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THERMO-HYDRAULIC PERFORMANCE ASSESMENT OF HEAT EXCHANGER TUBE WITH PERFORATED HOLLOW CIRCULAR DISK INSERT

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Abstract - Heat transfer enhancement by creating turbulence in the physical behavior of fluid flow inside the heat exchanger has become a very interesting area of research for the researchers. Although very significant results has been achieved in the thermal performance of heat exchangers, especially in the range of lower Reynolds number, but still these passive approaches of heat transfer enhancement is not effective for the range of higher Reynolds number. In the present work the effect of perforated circular disk turbulators on heat transfer, friction factor and thermal performance of heat exchanger is evaluated experimentally. The different parameters used for the experiment include fixed diameter ratio (d/D=0. 8), pitch ratios (l/D= 1, 2 & 3) and perforation Index (P_a/T_a = 0%, 8%, 16% & 24%). The experiment is done in the range of Reynolds number lying from 6,500 to 26,800. On the basis of experimental observation, there is 2.2-3.54 times improvement in heat transfer and around 1.18-1.47 times improvement in thermal performance factor over smooth tube heat exchanger.

Index Terms: Heat transfer, friction factor, thermal performance factor, perforated hollow circular disk, twiated tape, pitch ratio, diameter ratio.

1 Introduction

Enhancement of heat transfer in the heat exchanger is of prime focus for the researchers. Several methods of heat transfer enhancement are adopted in case of heat exchangers, in which passive method [1] of heat transfer enhancement has become a wide area of interest for researchers all over the world. In the

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passive method of heat transfer enhancement, turbulent promoters of different geometries with several flow and geometrical parameters are used to study heat transfer and fluid flow characteristics of heat exchanger tube. These turbulent promoters are placed between the fluid path in a systematic order to improve the heat transfer and thermal performance of the heat exchangers. The use of twisted tapes [2-4] as insert geometry showed significant improvement in the amount of heat transfer and thermal performance factor for the tube heat exchangers. Halit Bas et al. [2] used single twisted tape, Bhuiya et al [3-4] used double and triple twisted tape in their experiments. Eiamsa-ard et al. [5] used combination of circular ring with twisted tape as insert geometry, which also showed a significant improvement in heat transfer as compared to smooth tube heat exchangers. Perforation in insert geometry like perforated twisted tape and perforated conical rings were also used by some researchers (Bhuiya et al. [6], Kongkaitpaiboon et al. [7]). Kongkaitpaiboon et al. [8] and Promvonge et al. [9] used circular ring and inclined vortex ring respectively in their study. Surface modification by using coiled wire is also an area of interest for the researchers (Gunes et al. [10])

So, considering effect of perforation and surface modification, in this work perforated hollow circular disk (PHCD) is used as insert geometry with a range of geometrical and flow parameters as shown in Tab. 1.

TABLE 1: PARAMETERS RANGE

Serial number	Parameters	Range
1	Pitch ratio (l/D)	1-3
2	Diameter ratio (d/D)	0.8
3	Perforation index (P_a/T_a)	0% - 24%
4	Reynolds number	6,500 – 26,800

In this study, some assumptions are made in order to avoid minor losses during the experimentation. These assumptions are:

i. Test section is perfectly insulated hence there is no heat loss from the test section.

ii. Heat flux is uniformly distributed throughout the test section.

iii. Heat transfer is taking place only through forced convection.

