Thermal Behavior in a Square Channel with 45° Cross Baffle Insert

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Abstract

The paper presents a study of heat transfer and pressure loss in a heat exchanger channel inserted with inclined baffles on two opposite walls in cross arrangement. The channel has a square section with uniform wall heat flux conditions. The fluid flow and heat transfer characteristics are presented for Reynolds numbers based on the hydraulic diameter of the channel ranging from 4000 to 40,000. The inclined baffles with an axial pitch equal to one times of channel height and with the attack angle of 45° are mounted in tandem and cross arrangement on the upper and lower walls of the test channel. Effects of five baffle-to-channel height ratios (e/H = 0.1, 0.15, 0.2, 0.25 and 0.3) on heat transfer in terms of Nusselt number and pressure loss in the form of friction factor are experimentally investigated.

The experimental result shows that the insertion of cross inclined baffles with the e/H = 0.3 provides higher heat transfer and friction factor values than others. This is caused by higher baffle-to-channel height ratios of using e/H = 0.3 interrupting the flow and diverting its direction thus promoting high levels of mixing over others.

***Keywords**:* Square channel; Cross baffle; Thermal behavior; Friction factor; Turbulator

1. Introduction

Forced convection heat transfer is the most frequently employed mode of the heat transfer in heat exchangers or in various chemical process plants. In the cooling channel or duct heat exchanger design, rib, fin, groove or baffle turbulators are often employed in order to increase the convective heat transfer rate leading to the compact heat exchanger and increasing the efficiency. For decades, rib turbulators have been applied in high-performance thermal systems due to their high thermal loads. The cooling or heating air is supplied into the passages or channels with several ribs to increase the stronger degree of cooling or heating levels over the smooth wall channel. The use of rib turbulators completely results in the change of the flow field and hence the variation of the local convective heat transfer coefficient. The use of ribs increases not only the heat transfer rate both for the increased turbulence degree and for the effects caused by reattachment but also substantial the pressure loss. Another technique used inclined ribs for improving the performance of heat exchange devices is to set up periodic disturbance promoters along the streamwise direction. Such an arrangement of the channels might lead to the enhancement of the heat transfer due to flow mixing and periodic interruptions of thermal boundary layers, but often causes the increase of pressure drop penalty. The artificial roughened surfaces are widely used in modern heat exchangers, because they are very effective in heat transfer augmentation.

Several investigations have been carried out to study the effect of these parameters of turbulators on heat transfer and friction factor for roughened surface. Taslim et al. [1] conducted measurements of the heat transfer in a straight square channel with three e/H ratios (e/H=0.083, 0.125 and 0.167) and a fixed P/e = 10 using a liquid crystal technique. Various staggered rib configurations were studied, especially for the angle of 45°. Experimental data showed a significant increase in average Nusselt number for the increase of the e/H ratio. Mochizuki [2] studied numerically the heat transfer distribution in a ribbed square channel with a large eddy simulation method. The ribs were placed at 60°, e/D =0.1 and P/e =10. Their numerical result indicated that the flow reattachment at the midpoint between ribs caused a significant increase in the local heat transfer. Chandra et al. [3] carried out measurements on heat transfer and pressure loss in a square channel with continuous ribs on four walls. Ribs were placed superimposed on walls at the rib height ratio e/D = 0.0625; and the rib pitch ratio, P/e=8. They reported that the heat transfer augmentation found to increase with the rise in the number of ribbed walls was decreased with increasing Reynolds number while the friction factor augmentation increased with both cases. Lee et al. [4] studied experimentally the heat/mass transfer in rectangular channels with two different V-shaped ribs: continuous 60° V-shaped and multiple (staggered) 45° V-shaped ribs, and found that two pairs of counter-rotating vortices are generated in the channel. The effect of channel aspect ratio wasmore significant for the 60° V-shaped rib than for the multiple 45° V-shaped rib. Tanda [5] examined the effect of transverse, angled ribs, discrete, angled discrete ribs, V-shaped, V-shaped broken and parallel broken ribs on heat transfer and friction. It was found that 90° transverse ribs provided the lowest thermal performance while the 60° parallel broken ribs or 60° V-shaped broken ribs yielded a higher heat transfer augmentation than the 45° parallel broken ribs or 45° V-shaped broken ribs. Parallel angled discrete ribs were seen to be superior to parallel angled full ribs and its 60° discrete ribs performed the highest heat transfer. Promvonge and Thianpong [6] studied the thermal performance of wedge ribs pointing upstream and downstream, triangular and rectangular ribs with e/H = 0.3 and P/e = 6.67 mounted on the two opposite walls of a channel with AR = 15. They found that the inline wedge rib pointing downstream performed the highest heat transfer but the best thermal performance is the staggered triangular rib. Thianpong et al. [7] again investigated the thermal behaviors of isosceles triangular ribs attached on the two opposite channel walls with

