

Influences of pitch-length louvered strip insert on thermal characteristic in concentric pipe heat exchanger

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Abstract. The effects of pitch-length louvered strip inserts with a forward arrangement on heat transfer and friction factor characteristics in a concentric pipe heat exchanger were experimentally investigated. Louvered strip insert was installed inside the tube. For comparison, the experiment without louvered strip (plain tube) was conducted. All cases were tested under steady state condition for Reynolds number ranging from 5300-17,500. The results show that the utilization of louvered strip insert produced a higher heat transfer rate compare to a plain tube. Nusselt number, friction factor, and heat transfer coefficient ratio increased with decreasing pitch length. The Nusselt number for louvered strip insert with S of 40, 50, and 60 mm increasing in the value of 67% - 77%; 48% - 53%; and 21% - 24% than plain tube, respectively. While the tube fitted with louvered strip insert by S of 40, 50, and 60 mm, friction factor increased 2.91 - 3.84; 2.41 - 2.92; and 1.88 - 2.08 times greater than a plain tube. The addition of louvered strip insert with S of 40, 50, and 60 mm producing heat transfer coefficient ratio in the range of 1.02 - 1.12; 1.03 - 1.07 and 1.03 - 1.04.

1 Introduction

Enhanced heat transfer is a method to increase the thermal performance system by enhancing heat transfer and flow characteristics. In the last years, enhanced heat transfer technology has been developed and extensively applied to heat exchanger applications; such as cooling devices, automotive, chemical industries and others. Some investigations have been done to increase the heat exchanger performance with the aim to diminish the size and cost of a heat exchanger, which can lead to more economical design. This economical design can help to make energy saving related to heat exchange process. The required pumping power influences energy saving in this case.

The significant effect of using heat transfer enhancement technology is the rise of pressure drop that leads to the increase of required pumping power. Pressure drop in the tube flow occurs due to the frictional losses with the solid surface. Therefore, while selecting the enhancement method should be optimal with the consequence of the rising heat transfer coefficient and the pumping power because of the rising pressure drop.

Among three enhancement methods, passive, active, and compound methods, the passive methods are mostly used to achieve the highest heat transfer rate. There are many kinds of mechanism of enhanced heat transfer by using passive methods such as modified surfaces or geometries, extended surfaces, and insertion. The insertion considered as the most advantageous passive method. There are several advantageous of the use of insertion (1) can be fitted on the smooth tube; (2) can be

formed in various shapes; (3) easy to install and operate. Researchers have been conducted a large numbers of experimental investigation using insertion methods. Including louvered strip, twisted tape, wire coil and helical wire coil to examine the thermal characteristics on each insertion.

The experiment related to a modification of twisted tape insert, Yaningsih et al. [1] experimentally investigated the heat transfer and flow characteristics of heat exchanger by employing perforated twisted tape inserts with a different axial pitch ratio. At the given Reynolds number, heat transfer rate, friction factor and thermal performance factor increases with decreasing axial pitch ratio. Sibel et al. [2] developed a study by using equilateral triangle cross-section coiled wire employed inside the tube. The use of insertion guided to a significant increase in heat transfer and pressure drop over the smooth tube. Mohammed [3] and Fan [4] studied numerically on the convective heat transfer characteristics of heat exchanger fixed with louvered strip insert with various slant angle and pitch length. In the tested range, the Nusselt number increases with the increasing slant angle and tend to decrease with rising pitch length. Eswaraiah [5] studied numerically on heat transfer enhancement using louvered strip inserts in the oil cooler. Numerical modelling was using computational fluid dynamics (CFD). These experiments were carried to compare the theoretical correlations and CFD simulated result. The result shows that a good agreement was found between the theoretical correlations and CFD simulated result. Eimsa-ard [6] experimentally studied the influenced of louvered strip insert with backwards and forward arrangement on the

