm{Nu}}^2 = \left(\frac{\partial \mathrm{Nu}}{\partial h} U_h\right)^2 + \left(\frac{\partial \mathrm{Nu}}{\partial D_h} U_{D_h}\right)^2 + \left(\frac{\partial \mathrm{Nu}}{\partial k} U_k\right)^2 \tag{7}$$

3. Experimental results and discussion

The heat transfer coefficient for a smooth channel were determined before performing tests in channels with ribbed walls. Figure 4 shows the fully developed Nusselt numbers in a smooth channel at various Reynolds numbers. The empirical correlation by Sieder and Tate[10] is also plotted for a comparison. It is evident from Fig. 4 that there is a good agreement between the correlation and our results.

Figure 5 represents the Nusselt numbers normalized by Sieder and Tate's correlation[10] for smooth channels along axial distance.

In general, the Nusselt number decreased monotonically from its maximum value at the beginning of the test section with increasing axial distance, eventually reaching a constant value at $L/D_h=36$ where the flow was fully developed for the given Reynolds number. Hu and Shen[12] reported similar results in their study of square channel walls with 45° inclined discrete ribs.

The heat transfer rate ($\dot{Q}$) Fig. 6 was obtained from Eq. (2). The heat transfer rate ($\dot{Q}$) in the channel with the ribbed wall increased more

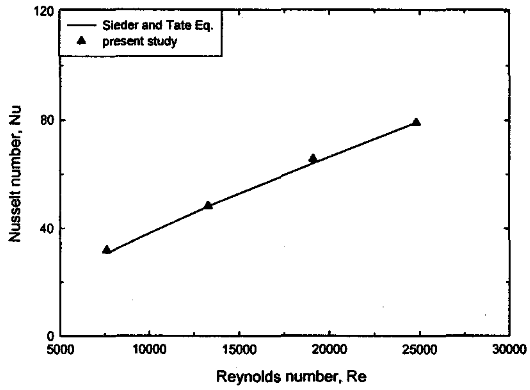


Fig. 4 Average Nusselt numbers for four heating walls-smooth channel.

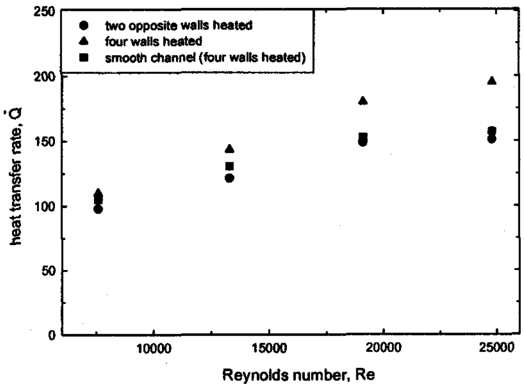


Fig. 6 Average heat transfer rate against Reynolds number for the one-ribbed wall channel.

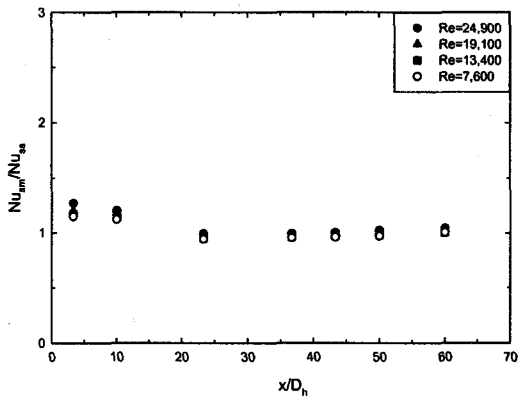


Fig. 5 Local Nusselt number for four heating walls-smooth channel.