NOMENCLATURE

Re	Reynolds number
Nu	Nusselt number
Pr	Prandtl number
f	Friction factor
η	Thermal performance factor
PR	Pitch ratio
DR	Diameter ratio
PI	Perforation index
P_{atm}	Atmospheric pressure
$\dot{m}$	Mass flow rate of fluid
D	Hydraulic diameter
d	Internal diameter of insert
C_d	Coefficient of discharge
L	Length of test section
l	Distance between two consecutive inserts
h	Convective heat transfer coefficient
K	Thermal conductivity of air
T_a	Total area of insert geometry
T_p	Perforated area of insert geometry
T_w	Local wall temperature
T_b	Bulk mean temperature
T_{wm}	Mean wall temperature
T_i	Fluid inlet temperature
T_o	Fluid exit temperature.
ΔP	Pressure drop across test section
ΔP_o	Pressure drop across orifice plate

2 Methodology

Experimental facility mainly comprised of inlet section also called as calming section, test section, heating arrangement and air supply system (blower).

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Inlet section is 2.5 m long and its main purpose is to allow fully developed flow to enter into the test section and to avoid flow separation and stratification. Inlet section is followed by test section which is 1.4 m long. It comprises of heating arrangement followed by insulation at the outer surface of tube. T-type thermocouples are placed near the tube wall, inlet and exit of flow stream to sense the wall and fluid temperature with help of data logger which is attached to computer with labview software. For measuring pressure drop across the test section, digital micro manometer is used, which is attached to knob placed at inlet and exit of test section. Test section is followed by flow arrangement which comprises of blower with 2 HP, 3 Φ power supply. In between blower and test section, flow measurement device is attached. The flow measurement device comprises of orifice plate with U-Tube manometer filled with water as manometric fluid. Adjacent to blower flow control valve is attached to control the flow rate in order to vary fluid flow rate during the experimentation. Shemetic diagram of experimental test setup is shown in Fig. 1.

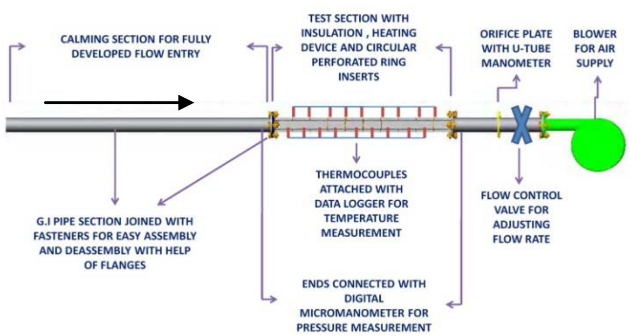
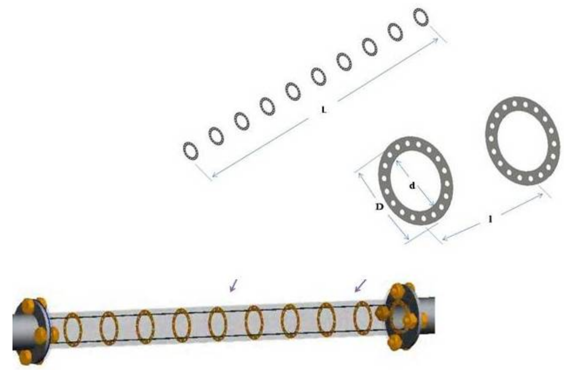


FIGURE 1. SCHEMATIC VIEW OF EXPERIMENTAL SETUP.

As discussed above perforated hollow circular disk is used as insert geometry with different geometrical and flow parameters in this study. A schematic diagram of PHCD assembly is shown in Fig. 2.



3. Data reduction and validation

At the steady state condition of the experiment, it is assumed that convective heat transfer from the wall surface of heat exchanger tube is equal to the heat gain by the fluid circulating in the tube. Hence the energy balance equation can be written as:

$$Q_{air} = Q_{conv.} \qquad (1)$$

Where,

$$Q_{air} = mC_p\,(T_o - T_i) \qquad (2)$$

And the convective heat transfer from the heat exchanger wall is given by:

$$Q_{conv} = hA(T_{wm} - T_b) \qquad (3)$$

Where T_b is bulk mean temperature of fluid and it is calculated by equation:

$$T_b = (T_o + T_i) / 2 \qquad (4)$$

And

$$T_{wm} = \Sigma\, T_w/12 \qquad (5)$$

Where, 'T_w' is the local wall temperature at which thermocouple was placed. 'T_{wm}' represents mean wall temperature, which was again the average temperature of the total thermocouple placed on the wall surface.

