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EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER ENHANCEMENT IN SHELL AND TUBE HEAT EXCHANGER USING DISCONTINUOUS CURVED AND LONGITUDINAL STRAIGHT FINS

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Abstract

In the present study, an experimental facility's design and performance evaluation was conducted to study the turbulent heat transfer for the annular side of the concentric tube in the shell and tube heat exchanger. The single-phase water and air were used as working fluids in the tube and annular core, respectively. The passive technique was used to enhance the heat transfer using two types of fins: discontinuous curved fins and longitudinally straight fins. The air mass flow rate of the annular side ranged between (0.03 to 0.13 kg/s), while the water flow rate of the tube side was fixed at 30 l/min. The turbulent flow was achieved on the annular side for all air flow rate tests, and the flow was laminar for the tube side. The experimental data was compared, and the results showed that extended surfaces on the inner tube of double pipe heat exchangers enhance heat transfer by increasing the flow Nusselt number. Discontinuous longitudinal fins increase the Nusselt number by 19.3% on the waterside and 40% on the airside. Discontinuous curved fins further increase the Nusselt number by 24.2% on the waterside and 42.3% on the airside. Curved fins provide higher heat augmentation than longitudinal fins, with a heat exchanger effectiveness of 52.8%. Longitudinal fins and smooth tubes have 52% and 48% effectiveness values, respectively. This study on extended surfaces in double-pipe heat exchangers provides valuable insights for improving heat transfer efficiency, leading to more efficient designs and cost savings in various industries.

Keywords: Curved fin, Heat exchanger, Extended surfaces, Nusselt number.

1. Introduction

A heat exchanger is a mechanism that uses convection and conduction to transfer energy between two fluids through a solid surface [1-3]. These devices have various applications in different industries, such as power generation, air conditioning systems, the automotive sector, and chemical processing [4-6]. Designing a new heat exchanger is commonly known as the sizing problem, which involves selecting the appropriate construction type, flow arrangement, tube and shell material, and physical dimensions to meet the specified heat transfer and pressure drop criteria of existing heat exchangers [7-9].

In the past decade, multiple numerical studies have examined the impact of various fin shapes on heat transfer enhancement. Jude et al. [10] performed a comprehensive numerical analysis on a shell and tube heat exchanger, comparing the obtained data with theoretical calculations based on the NTU method. This analytical investigation revealed a remarkable disparity between the computational fluid dynamics (CFD) simulation results and the corresponding theoretical calculations. This disparity amounted to a 7.3% deviation between the two approaches, indicating a notable disparity in their outcomes.

Kadhim et al. [11] examined the temperature difference in a cross-flow heat exchanger using Computational Fluid Dynamics (CFD) with ANSYS Fluent. Their results revealed that heat transfer and temperature difference were higher in a finned tube heat exchanger compared to a smooth tube heat exchanger. Another study by Melvinraj et al. [12] modelled parallel flow heat exchangers and ribbed tube heat exchangers using a commercial CFD package. They used ANSYS 14.5 to design and analyses the heat exchangers and found that the effectiveness of the heat exchanger can be improved significantly by using ribbed tubes. In a separate study, Kanti et al. [13] investigated the impact of the outer material (Stainless Steel and Galvanized Iron) on shell and tube heat exchangers without baffle plates. Using FLUENT 15.0, they discovered that the effectiveness and efficiency of Stainless Steel coupled with copper inner material were better than Galvanized Iron, making Stainless Steel a superior option for outer material in a heat exchanger.

