Heat Transfer Behaviors in a Square Channel with Different Turbulators Insert

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Abstract

This work presents a study of heat transfer in a heat exchanger channel inserted with different turbulator; stagger baffles and wire coil. The test channel cross-section is square with a uniform wall heat flux condition. The fluid flow and heat transfer characteristics are presented for Reynolds numbers based on the hydraulic diameter of the channel ranging from 4000 to 25,000. The case of inserted baffles are mounted in tandem and stagger arrangement with a rib-pitch to channel-height ratio, PR = 1 and the attack angle (α) of 45° on the lower and upper walls of the channel and the five baffle-to-channel height ratios, e/H = 0.1, 0.15, 0.2, 0.25 and 0.3. The case of inserted wire coil placed in tandem inside the channel with coil ratio, CR = 8 on heat transfer in terms of Nusselt number and pressure loss in the form of friction factor are experimentally investigated. The results obtained from turbulator inserts are also compared with the smooth channel. The experimental result indicates that the baffle with e/H = 0.3 provides the highest heat transfer and friction factor values but one with e/H = 0.25 provides the highest thermal enhancement factor.

***Keywords**:* Heat transfer; Square channel; Turbulator; Stagger baffles; Wire coil

1. Introduction

Over the past decades, many engineering techniques have been devised for enhancing the rate of convective heat transfer from the wall surface. Heat transfer enhancement techniques can be classified into two techniques: Passive techniques; do not require any type of external power for the heat transfer augmentation, such as coating of the surfaces, rough surfaces and extended surfaces, whereas, the active techniques; need some power externally, such as electric or acoustic fields and surface vibration. The effectiveness of both types depends strongly on the mode of heat transfer in order to increase the convective heat transfer rate leading to the compact heat exchanger and increasing the efficiency. The used of turbulators (baffles, ribs, fins, grooves and wire coil) are simple passive techniques for enhancing the rate of convective heat transfer. The use of turbulators completely results in the change of the flow field and hence the variation of the local convective heat transfer coefficient and 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 baffles 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. Promvonge et al. [1,2] studied experimentally and numerically the turbulent flow over 30° angle-finned tapes inserted diagonally in the square duct. They noted that at smaller fin pitch spacing, the finned tape with BR = 0.3 provides the highest heat transfer and friction factor but the one with BR = 0.2 and PR = 1 yields the best thermal performance. The thermal performance of the newly invented finned tape turbulator is found to be much higher than that of the wire coil/twisted tape turbulator. Thianpong et al. [3] 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. 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 was more significant for the 60° V-shaped rib than for the multiple 45° V-shaped rib. Promvonge and Thianpong [5] 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. Promvonge [6] investigated thermal characteristics of circular- and square-wire coils inserted tubes and found that the wire coil with square cross section provided higher thermal performance than the one with circular cross section. Tanda [7] 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. Yakut and Sahin [8] studied the heat transfer and friction loss characteristics by placing wire coil turbulators in a tube in terms of the shedding frequencies and amplitudes of vortices produced by wire coil turbulators. Wang and Sunden [9] for both laminar and turbulent flow regions. They found that the coiled wire performs effectively in enhancing heat transfer in a turbulent flow region, whereas the twisted tape yields a poorer overall efficiency. Inaba and Ozaki [10] presented that the turbulent flow induced by using a coiled wire enhances heat transfer even downstream of the coiled wire Chandra et al. [11] 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. Taslim et al. [12] 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°. Han et al. [13, 14] studied experimentally the heat transfer in a square channel with ribs on two walls for nine different rib configurations. Average heat transfer and friction factor were reported for the *P/e* = 10 and *e/H* = 0.0625 rib by heating either only one of the ribbed walls or both of them, or all four channel walls. The heat transfer augmentations and the friction factor were highest for the 60° orientation amongst the angled ribs. Experimental data showed a significant increase in average Nusselt number for the increase of the *e/H* ratio. An extensive literature review over hundred references on various rib turbulators was reported by Varun et al. [15].

The study on stagger baffles in square channels has never been reported in the literature. In the present work, the experimental data presented in turbulent channel flows over stagger baffles and wire coil in a square channel with the main aim being to study the changes in the flow pattern and heat transfer performance. The use of the turbulators attached in tandem is expected to create a 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 Fig. 1 while the detail of 45° stagger baffles and wires coil are inserted in a channel of the test section are depicted in Fig. 2 and Fig.3 respectively. In Fig. 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., the baffle pitch equal to 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 45x45 mm^2. and 1000 mm. long (*L*). The baffle strip dimensions were 4.5, 6.75 9.0, 11.25 and 13.5 mm. high (*e*) and 0.3 mm. thick (*t*) and the wire coil used was made steel wire with diameter (d) of 5.5 mm. The wire coil with coil diameter to channel height ratio (H/d, CR = 8).

