Study of the Heat Transfer Enhancement in a Double Pipe Heat Exchanger Using Segmental Perforated Baffles

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Abstract

This work experimentally investigates the characteristics of convective heat transfer and pressure drop of water flow in the annulus-side of horizontal double pipe heat exchangers. Eight heat exchangers of counter-flow configurations are constructed with/without Single Segmental Perforated Baffles (SSPBs), which are fabricated with different void, cut and pitch ratios. The experiments are performed for annulus-side Reynolds number from 1380 to 5700, and for Prandtl number from 5.82 to 7.86. The results revealed that increasing void ratio and inclination angle, and decreasing SSPBs cut and pitch ratios increases the annulus average Nusselt number \((\overline{Nu}_{an})\) as well as friction factor \((f_{an})\). Furthermore, there is a slight increase in \(\overline{Nu}_{an}\) with decreasing the annulus-fluid inlet temperature, while its effect on \(f_{an}\) can be neglected. The thermal performance index (TPI) is calculated to compare the thermal performance of perforated baffled double pipe heat exchangers with un-baffled one. It is observed that increasing SSPBs inclination angle and decreasing SSPBS void, cut and pitch ratios enhances the TPI. Finally, correlations for \(\overline{Nu}_{an}\) in addition to \(f_{an}\) for concentric tube heat exchangers with SSPBs as a function of the investigated parameters are obtained.

Keywords: Perforated baffles, Heat exchanger, Passive technique, Heat transfer enhancement, Thermal performance, Friction, Experimental.

1. Introduction

Numerous applications involve heat transfer processes, whereas conversion, utilization, and recovery of the energy in every industrial, commercial, and domestic application involve a heat transfer process. In most of these applications, heat is transferred through heat exchangers as in the chemical, electrical, power, petroleum, air-conditioning, refrigeration, cryogenic, heat recovery and manufacturing industries. Therefore, enhancing the thermal performance of heat exchange affects directly on energy, material and cost savings. Consequently, improving the heat exchange above that in the usual or standard practice, can significantly improve the thermal efficiency in such applications as well as the economics of their design and operation [1]. Double pipe heat exchangers are the simplest devices in which heat is transported from the hot fluid to the cold fluid through a separating cylindrical wall. They are primarily adapted to high temperature and high-pressure applications due to their small diameters. They are cheap, but the amount of space they occupy is relatively high compared to the other types [2]. To achieve the desired heat transfer rate in the given design and size of the heat exchanger at an economic pumping power, numerous techniques have been proposed. These enhancement techniques can be categorized as active and passive techniques, which are discussed briefly in the following two subsections. Furthermore, any two or more of these techniques (passive and/or active) may be employed simultaneously to obtain enhancement in heat transfer that is greater than that produced by only one technique itself. This simultaneous utilization is termed compound enhancement [3–8].

Baffles are one of the passive enhancement techniques, which are utilized to improve the thermal performance of heat exchangers that are widely used in various industries: chemical, biological, petrochemical, biomedical and others. The wide using of the baffles in heat exchangers is because they direct the shell-side fluid to move back and forth across the internal tube to increase the turbulence level and provide good mixing of the fluid layers. In addition, they eliminate dead spots and increase heat transfer rate. Furthermore, in shell and tube heat exchangers, they minimize tube-to-tube temperature differences, which reduces the thermal stresses. Additionally, they support the internal tubes during operation, which prevent bending and vibration. Due to the extensive use of the baffles, knowledge about the heat transfer and shell-side fluid flow characteristics are very important.

Hwang, 1997 [9] conducted experiments to study turbulent air flow and heat transfer in a rectangular channel with segmented solid and perforated-type baffles (with holes of mean diameter of 0.72 mm and porosity of 42%) in a staggered manner. The top and bottom horizontal walls of the test section were heat transfer surfaces, and the remaining two walls were thermally insulated. The results showed that the porous-baffled channel have a significantly lower friction, and the porous type-baffled channel could thermally perform better than the solid-type baffled channel at the constant power constraint.

Dutta and Dutta, 1998 [10] investigated experimentally frictional loss and heat transfer characteristics of turbulent air flow in a rectangular channel, subjected to constant heat flux from the upper surface for different sizes, positions, and orientations of inclined solid and perforated baffles. Results indicated that the friction factor ratio increases with the increase in the inclination angle of baffle and also with a decrease in the average Nusselt number ratio in addition to with a decrease in the perforation density for a given configuration.

Ko and Anand, 2003 [11] studied experimentally the heat transfer rate in heated rectangular channel subjected to constant heat flux with wall mounted perforated baffles in staggered arrangement. Compared to no baffles, the results showed a heat transfer enhancement of 300%, while this enhancement per unit increase in pumping power was less than one for the range of parameters studied in that work.

Yang and Hwang, 2003 [12] numerically presented predictions on the turbulent fluid flow and heat transfer characteristics in rectangular channel with solid and
porous baffles mounted in a staggered way. The simulations indicated that the flow patterns around the porous- and solid-type baffles are completely different due to different transport phenomena and it significantly influences the local heat transfer coefficient distributions. Compared with the solid baffle, the perforated baffled channel has a lower friction factor due to less channel blockage. 

Dutta and Hossain, 2005 [13] experimentally studied the local heat transfer and friction loss characteristics in a rectangular channel with inclined solid and perforated baffles. A combination of two baffles of same dimensions was used. The results showed that the local Nusselt number and friction factor are strongly dependent on the position, orientation, and geometry of the second baffle plate. 