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characteristics heat transfer and friction factor of concentric pipe heat exchanger. The results reveal that the using of louvered strip insert yielded the higher heat transfer rate compared with a plain tube. Pethkool [7] experimentally researched thermal characteristics of concentric pipe heat exchanger fitted with louvered strip insert with the different inclination angle. The results show that the Nusselt number and friction factor tend to increase with increasing inclination angle. Sarada et al. [8] using louvered square leaf inserts inside a horizontal tube heat exchanger to estimate the enhancement heat transfer rate for air in the presence of insert. The experimental investigation carried out by combination different arrangement of the direction of the airflow and inclined angle. The results show that Nusselt number and pressure drop increased. The overall enhancement ratio is highest for the highest inclined angle. Variation of the geometry louvered strip insert includes pitch and slant angle were investigated by Huisseune et al. [9]. They investigate the thermal and hydraulic performance of such a compound heat exchanger. They found that the combination between small fin pitch and large slant angle cause the best performance of the compound heat exchanger. Other experiment conducted by Raut and Farkade [10], they studied convective heat transfer

enhancements in a tube with three arrangements of louvered strip insert (forward, semi-forward, semi-backward, and forward-backward arrangements). The results show that the that the different arrangement has a various value of Nusselt number and friction factor.

The literatures above show that the using of insertion can increase the thermal performance of the heat exchanger. From literatures [3-10], there are many aspects that influence the use of louvered strip insert (geometry, the arrangement to the flow direction, and shape of louvered strip insert). The main mechanism of heat transfer enhancement due to insertion include the reduction of the hydraulic diameter and induced swirling flow makes a better fluid mixing [11]. Based on the literature this experiment will be carried out to investigate the thermal characteristics of concentric pipe heat exchanger installed with louvered strip insert. The innovative design and new materials of louvered strip insert were present in this experiment.

2 Experimental details

The schematic diagram of test rig is shown in Fig. 1.

