RESEARCH ARTICLE-CHEMICAL ENGINEERING

Experimental Investigation on the Effect of Semi-circular Perforated Baffles with Semi-circular Fins on Air–Water Double Pipe Heat Exchanger

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Abstract

The heat transfer performance of the air–water double pipe heat exchanger (HE) was experimentally investigated. Segmental baffles with semi-circular perforations were introduced in the annular side of HE. Each baffle contains semi-circular fins. Air was used as a working fluid in the annular while, water for the inner tube. Seven different air Reynold's numbers range from 2700 to 4000 were selected. While water side Reynold's number fixed at 34,159. Three different semi-circular perforation diameters were studied (30, 25, and 20 mm) to investigate their effect on HE thermal performance. The Nu number, overall heat transfer coefficient, friction factor, and thermal performance factor (TPF) were calculated. Also, compassion for the baffled and un-baffled pipe was done. It was observed that the thermal performance of the baffled pipe HE was better than the un-baffled one. The average overall heat transfer coefficient increases by 29.7%, 62%, and 80.6% by using perforated baffles with 30, 25, and 20 mm perforation diameters. The TPF of the heat exchanger with all studied perforated baffles cases is above unity and the best thermal performance obtained by using baffles was with 20 mm perforation diameter.

Keywords Double pipe heat exchanger · Heat transfer enhancement · Perforated baffles · Segmental baffles

List of symbols

A	Surface area, m^2
A_c	Cross-sectional area, m^2
Cp	Specific heat, kJ/kg.K
d	Diameter, m
D_i	Inside diameter of the annular pipe, m
D_o	The outside diameter of the annular pipe, m
D_H	Hydraulic diameter, m
F	Friction factor
H	Heat transfer coefficient, W/m^2.K
K	Thermal conductivity, W/m.K
L	Length of the tube, m
M	Mass flow rate, kg/s
Nu	Nusselt number
P	Perimeter, m
Pr	Prandtl number
Q	Heat transfer rate, W
Re	Reynolds number
T	Temperature, K
ΔT_m	Log mean temperature difference, K
ΔP	Pressure drop, bar
U	Overall heat transfer coefficient, W/m^2.K
Un	Uncertainty
V	Velocity, m/s

Greek symbols

μ	Dynamic viscosity, kg/m.s
ρ	Density, kg/m^3

Subscript

A	Air
an	Annular
I	Inlet
O	Outlet
W	Water

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1 Introduction

Heat exchangers (HEs) have numerous applications in industries. They have been using to transfer and regulate heat between different temperature media such as in gas turbine,

air conditioning, heat pump, food processing, and cryonic [1]. There were different kinds of HE according to their specific requirements like a double pipe, shell and tube, plate and frame, and heat sinks [2]. Double pipe HE is the simplest type that was used in a wide range of industries. It was the cheapest and easiest method to exchange or dissipate heat but it needs more space compared to other types.

There are many used techniques to enhanced HE heat transfer rate. These techniques can be classified as passive and active techniques [3, 4]. Generally, passive techniques provide heat transfer enhancement without the need for any external power or additional devices. It can be enhanced heat transfer by using extended surface, twisted tape inserts, additives, baffles, and different tube geometries[5–8]. Baffles consider an effective passive method used to generate vortex flow and provide good mixing in the shell side of a HE. Existing baffles increase turbulence and eliminate the dead regions leads to an increase in the heat transfer rate [9]. Furthermore, baffles are used to support the internal pipe to prevent vibration and bending. Many types of baffles were used like helical, segmental, ring, trapezoidal, and flower [10–13]. Moreover, many enhancement technique methods could be applied by baffles to increase the thermal performance. One of these methods is using porous Das et.al [14–16]. [17, 18] or by adding smooth or shaped baffles which increase the turbulence and temperature distributions along the heat exchange surface, Xiao et.al [19]. studied numerically the effect of using segmental and helical baffles with different tilts angle on the heat transfer performance of a shell and tubes HE. Five different fluids with various Nu numbers were studied and compared.