Karwa et al., 2005 [14] studied the heat transfer and friction in rectangular ducts with solid and perforated baffles (with void ratio from 18.4% to 46.8%) attached to one of the broad walls. The baffled wall of the duct was subjected to constant heat flux while the remaining three walls were insulated. The results revealed that the baffles with the highest open area ratio give the best performance compared with the smooth duct at equal pumping power. 

In Santos et al., 2005 [15] work, simulations were presented for turbulent flow in a channel containing solid and perforated baffles. The simulations showed that no advantages obtained when low porosity baffles were used in turbulent flow regime. El-Shamy, 2006 [16] carried out measurements to examine the turbulent flow and heat transfer characteristics in an annulus with perforated disc-baffles aligned along the inner heated tube surface, using air as a working fluid. The effects of the baffles pitch ratio, void ratio, and flow Reynolds number on the thermal performance were examined. The results revealed that Nusselt number increased with increasing the baffle void ratio and decreased with increasing the baffles pitch ratio. In addition, for a given Reynolds number and baffles pitch ratio, the friction factor decreased as the baffle void ratio increased. Moreover, the perforated-baffled annulus has a higher thermal performance than the solid baffled annulus for all studied baffles configurations. 

Ary et al., 2012, [17] experimentally and numerically studied the influence of a number of tilted (5°) perforated baffles on the turbulent flow patterns and heat transfer in the rectangular channel with different types of baffles. The results showed that the flow patterns around the holes are entirely different with different numbers of holes and it significantly affects the local heat transfer, and two baffles provide greater heat transfer performances than a single baffle. 

Chamoli and Thakur, 2014 [18] mathematically studied the performance of solar air heaters with V-down perforated baffles as roughness on the airflow side of the absorber plate. They indicated that the thermal and effective efficiencies differ only slightly at lower flow rates. In another work [19], they numerically investigated the effect of transverse perforated baffles attached on the heated wall of rectangular duct on heat transfer and friction factor. They observed that installing perforated baffles enhances the heat transfer, while friction loss increases over a smooth surface. 

Sheikholeslami et al., 2015 [20] and 2016 [21] conducted an experimental study on friction loss and heat transfer enhancement in an air to water double pipe heat exchanger. Typical circular-ring and perforated circular-ring turbulators. In their investigation, air was flowed in the annular pipe. Experimental analysis was carried out for open area ratio; 0, 0.0208–0.0833, Reynolds number; 6000–12000 and pitch ratio; 1.83–5.83. The results showed that installing perforated circular-rings decreased the heat transfer enhancement compared with the circular rings because of reduction of intersection angle between the velocity and the temperature field. In addition, the thermal performance increased with increase of number of rings but it decreased with increase of Reynolds number and pitch ratio. 

Sahel1 et al., 2016 [22] numerically examined the performance of SSPB (having a row of four holes placed at three different positions) aiming to enhance the heat transfer phenomenon in a rectangular channel. They observed that there was enhancement in the heat transfer rate from 2% to 65% compared with the simple baffle. 

Kumar and Kim, 2016 [23] numerically presented heat transfer and fluid flow characteristics in a solar air heater channel with multi V-type perforated baffles. The baffle height ratio, pitch ratio, baffle-hole position ratio, inclination angle, and baffle void ratio were 0.6, 8.0, 0.42, 60°, and 12%, respectively. Multi V-type perforated baffles were shown to have better thermal performance as compared to other baffle shapes in a rectangular passage. 

The aforementioned literature survey indicates that the majority of studies performed on perforated baffles that were mounted inside rectangular channels. In addition, it is found that the thermal performance due to using the perforated baffles is more than that of using solid baffles. Despite the importance of double pipe heat exchangers, but it is found that there have not been any investigations on enhancing the heat transfer in their annulus with SSPBs. Therefore, the objective of the present work is to investigate experimentally the convective heat transfer and pressure drop in an annulus with perforated SSPBs aligned along the inner heated tube surface, using water as a working fluid. The experimental measurements were performed to investigate the effect of the baffle open area ratio, holes spacing ratio, baffles cut and pitch ratios, in addition to their inclination angles at wide range of annulus-side operating conditions.

2. Experimental Apparatus

The apparatus used in this study comprises hot and cold loops. The hot circuit consists of heating unit, pump, valves, flow meter, straight tube and the connecting pipes. The cold circuit consists of cooling unit, pump, flow meter, valves, annular pipe with/without SSPBs and the connecting pipes. Fig. 1 is a schematic diagram of the experimental setup. 

The heating and cooling units were made of 50 liters stainless steel (2 mm wall thickness) cylindrical tank for each. Each tank was installed inside 2 mm thick
galvanized steel tanks with 2 cm gap, which was filled with spray foam insulation to minimize the heat gain/loss from/to the atmosphere. For the heating unit, an electric heater (has a maximum power rating of 6 kW) was fixed horizontally on the bottom of the heating tank and performed the function of heating the water to the required temperature. On the other side, the heat was removed from the water in the cooling tank by two cooling units of 10.5 kW cooling capacity. Sometimes, the two units operate in series, and in other times, in parallel manner to prevent thermal overloads. The operations of the electric heater and the cooling units were based on pre-adjusted digital thermostats, which were used to keep constant temperatures of the liquids directed to the heat exchanger, whether for the tube-side or the annulus-side. Additionally, there are four ports in each tank; two of them are in the top covers of the tanks, represent the inlet ports from the heat exchanger and from the by-pass line. The other two ports are in the bottom, which represent the exit ports to the drain and to the pump.

