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Enhancement of Heat Transfer rate using Axially Interrupted Rectangular Fins in Double Pipe Heat Exchanger

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Abstract: The current investigates the thermo-fluid behavior of a double pipe heat exchanger (DPHE) featuring axially interrupted rectangular fins (AIRF) on the annulus part. The inner tube under this study with AIRF represents an interruption of straight longitudinal fins. This modification introduces periodic breaks along the tube's surface, effectively disrupting the boundary layer of the fluid flow. Consequently, it enables a non-continuous fluid passage along the length of the tube, potentially enhancing heat transfer. The experimentation employs standard liquid water, for investigations conducted under varying cold water mass flow rate 0.136 Kg/s to with 0.374 Kg/s keeping hot water at constant flow rate of 0.34 Kg/s with a fin split interval of four different lengths 7mm,27mm,55mm,100mm. A comprehensive investigation of the AIRF arrangements is carried out in contrast to the plain pipe arrangement, concentrating on fluid flow parameters such as Nusselt number (Nu), friction factor, heat transfer rate, and overall performance factor. The findings reveal that heat transfer rates in an annulus equipped with 7mm AIRF exceed those of a plain pipe by 59.31% under similar fluid flow conditions. The Nusselt number shows 1.5 times increase in the 0.007 m AIRF arrangement compared to the plain pipe. Thermal performance factor for 7mm interrupted length of AIRF outperforms other models.

Keywords: Double Pipe Heat Exchanger, Axially Interrupted Rectangular Fins, Pressure Drop, Heat Transfer Rate, Thermal Performance Factor

1. Introduction

The importance of heat exchangers extends to various engineering processes and research applications. Industries like excess temperature retrieval, chemical progressions, adaptation structures, energy production, and cooling heavily rely on these crucial components. A prevalent and widely employed type of heat exchanger in diverse productions is the DPHE [1]. Its popularity is chiefly due to its remarkable design flexibility, cost-effectiveness, straightforward construction, ease of disassembly for cleaning, and overall adaptability to different usage scenarios. A DPHE comprises of two concentric tubes, typically facilitating a counter-current flow of fluids [2, 3]. Enhancing the heat transfer capacity of a DPHE can be achieved by incorporating heat transmission surface attachments known as fins. Fins prove beneficial in scenarios where convection film coefficients are low, especially in processes involving gases or viscid liquids [4]. For example, straight longitudinal fins stand commonly employed in condensing claims and for handling viscid liquids within DPHE. This utilization of fins allows for an augmented heat transfer to counteract the limitations associated with lower heat transfer efficiency [5, 6]. Dependent on the specific claim, the incorporation of fins could enhance the heat transmission zone by a quantity ranging from 5 to 30. The type, characteristics, and arrangement of fins constitute crucial parameters for enhancing thermo-fluid performance. Extensive literature exists on the use of fins in DPHEs, encompassing various fin types employed in thermo-fluid processes [7]. The purpose is to augment heat transmission and enhance the overall effectiveness of these heat exchangers.

The impact of longitudinal fins with varying tip thickness in a finned DPHE has been examined under completely established laminar circumstances. The findings suggest that the tip-to-base proportion plays a pivotal role as a decisive factor in optimizing factors such as mass, frictional losses, heat transfer amount, and cost [8]. In an experimental investigation involving a water-to-air DPHE with discontinuous helical turbulators, it was observed that thermal performance improves with an upsurge in open area ratio and a reduction in pitch fraction [9]. Additionally, an efficient approach was studied to optimize a finned DPHE utilized in manufacturing convalescent rings. The objective has been to exploit heat transfer in the convalescent heat exchanger while adhering to constraints related to cost, compactness, and pressure drop [10]. The results indicated that the optimum configurations could enhance the performance by up to 10.8%. An experiment was conducted on a pin fin DPHE to assess both energy transmission and stream features. A precise governing equation was subsequently developed, grounded in the reduction of entropy production, considering various flow conditions and pin distances [11]. The aim was to enhance rate of thermal capability and minimize pressure harm. The study also presented and deliberated on best fin profiles for a finned DPHE operating in entirely established viscous flow, with the specific goal of exploiting the Nu. The conclusion drawn was that the best arrangements remained contingent on the excellent of the typical distance utilized in defining the Nu. In an experimental study utilizing pierced tabulators to enhance thermal effectiveness in a DPHE, it was observed that efficiency enhances with the pitch proportion [12]. In a separate investigation, a contrast was complete among a simple smooth-walled DPHE and a DPHE where the inside pipe wall is helically ribbed. The findings indicated that the thermal transmission rate in the inclined ribbed DPHE was 3 times higher than that in the conventional DPHE [13, 14]. Though, it was noted that the pressure globule in the previous was complex compared to the concluding. In a study examining the effectiveness of an exponentially ribbed inside pipe in a DPHE, it was observed that the exponential fin outperformed the triangular fin by as much as 15%. Another investigation focused on thermal transmission expansion for annular stream by means of a revolving inside tube [15]. The outcomes revealed that the developments achieved through the turning of the tube were three times greater than those attained with traditional fins [16]. Experimental investigations have been conducted to know the thermal transmission of air in a DPHE. Notably, the impact of a fused lead fin on the development of the annulus thermal transmission factor has been investigated, leading to notable improvements in the overall thermal transmission rate [17]. An investigation was conducted to understand the thermohydraulic recital of various arrangements of a gas-to-liquid DPHE with helical fins. The study demonstrated the effectiveness of helical fins in attractive thermal transmission and refining thermohydraulic effectiveness [18]. Additionally, triangle winglet couple twister producers were introduced sideways the middle line of the helical conduit in a DPHE with quadrangular sectional view. The observed outcome of this arrangement indicated that thermal transmission was 16.6% larger compared to a helically ribbed DPHE starved of vortex producers. Numerous study clusters have explored diverse methods to thermally enhance DPHE [19, 20]. In a recent literature review, the authors extensively covered temperature augmentation in DPHXs, focusing on an examination of submissive improvement methods like fins. The review provided thorough statistics on thermofluidic effectiveness for various fin categories, offering valuable understandings for future researches [21]. Fins, owing to their advantages, find application not only in DPHEs but also in plentiful other heat argument scenarios wherever thermal transmission constants are relatively low.