The results showed that the best heat transfer coefficient when water is the working fluid in the shell side with 40° tilts helical baffles. Moreover, like the Nu Number increases on the shell side, a small tilt angle of baffles is the best choice. Similarly, Wang et.al. [20] proposed a numerical configuration optimization by using helical baffles with different tilts angle. The best thermal performance was obtained when the flow rate at the shell side with maximum tilts angle values, while Wen et.al [21] studied numerically the effect of helix angle and overlapped degree of the helical baffles on the thermal performance of HE. The results showed that the heat transfer coefficient and ΔP are greatest with less helix angle, and the overlap degree had no significant effect on the H.T coefficient. Dong et.al [8]conducted numerically the heat transfer and flow performance of HE with trisection helical baffles. Four different baffles geometries were modeled. Among these four geometries, the circumferential overlap helical baffles gave the highest heat transfer coefficient 16.5%, 27.3%, and 13.5% compared to the other baffles geometries. Also, Dong et.al. [22] studied a further improvement to the helical baffles geometries. They found the heat transfer rate was significantly increased by using the

trisection circumferential overlap scheme with 20° of inclination. Furthermore, Dong et.al. [23] showed that trisection helical baffles with fewer inclination angles had the best H.T coefficient and the highest pressure drop. The same observations were noticed by Lin et.al. [24] They enhanced the heat transfer rate of a condenser by using the trisection helical baffles which enhanced the heat transfer coefficient by 35% than the segmental baffles. Wen et.al [25] performed a CFD simulation to develop the velocity and heat distribution in the shell side by using ladder fold continuous baffles. The results were experimentally validated and compared to the results of plain helical baffles. They showed that the ladder baffles scheme enhanced the thermal performance by 19.5%. Also, Yang et.al. [26] investigated numerically the effect of unilateral ladder baffles, baffles pitch, and the baffle angle of inclination on the thermal performance of a HE. Among these different parameters, the ladder helical baffles with a smaller pitch, and the less inclined angle was the best choice. Besides, Xiao et.al [27] used a shell and tube HE with ladder fold baffles in an experimental investigation to enhance gasification process efficiency by preheating the coal water slurry before entering the gasifier. As well as Yang et.al. [28] designed numerically three models of ladder folded helical baffles to enhance the performance of the desuperheating zone of high-pressure feedwater heaters. They found that the ladder-type had no significant effect on the H.T coefficient but a high influence on the pressure drop. He and Li [29] compared the thermos hydraulic characteristics of a single and double pass HE with three types of baffles. They used a CFD method to simulate the thermal performance with segmental, helical, and flower baffles schemes. Results showed that helical baffles had the best H.T coefficient but the flower scheme produces the lowest pressure drop. Cao et.al. [30] made a numerical and experimental investigation to conduct the H.T behavior and flow characteristics of a shell side of a HE with eight different helical baffles configurations. Among these geometries, the sextant helical baffles perform the best thermos-hydraulic performance. El-Maakoul et.al. [31] performed a CFD simulation to evaluate the H.T coefficient, pressure drop, and velocity distribution in the shell side of a heat exchanger by using three different baffles configurations. The results indicated that helical baffles introduce the best velocity distribution with minimum pressure drop but, the greatest H.T coefficient obtained with trefoil segmental baffles.

It is obvious when heat exchanger surface area and turbulence increased the heat transfer rate will also increase.

The main object of this study is to investigate the thermal performance of a double pipe HE. Based on previous studies the researchers either used fins to increase heat transfer area or perforations to increase the system turbulence. The newly designed segmental perforated baffles were proposed to achieve maximum heat transfer rate by increasing

turbulence and heat transfer area. Many researchers worked on segmental baffles with circular perforation, but in this work, a newly designed semi-circular perforation shape with different diameters was used. The newly designed perforations will provide an extended surface which increases the contact area between the cold air and the hot baffle surfaces. These perforations are designed in such a way that the semi-circular cutting area is used as a fin by bending the cutting semicircular area on the opposite side. This type of baffles did not use before by other researchers. While the perforations were used to redistribute the flowing air velocity in the annular side and eliminate the dead zone regions.

2 Experimental Work

A complete air–water heat exchanger was designed. Figure 1 shows a completely constructed experimental heat exchanger rig. It mainly consists of three sections which are the test section (double pipe HE), hot water section, and cold air section. The hot water was flowing in a copper pipe. While cold air was flowing in the annular Perspex shell side. Water was heated in a tank with an electrical heater then circulated through closed pipes looping by a pump. The air pump was used to circulate the cold air through the annular side. Eight thermocouples were used to measure the system temperature distribution.