Two rotameters (1.7-18 l/min) were used to measure the volume flow rates of the two main loops fluids. The two flow meters have been calibrated by calculating the time required to fill a vessel with 20-litre capacity. Four K-type thermocouples (wires of 0.1 mm diameter) were directly inserted into the flow streams, at approximately 5 cm from the heat exchanger ports, to measure the inlet and exit temperatures of the annulus and internal tube fluids. The thermocouples were connected via switching box to a digital thermometer indicator with resolution of 0.1 °C to display the thermocouples outputs. All thermocouples were calibrated in the laboratory against a mercury-in-glass thermometer, which could be accurately read to ± 0.1 °C. A digital differential pressure transducer was employed for measuring the pressure drop of water between the annulus inlet and outlet. Two identical 1.5 hp power rating centrifugal pumps; were used; pump-1 was used to circulate the heating water on the internal tube-side, while pump-2 was used to circulate the cooling water on the annulus-side. Flexible nylon and PVC (Polyvinyl Chloride) tubing were used for all connections.

Eight concentric tube heat exchangers of counter-flow configurations were constructed; one was without any baffles, while seven heat exchangers were fabricated with different SSPBs void ratio (φ), cut ratio (δ) and pitch ratio (λ). The characteristic dimensions of the different configurations are revealed in Table (1). The SSPBs were formed from 0.6 mm thick copper sheet and laser was used during cutting and drilling process. All baffles have circular shape of the same diameter of the heat exchanger annular pipe; 50.8 mm except the inclined SSPBs which have parabolic shape to keep same cut ratio.

The annular pipes of the heat exchangers was made up of PVC tubes of 50.8 mm internal diameter and 4 mm wall thickness. Their ends were closed using PVC caps and adhesives to prevent any leakage. The internal tube of all exchangers is a copper tube of 26 mm internal diameter and 28 mm external diameter. The length of all heat exchangers annular pipes and their internal tube is 1200 mm. The outer surface of all annular pipes was thermally isolated with thick insulation consisting of layers of ceramic fiber, asbestos rope and glass wool.

### 3. Experimental Procedures

The experimental procedures were initiated after assembling the following equipments: the concentric tube heat exchanger, heating and cooling units, pumps, piping, flow meters, thermocouples and the differential pressure transducer. The thermocouples were attached at the inlet and outlet of the annulus and internal-tube sides. The first step to collect the data from the system was to fill the heating and cooling tanks with water from the local water supply. Then, the heater, the cooling unit and the pumps were operated. The inlet temperatures of the fluids in both sides were adjusted by regulating the temperatures of the heating and cooling tanks through their thermostats. The flow rates were adjusted through the flow meters and the installed valves, which were regulated to obtain the required flow rates in the primary lines and the remainder is bypassed to the reservoirs. The range of the operating conditions is given in Table (2).

### Table (1): Characteristic dimensions of the used perforated baffles.

<table>
<thead>
<tr>
<th>No.</th>
<th>H (mm)</th>
<th>D (mm)</th>
<th>L (mm)</th>
<th>H/D</th>
<th>D/L</th>
<th>φ (%)</th>
<th>c (°)</th>
<th>λ (%)</th>
<th>ΔH (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4.0</td>
<td>97</td>
<td>200</td>
<td>0.41</td>
<td>2.04</td>
<td>4.0</td>
<td>3.40</td>
<td>20.4</td>
<td>8.77</td>
</tr>
<tr>
<td>2</td>
<td>4.0</td>
<td>97</td>
<td>200</td>
<td>0.41</td>
<td>2.04</td>
<td>3.45</td>
<td>3.7</td>
<td>20.4</td>
<td>8.77</td>
</tr>
<tr>
<td>3</td>
<td>2.5</td>
<td>97</td>
<td>200</td>
<td>0.26</td>
<td>2.04</td>
<td>3.45</td>
<td>3.76</td>
<td>20.4</td>
<td>8.77</td>
</tr>
<tr>
<td>4</td>
<td>2.5</td>
<td>97</td>
<td>200</td>
<td>0.26</td>
<td>2.04</td>
<td>3.45</td>
<td>3.76</td>
<td>20.4</td>
<td>8.77</td>
</tr>
<tr>
<td>5</td>
<td>2.5</td>
<td>97</td>
<td>200</td>
<td>0.26</td>
<td>2.04</td>
<td>3.45</td>
<td>3.76</td>
<td>20.4</td>
<td>8.77</td>
</tr>
<tr>
<td>6</td>
<td>2.5</td>
<td>97</td>
<td>200</td>
<td>0.26</td>
<td>2.04</td>
<td>3.45</td>
<td>3.76</td>
<td>20.4</td>
<td>8.77</td>
</tr>
<tr>
<td>7</td>
<td>3.45</td>
<td>97</td>
<td>200</td>
<td>0.34</td>
<td>2.04</td>
<td>3.45</td>
<td>3.76</td>
<td>20.4</td>
<td>8.77</td>
</tr>
<tr>
<td>8</td>
<td>10.05</td>
<td>97</td>
<td>200</td>
<td>0.34</td>
<td>2.04</td>
<td>3.45</td>
<td>3.76</td>
<td>20.4</td>
<td>8.77</td>
</tr>
</tbody>
</table>

### Table (2): Range of fluids operating conditions.

<table>
<thead>
<tr>
<th>Parameters/operating conditions</th>
<th>Range or value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annulus-side water flow rate, l/min</td>
<td>6.02-18.36 (1380 ≤ Re ≤ 5700)</td>
</tr>
<tr>
<td>Annulus-side water inlet temperature, °C</td>
<td>15, 20, 25 (5.82 ≤ Pr ≤ 7.86)</td>
</tr>
<tr>
<td>Tube-side water flow rate, l/min</td>
<td>8.66</td>
</tr>
<tr>
<td>Tube-side water inlet temperature, °C</td>
<td>50</td>
</tr>
</tbody>
</table>
Totally, a series of 168 experiments was carried out on the twelve heat exchangers; 147 runs for the heat exchangers with SSPBs and 21 runs for the un-baffled one. During the test operation, the steady-state condition is conducted when a maximum variation of 0.5°C for each thermocouple reading within 20 minutes is recorded. Moreover, it is considered to be achieved when stable fluid inlet and outlet temperatures are obtained; variation of inlet and outlet temperatures of the two streams were within 0.2°C during a minute period before each measurement was taken.

