Turbulent Heat Transfer in a Channel with Rib-Groove Turbulators

S. Chokphoemphun[1], S. Skullong[1], T. Chompookham[2],
T. Limkul[1] and P. Promvonge[1,*]

[1]Department of Mechanical Engineering, Faculty of Engineering,
King Mongkut's Institute of Technology Ladkrabang, Bangkok 10520, Thailand
[2] Heat Pipe and Thermal Tools Design Research Unit (HTDR),
Faculty of Engineering, Mahasarakham University, Mahasarakham 44150, Thailand
*Corresponding Author: E-mail address: kppongje@kmitl.ac.th,
Tel: +662-3298351; fax: +662-3298352

Abstract

The paper presents a study on thermal performance in a solar air heater channel using rib and groove turbulators. The experiments are performed by varying the airflow rate for Reynolds numbers based on the hydraulic diameter of the channel in the range of 4000 to 21,000. To produce recirculation flow in the tested channel having a constant wall heat-flux on the upper plate, the ribs are placed on the tested grooved channel walls. Three test cases of different rib heights or blockage ratios (BR=e/H) are considered in the present work. The groove dimension is 5 mm deep and 10 mm wide while the rib size is 0.3 mm thick and 5 mm high. The transverse ribs are placed with the attack angle of 90° relative to main flow direction. Two types of mounting ribs on the grooved plate are taken into account: ribs placed on the upper wall only and on both the lower and upper walls. The experimental result shows that the use of the ribbed and grooved plates on the upper and lower walls of the test channel leads to the highest heat transfer rate and friction factor in comparison with the smooth channel with/without ribs but the grooved upper plate provides the highest thermal enhancement factor at lower Reynolds number.
Keywords*:* Rib; Groove; Turbulators; Recirculation; Thermal enhancement factor.

1. Introduction

Heat transfer coefficient in a duct flow can be increased by roughening the wall of the duct. Over the past decades, many engineering techniques have been devised for enhancing the rate of convective heat transfer from the wall surface. The uses of turbulators such as ribs, fins, grooves and baffles are often employed in order to increase the convective heat transfer rate leading to the compact heat exchanger and increasing the efficiency.

Several investigations have been carried out to study the effect of these parameters of turbulators on heat transfer and friction factor for roughened surfaces. Han et al. [1] also investigated the influence of the surface heat flux ratio on the heat transfer in a square ribbed channel with e/H=0.063 and P/e =10, by heating either only one of the ribbed walls or both of them, or all four channel walls. They reported that the former two conditions resulted in an increase in the heat transfer with respect to the latter one and the average Nusselt number tends to decrease for increasing Reynolds numbers and the thermal boundary condition becomes less relevant at higher Reynolds number. Zhu et al. [2] simulated the turbulent flow and heat transfer in a rectangular channel with rectangular winglets on one wall and rib-roughness elements on the other wall. The results revealed that the combined effect of rib-roughness and vortex generators could enhance the average Nusselt number by nearly 450%.Varun et al. [3] and Hans et al. [4] carried out the reviews of roughness surface geometry in solar air heaters. They reported on solar air heaters fitted with different roughness geometries and provided details of the concept of artificial roughness, effects of various roughness parameters on the flow pattern and also briefly discussed the roughness geometries used in the heat exchangers. Promvonge et al. [5, 6] also investigated numerically the laminar flow structure and thermal behaviors in a square channel with 30° or 45° inline baffles on two opposite walls. They found that two counter-rotating vortex flows appear along the channel and vortex-induced-impinging (VI) jets occurred on the upper, lower and side walls, leading to the maximum thermal enhancement factors of about 2.6 at BR=0.2 and Re=1000; and around 4.0 at BR=0.15, PR=2 and Re=2000 for the 45° and 30°baffles, respectively. Chompookham et al. [7] studied experimentally the heat transfer in a channel fitted with the combined WVGs and wedge ribs (right-triangular) mounted on the principal channel wall and found that the combination of staggered wedge rib and the WVGs has efficiently performed and should be applied to obtain higher thermal performance at about 17-20% of a single use of turbulators. Promvonge et al. [8] investigated effects of combined ribs and winglet type vortex generators (WVGs) on forced convection heat transfer and friction loss behaviors for turbulent airflow through a constant heat flux. The combined staggered rib and the WVGs with lower angle of attack should be applied instead of using the rib/WVGs alone to obtain higher heat transfer and performance of about 40–65%.