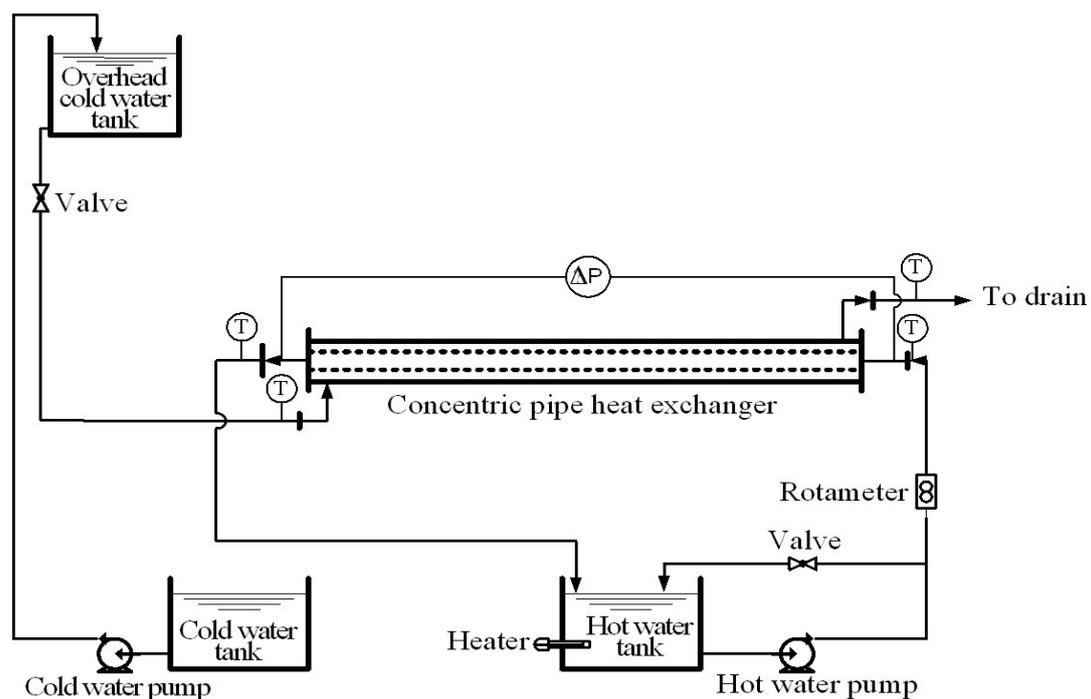


Fig. 1. Schematic diagram of concentric tube heat exchanger rig

The experimental concentric tube heat exchanger rig consists of concentric tube heat exchanger, cold water circulation system, hot water circulation system and instrumentation system to measure cold water and hot water flow rate, pressure and temperature. Concentric tube heat section made from the straight aluminum pipe consists of an outer pipe and inner pipe. The dimensions of the inner and outer tubes of the heat exchanger are: $d_i = 14.3$ mm, $d_o = 15.8$ mm, $L_i = 2500$ mm, $D_i = 23.4$ mm,

$D_o = 25.4$ mm and $L_o = 1950$ mm. The heat exchanger is oriented horizontally, and the fluids inside the pipe are in counter flow directions. Hot water was passed through the inner tube, while cold water was flowing through the annulus. The outer wall of the pipe is wrapped with the glass wool as insulation to minimize the heat losses to ambiances.

Cold-water circulation system consists of an overhead cold-water tank, cold-water tank, and cold-

water pump. The overhead water tank is used for supplying cold water using gravity flow with constantly flow rate at 0.103 kg/s. The inlet cold-water temperature was the ambient temperature that relative constant at $\pm 28^\circ\text{C}$. After passed through the system, cold water exposed to the surrounding. The hot water circulation system consists of hot water tank, flow regulation valves, heating system, and hot water pump. The heating system consists of a water tank with eight electric heaters, 4000 W for each electric heater. The inlet hot water temperature was kept constant at 60°C . Rotameter was used to measure the flow rates. The test runs are done at the hot water mass flow rates ranging between 0.033 kg/s and 0.099 kg/s. Data were collected at steady state conditions.

Temperature measurement was done by K-type thermocouples. K-type thermocouples were used for the measurement of inlet and outlet temperatures of cold and hot water fluids. Similar to the fluid, ten thermocouples were used to measurement of inner pipe outer wall. The temperatures were read by using a multi-channel digital thermometer. Pressure loss of the inner pipe side was determined by using U-tube manometer. The working fluid inside manometer was water. At present work, it is intended to find the changes in the convection heat transfer coefficients of the inner pipe side turbulent flow by affecting the regions near the wall of the pipe flow. For this objective, louvered strip insert was installed on the inner pipe side of the concentric double pipe heat exchanger as a tubulator. For comparison, this study also tested inner pipe without insert (plain tube).

The louvered strips that used in this experimental study were shown in Figs. 2 and 3.

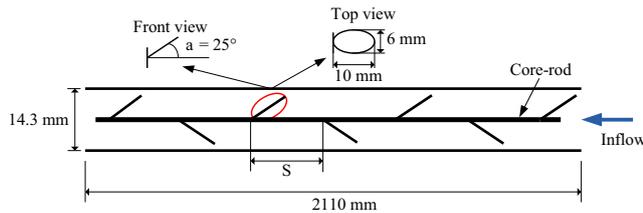


Fig. 2. Louvered strip with forward arrangements

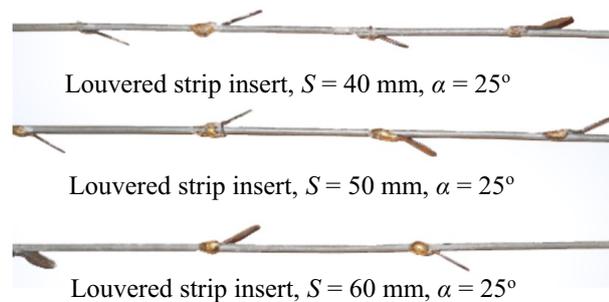


Fig. 3. Louvered strip with different pitch length (S)

Louvered strip inserts made from mild steel, 1 mm in thickness with an elliptical shape. The dimension of elliptical louvered strip insert was 6 mm, 10 mm respectively for minor and mayor axis. The louvered strip insert was connected to a central rod of 2 mm diameter as shown in the Fig. 2. The influenced of

different pitch length ($S = 40, 50, \text{ and } 60 \text{ mm}$) are studied in the present study. The slant angle (α) was kept constant at 25° . Louvered strips insert is on a forward arrangement with the direction of fluid flow.