4. Data Reduction
Excel sheets were prepared to process the experimental data for the heat transfer coefficients and pressure drop. It should be noted that for all calculations, the thermophysical properties of the water in the annulus and internal tube were calculated at the bulk temperatures, $T_{an,m}$ and $T_{tm}$, respectively, and were evaluated from Ralph [24].

$$T_{an,m} = \frac{(T_{an,i} + T_{an,o})}{2} \quad (1)$$

$$T_{tm} = \frac{(T_{tl,i} + T_{tl,o})}{2} \quad (2)$$

Where

- $T_{an,i}$ The inlet temperature of the annulus-side fluid, °C
- $T_{an,o}$ The outlet temperature of the annulus-side fluid, °C
- $T_{tl,i}$ The inlet temperature of the tube-side fluid, °C
- $T_{tl,o}$ The outlet temperature of the tube-side fluid, °C

4.1 Heat Transfer Calculations
The primary measurements in heat transfer calculations consist of six variables, namely the flow rates and the inlet and outlet temperatures of both streams of the heat exchanger. The heat transfer rates on the inner tube and annulus sides ($Q_t$ and $Q_an$) were calculated by;

$$Q_t = \dot{m}_t C_P t (T_{tl,i} - T_{tl,o}) \quad (3)$$

$$Q_an = \dot{m}_an C_P an (T_{an,o} - T_{an,i}) \quad (4)$$

Where

- $Q_t$ The rate of heat energy transferred from the tube-side fluid, W
- $\dot{m}_t$ The mass flow rate of the tube-side fluid, kg/s
- $C_P t$ The specific heat of the tube-side fluid, J/kg.°C
- $Q_an$ The rate of heat energy transferred to the annulus-side fluid, W
- $\dot{m}_an$ The mass flow rate of the annulus-side fluid, kg/s
- $C_P an$ The specific heat of the annulus-side fluid, J/kg.°C

Assuming that the measurements are sufficiently accurate without heat gain or loss, there is an energy balance between the two streams ($Q_t = Q_an$). While in the real experiments, there would always be some discrepancy between the two rates. Therefore, the arithmetical mean of the two, $Q_{ave}$, can be used as the heat load of the exchanger. For all experimental tests, the heating and cooling loads calculated from the hot and cold sides did not differ by more than ±5.1%.

$$Q_{ave} = \frac{|Q_t| + |Q_an|}{2} \quad (5)$$

The overall thermal conductance was calculated from this heat load, the temperature data and flow rates using Eq. (6);

$$U_i A_{tl} = \frac{Q_{ave}}{\Delta T_{LM}} \quad (6)$$

$$\Delta T_{LM} = \frac{(T_{tl,i} - T_{an,o}) - (T_{tl,o} - T_{an,i})}{\ln \frac{T_{tl,i} - T_{an,o}}{T_{tl,o} - T_{an,i}}} \quad (7)$$

Where

- $U_i$ The overall heat transfer coefficient based on the inner surface area of the internal tube, W/m².°C
- $A_{tl}$ Area of the inner surface of the internal tube, $A_{tl} = \pi d_{tl}L_m$, m²
- $\Delta T_{LM}$ The logarithmic mean temperature difference defined for counter flow configuration.
- $\Delta T_{tl}, \Delta T_{an}$ The temperature difference at each end of the heat exchanger.

Neglecting the thermal resistances of the tube wall and fouling, the overall thermal conductance can be expressed in terms of the thermal resistances.

$$\frac{1}{U_i A_{tl}} = \frac{1}{h_{an}A_{tl,o}} + \frac{1}{h_{an}A_{tl,i}} \quad (8)$$

The water flow in the inner tube is turbulent and fully developed where the ratio between the tube-length to the inner diameter is 46.2, which is more than 10. The average Nusselt number for the tube-side fluid, $\bar{Nu}_t$, can be calculated using Dittus-Boelter [25] correlation for fully developed turbulent flow, Eq. (9).

$$\bar{Nu}_t = 0.023 \ Re_t^{0.8} \ Pr_t^{0.3} \quad (9)$$

Where

- $Re_t$ Reynolds number of the tube-side fluid.
- $Pr_t$ Prandtl number of the tube-side fluid.

