Performance evaluation of simple DPHX with helical baffles in annulus side

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Abstract: In this paper, turbulent thermo-hydraulic analysis of water-to-water passively augmented DPHX with helical baffles is studied by varying the parameters, mass flow rate and baffle spacing. The baffles in annulus side of the inner pipe augments the rate of heat transfer on comparing with the simple DPHX. The influence of spacing between baffles on the heat transfer rate is investigated by varying the governing parameters. The Reynolds number is varied between 22,000 and 12,500 for three different baffle spacing (β) values of 75 mm, 100 mm and 125 mm. The thermo-hydraulic performance parameters pressure drop, Nusselt number and friction factor are examined and put forth corresponding to experimental data. From the results, it is inferred that the effect of baffles on the annulus side of the hot fluid flows through the inner pipe enhances heat transfer rate with increase in the Nu and decreases as the baffle spacing increases.

Keywords: Heat Exchanger, friction factor, helical baffles, heat transfer enhancement, pressure drop.

1. Introduction

The increase in demand to conceptualize, develop and increase the effectiveness of heat exchangers paved way for very wide range of numerous research investigations to improvise the rate of heat transfer without increasing the size and cost. Passive augmentation and active augmentation are two such approaches used and the usage of Nano fluids also used for heat transfer enhancement. The effective usage of augmentation methods resulted in improvisation in heat transfer rate but there was concern on the area of pressure drop. At present the devices, which can produce, swirl in the flow field is used vastly because of its economic aspect and easy installation. The heat exchanger is one such type of apparatus, which is having broad range of industrial applications, in which these types of enhancement techniques can used easily. The utilization of simple double pipe heat exchangers in industries extends areas like pasteurization, reheating and preheating, effluent heating processes and...
digester heating because it is economical effective, easily maintainable and flexibility towards various applications [1]. The compactness of the double pipe heat exchanger makes it as a suitable choice where high pressure, wide range of temperature and low mass flow rate is involved. In the DPHXs, there are two configuration viz. parallel flow and counter flow. Both fluid will flow in the same course in the former case and in the opposite direction in the latter case. There are research works which throw light on the characteristics of the working fluid and design modification [2-6].

Some other researchers studied the heat transfer augmentation like active, passive and other enhancement method. The outcome of these works shown there is significant rise in the pressure drop which affects the power required to drive the pump directly [7,8]. Studies shows that by incorporating different type geometrical modifications the life of equipment can be extended and energy saving, thermal rating rise are evident [9-11]. The augmentation techniques result in high pressure drop which in turn increase the power required for pumping. Hence, the enhancements techniques must be chosen wisely with intense care. Experimental investigation presented by [12] incorporating perforated tabulators in the annulus side shown the improvements in the thermal performance. Experimental investigation of DPHX with wavy strip tabulators inserted in the inner pipe exhibits much needed improvement in the heat transfer characteristics [13]. The usage of typical and perforated discontinuous helical turbulators with various open area ratio and pitch ratios showed that the heat transfer characteristics is increasing with open area ratio with decrease in the pitch ratio [14]. The effect of helical screw tape inserts with variation in ratio of twist and variation in the length of spacers was presented in [16] for laminar flow. The various important aspects, which influence the performance of the heat exchangers are turbulence, effective length of the DPHX, coefficient of heat transfer, flow rates and type of geometrical modification in the aspect of passive augmentation. Baffle is one of important element considered in this area of enhancement of heat transfer. The baffles plays the role of turbulator by creating turbulence hence increasing the velocity of the fluid, which in turn rises the coefficient of heat transfer. Baffles are predominantly utilized in the shell and tube type of heat exchanger. Dough nut type, flower, ring type and helical are some of the various forms of baffles [17].

There are very rare number research work, which studies the effect of helical baffles on the roundel side of DPHX. In this work, our main objective is to investigate the effect of passive augmentation by using helical baffles at different baffle spacing values on the thermo hydraulic performance of the DPHX. The counter flow pattern is considered for the study. The Figure 1 shows the flow pattern of the fluid where the cold fluid flows in the roundel side along the helical path (blue color) and the hot fluid flows along the inner pipe. The experimental setup and correlations used for calculating the various performance factors are presented in the following sections. The effect of Re value and spacing between the baffle on the coefficient of heat transfer and (Δp) pressure drop is studied here.