Steady state values of the fluid and wall temperatures of the experiments were obtained for given mass flow rate of the hot water fluid. These values were used to compute heat transfer rate to the fluid flowing inside a tube. Nusselt number and friction factor were also calculated to know the effect of louvered strip inserts on heat transfer and friction characteristics. The following expression were used for calculation of heat transfer rate of hot and cold water (Q), heat transfer coefficient (h), Nusselt number (Nu), friction factor (f), Reynolds number (Re) and thermal performance factor (η). Heat transfer rate from hot water in the inner tube:

$$Q_h = \dot{m} \cdot C_{p,h} \cdot (T_{h,i} - T_{h,o}) \quad (1)$$

Heat transfer rate from cold water in the annulus:

$$Q_c = \dot{m} \cdot C_{p,c} \cdot (T_{c,o} - T_{c,i}) = h_o \cdot A_o \cdot (\bar{T}_{w,o} - T_{b,o}) \quad (2)$$

where

$$T_{b,o} = \frac{T_{c,i} + T_{c,o}}{2} \quad \text{and} \quad \bar{T}_{w,o} = \frac{\sum T_{w,o}}{10} \quad (3)$$

$\bar{T}_{w,o}$ is the average of the outer wall inner tube temperature. There are ten point measurements located on the outer wall inner tube. $T_{b,o}$ is the bulk cold temperatures, computed from the average of inlet and outlet cold water temperature. The differences between equations (1) and (2) indicate the convective heat loss, which may be assumed to be negligible:

$$Q_{loss} = Q_h - Q_c \quad (4)$$

An average heat transfer rate between hot and cold water,

$$Q_{ave} = \frac{Q_h + Q_c}{2} \quad (5)$$

Overall heat transfer is:

$$U_i = \frac{Q_{ave}}{A_i \cdot \Delta T_{LMTD}} \quad (6)$$

while an average convection heat transfer coefficient, h_i , obtained from the overall thermal resistance consist of three resistances in series: convective thermal resistance of the inner tube, conductance thermal resistance of the inner tube wall, and convective thermal resistance on the annulus side:

$$\frac{1}{U_i \cdot A_i} = \frac{1}{h_i \cdot A_i} + \frac{\ln(d_o/d_i)}{2\pi k_p L} + \frac{1}{h_o \cdot A_o} \quad (7)$$

Thus h_i can be computed from equation (7) as:

$$h_i = \frac{1}{\left[\frac{1}{U_i} - \frac{d_i \cdot \ln(d_o/d_i)}{2k_p} - \frac{d_i}{d_o \cdot h_o} \right]} \quad (8)$$

The mean Nusselt is expressed as follows:

$$Nu_i = \frac{h_i \cdot d_i}{k_i} \quad (9)$$

The friction factor is defined as:

$$f = \frac{\Delta P}{(\rho u^2/2)(L/d_i)} \quad (10)$$

The Reynolds number can be calculated as:

$$Re = \frac{\rho \cdot u \cdot d_i}{\mu} \quad (11)$$

The fluid thermo-physical properties of the water are determined at the mean bulk fluid temperature (T_b).

Thermal performance factor (η) is defined as the ratio of the heat transfer enhancement ratio to the friction ratio at the same pressure drop. The thermal performance factor can be written as:

$$\eta = (Nu_t / Nu_p) / (f_t / f_p)^{1/3} \quad (12)$$

The present experimental data will be compared with standard correlations for Nusselt number and friction factor under a turbulent flow regime. The Nusselt number obtained from the present plain tube are compared with those from the proposed correlations by Petukhov (13) and Gnielinski (14) [12] and friction factor proposed correlation by Blasius (15), [13]. Nusselt number correlation from Petukhov is:

$$Nu = \frac{(f/8) Re \cdot Pr}{1.07 + 12.7(f/8)^{1/2} \cdot (Pr^{2/3} - 1)} \quad (13)$$

for $10^4 < Re < 5 \times 10^6$.