AR = 10 and suggested the optimum thermal performance of the staggered ribs could be at about e/H = 0.1 and P/H = 1.0. Promvonge et al. [8] studied the numerical computations for three dimensional laminar periodic channel flows over a 45° inclined baffle mounted only on the lower square-channel wall and found that the 45° baffle with BR = 0.4, the enhancement of heat transfer is about 2–3 fold higher than that for the 90° baffle while the friction loss is some 10–25% lower. However, the increase in heat transfer is accompanied by an increase in the resistance of fluid flow. An extensive literature review over hundred references on various rib turbulators was reported by Varun et al. [9].

The study on cross baffles in square channels has never been reported since most baffles found in the literature are square, rectangular, triangular and wedge shaped-baffles. In the present work, the experimental data presented in turbulent channel flows over a 45° cross baffle mounted on the upper and lower channel walls are conducted with the main aim being to study the changes in the flow pattern and heat transfer performance. The use of the cross baffle attached in tandem is expected to create a longitudinal vortex flow throughout the tested channel to better mixing of flows between the core and wall regimes leading to higher heat transfer rate.

2. Experimental Setup

A schematic diagram of the experimental apparatus is presented in Figure 1 while the detail of 45° cross baffles placed on the upper and lower channel walls is depicted in Figure 2. In Figure 1, 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 square channel including a calm section and a test section was employed following the settling tank. The square channel configuration was characterized by the channel height, *H* of 45 mm and a baffle pitch (P) of one times of channel height (pitch ratio, PR=1) and the attack angle of 45°. The overall length of the channel was 3000 mm. The test square channel made of 3 mm thick aluminum plates has a cross section of 45×45 mm^2 and 1000 mm long (L). The baffle strip dimensions were 4.5, 6.75, 9, 11.25 and 13.5 mm high (*e*) and 0.3 mm thick (*t*).

The test section consisted of the four walls. The AC power supply was the source of power for the plate-type heater, used for heating all walls of the test section in order to maintain a uniform surface heat flux.

Air as the tested fluid in both the heat transfer and pressure drop experiments, was directed into the systems by a 1.5 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 pre-calibrated by using hot wire and vane-type anemometers (Testo 445). The pressure across the orifice was measured using inclined manometer. In order to measure temperature distributions on the principal upper, lower and side walls, twenty eight thermocouples were fitted to the walls. 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 100 mm apart. To measure the inlet and outlet bulk temperatures, two thermocouples were positioned upstream and downstream of the test channel. All thermocouples were *K* type, 1.5 mm diameter wire. The thermocouple voltage outputs were fed into a data acquisition system (Fluke 2650A) and then recorded via a personal computer.

Two static pressure taps were located at the top of the principal wall 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 50 mm upstream of the test channel and the other is 50 mm downstream. The pressure drop was measured by a digital differential pressure and a data logger (Testo 1445 and Testo 350XL) connected to the 2 mm diameter taps and recorded via a personal computer.

To quantify the uncertainties of measurements, the reduced data obtained experimentally were determined. The uncertainty in the data calculation was based on ref. [10]. The maximum uncertainties of non-dimensional parameters were ±5% for Reynolds number, ±8% for Nusselt number and ±10% for friction. The uncertainty in the axial velocity measurement was estimated to be less than ±7%, and pressure has a corresponding estimated uncertainty of ±5%, whereas the uncertainty in temperature measurement at the channel wall was about ±0.5%.