Four thermocouples which measured the inlet and outlet temperatures of the hot and cold fluids, while the other four thermocouples were distributed in different positions along the test pipe. Two thermocouples are fitted on the surface of the inner pipe after and before the first and last baffles respectively to avoid the effect of the entrance and exit region. The other two thermocouples are fitted on the two middle baffles. Differential pressure digital air manometer connected to the shell side at 10% of the test section length after and before the entrance and air exit section to avoid the effect of entrance and exit region as shown in Fig. 2. To adjust the air and water flow rate to a certain value, two

Fig. 1 The Experimental test section

flow meters were used with two flow valves. To enhance the thermal performance of the heat exchanger and based on HE length, four segmental perforated baffles were added. According to the HE shell diameter, the number of perforations was designed. To investigate the effect of perforation diameters on HE thermal performance, three different perforation diameters were chosen (30, 25, and 20 mm). The details related to the rig specifications are shown in Table 1.

3 Experimental Procedure

A counter flow of the air–water system in a double pipe HE was designed. Hot water at 70 °C flow in a copper pipe while cold air at 22 °C flow in the shell side. The water flow rate was fixed at 14 LPM, while the cold air flowed in seven different flow rates ranging from 150 to 225 LPM. Twenty-eight experimental runs were done in this study (These experimental runs are repeated twice to ensure precision). The first seven runs were done for the un-baffled pipe. The temperatures and pressures were recorded after reaching steady-state conditions. The same procedures were repeated for the HE with semi-circular perforated baffles (SPB) with 30, 25, and 20 mm diameters. Figure 3 shows the perforated baffles geometries used in those experiments.

4 Data Reduction

The heat transfer across the annular and tube side can be estimated from Eq. (1) and (2), respectively, [32]:

$$Q_a = m_a C p_a (T_{ai} - T_{ao}) \tag{1}$$

$$Q_w = m_w C p_w (T_{wi} - T_{wo}) \tag{2}$$

The average heat transfer rate can be obtained from Eq. (3):

$$Q_{avg.} = \frac{Q_w + Q_a}{2} \tag{3}$$

Overall H.T coefficient for a double pipe HE can be calculated from. (4):

$$U = \frac{Q_{avg.}}{A \cdot \Delta T_m} \tag{4}$$

where A: is the outer surface area of the inner tube.

ΔT_m: Is the logarithmic mean temperature difference of the counter flow and calculated by using Eq. (5):
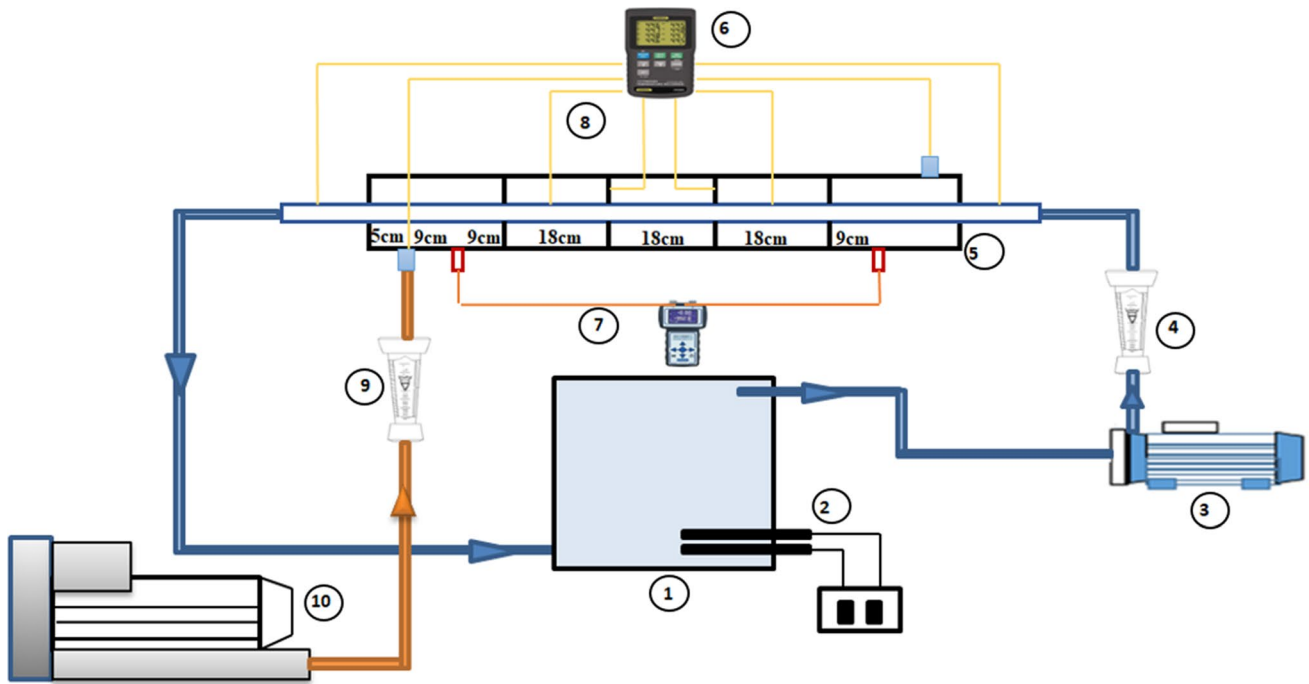
Fig. 2 Schematic experimental rig: (1) Water tank (2) Electrical heater (3) water pump (4) Flowmeter (5) Double pipe heat exchanger (6) Temperature recorder (7) Difference pressure recorder (8) Thermocouples (9) flow meter (10) Air pump