The study on the rib-grooved surfaces in a channel has rarely been reported since most ribs found in the literature are square, rectangular and wedge shaped-ribs. In the present work, the experimental data presented in turbulent duct flows over 90° rib-grooves placed on the upper and lower channel walls are conducted with the main aim being to study the flow friction and heat transfer performance. The use of the rib-groove turbulators is expected to create vortex flows throughout the tested channel to better mixing of flows between the core and the near-wall regimes leading to higher heat transfer rate.

2. Experimental Setup

The experiments were conducted to examine the effect of using ribs and grooves on heat transfer and friction characteristics of air flow in a rectangular channel. A schematic diagram of the experimental setup is illustrated in Fig. 1 and the detail of 90° rib-grooves placed on the upper and lower walls is depicted in Fig. 2. In the experimental setup, a circular pipe was used for connecting a high-pressure blower to a settling tank, which an orifice flow meter was mounted in this pipeline while a rectangular channel including a calm section and a test section was employed following the settling tank. The rectangular duct configuration was characterized by the channel height (*H*) of 20 mm and the channel width (*W*) of 200 mm. The overall length of the duct was 2000 mm which included length of the test section (*L*) of 650 mm with a constant heat-flux on the upper wall only.

The AC power supply was the source of power for the plate-type heater, used for heating the upper-plate of the test section only to maintain uniform surface heat flux. Air as the tested fluid, was directed into the systems by a 1.45 kW high-pressure blower. The operating speed of the blower was varied by using an inverter to provide desired air flow rates. The flow rate of air in the systems was measured by an orifice plate. The pressure across the orifice was measured using inclined manometer. In order to measure temperature distributions on the principal upper wall, twelve thermocouples were fitted to the wall. The thermocouples were installed in holes drilled from the rear face and centered of the walls with the respective junctions positioned within 2 mm of the inside wall and axial separation was 40 mm apart. To measure the inlet and outlet temperatures, two RTD thermocouples were positioned upstream and downstream of the test channel. The thermocouple voltage outputs were fed into a data acquisition system (Fluke 2650B) and then recorded via a personal computer.

Two static pressure taps were located at the top of the principal channel to measure axial pressure drops across the test section, used to evaluate average friction factor. These were located at the centre line of the channel. One of these taps is 120 mm downstream from the leading edge of the channel and the other is 50 mm upstream from the trailing edge. The pressure drop was measured by a digital differential pressure and a data logger (Testo 1445) connected to the 2 mm diameter taps and recorded via a personal computer.

3. Data Reduction

In the present work, the air is used as the test fluid and flowed through a uniform heat flux and insulation channel. The steady state of the heat transfer rate is assumed to be equal to the heat loss from the test section which can be expressed as:

$$Q_{air} = Q_{conv} = \dot{m}C_p\left(T_o - T_i\right) = hA\left(\tilde{T}_s - T_b\right) \quad (1)$$

where

$$h = \frac{Q_{conv}}{A\left(\tilde{T}_s - T_b\right)} \quad (2)$$

in which

$$T_b = (T_o + T_i)/2 \quad (3)$$

and

$$\tilde{T}_s = \sum T_s / 12 \quad (4)$$

The term A is the convective heat transfer area of the heated upper channel wall whereas $\tilde{T}_s$ is the average surface temperature obtained from

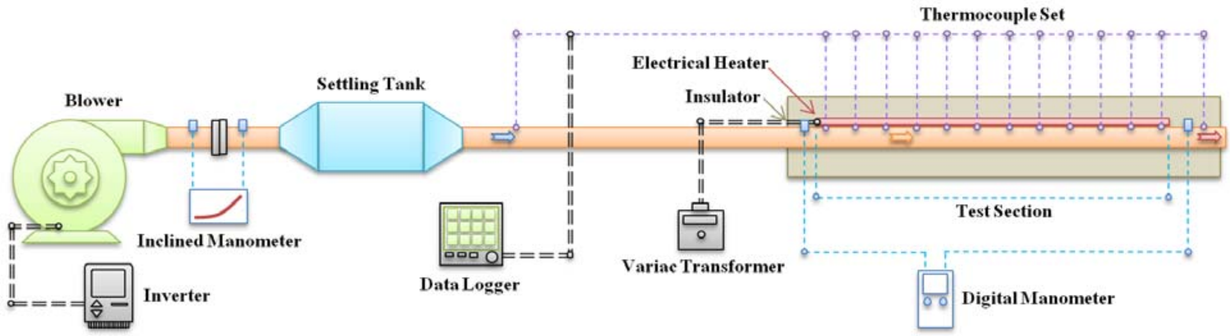

Fig. 1 A schematic diagram of the experimental set up.