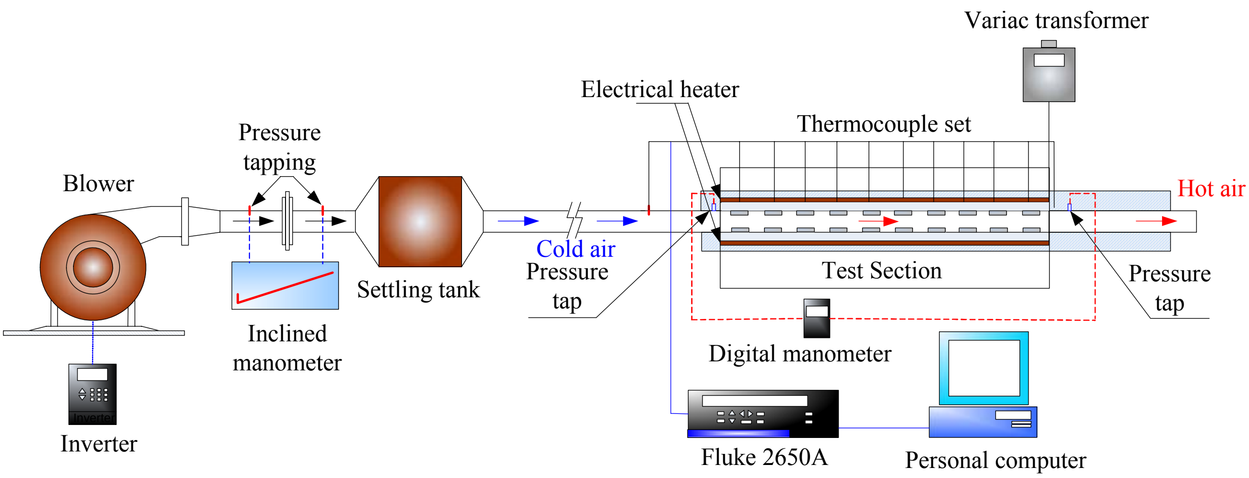


Fig. 1 Schematic diagram of experimental apparatus.

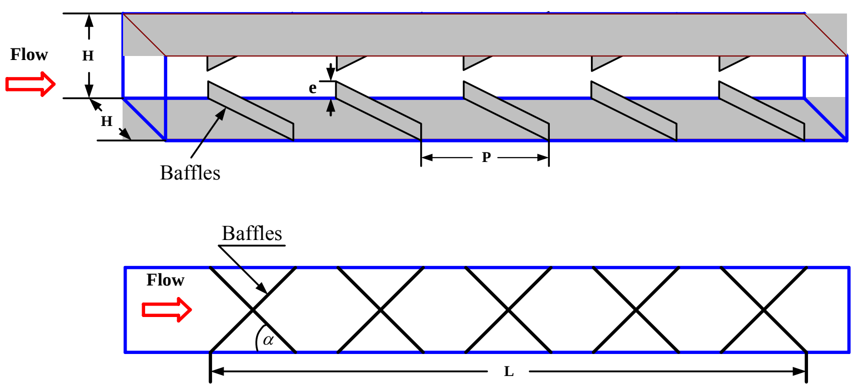


Fig. 2 Test section with cross baffle.

3. Data Reduction

The goal of this experiment is to investigate the Nusselt number in baffle channels. The independent parameters were Reynolds number and rib types. The Reynolds number based on the channel hydraulic diameter is given by

$$\mathrm{Re} = UD_h / \nu \tag{1}$$

The average heat transfer coefficients are evaluated from the measured temperatures and heat inputs. With heat added uniformly to fluid (Q_{air}) and the temperature difference of wall and fluid ($T_w - T_b$), average heat transfer coefficient will be evaluated from the experimental data via the following equations:

$$Q_{air} = Q_{conv} = \dot{m} C_p (T_o - T_i) = VI \tag{2}$$

$$h = \frac{Q_{conv}}{A(\tilde{T}_s - T_b)} \tag{3}$$

in which,

$$T_b = (T_o + T_i)/2 \tag{4}$$

and

$$\tilde{T}_s = \sum T_s / 28 \tag{5}$$

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 local surface temperatures along the axial length of the heated channel. Then, average Nusselt number is written as:

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

The friction factor is evaluated by:

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

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

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

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 and friction factor obtained from the present smooth channel are, respectively, compared with the correlations of Gnielinski and Petukhov found in the open literature [11] for turbulent flow in ducts.