It is assumed that temperature distribution is uniform throughout the wall surface. The average heat transfer coefficient, 'h, and the average Nusselt number, Nu are estimated as follows:

$$h = mC_p\,(T_o - T_i)/\,A(T_{wm} - T_b) \qquad (6)$$

And,

$$Nu = hD/k \qquad (7)$$

The value of Reynolds number is obtained by standard equations mentioned below:

$$Re = \rho VD/\,\mu \qquad (8)$$

and the value of Friction factor is obtained by:

$$f = \Delta P/\,\{(L/D)\,(\rho V^2/2)\} \qquad (9)$$

In which V is the mean velocity of the tube. All of thermal properties of the tested fluid are determined at the bulk mean temperature of fluid.

Finally the thermal performance factor is calculated by the equation:

$$\eta = (Nu/Nu_s)\,/\,(f/f_s) \qquad (10)$$

A thorough check of the instruments and the test set-up was followed by experimentation on conventional smooth pipe. The average values of Nusselt number and friction factor were determined. These values are compared with the values obtained from the standard correlation like Blasius equation for friction factor and Dittus-Boelter equation for Nusselt number in case of smooth tube. The standard equations for Friction factor and Nusselt number is given as:

Blasius equation

$$f_s = 0.316 \times Re^{-0.25} \qquad (11)$$

Dittus-Boelter equation

$$Nu_s = 0.023.Re^{0.8}.Pr^{0.4} \qquad (12)$$

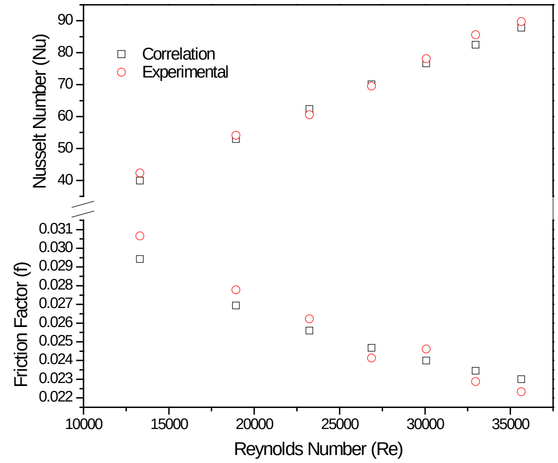


FIGURE 3. VALIDATION PLOT

Experimental calculated value, Fig. 3 shows the validation of friction factor and Nusselt number over the standard smooth tube equation. Deviation of ±5 % is seen in the case of Nusselt number and ±7% in case of friction factor, as compared to smooth tube heat exchanger. This small deviation in experimental result allows proceeding with the experimentation. Uncertainty in measurement for Reynolds number, friction factor and thermal performance factor were 6.13%, 9.83% and 5.04%.

4. Results and disscusssion

This study is carried out on the basis of experimental results obtained from heat exchanger tube fitted with perforated hollow circular disk inserts with different geometrical and flow parameters which is shown in Tab. 1. Firstly, on the basis of experimental result obtained for smooth tube is validated by standard correlations [Fig. 3], then further experimentation is carried out in order to study effect of each parameter on heat transfer, friction factor and thermal performance factor of heat exchanger tube.

Effect on heat transfer

Enhancement of heat transfer was of prime focus during the experimentation. This study showed 3.54 times improvement in heat transfer as compared to smooth tube heat exchangers for the lower range of Reynolds number. For higher range of Reynolds number this augmentation is found to be of order 3.4 times as compared to smooth tube heat exchanger.