Then the average heat transfer coefficient for the tube-side fluid, $\overline{h}_t$, can be obtained as follows;

$$\overline{h}_t = \frac{\bar{Nu}_t k_t}{d_{tl}} \quad (10)$$

Where $k_t$ is the thermal conductivity of the tube-side fluid, W/m.°C. The average heat transfer coefficient for the annulus-side fluid, $\overline{h}_an$, and then the average Nusselt number for the annulus-side fluid, $\bar{Nu}_an$, can be obtained as follows;

$$\overline{h}_an = \frac{\bar{Nu}_an d_{an,h}}{k_{an}} \quad (11)$$

Where

- $k_{an}$ The thermal conductivity of the annulus-side fluid, W/m.°C
- $d_{an,h}$ The hydraulic diameter of the annulus, $d_{an,h} = d_{an,i} - d_{tl,o}$

Tube and annulus Reynolds numbers can be written as follows;

$$Re_t = \frac{4 \dot{m}_t}{\pi d_{tl} \mu_t} \quad (12)$$
\[
Re_{an} = \frac{4h_{an}}{\pi d_{an,h} \mu_{an}} \tag{13}
\]

Where
\[
\mu_t \quad \text{The dynamic viscosity of the tube-side fluid, kg/m.s}
\]
\[
Re_{an} \quad \text{Reynolds number of the annulus-side flow.}
\]
\[
\mu_{an} \quad \text{The dynamic viscosity of the annulus-side fluid, kg/m.s}
\]

4.2 Friction Factor Calculation
In the present study, the measurement of the friction factor was conducted at the same time as the heat transfer measurements to show the effect of using the SSPBs in the annulus of the heat exchanger at different operating conditions. The Fanning friction factor for the fluid in circulation inside the annulus side was calculated with the following equation;
\[
f_{an} = \frac{\Delta P_{an}}{2L_{an} \rho_{an} \bar{V}_{an}^2} \tag{14}
\]
\[
u_{an} = \frac{\pi}{4}(d_{an,l}^2 - d_{co}^2) \tag{15}
\]

Where
\[
u_{an} \quad \text{The mean axial velocity of the annulus-fluid, m/s}
\]
\[
\bar{V}_{an} \quad \text{The volume flow rate of the annulus-fluid, m³/s}
\]

5. Uncertainty Analyses
In general, the accuracy of the experimental results depends on the accuracy of the individual measuring instruments and techniques. It should be noted that according to the manufacturer, uncertainty (\(\sigma\)) in the internal tube outer and inner diameters is ±0.01. The uncertainty in the measured annulus diameters and lengths, in addition to the pitch between the SSPBs were assumed to be ±0.5 mm; this was guessed quantity from meter scale. The uncertainty of all dimensions of baffles; diameter, window height, holes diameter and spacing, were ±0.01 mm.

Table (3): Average uncertainties in the main parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Average Uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annulus-side Reynolds number</td>
<td>±2.79</td>
</tr>
<tr>
<td>Annulus-side average Nusselt number</td>
<td>±0.74</td>
</tr>
<tr>
<td>Annulus-side average heat transfer coefficient</td>
<td>±0.35</td>
</tr>
<tr>
<td>Tube-side Reynolds number</td>
<td>±1.67</td>
</tr>
<tr>
<td>Tube-side average Nusselt number</td>
<td>±1.68</td>
</tr>
<tr>
<td>Tube-side average heat transfer coefficient</td>
<td>±1.68</td>
</tr>
<tr>
<td>Overall heat transfer coefficient</td>
<td>±2.86</td>
</tr>
<tr>
<td>Annulus-side Fanning friction factor</td>
<td>±4.48</td>
</tr>
</tbody>
</table>

In addition, the uncertainty applied to the thermal properties of water was assumed to be ±0.1%. The uncertainty of the parameters was calculated based upon the root sum square combination of the effects of each of the individual inputs as introduced by Kline and McClintock [26]. For all experimental runs, the average uncertainty in main parameters are summarized in Table (3). For the estimated uncertainties in the other variables and parameters used in the present study, additional information is given in Appendix A.

6. Apparatus Validation and Data Verification
Using the aforementioned experimental procedures and analysis methods, the validation of the methodologies in determining the heat transfer coefficients and friction factors was done by taking measurements for the flow in the annulus and comparing it with established heat transfer and friction factor correlations. For heat transfer and friction factor calculations, the experimental procedures were validated by comparing the results of \(Nu_{an}\) for the water flowing through the annulus with \(Nu_{an}\) for turbulent flow developed by Gnielinski [27], Eq. (16).

\[
Nu_{an} = \frac{f_{an}(Re_{an} - 1000)Pr_{an}}{1 + 12.7f_{an}^{(1/2)}(Pr_{an}^{2/3} - 1)} \tag{16}
\]

The Fanning friction factor in Eq. (16) was calculated according to Filonenko [28], through Eq. (17), which was also used for comparing the results of \(f_{an}\) for the water flowing through the annulus.

\[
f_{an} = 0.25(1.82 \log Re_{an} - 1.64)^{-2} \tag{17}
\]

Table (4): Validation operating conditions.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Operating conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annulus-side water flow rate, l/min</td>
<td>10.09-18.26 (2520 ≤ Re_{an} ≤ 5080)</td>
</tr>
<tr>
<td>Annulus-side inlet temperature, °C</td>
<td>9.86</td>
</tr>
<tr>
<td>Tube-side water flow rate, l/min</td>
<td>76</td>
</tr>
<tr>
<td>Tube-side inlet temperature, °C</td>
<td>50</td>
</tr>
</tbody>
</table>

Fig. (2): Validation of the experimental average Nusselt number for the annulus side.

Fig. (3): Validation of the experimental Fanning friction factor for the annulus side.

The results of these comparisons are shown in Figs. 2 and 3. The range of the operating conditions during validation is given in Table (4). It is indicated that the experimental results for both heat transfer and friction factor calculations are in good agreement with previous studies with average deviation of 10.1% and 3.1% for annulus average Nusselt number and Fanning friction factor, respectively. This good agreement in comparisons reveals the accuracy of the experimental setup and measurement technique.