2. Design optimization of helical baffle for DPHX

2.1 Assessment of passive augmentation by helical baffles with other baffles

The performance of DPHX with helical baffles are better than the segmental baffles [18–20]. The pressure drop will be comparatively less on comparing with the segmental baffles. In case of segmental baffle the rate of heat transfer is more but the overall functionality of the heat exchanger with helical baffle is better [21,22]. On comparison, the helical baffles have less pressure drop, better performance and fouling is less [23–26]. The helical baffles create spiral flow which eradicates losses in the dead zone and fouling also reduced. Initial investment cost is high for helical baffles but in a long run it is very economical because of its least operational and maintenance cost.

2.2 Impact of Baffle Spacing & Shape

There is reasonable amount of decrease in pressure drop with the increment in the space between baffles. At a fixed rate of flow the increase in the spacing will reduce the heat transfer rate
whereas at a constant pressure drop increase in the baffle spacing will leads to decrease in the heat transfer coefficient. The flow velocity will increase with respect to the increase in the baffle spacing which in turn increase the heat transfer coefficient. The shape of baffle is prime factor which will affect the performance. Wenjing et al concluded in their work that the usage of helical baffles results in comprehensively better performance than various other shapes like trisection, quadrant and sextant sector. The usage of helical baffles results in lower pressure drop [27,28].

3. Description of experimental arrangement and procedure

The detailed view of the experimental arrangement is depicted in the Figure 1. The various dimensions of the components used is given in the Table 1. The outer and inner tube materials are stainless steel and copper respectively. A geyser is used to heat the water which will be made to flow through the inner tube. The counter flow formation is considered in this work and three types of inner tube equipped with helical baffles with different spacing values of 75 mm, 100 mm and 125 is fabricated to improve the heat transfer characteristics. The three dimensional model of the pipe with baffle arrangement is shown in the Figure 2 (a) and Figure 2 (b). The inner copper tube is 31 (di) mm inner diameter with 2 mm thickness with thermal conductivity of 410 W/mK and outer stainless pipe is of 54.5 (do) mm inner diameter and 3 mm thickness with thermal conductivity of 16.3 W/mK. Hot water is allowed to pass by means of the inner pipe and the cold water pass through the ringlet side. An electric heater with a capacity of 2 kW is used to heat the water.

![Figure 1 Experimental setup arrangement](image-url)
Table 1. Structural parameters

| Parameter                              | Value          |
|----------------------------------------|----------------|
| Inner Tube Material                    | Copper         |
| Outer Tube Material                    | Stainless steel|
| Internal diameter of Inner Pipe        | 29 mm          |
| Thickness of inner pipe               | 2 mm           |
| Internal diameter of outer pipe       | 54.5 mm        |
| Thickness of outer pipe               | 2 mm           |
| Ringlet side baffles spacing          | 75 – 125 mm    |
| Effective length of Heat exchanger    | 600 mm         |

Our main aim is to investigate about various characteristics of heat transfer like coefficient of heat transfer, friction factor and pressure drop. J type temperature sensors are used to read the water temperature at the inlet and outlet of the inner pipe and outer pipe respectively. A U tube manometer is connected on the annulus side in order calculate the pressure drop as shown in the figure 3 (a) and figure 3 (b). Two pumps with head of 18 m are used to pump the water through the entire setup. The mass flow rate of the both the fluids are measured using rotameter with the range of 0 to 10 kg/s. Gate valves are used in the appropriate positions to control the flow. In the test section, inner pipe with
helical baffles are used to facilitate the rate of heat transfer. Mass flow rate of the cold fluid is varied from 1 kg/s to 3.6 kg/s whereas the mass flow rate of the hot fluid in the inner pipe is maintained at 1.2 kg/s. The change in the temperature is in the range of 42°C, 56°C and 62°C for the baffle spacing of 75 mm, 100 mm and 125 mm respectively. The experiment is reiterated for counter flow configuration at various mass flow rates.

4. Data reduction

The average velocity on the annulus side is given by

\[ \dot{v}_a = \frac{\dot{m}}{\rho A_c} \]

Where,
\[ \dot{m} – \text{Mass flow rate}, \rho – \text{Density of the water}, A_c – \text{Cross Sectional Area} \]

The area of cross section in the annulus side is given by

\[ A_c = 0.5\beta (d_i - d_o) \]

Where,
\[ \beta – \text{Length between the baffles}, d_i – \text{the internal diameter of the outer pipe}, d_o – \text{the external diameter of the inner tube}. \]

The Reynolds number can be find out using the average velocity value using the formula

\[ Re = \frac{\rho \dot{v}_a d_a}{\mu} \]

Where \( \mu \) is the dynamic viscosity on the annulus side.