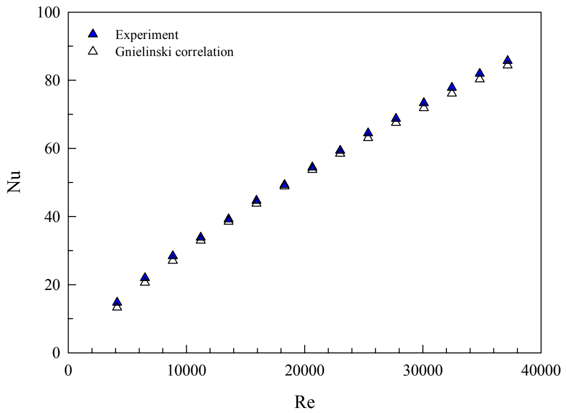
Correlation of Gnielinski,

$$Nu = \frac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7(f/8)^{1/2}(\mathrm{Pr}^{2/3}-1)} \tag{9}$$

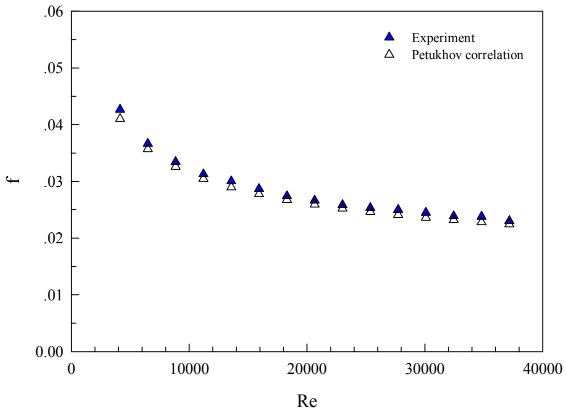
Correlation of Petukhov,

$$f = (0.79 \ln \mathrm{Re} - 1.64)^{-2} \tag{10}$$

Figure 3a and b shows, respectively, a comparison of Nusselt number and friction factor obtained from the present work with those from correlations of Eqs. (9) and (10). In the figure, the present results agree very well within ±3% for both friction factor and Nusselt number correlations.



(a)



(b)

Fig. 3 Verification of (a) Nusselt number and (b) friction factor for smooth channel.

4.2 Effect of blockage ratio

The present experimental results on heat and flow friction characteristics in a uniform heat flux channel with cross baffle, placed on the upper and lower wall are presented in the form of Nusselt number and friction factor. The Nusselt numbers obtained under turbulent flow conditions for all cases are presented in Figure 4. In the figure, the cross baffle turbulators yield considerable heat transfer enhancements with a similar trend in comparison with the smooth channel and the Nusselt number increases with the rise of Reynolds number. This is because the cross baffle turbulators interrupt the development of the boundary layer of the fluid flow and increase the turbulence degree of flow. It is worth nothing that the heat transfer coefficient for baffle-to-channel height ratio, e/H=0.3 is considerably higher than those for e/H = 0.25, 0.2, 0.15 and 0.1. This is caused by higher blockage of using e/H = 0.3 interrupting the flow and diverting its direction thus promoting high levels of mixing over others.

The effect of using the baffle turbulators on the isothermal pressure drop across the tested channel is presented in Figure 5. The variation of the pressure drop is shown in terms of friction factor with Reynolds number. In the figure, it is apparent that the use of baffle turbulators leads to a substantial increase in friction factor over the smooth channel. This can be attributed to flow blockage, higher surface area and the act caused by the reverse flow. As expected, the friction factor of baffle-to-channel height ratio, e/H=0.3 is considerably higher than those of e/H=0.25, 0.2, 0.15 and 0.1. For the baffle-to-channel height ratio e/H=0.3, the losses mainly come from the dissipation of the dynamical pressure of the air due to high viscous losses near the wall, to higher friction of increasing surface area and the blockage ratios because of the presence of the baffles.