Nusselt number correlation from Gnielinski:

$$Nu = \frac{(f/8)(Re-1000) \cdot Pr}{1 + 12.7(f/8)^{1/2} \cdot (Pr^{2/3} - 1)} \quad (14)$$

for $10^3 < Re < 5 \times 10^6$.

Friction factor correlation from Blasius:

$$f = 0.3164 \cdot Re^{-0.25} \quad (15)$$

for $4 \times 10^3 < Re < 10^5$

3 Results and discussion

3.1 Validation

The present data (Nusselt number and friction factor) of plain tube were compared with the empirical

equations (13-15) to confirm the accuracy and reliability of the experimental setup. Figure 4 demonstrated the comparison between experimental data and the standard equation for Nusselt number.

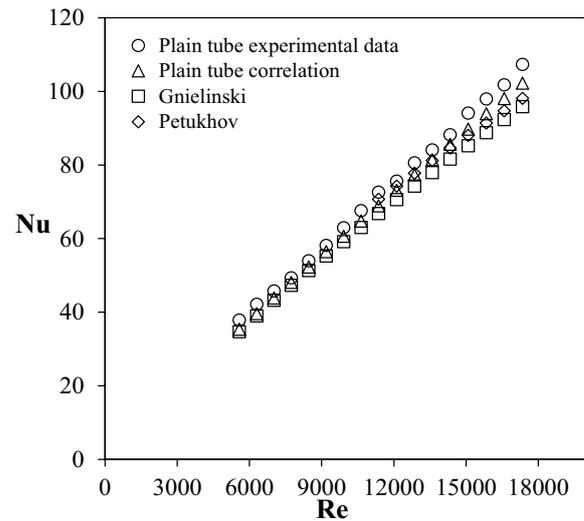


Fig. 4. Nusselt number validation for plain tube

From Fig. 4, it appears that the Nusselt number of the present data deviated from those of Dittus-Boelter and Pethukov empirical expression within $\pm 2.78\%$ and $\pm 7.05\%$ respectively. Figure 5 shows the comparison between friction factors from the present data with Blasius correlation.

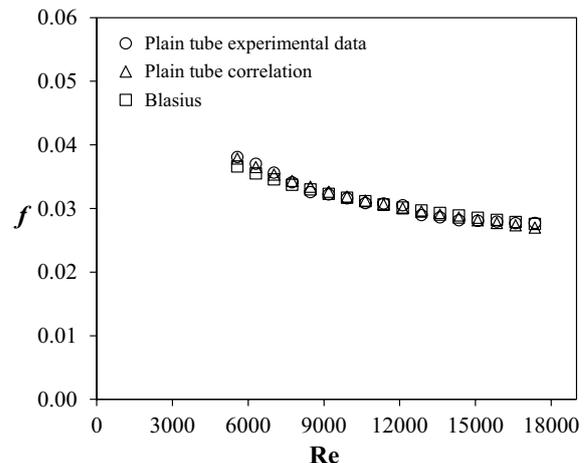


Fig. 5. Friction factor validation for plain tube

The friction factor deviated from the proposed correlation by Blasius within $\pm 2.46\%$. The comparisons imply that the present data accord well with the correlations, validating the reliability of the present experimental setup and method.

The present results are correlated with the Nusselt number and friction factor for the plain tube as follows:

$$Nu_i = 0,007 Re^{0,946} Pr^{0,3} \quad (16)$$

$$f = 0,495 Re^{-0,298} \quad (17)$$

Equations (16) and (17) are found to represent the experimental data within $\pm 6.6\%$ for Nusselt number and $\pm 2.6\%$ for friction factor, as shown in Figs. 3 and 4 respectively.

3.2 Heat transfer characteristic

Figure 6. depicted the heat transfer characteristics of the tube in term of Nusselt number (Nu) with Reynolds number (Re).