Sivalakshmi et al. [14] experimented to assess the performance of a double-pipe heat exchanger with and without helical fins. They evaluated the heat transfer rate, heat transfer coefficient, and effectiveness of the heat exchanger. They varied the flow rate of the hot fluid in the range of 0.01 kg/s to 0.05 kg/s while keeping a constant hot fluid temperature of 80 °C at the inlet. They showed that the use of fins increased the heat transfer coefficient. At higher flow rates, the heat exchanger's average heat transfer rate and effectiveness improved by 38.46% and 35%, respectively. Kundu et al. [15] performed an experimental study to investigate the beneficial design of shell and tube heat exchangers with longitudinal fins of different shapes. They attached rectangular and trapezoidal fin shapes longitudinally to the fin tubes. The experiment's results revealed that, under the constraints of a heat exchanger, the heat transfer rate was lower with the rectangular cross-section while keeping the outer shell diameter constant. However, using trapezoidal fins resulted in an increased heat transfer rate and a reduced pressure drop. This suggests that the trapezoidal fin design is more effective for heat transfer.

Lee et al. [16] carried out a study to examine the effect of cribriform annular finned tubes on the heat transfer characteristics of a heat exchanger. The study discovered that for the two-hole and four-hole cases, the convective heat transfer coefficient increased by 3.55% and 3.31%, respectively, while the pressure drop rose by 0.68% and 2.08%, respectively. Another research study by Iqbal et al. [17] aimed to determine the optimal shape for longitudinal triangular fins that fit closely to the outer surface of the inner pipe. The study focused on laminar, incompressible, and fully-developed flow in the shell of two concentric circular pipes with regular heat flux. The fins were straight, non-porous, and made of highly conductive material, with regular distribution around the periphery of the inside pipe. Using the Piecewise Cubic Hermite Interpolating Polynomial (PCHIP), the study found that the optimal fin shape was significantly influenced by the ratio of radii, the number of fins, the number of control points, and the characteristic length. The Nusselt number increased by up to 138%, 263%, and 312% for the trapezoidal, parabolic, and triangular shapes, respectively, for the equipollent diameter, and by 212%, 90%, and 59%, respectively, for the hydraulic diameter.

The impact of internal aluminum baffles on heat transfer and pressure drop in a double-pipe heat exchanger was investigated by Kanade et al. [18] using computational fluid dynamics (CFD). The study utilized semicircular and quarter-circular geometries for the baffles on the inner pipe. The quarter-circular baffles produced more consistent turbulence, resulting in a higher heat transfer rate than the semicircular baffles and the heat exchanger without baffles. Bhola et al. [19] studied an annular tube heat exchanger with a rectangular insert, modelled and simulated using SOLIDWORKS and ANSYS Workbench. CFD computations were performed under different mass flow rates and flow conditions. The results showed that the rectangular insert increased the inside convective heat transfer coefficient compared to a smooth tube. Counterflow was more efficient than parallel flow, resulting in a higher heat transfer coefficient. Sheikholeslami et al. [20] experimentally studied the flow and heat transfer characteristics of a horizontal annular tube double pipe heat exchanger with a copper inner tube and PVC outer tube. The study revealed that the Nusselt number on the waterside increased with the temperature and flow rate of water, while the Nusselt number on the air side showed the opposite trend. An increase in the inlet temperature and velocity of water increased by the friction factor.

Córcoles et al. [21] conducted a numerical analysis to investigate how different geometric factors affect a heat exchanger with two pipes of inner corrugated walls. The study focused on turbulent flow conditions and used the Realizable k-ε turbulence model with a three-dimensional unstructured tetrahedral mesh scheme. They showed that the configuration with the lowest helical pitch (P/D = 0.682) and the highest corrugation height (H/D = 0.05) experienced the highest pressure drops in both the annular and inner tubes. Compared to a flat tube, the heat transfer increased by 29%. Similarly, Ali et al. [22] performed an experimental investigation to examine the impact of twisting the inner pipe on the thermal efficiency of a double-pipe heat exchanger. Different twisting numbers (3, 5, and 7 twists per unit length) were tested under turbulent flow conditions. The results showed enhanced performance for all three configurations. The highest augmentation for the Nusselt number was 2.2 for counterflow and 1.8 for parallel flow, compared to a flat pipe. The maximum heat enhancement was observed at Re = 26,000 for both flow directions.