In heat exchange processes, achieving a turbulent flow regime is highly desirable as it offers the most effective heat transfer. However, in the case of high Prandtl number liquids, practical considerations may limit the feasibility of maintaining turbulent flow [22]. The elevated viscosity and constraints on tube velocity to prevent tube wall erosion often result in laminar flow. This is particularly relevant in areas like oil chillers for great manufacturing vapor turbine engines or maritime systems, whereas engine oil is chilled utilizing aquatic liquid [23]. In these scenarios, the convection film constants are characteristically insignificant, necessitating a larger thermal transmission zone to compensate for the reduced efficiency of heat transfer in laminar flow conditions. Cylindrical turbulators are used for heat transfer enhancement by placing circumferentially separated by 90^0 in inner pipe at regular pitch. A model with alternative arrangement in orientation outperformed [24]. Numerical investigation is carried on helical baffle spacing in annular space of double pipe heat exchanger with 0.025-0.1m spacing. It is observed that thermal performance and pressure drop is increasing with spacing and Reynolds number [25]. CFD simulation is carried out by FLUENT software with helical fin spacing varying from 0.05-0.2m in annulus side of DPHE. Within the scope of the present investigation, the helical fin configuration with a fin spacing of 0.1 m provides the optimal thermohydraulic performance [26]. The fin geometries included interrupted rectangular fins, circular fins and helical ribs in double pipe heat exchanger. The maximum heat transfer enhancement was obtained for a rectangular fin and the minimum was for a circular fin [27]. Minichannel heat sink is widely used in waste heat recovery systems for their compactness and ability to recover heat effectively from high heat flux applications [28].

Numerous researchers have conducted extensive work on double pipe heat exchanger with rectangular fins to enhance their thermal performance through the use of longitudinal rectangular fin configurations. However, there has been limited study on the use of interrupted rectangular fins experimentally in double-pipe heat exchangers, which are widely used in industries, to improve their thermal performance.

Therefore, the prime objective of this research is to investigate the heat transfer thermal performance DPHEs. In prior studies, most of researchers studied numerically about interrupted fins considering the recent developments in which interrupted fins dimensions are not constant as well in studies interrupted fins are considered on different diameter size pipes. The newly designed interrupted rectangular fins were proposed to enhance heat transfer rates by increasing the turbulence and heat transfer area. Inner pipe with same diameter for different interrupted lengths are affixed to the annulus side of a conventional plain pipe, strategically expurgated and counterbalance at detailed intermissions to enhance the thermo fluidic recital of the DPHE. The investigation focuses on differentiating between a plain pipe DPHE and various configurations of AIRF double pipe heat exchanger. Four distinct AIRF double pipe heat exchanger configurations with interrupted lengths as 100 mm, 55 mm, 27 mm and 7 mm AIRFs, have been designed to enhance the heat transfer rate of the DPHE. The relevant heat exchanger domains are fabricated to represent each configuration, and thermo fluidic effectiveness is assessed underneath varying mass flow rates. Experimental observations are conducted to assess fluid stream, thermal transmission rate, and overall performance indicators. Additionally, integral measures like heat transfer rate, friction factor, pressure drop, and Nusselt number are investigated and thoroughly deliberated. To validate the dependability and precision of the experimental observations, the friction factor and Nu outcomes are contrasted with established correlations such as Petukhov's and Dittus-Boelter, respectively. Subsequently, the experimental results obtained from the newly developed models of AIRF configurations are compared with those of the plain pipe.