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 1.45 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 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. [16]. 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%.

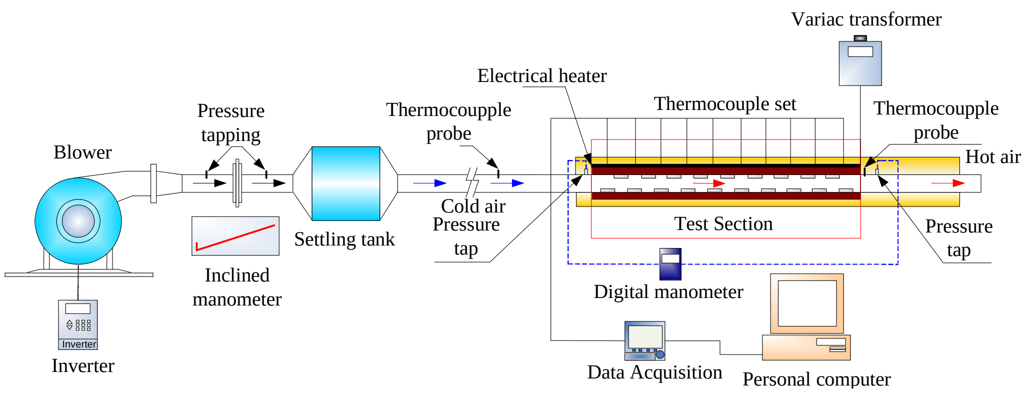

Fig. 1 Schematic diagram of experimental apparatus.

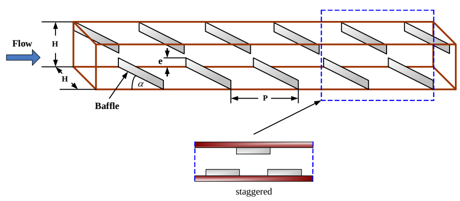

Fig. 2 Test section with stagger baffles insert.

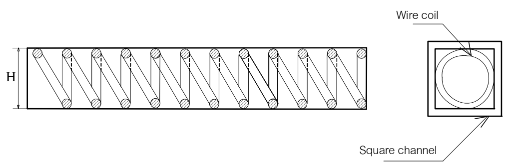

Fig. 3 Test section with wire coil inserts.

3. Data Reduction

The goal of this experiment is to investigate the Nusselt number and the friction factor in a square channel with different turbulators insert. 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:

$$Q_{air} = Q_{conv} = \dot{m}C_p(T_o - T_i) = VI \qquad (1)$$

where

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

in which,

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

and

$$\tilde{T}_s = \sum T_s / 28 \qquad (4)$$

The term A is the convective heat transfer area of the heated 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:

$$Nu = \frac{hD_h}{k} \qquad (5)$$

The Reynolds number based on the channel hydraulic diameter is given by:

$$\mathrm{Re} = UD_h / \nu \qquad (6)$$

The friction factor is evaluated by:

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

where ΔP is the pressure drop across the test section and U is the mean air velocity of the channel. All of the thermo-physical properties of the air are determined at the overall bulk air temperature.

The thermal enhancement factor (η) defined as the ratio of the heat transfer coefficient of an augmented surface, h to that of a smooth surface, h_0, at a constant pumping power, Webb. [17]

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

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 [18] for turbulent flow in ducts.

Correlation of Gnielinski,

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

Correlation of Petukhov,

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

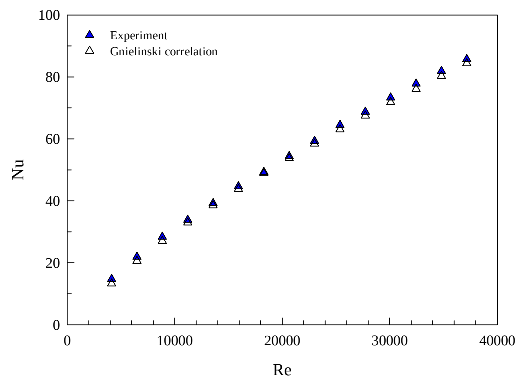

Fig. 3a Verification of Nusselt number for smooth channel.