$$\Delta T_m = \frac{(T_{wi} - T_{ao}) - (T_{wo} - T_{ai})}{ln\frac{(T_{wi}-T_{ao})}{(T_{wo}-T_{ai})}} \tag{5}$$

The surface area was equal to the sum of the un-baffled tube calculated by Eq. (6) and that of the baffled tube which calculated Eq. (7):

$$A = \pi . d_o . L \tag{6}$$

$$A = \pi . d_o . L + 2 \times N_B \times \frac{\pi}{4} (D_B - d_o)^2 \tag{7}$$

To calculate the Nu number for the annular side it is essential to estimate the H.T coefficient of the annular side which can be calculated from Eq. (8):

$$h_a = \frac{Q_a}{A . \Delta T} \tag{8}$$

where ΔT is the temperature difference between the outer surface of the inner pipe and the average air temperature of the annular side.

$$Nu_a = \frac{h_a D_H}{k_a} \tag{9}$$

where k_a is the air thermal conductivity at bulk temperature, and

D_H is the annular hydraulic diameter.

For plain pipe

$$D_H = D_i - d_o \tag{10}$$

For baffled pipe:

$$D_H = \frac{4 . A_c}{P} \tag{11}$$

At each side of HE fluids Reynold's number could be calculated from the following equations:

$$Re_a = \frac{D_H \rho_a v_a}{\mu_a} \tag{12}$$

$$Re_w = \frac{d_i \rho_w v_w}{\mu_w} \tag{13}$$

To calculate the annular friction factor Eq. (14) can be used:

$$f = \frac{\Delta P . 2 D_H}{\rho . L . v^2} \tag{14}$$

Finally,

The theoretical Nusselt's number was estimated by using the correlation of Dittus-Boelter [33]

Table 1 The Experimental Rig Specifications

Test section	
Length	1 m
Internal pipe (inner diameter, d_i)	0.02 m
Internal pipe (outer diameter, d_o)	0.022 m
Annular pipe (inner diameter, D_i)	0.078 m
Annular pipe (outer diameter, D_o)	0.088 m
Baffles	
Materials	Copper
Thickness	1 mm
Perforation shape	Semi-circular
Perforation diameters	30, 25, and 20 mm
Baffles No	4
Perforation No	8
Water Pump	
Type	End suction pump
Max. flow	35 L/min
Power	370 W
Air pump	
Type	Vortex air blower
Power	750 W
Max. airflow	140 m3/hr
Max positive press	24 kPa
Max negative press	− 21 kPa
Temperature recorder	
Channel	4 channels
Sensor type	(K type)
Operation temperature range	− 20 to 150 ºC
Flowmeter	
Water flow meter, Range	0 to 18 l/min
Air flow meter, Range	0 to 250 l/min