2. Experimental Methodology

The aim of this study is to experimentally assess and contrast the heat transfer rate of DPHEs featuring various arrangements of AIRF in the annulus. Specifically, five annulus configurations have been devised, including one plain pipe configuration utilized for experimental verification, and four AIRF arrangements with distinct split intermissions. The objective is to analyze and contrast the thermal performance of these DPHEs under different configurations. Figure 1 represents experimental set up setup involves two sets of water tanks, each with a 25-litre capacity for hot and cold water. The test section consists of 1.5 meters long, with inner tube having a 22 mm inside diameter and 25 mm outside diameter. The outer pipe is 57 mm inside diameter and 60 mm outside diameter, both tubes are made of stainless steel 304. Inlet and outlet temperatures are measured with digital temperature indicators at the entry sections of hot and cold liquids, as well as the exit sections of hot and cold fluids. A U-tube manometer is fitted at the entry and exit of cold liquid to measure pressure drop.

The hot water is pumped through the inner pipe of double tube heat exchanger where as cold water flows through the annulus region, re-circulated into the hot reservoir, while the cold-water outlet is drained into a sump. The outlet temperature of cold water is measured using a digital temperature indicator (0^oC - 100^oC) installed in the control panel, with K-Type thermocouples utilized for temperature measurements.

Figure 1. Experimental set up

Bypass valves and controlling water are employed for both hot and cold water to maintain the correct flow rates. A 0.5 HP pump is utilized for pumping hot water, while a submersible pump is utilized for cold water. The hot liquid is heated by a 2000W heater fitted to the hot water tank, and the temperature is maintained constant with a thermostat.

3. Experimental Procedure

Hot water is pumped through inner pipe with a constant flow rate of 0.34 kg/s at 353K. Cold water is pumped in the annulus region with varying mass flow rates from 0.136 Kg/s to 0.374 Kg/s at 303K. Total of 40 experiments were run, 8 flow rates are considered for each configuration, of which first 8 flow rates are run for plain pipe .Flow rates are regulated according to the investigations using a flow meter and values for different configurations as shown in schematic Figure 2.

Four different AIRF configurations are considered as shown in Figures 3, 4 configurations 1 involves six circumferential fins, introducing interrupted fins along the length of plain pipe with fin length as 100mm and height as 10mm with interruption length of 100 mm. Further addition of a pair of fins axially with 6 circumferentially is introduced to enhance turbulence, with uniform spacing between each set of fins. Configurations 2, 3, and 4 are considered with axial discontinuous fins, featuring interruption lengths of 55 mm, 27 mm, and 7 mm, respectively. The outside pipe is shielded using asbestos cord to diminish heat flow to the environments.

1 Heater; 2 Hot water tank;3 pump;4 over flow;5 flow controlvalue; 6 rotameter; 7 inelt hotwater temperature; 8 finned tube;9 outlet hot water temperature; 10 inlet cold water temperature; 11 inlet cold water pressure indicator; 12 oulet coldwater temperature; 13 outlet coldwater pressure indicator; 14 rotatmeter;15 flow controlvalue; 16 overflow; 17 pump; 18 coldwater tank; 19 sump.

Various parameters, including temperature, flow rate, pressure, and heating units, are monitored and recorded at corresponding points in the setup. Before commencing experiments, all parameter reading devices are calibrated using uncertainty analysis to ensure accurate readings. The system reaches steady-state conditions within 20-25 minutes.

4. Data Reduction

The DPHE estimated values from the analysis were based on outlet temperatures and pressure drops. The subsequent calculations were applied to determine the heat transfer rate, Nu, and friction factor [29].

For hot water, $Q_h = \dot{m}_h C_{ph}(T_{h1} - T_{h2})$ (1)

For cold water, $Q_c = \dot{m}_c C_{pc}(T_{c2} - T_{c1})$ (2)