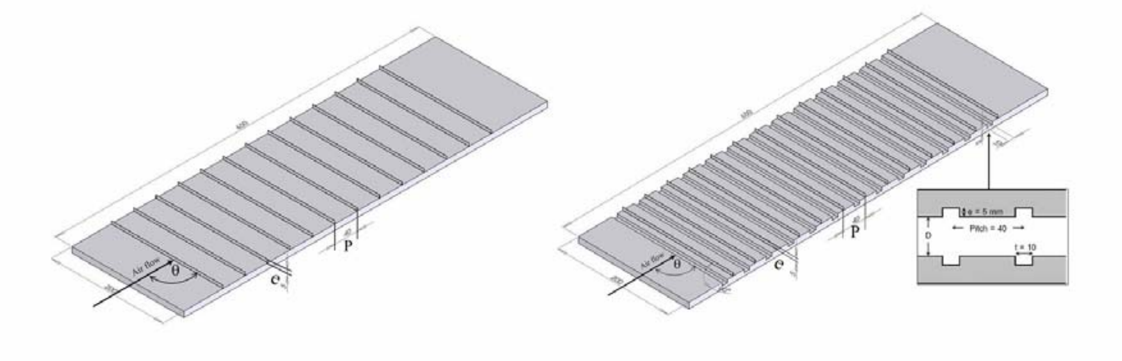

Fig. 2 Detail of 90° rib-groove placed channel wall.

local surface temperatures, $T_{s,}$ along the axial length of the heated channel, T_i and T_o are inlet and outlet of airs temperature, respectively. The average Nusselt number is written as:

$$Nu = \frac{hD_h}{k} \tag{5}$$

The Reynolds number based on the channel hydraulic diameter (D_h) is given by

$$\mathrm{Re} = UD_h / \nu \tag{6}$$

When U is average velocity of airs, while ν is kinematic viscosity
The friction factor is evaluated by:

$$f = \frac{2}{(L/D_h)} \frac{\Delta P}{\rho U^2} \tag{7}$$

The thermal enhancement factor is defined as the ratio of the, h of an augmented surface to that of a smooth surface, h_0, at a constant pumping power:

$$\eta = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{Nu}{Nu_0}\right|_{pp} = \left(\frac{Nu}{Nu_0}\right)\left(\frac{f}{f_0}\right)^{-1/3} \tag{8}$$

4. Results and Discussion

4.1 Verification of smooth channel

The present experimental results on heat transfer and friction characteristics in a smooth wall channel are first validated in terms of Nusselt number and friction factor. The Nusselt number (Nu) obtained from the present smooth channel are compared with the Dittus-Boelter correlations and the friction factors (f) are compared with the Blasius correlations that found in the open literature [9] for turbulent flow in ducts.
Correlation of Dittus-Boelter,

$$Nu = 0.023\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \tag{9}$$

Correlation of Blasius,

$$f = 0.316\,\mathrm{Re}^{-0.25} \tag{10}$$

Fig. 3 show a comparison of Nusselt number obtained from the present work with those from correlations of Eq. (9). In the figure, the present results agree very well within ±5% for Nusselt number.

A friction factor obtained from the present work and compared with Eq. (10) is presented in Fig. 4. The results show error within ±3% for friction factor.

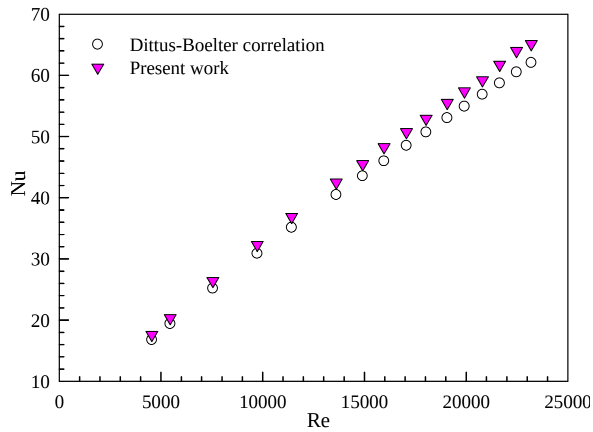

Fig. 3 Verification of Nusselt number for smooth channel.