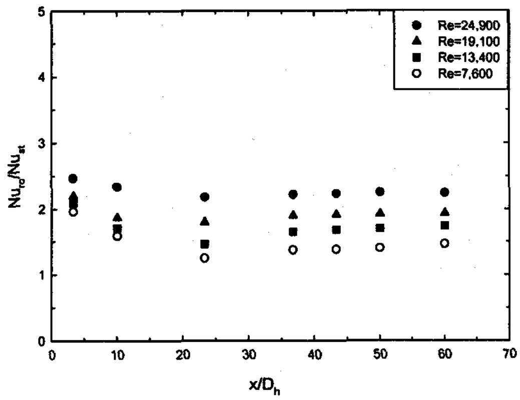


Fig. 7 Axial heat transfer distribution in the one-ribbed wall channel with four heating walls.

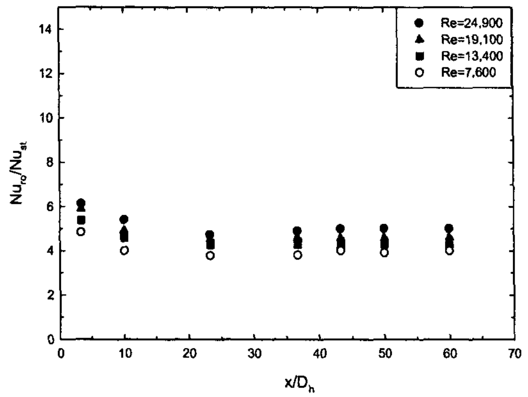


Fig. 8 Axial heat transfer distribution in the one-ribbed wall channel with two opposite heating walls.

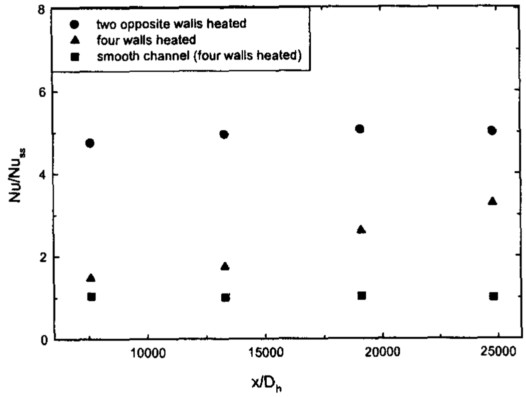


Fig. 9 Average Nusselt number for the one-ribbed wall channel.

rapidly than in the smooth channel on the condition of four wall constant heat flux. This tendency is attributed to the fact that, as Reynolds number increases, the turbulent mixing in terms of the rib on the wall is higher. This leads that a decrease in temperature difference from inlet to exit of the test section becomes lower than an increase in the flow rate; i.e., when the air bulk temperature difference from inlet to exit of the test section decreases 0.7 times from 18.2℃ to 14.2℃, Reynolds number increases 1.87 times from 13,000 to 24,900.

Figure 7 shows the normalized Nusselt number in the one sided ribbed wall channel with four heating walls. All the values were over unity. It was because the turbulent mixing due to the rib on the wall enhances the heat transfer. Figure 8 represents the axial heat transfer distribution in the one sided ribbed wall channel with two opposite heating walls.

Values of Nu_{ra}/Nu_{ss} with heating applied to two opposite walls were greater than those obtained with heating applied to all four walls of Fig. 7 at the same Reynolds number. This occurred because the colder fluid from the two unheated walls moved toward the two heated walls, resulting in higher heat transfer coefficients. Figure 9 shows a comparison of the normalized channel average Nusselt numbers with two sided and four sided heating walls at the fully developed region. Nusselt numbers of two opposite heating walls were greater than those obtained with heating applied to all four walls like Figs. 7 and 8. This means that the design of two opposite heating walls was advantageous over that of four heating walls.

4. Conclusions

(1) The normalized Nusselt numbers in the one sided ribbed wall channel with four or two heating walls were over unity.

(2) Values of Nu_{ra}/Nu_{ss} with heating applied to two opposite walls were greater than those obtained with heating applied to all four walls at the same Reynolds number.

(3) The heat transfer rate in the channel with the ribbed wall increased more rapidly than in the smooth channel.

Acknowledgement

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