$$Q_{avg} = \frac{(Q_c+Q_h)}{2} \quad (3)$$

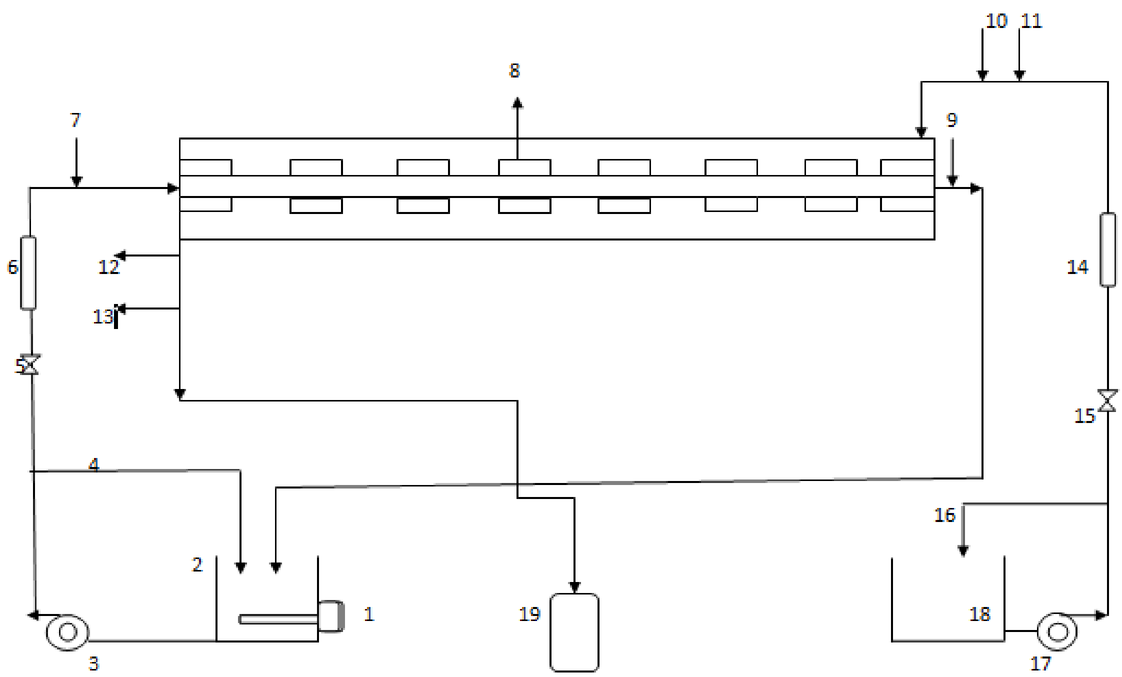

Figure 2. Schematic representation of the Experimental apparatus

Figure 3. Design of Plain pipe for double tube heat exchanger

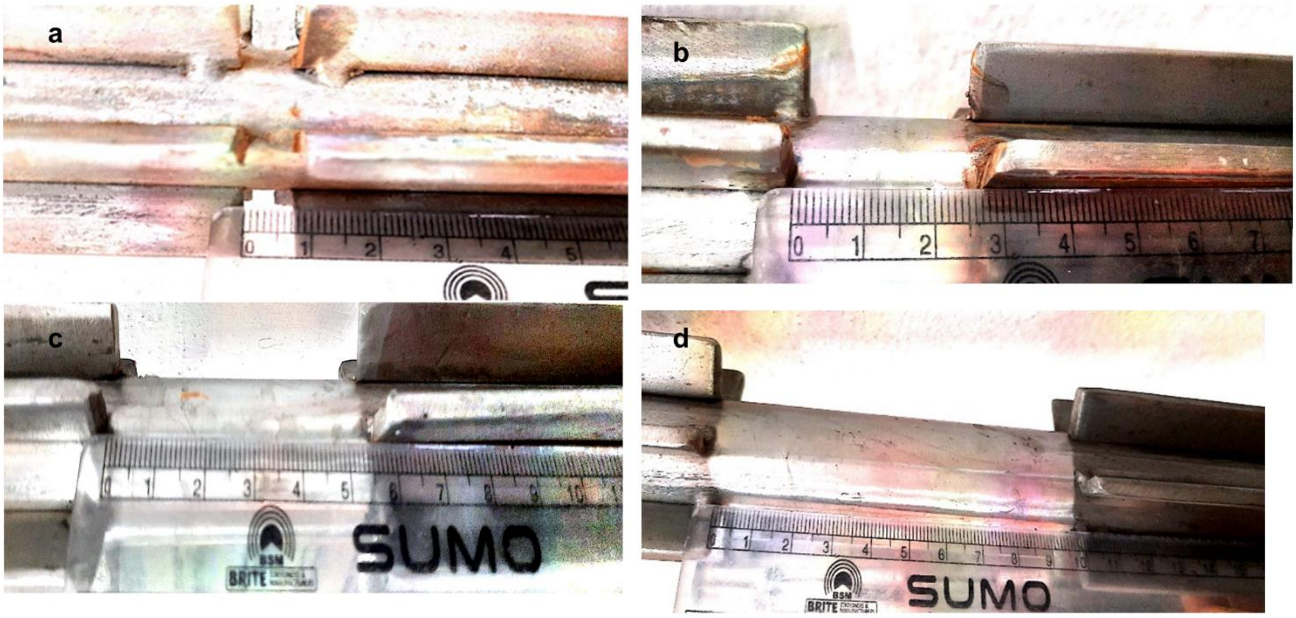

Figure 4. Design of finned tube with **a)** 7mm, **b)** 27, **c)** 55, **d)** 100mm interrupted

In the given expressions, $\dot{m}$denotes the mass flowrate, T signifies the temperature, and $C_p$represents the specific heat. The subscripts 1 and 2 correspond to the entry and exit of hot and cold liquid, correspondingly. Similarly, the subscripts h and c represent hot and cold liquid, correspondingly. The hydraulic diameter can be computed using the following equation:

For the plain pipe,

$$D_h = D_i \tag{4}$$

For finned pipe, $D_h = \frac{4A_f}{P}$ (5)

For each side of the DPHE, the characteristics are assessed at the mean temperature of the entry and exit. Concerning the annulus area,

Nusselt number,

$$Nu = \frac{h_o D_h}{k_o} \tag{6}$$

Here, k_o represents the thermal conduction of cold liquid, and h_o denotes the thermal transmission coefficient on the annulus part.