The rate of heat transfer is given by the subsequent equations for cold fluid and hot fluid respectively

\[ Q_c = m_c C_p (T_{out} - T_{in}) \]
\[ Q_h = m_h C_p (T_{in} - T_{out}) \]

mc and mh represents the mass flow rate of cold and hot fluid respectively, Tin and Tout represents the temperature of fluid at inlet and outlet, \( C_p \) represents the specific heat capacity. The heat transfer characteristics are calculated at the mean temperature.

The Nusselt Number, pressure drop (Δp) and friction factor are determined as described in [1] for plain tube using the relations,

\[ Nu = 0.023 Re^{0.8} Pr^{1/3} \]
\[ \Delta p = 4f \frac{L}{D} \rho \frac{u^2}{2} \]
\[ f' = 0.0014 + \frac{0.125}{Re^{0.82}} \]

For calculating the Nusselt number and friction factor for annulus side with baffle the correlation given by R.M Manglik and A. E. Bergles [29]

\[ Nu = 0.023 Re^{0.8} Pr^{0.4} \left( \frac{\pi}{\pi - 4\delta/d} \right)^{0.8} \left( \frac{\pi + 2 - 2\delta/d}{\pi - 4\delta/d} \right)^{0.2} \]
\[ f' = \frac{0.0791}{Re^{0.25}} \left( \frac{\pi}{\pi - 4\delta/d} \right)^{1.75} \left( \frac{\pi + 2 - 2\delta/d}{\pi - 4\delta/d} \right)^{1.25} \left( 1 + \frac{2.752}{y^{1.125}} \right) \]
the term \(\left(1 + \frac{L}{B}\right)\)

present in pressure drop take into account considering the changes in the baffle spacing with reference to the method adopted by Bell-Delaware for designing the shell and tube heat exchangers [30].

5. Results and discussion

This work investigates the effect of baffle spacing on heat transfer rate and the friction characteristics in a DPHX where water is used as the working fluid. The results obtained in this work is endorsed by means of Nusselt Number and friction factor. The results obtained for plain heat exchanger is compared with the results obtained for various baffles spacing. The helical Baffles are incorporated in the annulus side in order to improve the heat transfer rate. The surface are exposed in simple DPHX is comparatively less as with the heat exchanger with helical baffles. The inclusion of helical baffles as augmentation technique increases the surface area for heat transfer with the same length, with very negligible rise weight of heat exchanger. The ringlet side heat transfer area for all the designs is given in the Table 2. The heat transfer area surges with the reduction in baffle spacing.

| S.No. | Configuration                  | Heat transfer surface area in \(\text{m}^2\) |
|-------|--------------------------------|---------------------------------------------|
| 1     | Simple Double pipe             | 0.00157                                    |
| 2     | Helical baffle with \(\beta = 0.075\) | 0.1132799                                  |
| 3     | Helical baffle with \(\beta = 0.1\) | 0.1153349                                  |
| 4     | Helical baffle with \(\beta = 0.125\) | 0.1195679                                  |

The surface area values of baffle spacing with 0.075 m, 0.1 m and 0.125 m are 13.5, 12 and 11.3 times larger than the simple heat exchanger. From this we can say, the external heat transfer area increases for the same inner pipe length. These data advocates geometrical modification in the outer heat transfer surface of the inner pipe. It makes the heat exchanger to transfer more heat without any major modification in its weight and size. We can say the heat exchanger is more compact and effective. Figure 4 depicts annulus side change in average temperature along the length of the DPHX at the same mass flow rate. It is found that the change in temperature between the entry and exit is maximum for the BF 0.075 configuration with the \(\Delta T\) value of 13.7 K which is due to more heat transfer surface area, whereas 8.6 K, 10.4 K and 12.2 K for Plain, BF 0.1 and BF 0.125 respectively. It shows that the rate of heat transfer for the DPHX with helical baffles is more than the plain DPHX.

Figure 4. Comparison of heat transfer for the DPHX with helical baffles
This due to the fact that lesser the space between the baffles there will be more turbulence and eddy motion, the convective heat transfer is more active when the turbulence level is high. The DPHX with spacing between the baffles of 0.075 m, 0.1 m and 0.125 m provide about 8.73, 7.05 and 5.9 times greater heat transfer coefficient respectively than the simple DPHX configuration. The gradual rise the mass flow rate and noteworthy improvement in the annulus side convective heat transfer coefficient results in a substantial heat transfer augmentation on the annulus side particularly for smaller baffle spacing (BF 0.075).