According to graph obtained from experimental results as shown in Fig. 4, Pitch ratio (PR) showed significant effect on the heat transfer as compared to smooth tube heat exchanger. As the value of pitch ratio decreases, heat transfer increases significantly and for PR=1 heat transfer is maximum. Because of closely spaced PCDI in case of PR=1, fluid stream detachment and reattachment is frequent. This detachment and reattachment of fluid stream causes proper mixing of fluid, which results to high heat transfer.

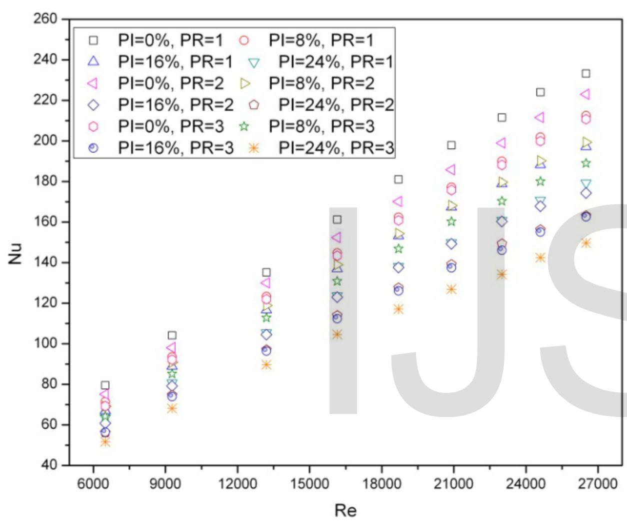


FIGURE 4 (a) Variation in Nusselt number (Nu) with Reynolds number (Re)

Phenomenon of jet impingement during the fluid flow is also dominant reason for heat transfer enhancement. Because of jet impingement flow obstruction takes place, which causes generation of eddy and vortex flow near the insert geometry, as the number of eddy and vortex formulation increases, heat transfer rate increases.

With increase in amount of perforation index (PI) increases, heat transfer decreases significantly. The effect of jet impingement is reduced due to higher hole diameter and thus corresponds to lower heat transfer rate. Maximum heat transfer is obtained in case of 0% perforation index and pitch ratio 1, which was 3.54 times as compared to smooth tube heat exchanger.

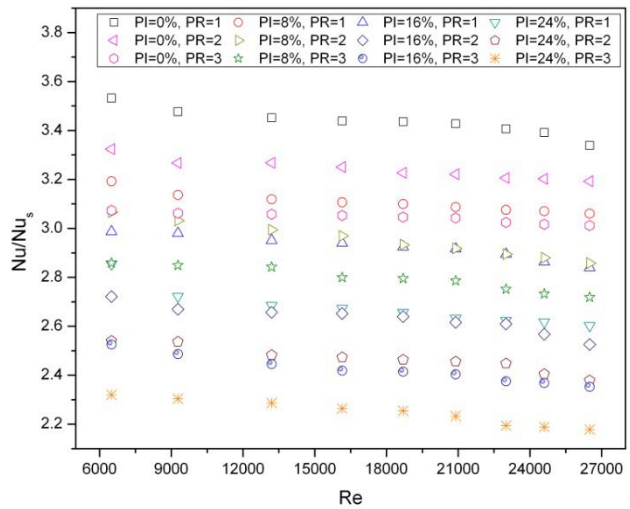


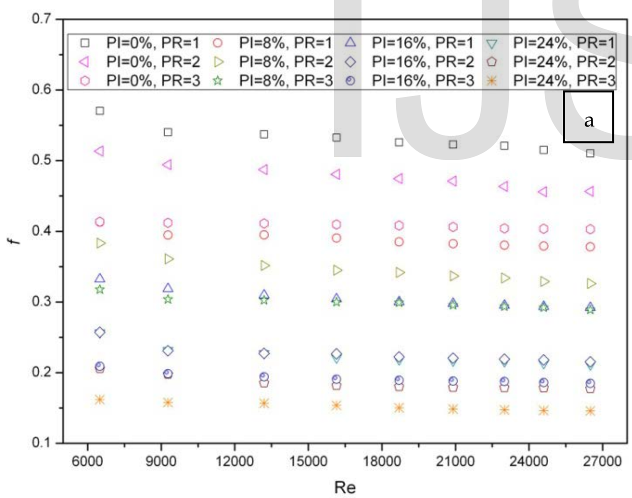
FIGURE 4. (b) Variation in Nusselt number enhancement (Nu/Nu_s) with Reynolds number (Re)