Fadaei et al. [23] investigated the heat transfer enhancement in a shell and tube heat exchanger using Carreau-Yasuda non-Newtonian fluid. They analysed the heat transfer rate in both horizontal and vertical directions. The results indicated that higher porosity parameter values led to slower heat transfer rates, weakening the melting front of the phase change material. Huu-Quan et al. [24] utilized computational fluid dynamics (CFD) to study the turbulent flow in double-pipe heat exchangers. They specifically investigated the influence of inner pipe geometry on flow and heat transfer. The thermal characteristics were strongly dependent on the Reynolds number parameter. Flat inner pipes with a reduced aspect ratio improved heat efficiency and performance. These outcomes indicate that employing flat inner pipes featuring smaller aspect ratios enhances the performance of heat exchangers in low Reynolds number scenarios. In contrast, circular inner pipes remain more favourable for high Reynolds numbers.

Based on the existing literature and to the authors' best knowledge, limited experimental research has studied the performance of heat exchangers using various fin configurations. Consequently, this study introduces a new experimental setup for a double-pipe heat exchanger to examine the impacts of discontinuous extended surfaces on device performance. Three inner tube configurations were considered and compared: smooth tube, tube with discontinuous curved fins, and tube with discontinuous longitudinally straight fins.

2. Experimental Methodology

2.1. Experimental setup.

As shown in Fig. 1, the experimental apparatus consists of a test section (double pipe heat exchanger), an air supply system for the annular tube, a water supply system for the inner tube, measurement devices (air flow meter, water flow meter, and temperature recorder), and control valves. The outer tube of the test section is constructed from a 1.0 m long transparent Pyrex tube of 67 mm inner diameter with a wall thickness of 3mm. The inner tube is an internal copper tube of three different configurations, without fins (smooth tube) or with discontinued curved or longitudinal straight copper fins. The tube is 1 m long with 14 mm and 15 mm inner and outer diameters, respectively. Using the silver welding technique, pairs of discontinued copper fins are attached to the outer surface of the inner tube. Figure 2 summarizes the dimensions of the heat exchanger test section.

The discontinuous fins were distributed as a pair in horizontal and vertical positions alternately along the tube, as illustrated in Fig. 2(a). The longitudinal fins are 25 mm long, 10 mm high, and 1.0 mm thick, with a 100 mm distance between each pair. Fourteen fins are soldered on the tube to form seven pairs, as shown in Fig. 2(c). Similar dimensions and arrangements were considered for the discontinuous curved fins. Fourteen longitudinal sheets were bent to a half-circle shape and attached to the outer surface of another inner copper tube.

A 40 W centrifugal blower is used for feeding the test section with air. After inserting the rubber seal, the blower outlet is joined directly with the outer tube. The blower is used to supply air velocity up to 2.35 m/sec to the test section. Various velocities are obtained by changing the blower gate, and the blower was fixed on the mainframe by bolts and rubbers between the blower and frame to dampen the vibration under operation status. Vane–type probe portable (Intell instrument plus AR–826) anemometer is used to measure the average air velocity as it gives accurate readings with digital readability and a convenient remote sensor separately.

Fig. 1. The experimental facility (double piper heat exchanger).