$$Nu_a = 0.023 \, (Re_a)^{0.8}(Pr_a)^{0.4} \qquad (15)$$

$$Pr_a = \frac{Cp_a \, \mu_a}{k_a} \qquad (16)$$

And the theoretical friction factor was calculated by using Petukhov [34] correlation,

$$f = (-1.64 + 0.79 \ln (Re))^{-2} \qquad (17)$$

5 Results and Discussion

The effect of segmental baffles with semi-circular perforations on the thermal performance of a double pipe HE was investigated. This study aimed to examine a new baffle geometry which was done by using semi-circular holes.
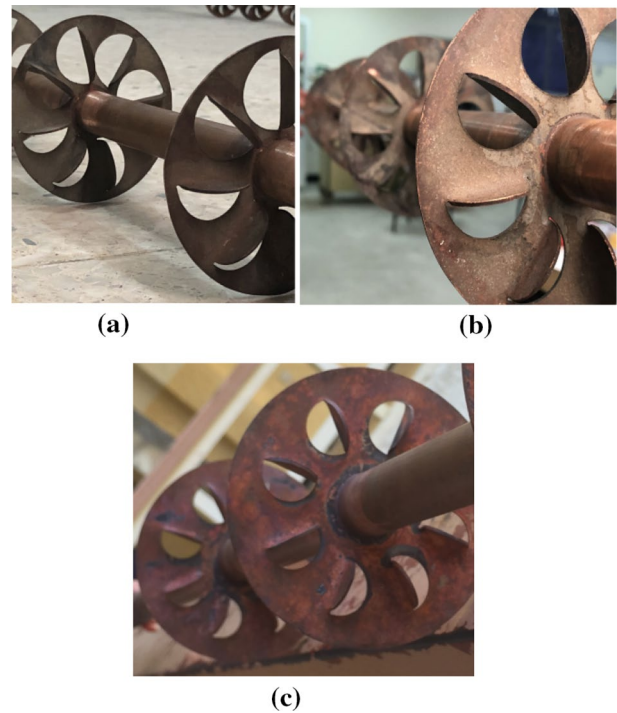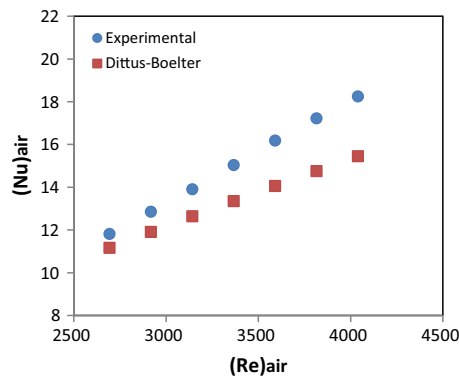

(a)   **(b)**

(c)

Fig. 3 The semi-circular perforated baffles with **a** 30 mm **b** 25 mm and **c** 20 mm diameter

The semi-circular cutting area was used as a fin by bending the cutting semicircular area on the opposite side and let it contact with the airflow on the annular side. Twenty-eight experiments were done to calculate the Nu number, overall heat transfer coefficient, and friction factor under various air velocity and Re numbers. The obtained results could be summarized in the following sections.

5.1 Model Validation

Experimental Nu numbers and friction factors were calculated for un-baffled HE (smooth traditional pipe). These experimental values were validated with the literature. Figure 4 represented a comparison study between this work Nussle's numbers and that by applying Dittus-Boelter correlation [33].

The HE thermal performance in terms of Nu number was shown in Fig. 4 for seven Reynolds numbers values. The development of a thermal field prevails and its values are fully developed. However, at the exit of the HE section, the heat transfer coefficients are slightly higher than the asymptotic calculated values. Adding baffles to the HE increase turbulence through the test section in addition to the mass flow rate measured turbulence which may cause a slight deviation from the empirical correlation calculated Nu number.