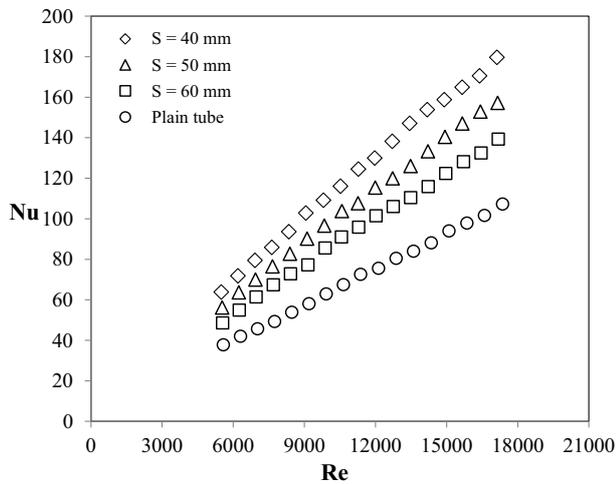


Fig. 6. Nusselt number vs Reynolds number

For all cases, the results present that the Nusselt number (Nu) rises with the increase of Reynolds number. The results are similar with previous experiment [1]. For different pitch-length (40, 50, 60 mm), the Nusselt number tend to increase with decreasing the pitch-length. The experimentation showed that pitch-length 40 mm provided the maximum value. However, pitch-length 60 mm has the lowest value of Nusselt number. The reasons why the excellent heat transfer was produced by smallest pitch-length is with the smaller pitch-length value resulting the higher compactness of the leaves of the louvered strip insert. The effects of the higher compactness are there was more streamlined pattern breaks up of fluid flows, therefore the flow between the elements of the leaves produces higher turbulence intensity due to rapid fluid mixing. Particularly on a pitch that was getting closer [3]. At given Reynolds number, the comparison between the plain tube and louvered strip insert with different pitch-lengths the Nusselt number found to be 67% - 77%; 48% - 53%; and 21% - 24%, respectively.

3.3 Friction factor

Friction factor characteristic is shown in Fig. 7. In term of friction factor with Reynolds number. The friction factor is inversely proportional to Reynolds number for all conditions. These results are in accordance with previous experiment [1]. The friction factor becomes greater for smaller pitch-lengths and decreases for the higher pitch length. The highest value of friction factor was found at pitch-length 40 mm, while the plain tube gave the minimum. Concentric tube fitted with louvered strip insert, the values of friction factor are decreasing with the rise of pitch-length [4]. In the range of $5.300 < Re < 17.500$ friction factor of the inner tube fitted with louvered strip insert with different pitch

length (40, 50, 60 mm) were 2.91 – 3.84; 2.41 – 2.92; and 1.88 – 2.08 times greater than the plain tube.

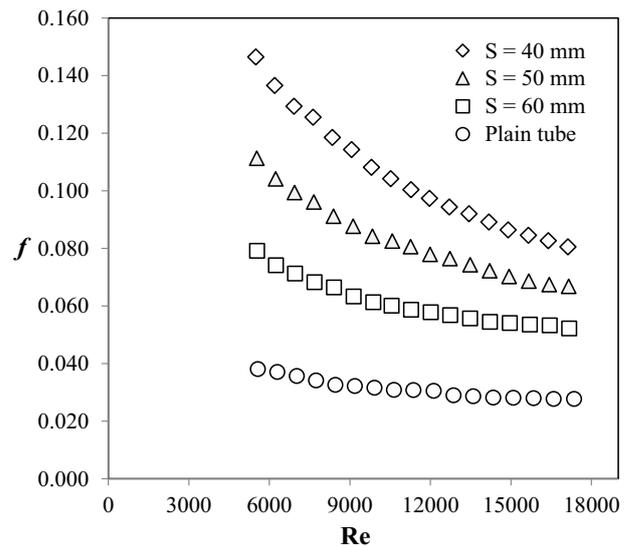


Fig. 7. Friction factor vs Reynolds number

3.4 Thermal performance factor

Thermal performance factor is used to evaluate the quality of the enhancement technique used. The thermal performance factor gradually increased with the rise of Reynolds number as shown in Fig. 8.