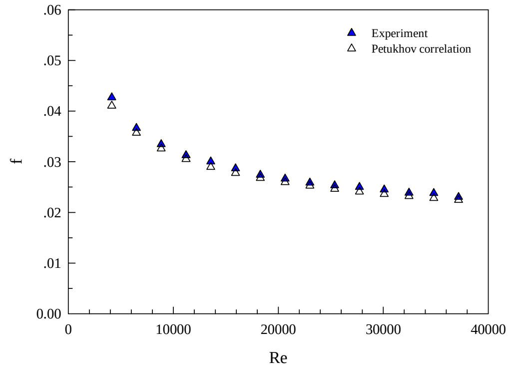

Fig. 3b Verification of friction factor for smooth channel.

Fig. 3a and 3b 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 figures, the present results reasonably agree very well within ±3% for both friction factor correlations of Petukhov and Nusselt number correlations of Gnielinski.

4.2 Effect of blockage ratio

The present experimental results on heat and flow friction characteristics in a uniform heat flux channel equipped with turbulators are presented in the form of Nusselt number and friction factor.

Fig. 4 shows the used of turbulators yield the considerable heat transfer enhancement with a similar trend in comparison with the smooth channel and the Nusselt number values increase with the rise of Reynolds number. This is because the 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 provides higher than those for *e/H* = 0.25, 0.2, 0.15, 0.1 and wire coil (*CR* = 8). 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 the others.

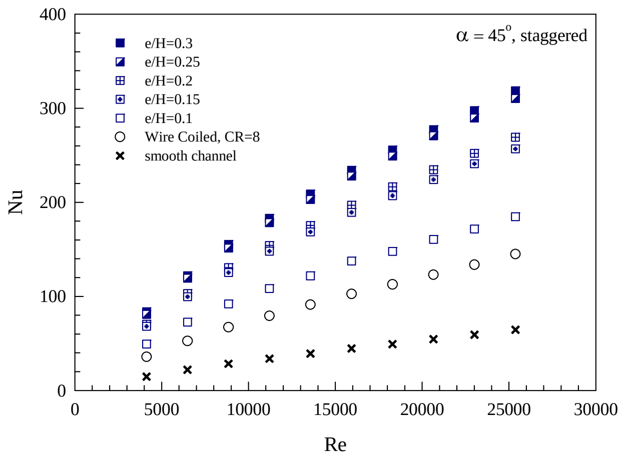

Fig. 4 Variation of Nusselt number with Reynolds number.

Fig. 5 displays the variation of the pressure drop shown in terms of friction factor with Reynolds number. In the figure, it is apparent that the use of 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 *e/H* = 0.3 is considerably higher than that of e/H = 0.25, 0.2, 0.15, 0.1 and *CR* = 8. For the baffle turbulators with *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.

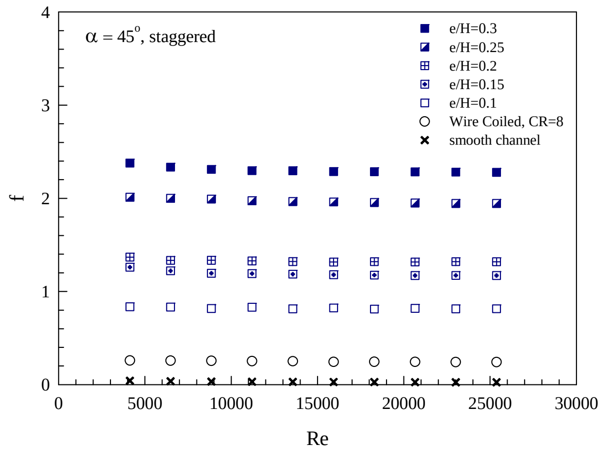

Fig. 5 Variation of friction factor with Reynolds number.

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 Fig. 6. In the figure, the Nusselt number ratio decrease with the rise of Reynolds number from 4000 to 25,000 for all cases of *e/H* and *CR* values. The mean Nusselt number ratio values are found to be about 5.29, 5.15, 4.46, 4.29, 3.10 and 2.32 times above the smooth channel for using the *e/H* = 0.3, 0.25, 0.2, 0.15, 0.1 and *CR* = 8, respectively.

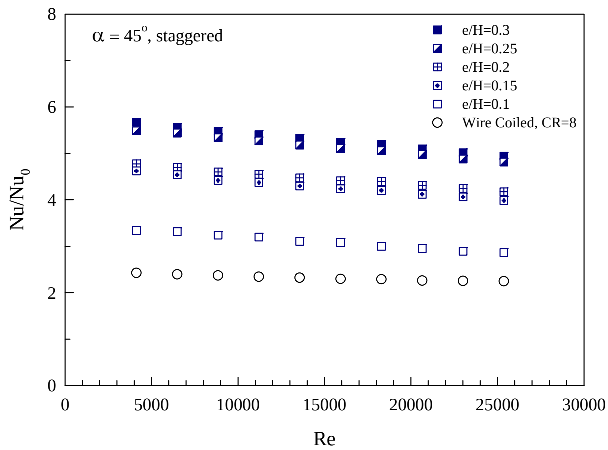

Fig. 6 Variation of Nusselt number ratio, Nu/Nu_0 with Reynolds number.