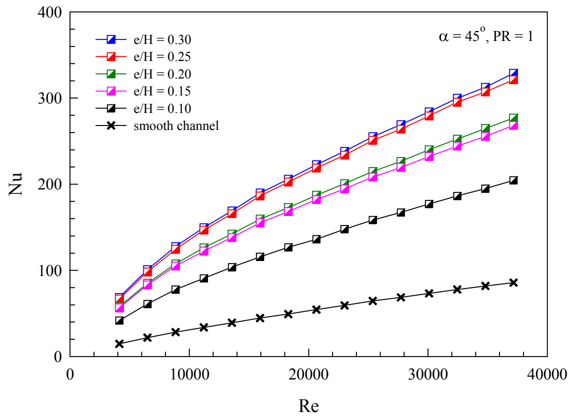


Fig. 4 Variation of Nusselt number with Reynolds number for various baffle heights.

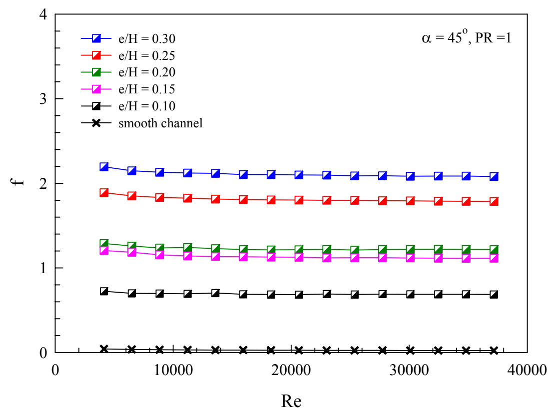


Fig. 5 Variation of friction factor with Reynolds number for various baffle heights.

4.3 Performance evaluation

The Nusselt number ratio, Nu/Nu_0, defined as a ratio of augmented Nusselt number to Nusselt number of smooth channel, plotted against the Reynolds number value is displayed in Figure 6. In the figure, the Nusselt number ratio tends to slightly decrease with the rise of Reynolds number from 4000 to 40,000 for all cases of cross baffle. The mean Nusselt number ratio values are found to be about 4.15, 4.07, 3.49, 3.39 and 2.55 time over the smooth channel for using the e/H = 0.3, 0.25 0.2, 0.15 and 0.1, respectively.

The variation of isothermal friction factor ratio value with Reynolds number for five baffle cases is also depicted in Figure 7. In the figure, the friction factor ratio value is found to be increased with the rise of Reynolds number. The mean friction factor ratio values are around 75.92, 65.22, 44.21, 40.82 and 24.96 for e/H=0.3, 0.25, 0.2, 0.15 and 0.1, respectively. This result indicates that the use of low blockage ratio can help to reduce the pressure loss considerably.

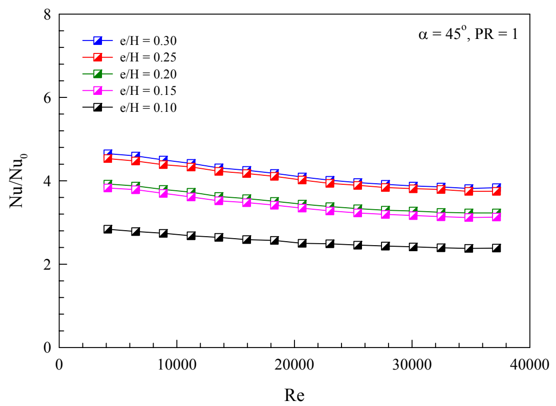


Fig. 6 Variation of Nusselt number ratio, Nu/Nu_0 with Reynolds number.

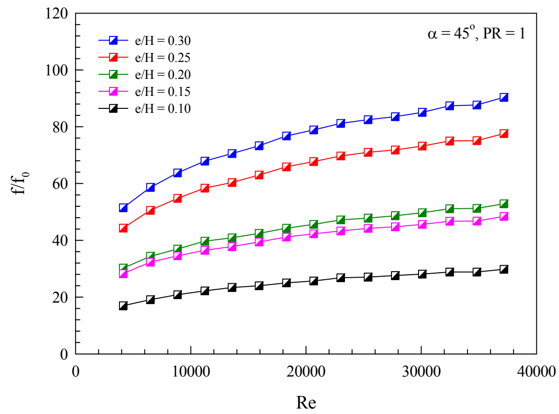


Fig. 7 Variation of friction factor ratio, f/f_0 with Reynolds number.