7. Results and Discussions
Totally, a series of 168 experiments was carried out on the twelve heat exchangers; 21 runs for case of no baffles and 147 runs for the heat exchangers with SSPBs, which were constructed with different cut, void,
and pitch ratios. These configurations were tested at different flow rates (6.01 to 18.26 l/min) and inlet temperatures (15, 20 and 25°C) for the annulus side of heat exchanger. All specifications and operating parameters are revealed in Table (2). The corresponding dimensionless are $1380 \leq Re_{an} \leq 5700$, $5.82 \leq Pr_{an} \leq 7.86$, $6.72\% \leq \phi \leq 20.4\%$, $6.6\% \leq \delta \leq 19.8\%$, and $5.68\% \leq \lambda \leq 13.16\%$. Heat transfer results in terms of annulus-side average Nusselt number and overall heat transfer coefficient in addition to the annulus-side Fanning friction are presented in the following subsections for the different governing parameters at $T_{an,1} = 20^\circ$C as a sample of the results.

### 7.1 Influence of SSPBs Void Ratio

In this analysis, four heat exchangers (SSPBs no. 1, 2, 3 and 4) are considered. Five SSPBs are inserted in the baffled heat exchangers with same pitch ratio ($\lambda = 8.77$) and same cut ratio ($\delta = 13.2\%$) while the void ratio ranges from 6.72% to 20.4%.

![Fig. 4: Influence of SSPBs void ratio at different annulus Reynolds numbers.](image)

Fig. 4 illustrates the obtained results. It is clear that increasing $\phi$ increases $\overline{Nu}_{an}$, $U_1$ and $f_{an}$. Compared with no baffles for all tested annulus-side operating conditions, the average increase in the annulus-average Nusselt number is of 23.3% to 73.8%; and the average increase in the overall heat transfer coefficient is of 9.9% to 25.5% and the average increase in the annulus-side friction factor is of 18.1% to 51% when $\phi$ increases from 6.72% to 20.4%. This can be attributed to increasing $\phi$ requires also increasing number of the holes in the baffle with decreasing their diameters, for same baffle diameter and other geometrical parameters, which increases the throttling for the annulus flow and produces better impingement where they create a flow blockage and increase the flow velocity. This breaks the fluid boundary layer and creates a significant enhancement of heat transfer in addition to significant increase in the pressure drop.

### 7.2 Influence of SSPBs Cut Ratio

In this analysis, also four heat exchangers (SSPBs no. 1, 5, 3 and 6) are considered. Five SSPBs are inserted in the baffled heat exchangers with same void ratio ($\phi = 12.14\%$), same pitch ratio ($\lambda = 8.77$) while the cut ratio ranges from 6.6% to 19.8%.

![Fig. 5: Influence of SSPBs cut ratio at different annulus Reynolds numbers.](image)

From Fig. 5, it is obvious that decreasing $\delta$ increases $\overline{Nu}_{an}$, $U_1$ and $f_{an}$. Compared with no baffles for all studied annulus-side operating conditions, the average increase in the annulus-average Nusselt number is of 16.8% to 94.6%, and the average increase in the overall heat transfer coefficient is of 7.8% to 30.7% and the average increase in the annulus-side friction factor is of 12.3% to 59.2% when $\delta$ decreases from 19.8 to 6.6%. This is due to decreasing $\delta$ increases the throttling for the annulus flow and produces better impingement that creates a significant enhancement of heat transfer in addition to significant increase in the pressure drop.

### 7.3 Influence of SSPBs Pitch Ratio

In this analysis, also four heat exchangers (SSPBs no. 1, 7, 3 and 8) are considered. The SSPBs are inserted in the baffled heat exchangers with same void ratio ($\phi =
12.14%), same cut ratio ($\delta = 13.2\%$) while the pitch ratio varies; 6.58%, 8.77% and 13.16%, and the corresponding number of the inserted SSPBs is 7, 5, 3 baffles.

From Fig. 6, it is obvious that decreasing $\lambda$ increases $\overline{Nu}_{an}$, $U_i$ and $f_{an}$. Compared with no baffles for all studied annulus-side operating conditions, the average increase in the annulus-average Nusselt number is of 10.2% to 81.5%, and the average increase in the overall heat transfer coefficient is of 4.4% to 26.8% and the average increase in the annulus-side friction factor is of 10.8% to 63.1% when $\lambda$ decreases from 13.16 to 6.58%. This is due to decreasing $\lambda$ increases the chance of turbulence for the annulus flow and produces better impingement as a result of increasing number of baffles that creates a significant enhancement of heat transfer in addition to significant increase in the pressure drop.

7.4 Influence of Annulus-Side Operating Conditions

In this analysis, effect of annulus-side operating conditions on the thermal performance of the studied heat exchangers is presented. It is obvious from Figs. 4 to 7 that increasing $Re_{an}$ increases both $\overline{Nu}_{an}$ and $U_i$. This can be backed to increasing the fluctuations level and fluid layers mixing around the internal tube by increasing Reynolds number. On contrary, increasing $Re_{an}$ decreases $f_{an}$, which can backed to that momentum forces overcomes viscous forces as $Re_{an}$ increases.

For the effect of annulus-side inlet temperature, the results are presented here in Fig. 7 for heat exchanger with SSPB no. (3), as a model of the results. It is evident that increasing annulus fluid inlet temperature leads to a slight decrease in both $\overline{Nu}_{an}$ and $U_i$. This can be attributed to decreasing Prandtl number with increasing fluid temperature. While the effect of $T_{an,1}$ on the annulus side friction factor is nearly negligible. This is due to lower effect of viscosity variation compared with the inertia force.