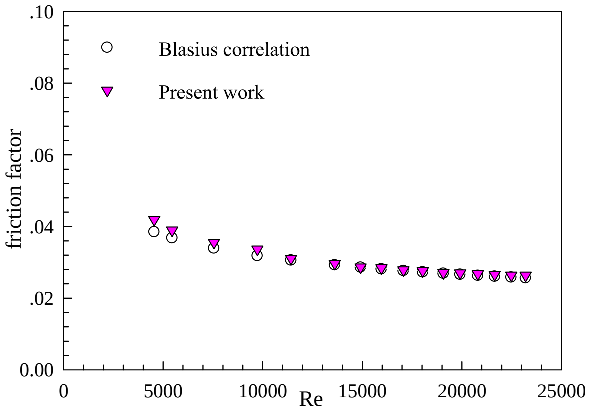

Fig. 4 Verification of friction factor for smooth channel.

4.2 Effect of rib and groove on heat transfer.

Fig. 5 shows the variation of *Nu* with *Re* values for different types of ribs on the plate. In the figure, the Nusselt number increases with the increment of Reynolds number. The use of ribs and groove turbulators improves heat transfer better than the smooth channel. This is because the rib and groove turbulators cause the reduction of boundary layer thickness and the increase the velocity of air flow. The heat transfer for using rib-groove on the upper-lower plates is peak at higher *Re*.

The variation of Nusselt number ratio, Nu/Nu_0 with Reynolds number is presented in Fig. 6. The Nu/Nu_0 is defined as a ratio of augmented Nusselt number to Nusselt number of the smooth channel. In the figure, the Nu/Nu_0 tends to be slightly decreased with the rise of Reynolds number for all cases. The mean Nu/Nu_0 values are found to be 1.46-2.17 times above the smooth channel.

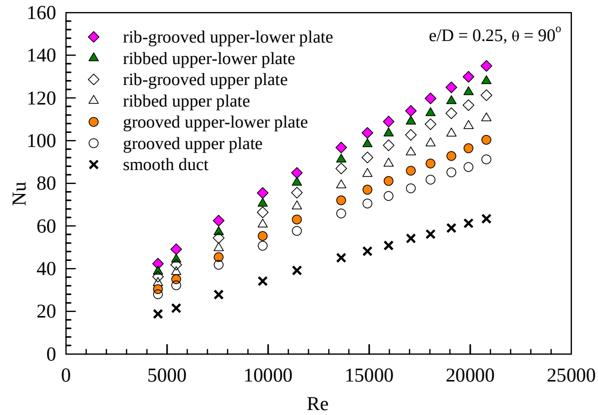

Fig. 5 Variation of Nusselt number with Reynolds number for various turbulators.

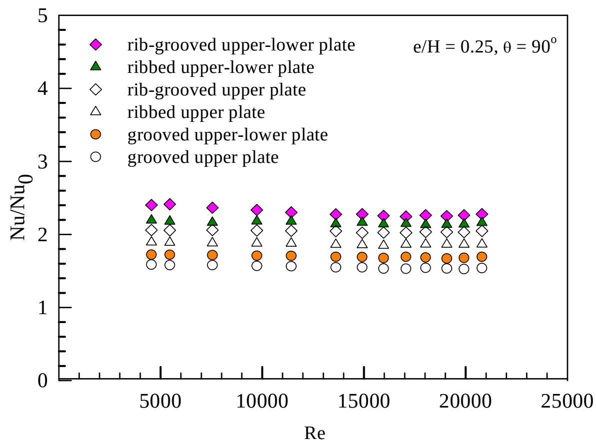

Fig. 6 Variation of Nusselt number ratio, Nu/Nu_0 with Reynolds number.

4.3 Effect of rib and groove on pressure drop.

The effect of using the rib-groove turbulators on the isothermal pressure drop across the tested channel is depicted in Fig. 7. The variation of the pressure drop is shown in terms of friction factor against Reynolds number. In the figure, it is found that the use of rib-groove turbulators leads to a drastic increase in friction factor over the smooth channel. This can be attributed to the flow blockage of ribs and the act caused by the reverse flow. The friction factors from the rib-grooved, upper-lower plates are found to be maximum and higher than the smooth channel at about 18 times.