The theoretical Nusselt number was calculated using the Dittus-Boelte correlation [30]

Nu=0.023 (Re)0.8 (Pr)0.4 (7)

The theoretical friction factor was determined using the Petukhov correlation [30]

f =(-1.64+0.79ln(Re))-2 (8)

5. Results and Discussion

The effect of rectangular fins with interrupted lengths on the thermal performance of double tube heat exchanger was investigated. The objective of this study was to investigate a novel rectangular fin geometry featuring varying the interrupted lengths. The results obtained can be summarized in the following sections.

5.1 Model Validation

To validate the experimental findings of this study, a comparison was made between the experimental values and standard equation. The results were compared with those obtained from an Dittus-Boelter equation as referenced [29] in the study, as illustrated in Figure 5 .This figure illustrates the validation of the Nusselt number for the traditional model across different mass flow rates. Figure 5 represents the Nusselt number is increasing with the mass flow rate, and closely follows dittus-boelter equation it is deemed satisfactory, with a maximum error of less than 14.42%.

The experimental friction factor values and correlation values for different mass flow rates in comparison is shown in figure 6, indicating that the average difference between them remains within an acceptable range. The maximum error in the friction factor from the Petukhov's correlation is 7.98%. Upon further scrutiny of the comparison between Nu and friction factor, it is evident that experimental values closely align with the correlated values.

The variation in results may be from several factors, such as heat losses, friction losses, instrument accuracy, etc. Therefore, it is concluded that the established new configuration of varying interrupted lengths approach is sufficiently accurate and can be used to analyze the parameters considered in this study, as detailed in the following section. Consequently, an experimental analysis can be conducted for interrupted fins with AIRFs of interrupted lengths 100 mm, 55 mm, 27 mm, and 7 mm in the DPHE to evaluate its thermal performance.

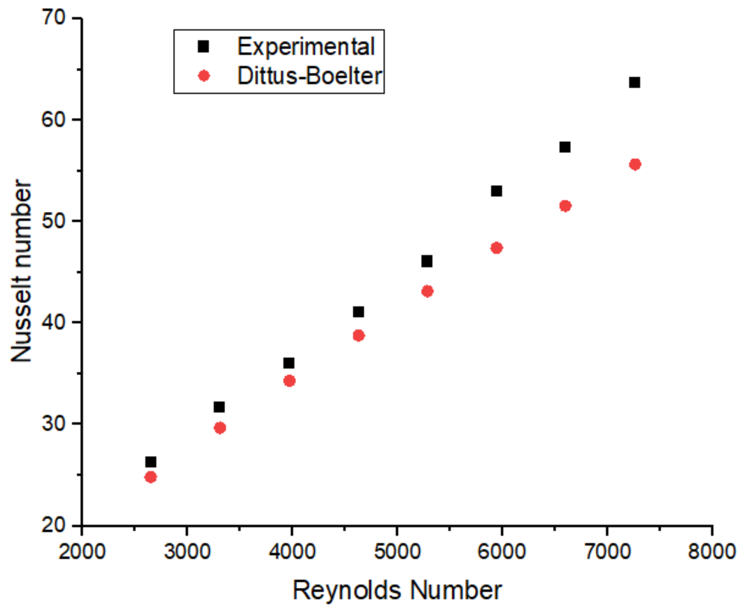

Figure 5. Comparison of Nu of DPHE plain pipe with experimental and Dittus-Boelter Correlation

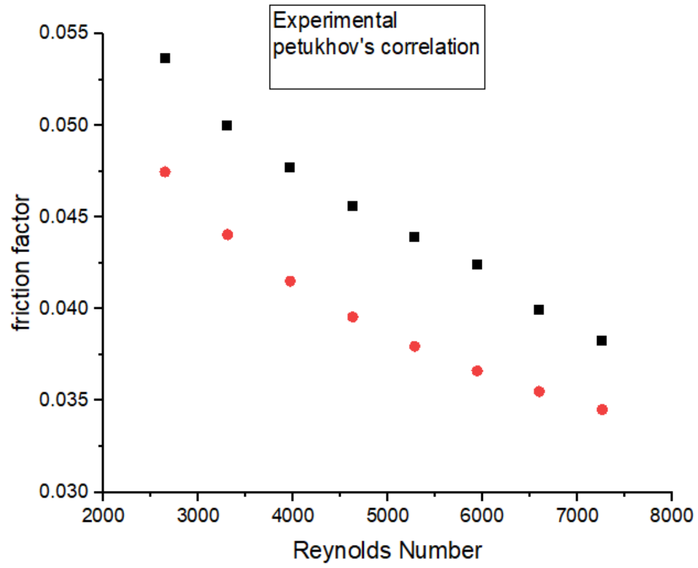

Figure 6. Comparsion of friction factor of DPHE plain pipe with experimental and Petukhov's Correlations