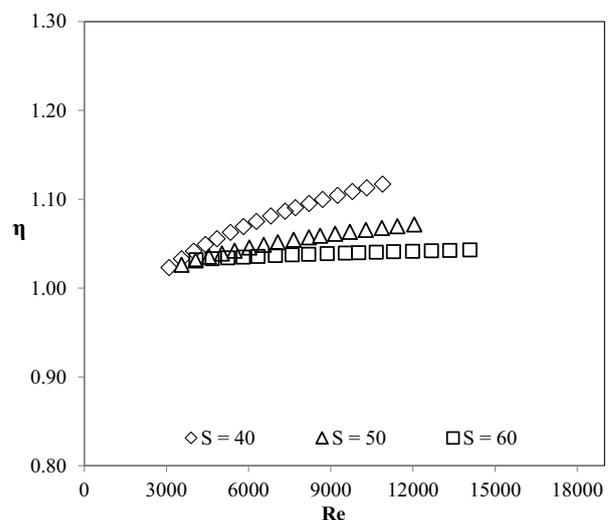


Fig. 8. Thermal performance factor vs Reynolds number

Thermal performance factor is observed highest for pitch-lengths 40 mm. The values of thermal performance factor of the inner tube with louvered strip insert at various pitch-lengths (40, 50, 60 mm) were found 1.02 – 1.12; 1.03 – 1.07 and 1.03 – 1.04 times greater than the plain tube.

4 Conclusions

The experimental study is conducted to enhance the performance of concentric pipe heat exchanger by

installing louvered strip insert with various pitch-lengths. The results can be concluded as follows:

- 1) Louvered strip enhanced the thermal performance of the concentric tube heat exchanger. Nusselt number, friction factor and thermal performance factor tends to be higher than the plain tube.
- 2) The influenced of the different pitch-length on the thermal performance, the smallest pitch length gave the higher Nusselt number, friction factor and thermal performance factor.
- 3) The 40, 50, 60 mm of pitch lengths augmented average heat transfer 67% - 77%; 48% - 53%; and 21% - 24%, respectively at Reynolds number 5300 – 17,500 compared to the plain tube.

Nomenclature

A_i	inner surface area of inner pipe (m ²)
A_o	outer surface area of inner pipe (m ²)
C_p	specific heat capacity (J/kg.°C)
d_i	inner diameter of inner pipe (m)
d_o	outer diameter of inner pipe (m)
f	friction factor (dimensionless)
f_p	friction factor of the plain tube (dimensionless)
f_i	friction factor of the inner pipe with louvered strip insert (dimensionless)
h_i	average convective heat transfer coefficient of the inner pipe side (W/m ² .°C)
h_o	average convective heat transfer coefficient of the annulus side (W/m ² .°C)
h_p	average convective heat transfer coefficient of the plain tube (W/m ² .°C)
k_i	thermal conductivity of hot water (W/m.°C)
k_p	thermal conductivity of the inner pipe material (W/m.°C)
L_i	length of the inner pipe (m)
L_o	length of the outer pipe (m)
Nu_i	average Nusselt number of the inner pipe side (dimensionless)
Nu_p	average Nusselt number of the plain tube (dimensionless)
Nu_t	average Nusselt number of the inner pipe side with louvered strip insert (dimensionless)
Pr	Prandtl number (dimensionless)
Q_h	heat transfer rate from the hot water in the inner pipe (Watt)
Q_c	heat transfer to the cold water in the annulus (Watt)
Re	Reynolds number (dimensionless)
S	pitch length
T_b	bulk water temperature (°C)

T_c	cold water temperature (°C)
T_h	hot water temperature (°C)
T_w	outer wall inner pipe temperature (°C)
u	velocity of hot water in the inner pipe (m/s)
U_i	overall heat transfer coefficient based on the inner pipe inside surface area (W/m ² .°C)

Greek symbol

α	slant angle (°)
ρ	density of hot water (kg/m ³)
μ	dynamic viscosity of hot water (kg/m.s)
η	thermal performance factor (dimensionless)
ΔP	pressure drop across the inner pipe (Pa)
ΔT_{LMTD}	logarithmic mean temperature difference (°C)

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