Figure 8 shows the variation of friction factor ratio, f/f_0 with Reynolds number. In the figure, the f/f_0 is increased with the rise of Reynolds number for the case of rib and rib-groove plates but for the case of grooved plates, the trends are relatively uniform. The mean f/f_0 values for all cases studied are in a range of 1.49-17.83 times above the smooth channel.

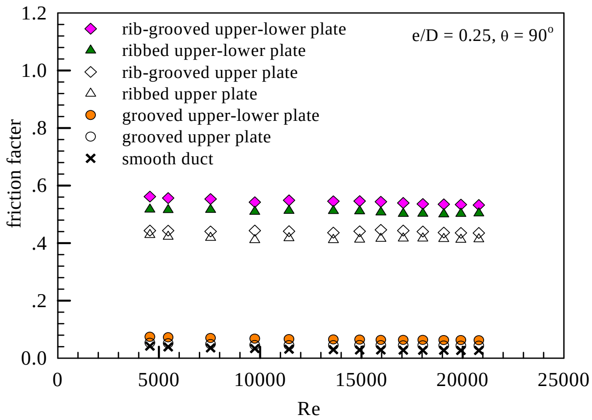

Fig. 7 Variation of friction factor with Reynolds number for various turbulators.

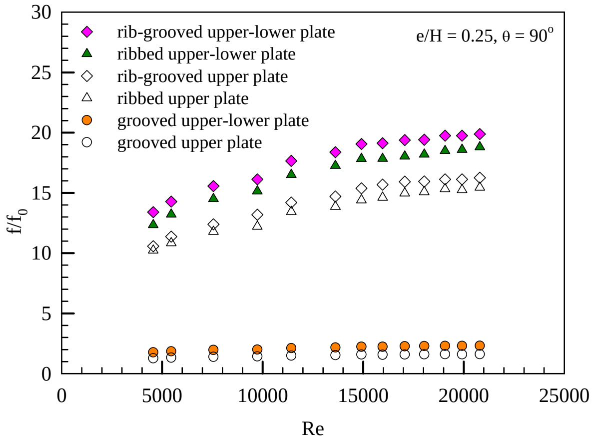

Fig. 8 Variation of friction factor ratio, f/f_0 with Reynolds number.

4.4 Performance evaluation

The variation of the thermal enhancement factor (η) with Reynolds number is depicted in Fig. 9. The Nusselt number and friction factor values obtained from the experimental data are compared at a similar pumping power in order to determine the net heat gain. The figure shows that the η value tends to decrease with the rise of Reynolds number for all. There will be a net heat gain only for η greater than unity. For the cases of using the grooved upper plate and the grooved upper-lower plate, the average η values are at 1.36 and 1.32, respectively.

5. Conclusion

An experimental study has been carried out to investigate heat transfer characteristics and airflow friction in a solar air heater channel using rib and groove turbulators for the turbulent regime, Reynolds number from 4000 to 21,000. The 90° ribs with *e/H*=0.25 mounted on the upper and upper-lower smooth/grooved plates are provide higher pressure drop and heat transfer than the smooth channel. The mean thermal enhancement factor is greater than unity for using both the grooved upper plate and the grooved upper-lower plates.

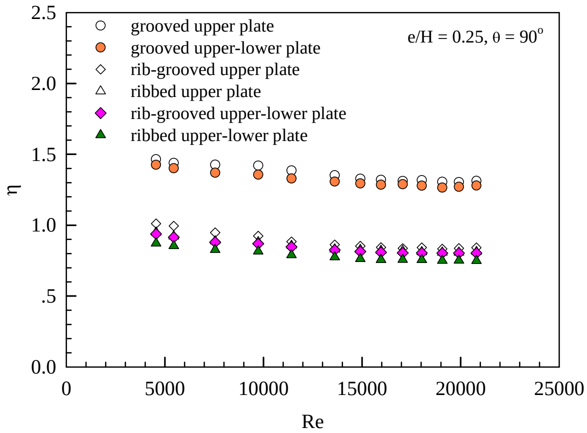

Fig. 9 Variation of thermal enhancement factor with Reynolds number.

6. Acknowledgement

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