The variation of friction factor ratio value with Reynolds number values is depicted in Fig. 7. In the figure, it is visible that the friction factor ratio tends to increase with raising the Reynolds number value for all turbulators. The baffle turbulators provides a considerable increase in the friction factor ratio than that with the wire coil under the same operating conditions. The mean friction factor ratio values are around 76.55, 65.48, 44.15, 39.59, 27.31 and 8.28 fold for using the baffles with *e/H* = 0.3, 0.25, 0.2, 0.15, 0.1 and wire coil (*CR* = 8), respectively. This result indicates that the use of low blockage ratio can help to reduce the pressure loss considerably.

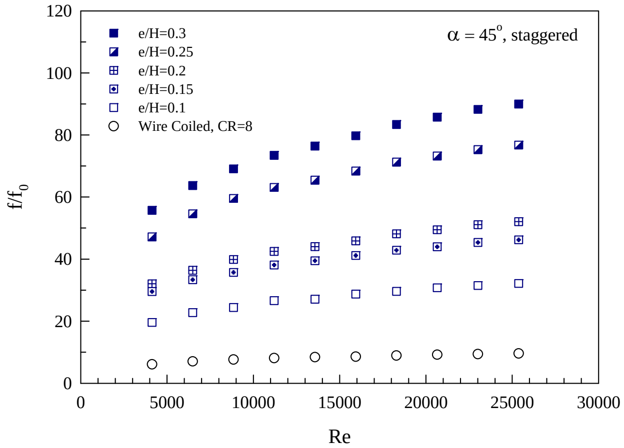

Fig. 7 Variation of friction factor ratio, f/f_0 with Reynolds number.

Fig. 8 shows the variation of the thermal enhancement factor (η) with Reynolds number for all turbulator cases. For all, the data obtained by Nusselt number and friction factor values are compared at a similar pumping power in order to determine the net final gain. There will be a net gain only is greater than unity. The thermal enhancement factor tends to decrease with the rise of Reynolds number values for all. It is seen that the baffle-to-channel height ratio or the blockage ratio of *e/H* = 0.25 shows the highest value of mean the thermal enhancement factor. It is worth noting that the thermal enhancement factor of the baffles *e/H* = 0.15 – 0.3 is higher than that with the wire coil for. The mean thermal enhancement factor values are around 1.29, 1.27, 1.26, 1.25, 1.04 and 1.13 for using the baffles with *e/H* = 0.25, 0.2, 0.15, 0.3, 0.1 and wire coil (*CR* = 8), respectively. The results are for Reynolds number of 4000 - 25,000 for the 45° stagger baffle and wire coil, the maximum thermal enhancement factor is found at the lowest value of Reynolds number.

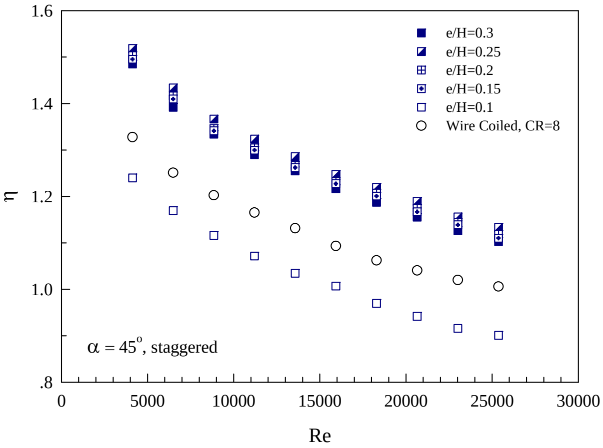

Fig. 8 Variation of thermal enhancement factor with Reynolds number.

5. Conclusion

An experimental study has been carried out to investigate airflow friction and heat transfer characteristics in a square channel fitted with different turbulators (baffles and wire coil) for the turbulent regime, Reynolds number of 4000-25,000. The use of the stagger baffles provide both of pressure drop increase and heat transfer augmentations higher than the wire coil. In comparison, the use of the baffle with *e/H* = 0.3 leads to the highest pressure drop and heat transfer rate but one with *e/H* = 0.25 provides the highest thermal enhancement factor at lower Reynolds number.

6. Acknowledgment

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7. References

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