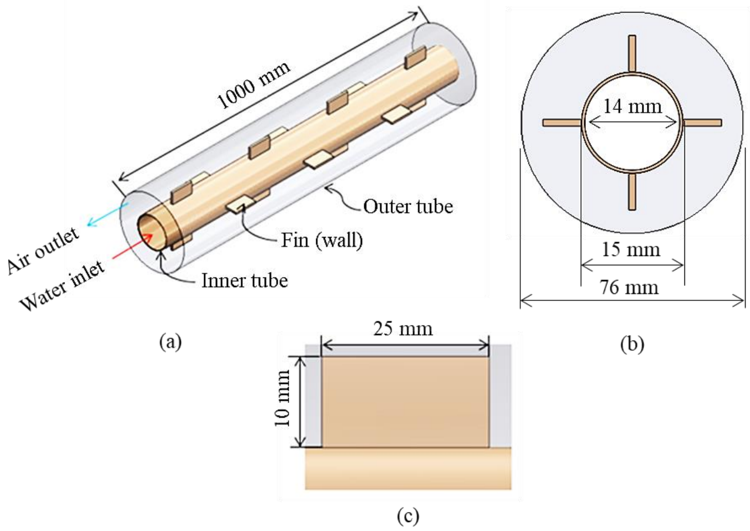

Fig. 2. (a) Schematic diagram of the test section,
(b) side view of the test section, (c) dimensions of a copper fin.

The water supply system consists of 100 Liters of a cold-water storage tank used to supply the electrical water heater with water. An electrical water heater of 60-liter capacity has been connected to the cycle for supplying hot water to the inner tube of the heat exchanger. A water pump (370 W) feeds water to the system with a maximum volumetric flow rate of 30 l/min and a maximum head of 30 m.

2.2. Measurements and control.

A system of measurement and control has been designed and implemented to control the operational conditions and record the needed parameter to estimate the performance of the heat exchanger. A water flow meter (LZS-15), which has a range of (10 to 100) Liter/hour, is used to measure the flow rate of the water. Three gate valves were used in the experimental test rig (waterside) to regulate the water flow rate (one at the main flow loop, one for the bypass line, and one between the tank and the water pump. Eight thermocouple wires (type K) of 0 to 100 ˚C range were mounted in the test section to measure the water and air temperature. Two thermocouples are placed at the inlet and outlet of the inner tube to measure the water temperature. Similarly, two thermocouples are placed inlet and outlet of the external tube to measure the air temperature. Finally, four thermocouples were attached to the outer surface of the inner tube to record the surface temperature at axial locations along the flow direction.

2.3. Experimental procedure.

A pre-identified strategy and operational conditions have been decided for each flowrate. The following procedure has been carried out during the experiments:

- Operate the water pump to push the water from the tank to the electrical water heater.
- Operate the electrical water heater until reaching the desired temperature of (50 ˚C) to equip the system with hot water.
- Operate an air blower to equip the test section with air.
- Adjusting the water flow rate by controlling the main and bypass valve.
- Recording the temperatures at the inlet and outlet of water and air on the two sides of the test section. Also, record the surface temperature.
- Repeat the above steps by adjusting the air mass flow rate at a fixed water flow rate.

The procedure was repeated for smooth and finned tubes.

3. Mathematical Analysis

During the test procedure, the cold fluid (air) flowing in the annular space absorbs heat from the hot water in the inner tube side. The heat transfer rate through the heat exchanger can be estimated as follows:

The heat dissipation of both sides:

$$Q_c = \dot{m}_c c_{p_c}(T_{co} - T_{ci}) \tag{1}$$

$$Q_h = \dot{m}_h c_{p_h}(T_{hi} - T_{ho}) \tag{2}$$

The product of mass flow rate and specific heat (heat capacity rate) is found from:

$$C_c = \dot{m}_c c_{p_c} and\ C_h = \dot{m}_h c_{p_h} \tag{3}$$

The heat dissipation in the heat exchanger can be expressed as:

$$Q = UA_s LMTD\ F \tag{4}$$

The correction factor equals 1, since counterflow arrangement was considered within the present experimental work. The properties of the test fluid during this study are calculated at the mean fluid temperature. The following equation is used to estimate the log mean temperature difference:

$$LMTD = \frac{\Delta T_{out} - \Delta T_{in}}{In(\Delta T_{out}/\Delta T_{in})} \tag{5}$$

where $\Delta T_{in} = T_s - T_{in}$, and $\Delta T_{out} = T_S - T_{out}$

The heat exchanger contains two flowing fluids separated by a solid wall. This mechanism is expressed by the overall heat transfer coefficient as follows:

$$\frac{1}{UA} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{ln(\frac{d_o}{d_i})}{2\pi k_t l} + \frac{1}{h_o A_o} \tag{6}$$

where $A_i = \pi d_i l$, $A_o = A_u + A_f$, $A_u = \left(\pi d_o l - N_f l\delta\right)$, and $A_f = \left(2H_f + \delta\right) N_f l$