5.2 Uncertainty Analysis

Uncertainties in empirical assessments are inevitable owing to factors such as unfitting apparatus assortment, atmospheric constrictions, apparatus accuracy, readability, and hominid errors [30]. Hence, it is crucial to identify the maximum possible uncertainty values in independent features (temperature, mass flow rate, pressure drop) and assessed measures through the DPHE investigation (friction factor, Nu, and overall performance factor) [31]. The parameters with uncertainty include measurements of temperature (U_T $\pm$ 10 °C), mass flowrate (U_M $\pm$0.01 kg/s), and pressure drop from manometer readings (U_P $\pm$0.1 kg/m²). Let U_R define the uncertainty value from the result, and $U_1, U_2, \dots, U_n$ define the uncertainty values from the independent features. The result R is a demarcated formulation in the framework of the independent features $X_1, X_2, \dots, X_n$. The absolute (U_{Ra}) and relative (U_{Rr}) uncertainties in the result could be calculated with the following formulations [32, 33], and the uncertainty of assessed measures through the DPHE experimentation is obtainable in Table 1.

$$U_{Ra} = \pm\sqrt{\left(\frac{\partial R}{\partial X_1}U_1\right)^2 + \left(\frac{\partial R}{\partial X_2}U_2\right)^2 + \cdots + \left(\frac{\partial R}{\partial X_n}U_n\right)^2} \quad (9)$$

$$U_{Rr} = \frac{U_{Ra}}{Estimatedquantity} \quad (10)$$

Table 1. Uncertainty vaues of measured features during DPHE analysis

Estimated quantities	Absolute uncertainty	Relative uncertainty
Friction factor	$\pm$0.0024	0.095 %
Nusselt number	$\pm$0.1923	0.281 %
Overall performance factor	$\pm$0.2279	0.314 %

5.3 Enhancement of Nusselt Number

Experimentally, the thermal characteristics between finned tubes of different interrupted lengths and plain tube of DPHE are assessed at various mass flow rates as shown in figure 7. Cold water flows on the annulus side at different rates, while hot water at a constant rate of 0.34 kg/s at 80^{0}C on the tube side. The heat transfer rate increases at higher reynolds number values due to increased turbulence. This leads to a higher heat transfer coefficient, as the fluid velocity increases, promoting intense mixing in the turbulent boundary layer and a thinner laminar sub-layer. Turbulent flow accelerates thermal mixing, resulting in an increased Nusselt number. The interrupted finned heat exchanger exhibited a higher Nusselt number than the plain pipe at a given mass flow rate. The interrupted fins act as disrupts the thermal boundary layer on the annular side. Furthermore, the interrupted fins increase the contact surface area between the hot surface of the tube side and the cold water in the annular side, thereby enhancing the convective heat transfer coefficient.

The findings indicate that increase in average Nusselt number for a finned tubes with a 7 mm interrupted length is 28.79% higher than that for a 27 mm , 37.35% higher than that for a 55 mm ,62.77 % higher than that for a 100 mm, it is 1.5 times the value observed in the plain pipe. Because of smaller interrupted length cause more maximum disruption to the boundary as well increase in contact area.

5.4 Enhancement of Heat Transfer Rate

The heat transfer rate for the various finned configurations is presented in Figure 8 illustrating the variation in the heat transfer rate for a DPHE with and without AIRFs. The heat transfer values are improved at higher Reynolds numbers due to increased mixing and turbulence in the cold water side, which prevent the formation of thick boundary sub-layers. Also the average increase in the heat transfer of 7mm, 27mm, 55mm, and 100mm interrupted lengths is 59.31%, 48.01%, 41.61%, and 27.26% as compared with the plain pipe. This occurs because the temperature difference between the input and output fluids increases in a heat exchanger with a greater contact area and turbulence generated by interrupted fins.

5.5 Friction factor

The effect of mass flow rate, varying the interrupted lengths of a finned tube on friction factor is shown in Figure 9. Whereas friction factor decreases with the increase of mass flow rate. This behavior is opposite to that of heat transfer behavior. Furthermore, the friction factor is higher for a finned tube compared to other plain pipe which increases with decrease in interrupted length. This is due to as interrupted length decreases more number of intervals come across the fluid flow in the annulus side.