Fig. 4 Comparisons of experimental Nu values and empirical correlations values of the smooth pipe
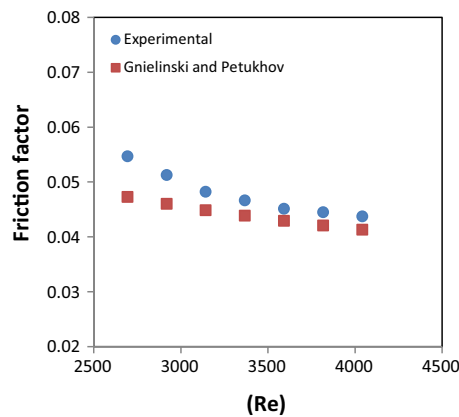

Fig. 5 Comparisons of experimental friction factor values and empirical correlations values of the smooth pipe

Also, Fig. 5 represented a comparison between this work friction factor and that by Gnielinski and Petukhov correlation [34]

The results show a good agreement within 12.33% and 8.25% average absolute deviation for Nu and friction factor, respectively, compared to the theoretical correlations. The possible sources of these deviations are back to slight uncertainty experimental runs. Also, it back to some of the theoretical not identical assumptions on the experimental HE search rig. Where Dittus-Boelter correlation did not take the new baffle shape geometry effect and change of fluid physical properties under consideration causing a slight calculated deviation compared to the experiment measured data points.

5.2 Uncertainty Analysis

The accuracy of experimental results related to the accuracy of the measuring instruments and the reading data precision. A calibrated measuring instrument like thermocouples

and flow meters was used. To ensure precision, all the experimental runs were repeated twice. The uncertainty is ($\pm 0.5°°C$) for the temperature probe and ($\pm 0.05\%$) for flow meters. The uncertainties of the calculated Re number, Nu number, heat load, and friction factor in an annular side were calculated according to the following equations. In the present study, the uncertainty of the parameters across the annular side is calculated as follows [35]:

$$\frac{Un_{Qan}}{Qan} = \mp \sqrt{\left(\frac{Un_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{Un_{cp}}{Cp}\right)^2 + \left(\frac{Un_{Ti}}{Ti - To}\right)^2 + \left(\frac{Un_{To}}{Ti - To}\right)^2} \tag{18}$$

$$\frac{Un_{Qan}}{Qan} = \mp \sqrt{\left(\frac{Un_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{Un_{Ti}}{Ti - To}\right)^2 + \left(\frac{Un_{To}}{Ti - To}\right)^2} \tag{19}$$

$$\frac{Un_{Re_{an}}}{Re_{an}} = \mp \sqrt{\left(\frac{Un_{\dot{m}}}{\dot{m}}\right)_{an}^2 + \left(\frac{Un_{d_{an}}}{d_{an}}\right)^2 + \left(\frac{Un_{\mu_{an}}}{\mu_{an}}\right)^2} \tag{20}$$

$$\frac{Un_{Nu_{an}}}{Nu_{an}} = \mp \sqrt{\left(\frac{Un_{h_{an}}}{h_{an}}\right)^2 + \left(\frac{Un_{d_{an}}}{d_{an}}\right)^2 + \left(\frac{Un_{k_{an}}}{k_{an}}\right)^2} \tag{21}$$

$$\frac{Un_{f_{an}}}{f_{an}} = \mp \sqrt{\begin{array}{c}\left(\frac{Un_{\Delta Pan}}{\Delta p_{an}}\right)^2 + \left(\frac{Un_{d_{an}}}{d_{an}}\right)^2 + \left(\frac{Un_{L_{an}}}{L_{an}}\right)^2 \\ + \left(\frac{Un_{\rho_{an}}}{\rho_{an}}\right)^2 + \left(\frac{Un_{u_{an}}}{u_{an}}\right)^2\end{array}} \tag{22}$$

The results by applying the above equations are shown in Table 2. The uncertainty in a property such as specific heat, dynamic viscosity, density, and thermal conductivity can be neglected. It should be noted that according to the manufacturer, uncertainty in the annular diameters and length is ± 0.5 mm.