In Equation (6), the overall heat transfer coefficients, U_i and U_o appear due to the difference between the two heat transfer areas, A_i and A_o. Owing to the high value of thermal conductivity of the inner tube, the thermal resistance, $ln(\frac{d_o}{d_i}) /2\pi k_t l$ could be neglected in Eq. (6).

Newton's law of cooling achieves the calculation of the inner side heat transfer coefficient (water):

$$Q_h = h_i A_i (T_m - T_s) \tag{7}$$

where $T_s = \frac{T_1 + T_2 + \cdots + T_n}{n}$

The air side heat transfer coefficient, h_o in annuli, could be obtained for smooth and finned tubes.

To calculate the annuli side heat transfer coefficient with a smooth outer surface of the inner tube, the Wärmeatlas equation [25] was used. In this equation, estimation for the Nusselt's number in annuli with fully developed in the turbulent flow regime as follows:

$$Nu = 0.86\left(\frac{d_0}{D_I}\right)^{-0.16} \frac{(£/_8)Re\ pr}{1 + 12.7\sqrt{£/_8}\left(pr^{\frac{2}{3}} - 1\right)} * [1 + (\frac{D_{hu}}{l})^{\frac{2}{3}} \tag{8}$$

where $£ = (1.8 \log \mathrm{Re} - 1.5)^{-2}$

Reynold's number is expressed as:

$$\mathrm{Re} = \frac{\rho_c \upsilon_c D_{hu}}{\mu_c} \tag{9}$$

where $D_{hu} = D_i - d_o$

Then cold side heat transfer coefficient can be estimated:

$$h_o = \frac{Nu k_c}{D_{hu}} \tag{10}$$

Annuli heat transfer coefficient with discontinued fins on the outer surface of the inner tube can be calculated as follows:

$$h_o = \frac{1}{\frac{1}{U_o} - \frac{A_O}{h_i A_i}} \tag{11}$$

Reynold's number could be computed after calculating the hydraulic diameter from the equation:

$$D_{hu} = \frac{4A_c}{P_w} \tag{12}$$

where $A_c = \frac{\pi}{4}(D_i^2 - d_o^2) - (\delta H_f N_f)$, and $P_w = \pi(D_i + d_o) + 2H_f N_f$

The annuli side Nusselt number can be estimated as:

$$Nu = \frac{h_o D_e}{k_c} \tag{13}$$

where $D_e = \frac{4A_c}{P_h}$, and $P_h = \pi d_o + 2H_f N_f$

4. Results and Discussion

This section compared the experimental Nusselt number values for the water and air side for the three different test section configurations. Figure 3 depicts the variation of the Nusselt number for the three cases at water flow rates ranging from *Re* = 260 to *Re* = 1900, while the air mass flow rate is fixed at 0.03 kg/s in the annular tube. From this figure, the waterside Nusselt's number increases by increasing Reynold's number for all cases, as expected. Additionally, it is observed that the experimental Nusselt number values are higher for the tube with discontinuous curved fins. The Nusselt number increased by 42.8% compared to the smooth tube.

Similarly, a 38.2% increase in the Nusselt number values was recorded after using discontinuous longitudinal fins on the inner copper tube of the facility. These values were measured at *Re* = 260, while the percentage increase is smaller for higher values of Reynold's number. At *Re* = 1900, a 19.3% and 24.2% increase in *Nu* was obtained for curved and longitudinal fins, respectively.