The friction factor can be determined by utilizing the experimental pressure drop, and the following equation can be applied [29]:

$$f = \frac{\Delta P}{\left(\frac{L}{Dh}\right)\frac{1}{2}\rho V^2} \tag{11}$$

Where ΔP is the pressure drop in the annulus side, L is the test section of the pipe, ρ is the density of the liquid, and $V$is the velocity of the liquid, D_h hydraulic diameter.

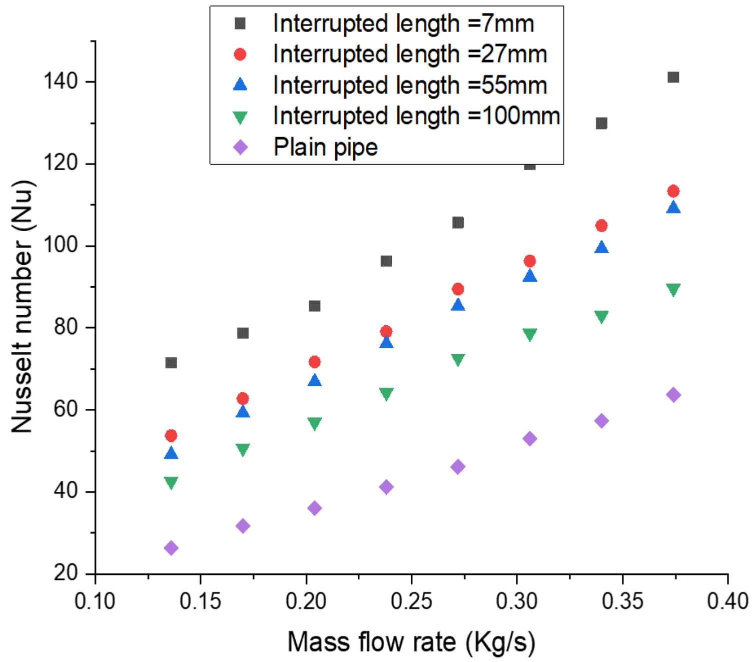

Figure 7. Comparison of Nusselt number of plain pipe with different AIRFs DPHE

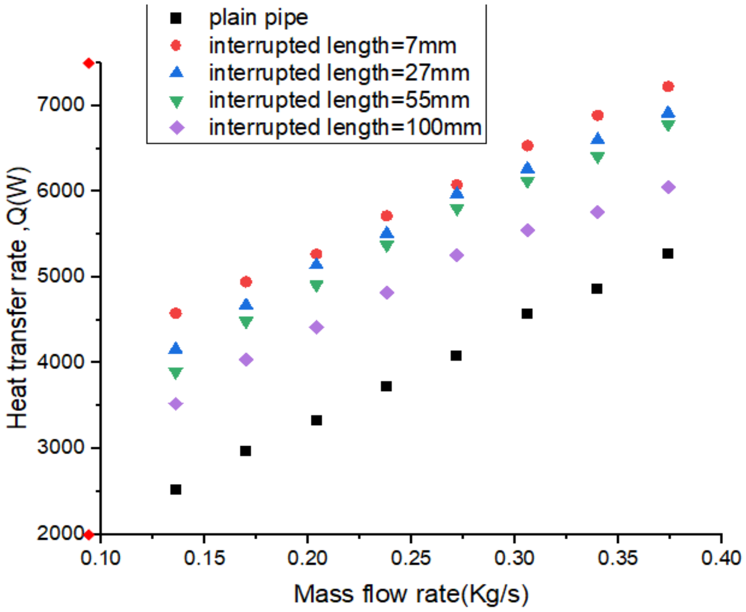

Figure 8. Comparison of heat transfer rate of plain pipe with different AIRFs DPHE

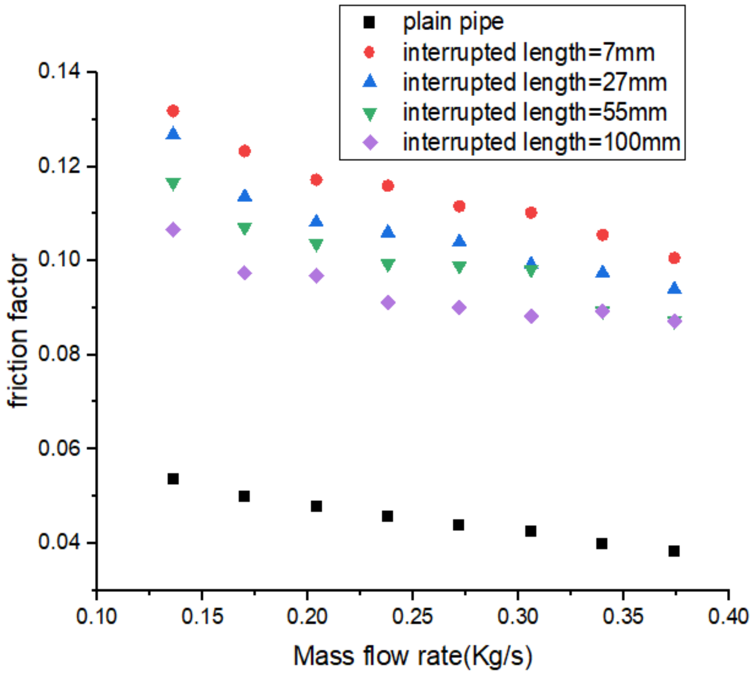

Figure 9. Comparison of friction factor of plain pipe with different AIRFs of DPHE