Effect on friction factor

In order to improve thermal performance factor of heat exchangers, it is very necessary to control its friction factor. The friction factor is directly dependent upon the pressure drop across the test section and inversely proportional to the square of fluid velocity. So for the value of higher Reynolds number the friction factor is minimum, but as the Reynolds number decreases the value of friction factor increases as velocity is lower in case of lower Reynolds number. Since, there is use of inserts in the heat exchanger; hence, increase in friction factor is obvious because of disturbance and obstruction in flow stream. Therefore, it is necessary to find a insert parameter which shows minimum impact on the value of friction factor.

From Fig. 5, it is observed that all the parameters have their dominant influence on the value of friction factor. With increase in the value of pitch ratio, friction factor decreases. This is because of increase in PCDI spacing, as the spacing between the perforated circular disk increases, flow obstruction decreases, which results in decrease of pressure drop. Lowest value of friction factor was observed in case of PR=3.

Perforation index (PI) plays a major role in controlling the friction factor value. In case of 0% perforation, there is huge flow blockage. Because of this high flow blockage, pressure drop across the test section increases in large extent. This results in high value of friction factor. In order to control this high friction factor perforation index parameter was introduced in the experimentation. There was a drastic decrease in the value of friction factor as perforation index increases. Minimum friction factor was obtained for 24% perforation index and pitch ratio 3, which was only 4 times more in the lower range of Reynolds number as compared to smooth tube heat exchangers. As the amount of perforation increases flow disturbance decreases with results in a lower value of pressure drop, which decreases the friction factor.



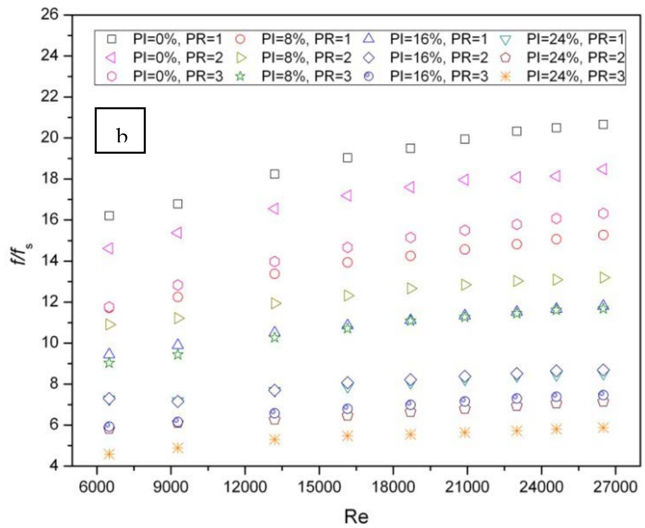


FIGURE 5. VARIATION IN FRICTION FACTOR WITH REYNOLDS NUMBER (Re), (a) f and (b) $f\,/\,f_s$

Effect on thermal performance factor

Thermal performance factor is a very important aspect for the utilization of heat exchanger. There is a significant enhancement in the value of heat transfer and friction factor is observed for different set of geometries used in the experimentation. So, it is very important to use only such geometry in which thermal performance factor is maximum. As represented in Fig. 6, different insert and flow parameter showed different impact on the value of thermal performance factor. It was found that for the lower value of the Reynolds number, thermal performance factor is high, and as the value of flow parameter increases, the thermal performance factor decreases to a great extent.