5.3 The Effect of New Enhancement on Nu Number

The effect of Re number, baffles and semi-circular perforation diameter on the Nu number is shown in Fig. 6. The heat transfer rate is enhanced at high Re number values (high fluid velocity) because of increasing turbulence. The heat transfer coefficient increases when the fluid velocity increased due to intense mixing in the turbulent boundary layer, thinner laminar sub-layer. Turbulent flow also accelerates thermal mixing. This lead to an increase in the Nu number value as well. Also, the baffled HE had a higher Nu number than the un-baffled one for a given Re number. The baffles with perforations work as a distractor to the thermal boundary layer on the annular side. Besides, the semi-circular fins increased the contact surface area between the hot surface of the baffles and cold air in the annular sides which increase the convective H.T coefficient. Moreover, the best

Table 2 Uncertainties of annular heat load

Studied Parameter	%Average Uncertainty			
	Plain tube	Baffle perforation D = 30 mm	Baffle perforation D = 25 mm	Baffle perforation D = 20 mm
Re number	5.07	5.08	5.08	5.08
Nu number	6.257	6.254	6.255	6.254
	6.017	5.842	5.661	5.584
Friction Factor	3.907	3.737	3.551	3.474
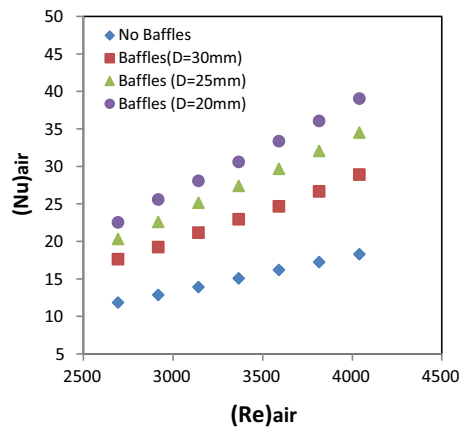
Fig. 6 Nu number vs. Re of different H.E geometries

Fig. 7 Overall heat transfer coefficients vs. Re number of different H.E geometries

Nu number was for the baffles with minimum perforations diameter. The results reveal that the increase in Nu number for a baffled pipe with 20 mm perforation diameter is 12% and 33% higher than 25 mm, 30 mm perforation diameter, respectively, and twice its value in the un-baffled pipe. Because a smaller diameter causes maximum destruction for the boundary layer. As the small diameter woks as throttling to the flow and increase the impingement.

5.4 Influence of Enhancement on Overall Heat Transfer Coefficient (U)

Figure 7 depicts the effect of Re number, baffles, and semi-circular perforation diameter on the overall heat transfer coefficient (U). The U values were improved with a higher Re number, which is due to high mixing and increase turbulence in the cold air layers region which prevents the forming of thick boundary sub-layers. Also, the increase in average U for perforated baffles with 20, 25, and 30 mm perforation diameter is 80.6%, 62%, and 29.7% than the un-baffled one, respectively. This is because the temperature difference between the input and output fluids increases in HE with an increase in the contact area and
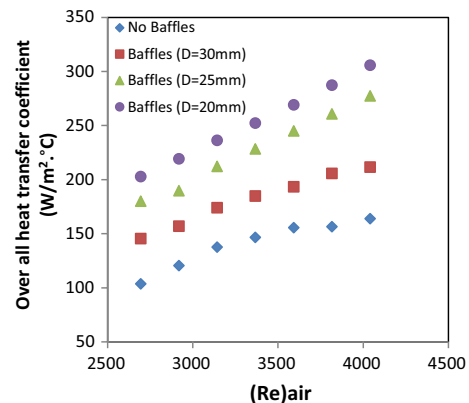
turbulence of perforated baffles, especially for high thermal conductive materials.

5.5 Influence of Enhancement on Friction Factor

The effect of Re number, baffles and semi-circular perforation diameter on the friction factor illustrates in Fig. 8. Where the friction factor decreases with the Re number increase. This behavior is contrary to the heat transfer behavior. Moreover, the friction factor is higher for a baffled pipe compared to an un-baffled pipe which is increases as the perforation diameter decrease. The semi-circular perforation caused turbulence inflow, vortex generation, and flow blockage inside the annular pipe.