Figure 4 introduces a similar trend, as it demonstrates the Nusselt number values against the mass flow rate of air in the annular tube (0.03 kg/s to 0.13 kg/s) at constant water flow Reynold's number inside the copper tube of (*Re* = 1900). Similarly, the airside Nusselt number increases by increasing the airflow rate for all cases. The heat transfer coefficient increased by the turbulence induced by the increase in air velocity. The values of the Nusselt number for finned tubes are significantly greater than those for smooth tubes. This is caused by the increase in the surface area introduced by the fins.

Furthermore, the curved fins showed a relatively higher Nusselt number than the other two cases. The shape of the fins influences more turbulence in the air side of the test section, thus providing a higher heat transfer rate. The figure shows that up to a 42.3% increase in Nusselt number is possible using curved fins at a high air mass flow rate of 0.13 kg/s.

In addition, the effectiveness of the heat exchanger was calculated from the experimental data and compared in Fig. 5 for the three configurations. The facility effectiveness increases from 32.1% to 48% as the air mass flow rate increases from 0.03 to 0.13. A similar trend was observed by Sivalakshmi et al. [14] when implementing helical fins in a double-pipe heat exchanger. In their study, the effectiveness of the heat exchanger improved by 38.46% at a higher flow rate when compared to a smooth tube. Furthermore, in the current study maximum value of 52.8% was obtained when using curved fins on the copper tube. Accordingly, discontinuous curved fins enhance the facility performance better than longitudinal fins in heat transfer rate. However, the effect of the extended surface on the pressure drop in the facility was not monitored, and further experimental and numerical investigations are required to study the frictional effects in the experimental facility.

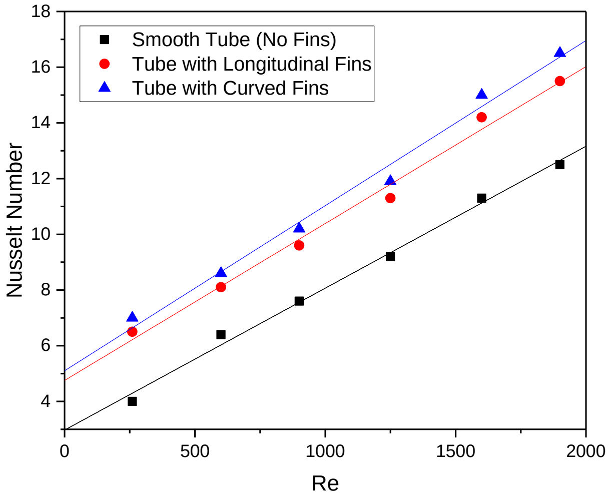

Fig. 3. Waterside Nusselt number vs water flow Reynold's number for smooth tube, tube with curved fins, and tube with longitudinal fins at air mass flow = 0.03 kg/s.

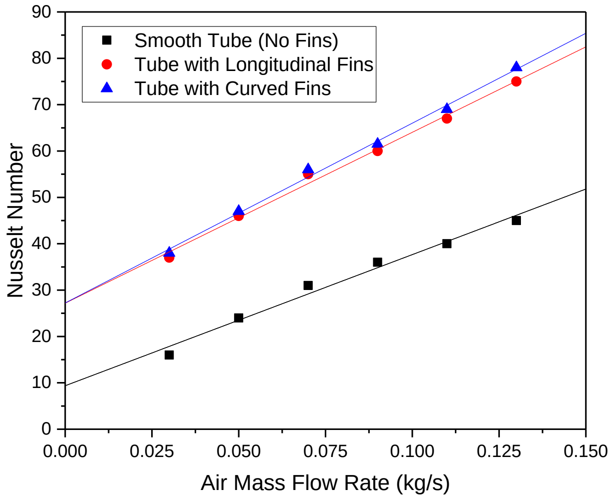

Fig. 4. Airside Nusselt number versus air mass flow rate for smooth tube, tube with curved fins, and tube with longitudinal fins at water flow Re = 1900.