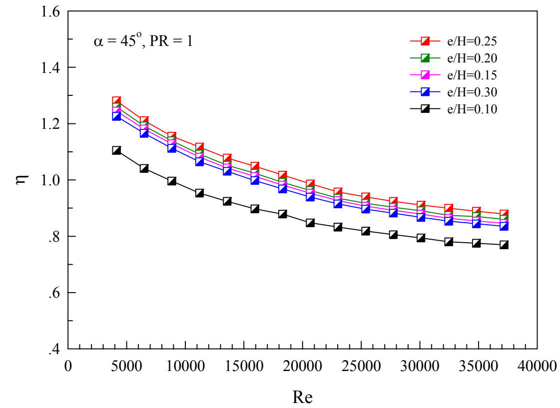


Fig. 8 Variation of thermal enhancement factor with Reynolds number.

Figure 8 shows the variation of the thermal enhancement factor (η) with Reynolds number for all cases. For all, the data obtained by Nusselt number and friction factor values are compared at similar pumping power. The thermal enhancement factor tends to decrease with the rise of Reynolds number values for all. It is seen that the blockage ratio of 0.25 shows the highest value of mean the thermal enhancement factor. The mean thermal enhancement factor values are around 1.1, 1.0, 0.99, 0.97 and 0.88 times for using the baffles with e/H=0.25, 0.2, 0.15, 0.3 and 0.1, respectively. The results are for Reynolds number of 4000-40,000 for the 45° cross baffle, the maximum thermal enhancement factor is found at e/H = 0.25. This can be attributed to considerably lower friction loss.

5. Conclusions

An experimental study has been carried out to investigate the airflow friction and heat transfer characteristics in a square channel fitted with different blockage ratio turbulators for the turbulent regime, Reynolds number from 4000-40,000. The use of the cross baffles with e/H = 0.3 causes a very high pressure drop increase and also provides considerable heat transfer augmentations, Nu/Nu_0 = 4.15. Nusselt number augmentation tends to increase with the rise of Reynolds number. In comparison, the use of baffle leads to the higher heat transfer rate but the e/H = 0.25 provides the higher thermal enhancement factor due to lower friction loss.

6. Acknowledgement

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7. References

[1] M.E. Taslim, T. Li, D.M. Kercher, Experimental heat transfer and friction in channels roughened with angled, V-shaped, and discrete baffles on two opposite walls, ASME, Journal of Turbomachinery 118 (1996) 20–28.

[2] A. Murata, S. Mochizuki, Comparison between laminar and turbulent heat transfer in a stationary square duct with transverse or angled rib turbulators, International Journal of Heat and Mass Transfer 44 (2001) 1127–1141.

[3] P.R. Chandra, C.R. Alexander, J.C. Han, Heat transfer and friction behaviour in rectangular channels with varying number of ribbed walls, International Journal of Heat and Mass Transfer 46 (2003) 481–495.

[4] D.H. Lee, D.H. Rhee, K.M. Kim, H.H. Cho, H.K. Moon, Detailed measurement of heat/mass transfer with continuous and multiple V-shaped ribs in rectangular channel, Energy 34 (2009) 1770–1778.

[5] G. Tanda, Heat transfer in rectangular channels with transverse and V-shaped broken ribs, International Journal of Heat and Mass Transfer 47 (2004) 229–243.

[6] P. Promvonge, C. Thianpong, Thermal performance assessment of turbulent channel flow over different shape ribs, Int. Commun. Heat Mass Transfer 35 (2008) 1327–1334.

[7] C. Thianpong, T. Chompookham, S. Skullong, P. Promvonge, Thermal characterization of turbulent flow in a channel with isosceles triangular ribs, Int. Comm. Heat Mass Transfer 36 (2009) 712–717.

[8] P. Promvonge, S. Sripattanapipat, S. Tamna, S. Kwankaomeng, C. Thianpong, Numerical investigation of laminar heat transfer in a square channel with 45° inclined baffles, Int. Commun. Heat Mass Transf 37 (2010) 170–177.

[9] Varun, R.P. Saini, S.K. Singal, A review on roughness geometry used in solar air heaters, Solar Energy 81 (2007) 1340–1350.

[10] ANSI/ASME, Measurement uncertainty, PTC 19, 1-1985. Part I, 1986.

[11] F. Incropera, P.D. Dewitt, Introduction to heat transfer, 5th edition, John Wiley & Sons Inc; 2006.