With increase in the value of pitch ratio, the thermal performance factor decreases, and this is observed due to higher Nu/Nu_s and minimum f/f_s value for the lower range of Reynolds number. And as pitch ratio increases thermal performance factor decreases because of minimum Nu/Nu_s and maximum f/f_s value in case of higher Reynolds number.

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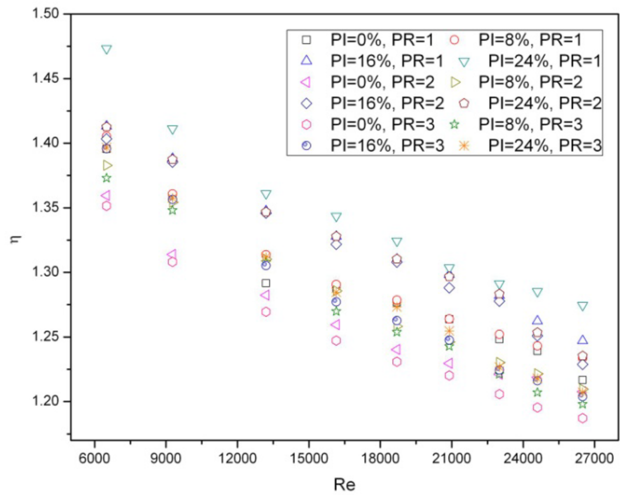

FIGURE 6. VARIATION IN THERMAL PERFORMANCE FACTOR (η) WITH REYNOLDS NUMBER (Re).

There is a significant effect of perforation index recorded on thermal performance factor. As the value of perforation index increases thermal performance factor also increases. This is because of drastic decreases in the value of friction factor with increase in perforation index. Maximum thermal performance factor is observed for PR=1 and PI=24%, in the lower range of Reynolds number. There is 1.47 times improvement found in thermal performance factor as compared to smooth tube heat exchangers.

Conclusion

On the basis of experimental study on heat exchanger tube with perforated circular disk insert, it is observed that each geometrical and flow parameters used in experimentation showed there independent impact on the value of heat transfer, friction factor and thermal performance factor. On the basis of results obtained following conclusions are made:

- The validation test of the experimental setup is carried out and, error in the value of Nusselt number and Friction factor were recorded as ±4% and ±7% respectively. This shows that the error was in the permissible range for conducting experimentation.
- For heat transfer, as the value of the Reynolds number increases heat transfer rate also increases and vice-versa. At the maximum value of the Reynolds number, the amount of heat transfer was found maximum. When compared to the smooth tube, maximum enhancement in heat transfer was obtained for the lower value of Reynolds number i.e. 6,500 and the geometrical parameter for which maximum heat transfer was obtained is for PR=1 and PI=0%. Enhancement in heat transfer for this case was around 3.54 times as compared to smooth tube heat exchanger and for the higher value of the perforation index heat transfer decreases significantly. In case of pitch ratio also similar result was obtained. For lower PR, heat transfer is maximum and vice-versa.
- For Friction factor, as the Reynolds number increases the value friction factor decreases and vice-versa. For the higher value of flow parameter friction factor is minimum. As compared to the smooth tube heat exchanger minimum value of friction factor was obtained in the lower range of flow parameter and it was 4 times higher, this minimum value of friction factor was obtained for the value of geometrical parameter, PR=3 and PI=24%. As perforation index and pitch ratio decreases value of friction factor increases significantly.
- Thermal performance factor was found maximum for the given value of geometrical parameter, i.e. PI=24%, PR=1. It was also observed that in the lower range of flow parameter, i.e. for Reynolds number 6,500, Thermal performance factor was maximum and as the value of Reynolds number increases, Thermal performance decreases respectively. Maximum Thermal performance factor obtained was observed 1.47 times higher than smooth tube heat exchangers. As PR decreases thermal performance factor increases and vice-

versa, and as PI increases thermal performance factor also increases as compared to lower value of PI.

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