8. Thermal Performance Index

To be a successful heat transfer enhancement tool, the rise in convective heat transfer given due existing perforated baffles in heat exchangers should be higher than the rise in the fluid pressure drop. The thermal performance index (TPI) is determined using $\overline{Nu}_{an}$ ratio and the pressure drop ratios [29–31] that are calculated using the values obtained for existing perforated baffles and no baffles, as follows;

$$TPI = \frac{\overline{Nu}_{an, baffles}}{\overline{Nu}_{an, no baffles}}$$  \hspace{1cm} (18)

Where

- $\overline{Nu}_{an, baffles}$: Annulus-side average heat transfer coefficient when perforated baffles inside, $W/m^2\cdot^\circ C$
- $\overline{Nu}_{an, no baffles}$: Annulus-side average heat transfer coefficient when no baffles inside, $W/m^2\cdot^\circ C$
- $\Delta P_{an, baffles}$: Annulus-fluid pressure drop when perforated baffles inside, Pa
Annulus-fluid pressure drop when no baffles inside, Pa

Over the studied range of annulus-operating conditions, the average TPI was calculated and the results are illustrated at different SSPBs geometrical parameters in Fig. 8 for different perforated baffles characteristics.

![Fig. 8: Variation of thermal performance index with SSPBs geometrical parameters.](image)

It is obvious from Fig. 8 that the TPI is more than unity for all ranges of tested perforated baffles spacing, void and cut ratios, and increases with decreasing of φ and δ. While for the studied range of baffles pitch ratio, the TPI is more than unity for λ < 13.16%, and increases with decreasing λ. For λ = 13.16%, the average TPI is around unity and there is no benefits with using SSPBs with this condition.

9. Correlations for Annulus Average Nusselt Numbers and Friction Factor

Using the present experimental data, correlations were developed to predict the annulus average Nusselt number and its Fanning friction factor with using SSPBs inside. The annulus average Nusselt number is correlated as a function of annulus-side Reynolds and Prandtl numbers, baffles void ratio, cut ratio and pitch ratio as follows:

\[
\overline{Nu}_an = 0.0019 \text{Re}_an^{1.25} \text{Pr}_an^{0.31} \phi^{-0.45} \delta^{-0.73} \lambda
\]  \hspace{1cm} (19)

Additionally, a correlation for annulus-side Fanning friction factor was obtained as follows:

\[
f_{an} = 4.22 \text{Re}^{-0.59}_an \phi^{-0.22} \delta^{-0.3} \lambda^{-0.5}
\]  \hspace{1cm} (20)

![Fig. 9: Comparison of experimental values for annulus-average Nusselt number with that correlated by Eq. (19).](image)

![Fig. 10: Comparison of experimental values for HCT-Fanning friction factor with that correlated by Eq. (20).](image)

Eqs. 19 and 20 are applicable for 1380 ≤ Re_{an} ≤ 5700 , 5.82 ≤ Pr_{an} ≤ 7.86 , 6.72% ≤ φ ≤ 20.4% , 6.6% ≤ δ ≤ 19.8% , and 6.58% ≤ λ ≤ 13.16% . Comparisons of the experimental annulus average Nusselt number and annulus-friction factor with those calculated by the proposed correlations are shown in Figs. 9 and 10. From these figures, it is evident that the proposed correlations are in good agreement with the present experimental data. It is clearly seen that the data falls of the proposed equations within maximum deviation of ±13.9% and ±12% for \( \overline{Nu}_an \) and \( f_{an} \) respectively.

10. Summary

The present work was carried out to investigate experimentally the heat transfer characteristics and the pressure drop in the annulus of concentric tube heat exchangers with SSPBs. The SSPB-geometrical parameters and operating conditions of the annulus side are the main parameters throughout this study. Therefore, eleven baffled heat exchangers of counter-flow configuration were constructed with different SSPBs geometries and tested at different water flow rates and inlet temperatures. In experiments, the investigated operating parameters were 1380 ≤ Re_{an} ≤ 5700 , 5.82 ≤ Pr_{an} ≤ 7.86 , 6.72% ≤ φ ≤ 20.4% , 6.6% ≤ δ ≤ 19.8% , and 6.58% ≤ λ ≤ 13.16% . From the previous sections and according to the results obtained using the experimental investigation, the following conclusions can be expressed:

- Installing segmental perforated baffles inside double pipe heat exchangers increases the heat transfer rate in addition to the pressure drop in the annulus side when compared with that in un-baffled heat exchangers
- The annulus average Nusselt number and friction factor increases with increasing SSPBs void ratio, and with decreasing SSPBs cut ratio and pitch ratio.
- There is a slight increase in annulus average Nusselt number with decreasing the annulus-fluid inlet temperature, while its effect on the annulus side friction factor can be neglected.
- Decreasing SSPBs void ratio, cut ratio and pitch ratio enhances the thermal performance index.
- Correlations for the average Nusselt numbers in addition to the Fanning friction factor for the annulus side of the concentric tube heat exchangers with SSPB as a function of the investigated parameters were obtained.
11. Appendix

In the present study the root sum square combination of the effects of each of individual inputs as introduced by Kline and McClintock [26] were applied to determine the uncertainty in all parameters as;