The Figure 5 shows the deviations in the friction factor value with respect to the Re. The value of friction factor increases with respect to the increase in the spacing between the baffles. It indicates that the increase in the spacing between baffles reduces the disturbance of flow stream. Evidently, from the figure 5 we can say that the friction factor value for BF 0.075 configuration on comparing with the other two configurations is high and its value decreases with the increase in the Reynolds number. The Reynolds number value decreases with the increase in baffle spacing due to the reduction of turbulent fluctuation or eddy motion. Hence the friction factor values are comparatively low for BF 0.075, BF 0.1 and BF 0.125 on comparing with the plain DPHX as the Re value increases.

![Figure 5. Deviations in the friction factor value with respect to the Re](image)

Pressure drop plays a vital role in designing the heat exchangers. The people will show utmost care in the pressure drop values. Because it decides the pumping power requires to maintain a stable flow of the fluid. Figure 6 depicts the variation of pressure drop with Re for the different configurations.

![Figure 6. variation of pressure drop with Re](image)
The pressure drop in the annulus region rises gradually as the Reynolds number value and the spacing between baffles increases. The augmentation technique using helical baffles leads to remarkable rise in the pressure drop. The pressure drop values are 11.3, 5.1 and 2.7 times larger for spacing between baffles 0.075, 0.1 and 0.125 m respectively. The cause for the rise in the pressure drop is increase in the annulus side fluid velocity due to geometry modification, large annulus surface area particularly for lesser baffle spacing value and abrupt modification in the flow pattern. Figure 7 depicts the rise in the pressure drop along the length of the fluid flow in the annulus side at constant mass flow rate of 3.6 kg/s. The higher values of Reynolds number are the prime contributor for the rise in the pressure drop, particularly for the low baffle spacing values. The aforementioned factor is clearly visible from the Figure 7. A huge quantity of fluid energy is spent in the region of 0.1 m to 0.3 m which is ≈ 65 % for β = 0.075 mm. The rise in the pressure drop considerably increase the value of friction factor in the annulus side. Here in this work since we are using 0.6 m as the effective length so we can neglect the contribution of entrance region. The pressure drop become a linear function after 0.2 m. The design modification in the DPHX by passive augmentation will always results in a compromise between pressure drop and coefficient of convective heat transfer.

Figure 7. Pressure drop Vs Postion

These two parameters are reliant each other very heavily. The heat transfer coefficient values can be amplified by improving the mass flow rate of the working fluid i.e. velocity. But it leads to scenario where more pumping power is required. Due to the increase in pressure drop. So the parameter called thermal augmentation factor which is non dimensional number (η) is taken into account in order to have better functional assessment. This variable is calculated using the correlation given below which is presented in [31].

\[ \eta = \left( \frac{Nu}{Nu_0} \right) \left( \frac{\Delta P}{\Delta P_0} \right)^{-1/3} \]

Figure 8. Thermal performance augmentation factor with mass flow rate.
Figure 8 shows the deviation of the thermal performance augmentation factor with the mass flow rate on the annulus side with all three baffle spacing values. As the baffle spacing values decreases, the thermal augmentation factor value also decreases but its value improves significantly with the increase in the working fluid mass flow rate in tao. The average thermal augmentation factor values for BF 0.075 m, BF 0.1 m and BF 0.125 m are 3.94, 4.13 and 4.26 respectively. The BF 0.075 has the minimum thermal augmentation factor but it provides heat transfer enhancement. Hence BF 0.075 can be considered for the applications which are having better priority for heat transfer rate on comparing with the pressure drop. In an overall view we can say the DPHX with helical baffles shows very decent thermo-hydraulic performance than that of a plain DPHX.

6. Conclusions

In this research work, the thermo-hydraulic performance of three different structures of helical baffles on the annulus side of DPHX is studied. The conclusive results obtained from the study is presented below. The presence of helical baffles provides larger effective surface area for heat transfer at the same size of simple DPHX. In the presence of helical baffles the rise in the heat transfer area is 11 to 14 times more than the simple heat exchanger. The passive augmentation in terms of helical baffles does not have a major impact on the size and weight. The heat transfer rate is much better for the BF 0.075 configuration at same mass flow rate. From this study we can conclude that the BF 0.125 configuration is have better thermal enhancement which results in better thermo-hydraulic performance. The BF 0.075 configuration is suitable is having high heat transfer capacity. This type can be considered for applications where pressure drop in not a major element to be considered. In a complete overview of this work we can conclude in such a way that the DPHX with helical baffles is very efficient in augmenting the rate of heat transfer and exhibiting better thermo-hydraulic performance.

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