5.6 Thermal Performance Factor (TPF)

To obtain a successful thermal performance in HE enhancement a heat transfer coefficient must be higher than the pressure drop across the fluid flow as shown in Eq. 23.
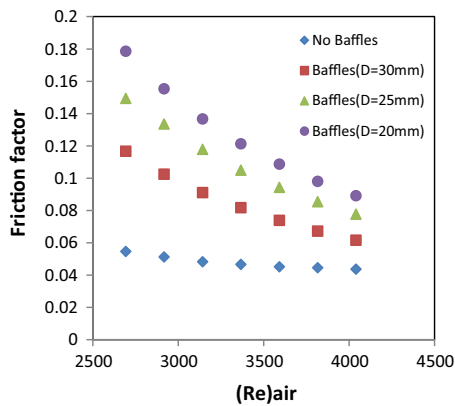
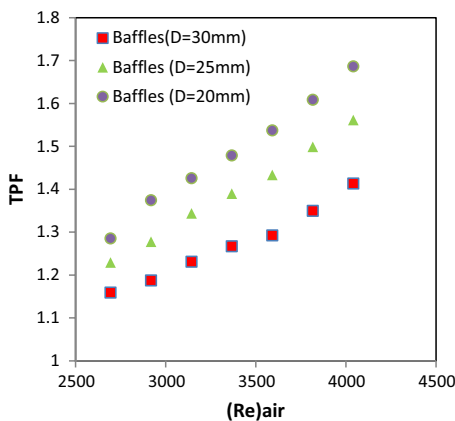**Fig. 8** Friction factor vs. Re number of different H.E geometries

Fig. 9 thermal performance factor vs. air Reynold's number
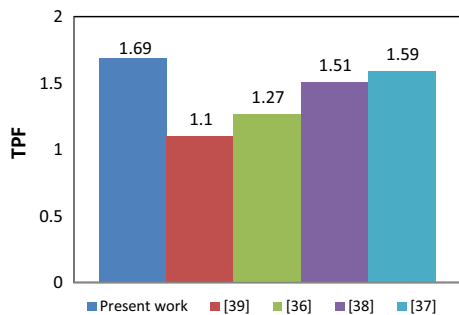

Fig. 10 Comparison of the present TPF with previous works

$$TPF = \frac{\dfrac{Nu_{air,baffled}}{Nu_{air,unbaffled}}}{\left(\dfrac{f_{air,baffle}}{f_{air,unbaffled}}\right)^{1/3}} \tag{23}$$

Figure 9 shows the variation of the TPF with the Re number. TPF is greater than unity for all the enhanced HE. But,

the HE with 20 mm semi-circular perforation baffles provides 1.69 which is the highest TPF value. Followed by the 25 mm and 30 mm perforations diameter with TPF values 1.56 and 1.41, respectively. All these TPF values for the Re number equal 4041. The TPF increased as the Re number increased in all HE geometries. Although the high Re number caused a high-pressure drop, it caused a high heat transfer rate due to high turbulence. Figure 10 compared the TPF maximum values obtained in the present work and those from previous work by using an air–water heat exchanger with enhancement technologies in the air side [36–39].

6 Conclusions

An experimental investigation was carried out at various conditions to calculate the overall H.T coefficient, Nu number, and friction factor in the annular side of air–water double pipe HE. Twenty-eight experiment runs were done for un-baffled and baffled pipe with three different perforation diameters (20 mm, 25 mm, and 30 mm). These perforations are designed in a way to provide semi-circular fins by bending their cutting opposite sides. This new design increases the contact surface area and fluid turbulence.

According to the obtained results, the conclusions can be summarized as follows:

a) The average Nu number for perforated baffles increases as the perforations diameter size is decreasing, Nu number for a baffled pipe HE with 20 mm perforation diameter is 12% and 33% higher than 25 mm, 30 mm perforated diameter, respectively.

b) The heat transfer rate of HE with perforated baffles showed a very good enhancement as compared with un-baffled HE. The average overall heat transfer coefficient increases by 29.7%, 62%, and 80.6% by using perforated baffles with 30, 25, and 20 mm perforation diameters.

c) The friction factor in the annular side increased when perforated baffles were used. The maximum friction factor values were obtained for minimum perforation diameter.

d) Maximum turbulence and vortices are obtained when baffles with minimum baffles perforations HE diameter are used.

e) The thermal performance factor was above unity for all selected baffled HE. The best value was 1.7 for baffles with 20 mm perforations diameter at 4000 Reynolds number

f) The baffles with small size perforations are preferred in use rather than the one with large diameter since they give maximum TPF compared to other studied cases concerning other operating conditions and system pressure drop.

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