5.6 Thermal performance factor

To achieve successful thermal performance enhancement in a heat exchanger, the heat transfer coefficient must exceed the pressure drop across the fluid flow, as demonstrated in below equation [29]. Figure 10 illustrates the variation in thermal performance of various interrupted lengths of finned tube with the mass flow rate. Thermal performance factor is greater than one for all configurations. Of all the configurations of varies interrupted lengths 7mm interrupted length finned tube show highest TPF value of followed by 27mm ,55mm and 100mm interrupted lengths. It is observed that as mass flow rate increases TPF decreases. Due to increase in area of contact and heat transfer coefficient at low mass flow rates TPF at low mass flow rates is higher and at higher mass flow rates it is reverse. TPF values obtained in this current experimental work and those with previous work [34-35].

The thermal performance factor can be computed utilizing the below formula [29]:

$$\eta = \frac{(\frac{Nuf}{Nu_s})}{(pf/p_s)^{1/3}} \quad (12)$$

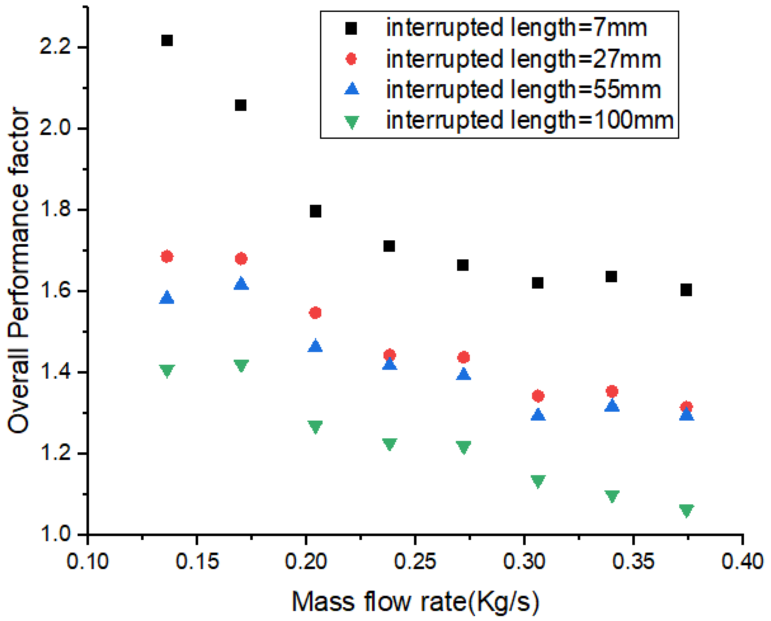

Figure 10. Comparison of overall performance factor of plain pipe with different AIRFs DPHE

6. Conclusion

An experimental study was conducted under various conditions to determine the Heat transfer rate, Nusselt number, and friction factor in the annular side of an air-water double-pipe heat exchanger. A total fourty experimental runs were conducted for both finned and unfinned pipes, of different interrupted lengths (7 mm, 27 mm, 55mm and 100 mm). This split fin configurations enhances both the contact surface area and fluid turbulence. Based on the results obtained, the following conclusions can be drawn:.

- The average Nusselt number for finned tube heat exchanger increases as the interrupted length decreases. Specifically, the Nusselt number for a finned tube with interrupted length of 7mm is 28.79%, 37.35% and 62.77% higher than that for 27 mm, 55mm and 100 mm respectively.
- Within the possibility of this research, the minimum split interval with 7 mm provides the maximum thermo fluidic performance.
- The heat transfer rate of a heat exchanger with interrupted lengths exhibited significant enhancement compared to an un finned heat exchanger. The average overall heat transfer rate increased by 59.31%, 48.01%, ,41.61%, 27.26% when using finned tube heat exchanger of interrupted lengths of 7 mm, 75 mm, 55mmand 100 mm, respectively.
- The friction factor in the annular side increased with the use of finned tubes, with the maximum values observed for the smallest interrupted length.
- The thermal performance factor was found to be greater than one for all selected finned tube heat exchangers, with the highest value of 2.21 observed for interrupted length of 7mm for a mass flow rate 0.136 kg/s.
- Overall, the thermal performance of the DPHE is enhanced with AIRFs, and the increase in friction loss is counterbalance by the greater heat transfer rate

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Data Availability

The data supporting the findings of this study can be obtained from the corresponding author upon reasonable request.

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