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \left( \frac{\partial \text{d} \text{m} \text{h}}{\partial \text{d} \text{m} \text{h}} \omega_{\text{d} \text{m} \text{h}} \right)^2 + \left( \frac{\partial \text{d} \text{m} \text{h}}{\partial \text{d} \text{m} \text{h}} \omega_{\text{d} \text{m} \text{h}} \right)^2 \]  \hspace{1cm} (21)

\[ \omega_\theta = \pm \left( \frac{\partial \theta}{\partial \text{d} \text{m} \text{h}} \omega_{\theta} \right)^2 + \left( \frac{\partial \theta}{\partial \text{d} \text{m} \text{h}} \omega_{\theta} \right)^2 \]  \hspace{1cm} (22)

\[ \omega_\theta = \pm \left( \frac{\partial \theta}{\partial \text{d} \text{m} \text{h}} \omega_{\theta} \right)^2 + \left( \frac{\partial \theta}{\partial \text{d} \text{m} \text{h}} \omega_{\theta} \right)^2 \]  \hspace{1cm} (23)

\[ \omega_\Delta T = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (24)

\[ \omega_\Delta T = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (25)

\[ \omega_\Delta T = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (26)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (27)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (28)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (29)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (30)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (31)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (32)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (33)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (34)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (35)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (36)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (37)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (38)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (39)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (40)

\[ \omega_{\text{d} \text{m} \text{h}} = \pm \frac{\omega_{\text{d} \text{m} \text{h}}}{\Delta T} \]  \hspace{1cm} (41)

References


Experimental Study of the Influence of γ

Superscripts and Subscripts

Greek Letters

Dimensionless Groups

Acronyms and Abbreviations

Nomenclature
خلاصة البحث

يتناول هذا البحث دراسة تجريبية لخصائص إنتقال الحرارة بالجملة وهبوط الضغط لماء نقي يتدفق في المجرى الحلقى لمبادلات حرارية من نوع أنبوبة بداخل أنبوبة في وضع أفقى، وذلك في حالة وجود وعاء حقلي متصلية كأحد أنواع تقنيات تحفيز إنتقال الحرارة الفيغ ميثر، وقد تم إجراء هذا العمل تبخير معاملات هندسية عدة للعوائق المثقبة عن طريق رفع التشغيل في الجانب الحلقى للمبادل الحراري، وذلك ب겸 جيرون من أهم المعاملات التي تؤثر على الأداء الحراري لمثل هذا نوع من المبادلات الحرارية.

تم تصنيف ثمانية مبادلات حرارية ذات جريان متعاكس، أخذت من دون أنعكاس، أخذت بداخل مجراهم الحلقى عوائق مثقبة مع تغيير النسبة بين مساحة الثقوب ومساحة العائق (φ) من 6.72% إلى 20.4%، والنسبة بين أقصى طول لشباك العائق إلى قطر العائق (δ) من 19.8% إلى 13.16%، بالإضافة إلى نسبة الحطاء بين العوائق إلى قطر اليندرزليك للمجرى الحلقى (λ) من 2.58% إلى 18.16%، والذين تم دراسة تأثير ظروف التشغيل في المجرى الحلقى للمبادل الحراري تبخير معالل التدفق من 0.01 إلى 0.21 لتر/دقيقة والذي يتناسب مع رقم رينولدز من 1380 إلى 5700، وكذلك تغيير درجة حرارة دخول الماء من 15 إلى 25 درجة مئوية والذي يتناسب مع رقم براندل من 5.82 إلى 7.86.

تم إجراء كل التجارب بمرور ماء ساخن بدرجة حرارة دخول إلى الأنبوبة الداخلية ثابتة عند 50 درجة مئوية ومعدل تدفق حمطي 8.06 لتر/دقيقة بينما كان يمر الماء البارد بداخل المجرى الحلقى للمبادل الحراري. تم تنفيذ 168 تجربة معملية وكان منها 147 تجربة على مبادل حراري بدون أي عوائق. وتم عرض نتائج الأداء الحراري للمبادلات الحرارية في صورة حساب معدل إنتقال الحرارة بالجملة المتوسط ورقم ناثيلت المتوسط ومعامل الإحتكاك داخل النفق الحلقى في المبادلات الحرارية بالإضافة إلى معامل إنتقال الحرارة الكلية.

أوضح النتائج في كل التجربة أن تثبيت العوائق المثلية داخل المجرى الحلقى للمبادل الحراري (أنبوبة بداخل أنوابة) أدى إلى زيادة معدل إنتقال الحرارة بالإضافة إلى زيادة معدل الانزلاق في المجرى الحلقى بالمقارنة مع عدم وجود العوائق، كذلك أظهرت النتائج أن رقم ناثيلت المتوسط وعامل الإحتكاك داخل المجرى الحلقى يزيدان بزيادة في رقم ناثيلت وعامل الانزلاق. بالإضافة إلى ذلك فقد أظهرت النتائج زيادة طفيفة في رقم ناثيلت وعامل انزلاق العوائق المثلية داخل المجرى الحلقى، يعكس ذلك تأثير العوائق المثلية على زيادة حرارة دخول الماء بينما يعكس تأثير درجة الحرارة على معامل الإحتكاك.

كانت تؤثر الأداء الحراري الذي يعكس نسبة الزيادة في معدل إنتقال الحرارة بالجملة في نوع العوائق المتصلة بوجود القيمة في القيمة المثلية. وقد أوضح النتائج أن مؤثر الأداء الحراري يزيد مع نقصان في رقم رينولدز. فما ينطبق في النهاية يتم استنتاج معالجات إنتاجية من خلال النتائج العملية، وذلك لحساب رقم ناثيلت المتوسط ومعامل الإحتكاك في المجرى الحلقى للمبادل الحراري في مدى المعاملات التي تم دراستها.