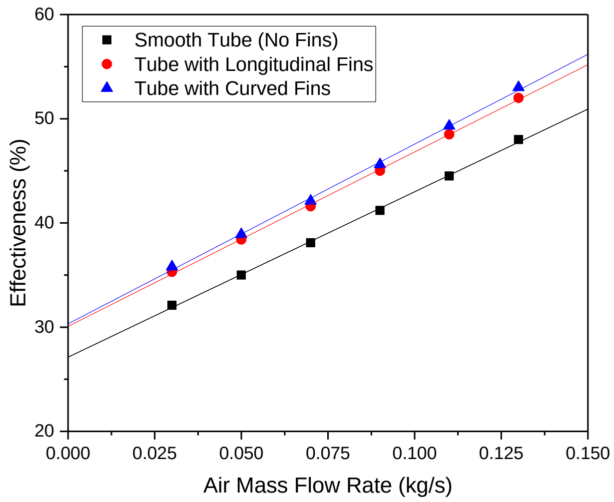

Fig. 5. Heat exchanger effectiveness versus air mass flow rate for smooth tube, tube with curved fins, and tube with longitudinal fins at water flow Re = 1900.

5. Conclusions

In this study, experimental investigations were carried out on the effects of curved and longitudinal fins on the performance of a double-pipe heat exchange. The outcomes from the study can be concluded as follows:

- Extended surfaces on the inner tube of double-pipe heat exchangers increase the flow Nusselt number and enhance the facility's heat transfer phenomenon.
- Using discontinuous longitudinal fins at high Reynold's number, the Nusselt number increased by 19.3% and 40% in the water and air side of the tube, respectively.
- In addition, using discontinuous curved fins, the Nusselt number further increases to 24.2% and 42.3% in the water and air side of the tube, respectively.
- Discontinuous curved fins have higher heat augmentation compared to longitudinal.
- The highest heat exchanger effectiveness of 52.8% was recorded for curved fins, while the effectiveness of 52% and 48% were observed for tubes with longitudinal fins and smooth tubes, respectively.

This study on the utilization of extended surfaces in double-pipe heat exchangers holds significant importance as it provides valuable insights into enhancing heat transfer efficiency. By examining the effects of different fin configurations, such as discontinuous longitudinal and curved fins, the study contributes to developing more efficient heat exchanger designs, ultimately enabling improved energy utilization and cost savings in various industrial applications. In addition, further investigations are required to analyses the pressure drop and friction factor within the experimental facility.

Nomenclatures

A_s	Surface area, m^2
A_c	Cross-section area of annuli, m^2
C	Heat capacity, W/ C$^\circ$
Cp	Specific heat, J/kg. C$^\circ$
D_e	Equivalent diameter of annuli, m
D_{hu}	Hydraulic diameter of annuli, m
d	Diameter of inner tube, m
F	Correction factor
h	Heat transfer coefficient, W/m^2.C$^\circ$
H	Height, m
k	Thermal conductivity, W/m. ^{o}C$^\circ$
l	Length, m
$\dot{m}$	Mass flow rate, kg/sec
N_f	Number of fins
Nu	Nusselt's number
P	Perimeter, m
Q	Heat dissipation, W
Re	Reynold's number
T	Temperature, C$^\circ$
T_{ci}	Inlet temperature of cold fluid, C$^\circ$
T_{co}	Outlet temperature of cold fluid, C$^\circ$
T_{hi}	Inlet temperature of hot fluid, C$^\circ$
T_{ho}	Outlet temperature of hot fluid, C$^\circ$
U	Overall heat transfer coefficient, W/m^2.C$^\circ$
Greek Symbols	
δ	Fin thickness, m
μ	Dynamic viscosity, kg/m . s
ρ	Density, kg/m^3

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