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Research Article

Thermal Performance of a Laminar Flow Double Pipe Heat Exchanger with Spiral Airfoil Inserts

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Abstract

Objective: This study evaluates the thermal performance of a double-pipe heat exchanger using a spiral airfoil insert in the inner pipe.

Methods: The enhancement technique is classified as a passive method, as it does not require the direct application of external power to improve the heat transfer coefficient. Spiral pitch ratios of P/D=7, 5, and 3 were used, with the Reynolds number (*Re*) varying within the laminar flow regime ($Re \leq 2{,}100$). Cold water flows inside the inner pipe, while hot water flows counter to it in the annulus.

Results: The experimental results show that the Nusselt number increased by 159%, 161%, and 171%, respectively, compared to a smooth pipe alone, for flow rates of 2L/min, 2.5L/min, and 3L/min. Additionally, the highest friction factor reached 142%.

Conclusion: The results indicate that the best performance evaluation criterion (PEC) was achieved within the flow range of 2L/min to 3L/min, with a pitch insertion of 3, yielding a PEC of 158% to 152%, respectively.

Keywords: thermodynamic analysis, heat exchanger, laminar flow, air foil, Reynolds number

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1 INTRODUCTION

The insertion enhancement in cold fluid side used in heat exchanger to improve. Insertion enhancement on the cold fluid side of a heat exchanger is used to improve the heat transfer coefficient, particularly in laminar flow conditions. This technique works by increasing fluid mixing near the surface and reducing boundary layer thickness, which in turn decreases thermal resistance. There are two main types of enhancement methods: passive and active. Passive methods do not require external energy and are the simplest to implement; they have been extensively studied and generally yield good performance results. Active methods, on the other hand, require external energy to achieve enhancement. Garcia et al.[1] conducted experimental investigations on helical wire inserts in round pipes across laminar, transitional, and turbulent flow regimes at various Prandtl numbers.

The experiments utilized water and water-propylene glycol mixtures at different temperatures. Six wire coil pitches (P/D) ranging from 1.17 to 2.68 and wire diameters (e/d) between 0.07 and 0.10 were tested, resulting in a 200% improvement in heat transfer for transitional flow compared to smooth pipes. Kumar R et al.[2] also performed experimental studies on heat transfer and pressure drop using wire coil inserts in horizontal pipes with engine oil flow. Their findings led to the development of two empirical correlations with an error margin of ±20%. Naphon P[3] investigated heat transfer and pressure drop in a dual pipe heat exchanger and demonstrated that coil wire inserts significantly enhanced heat transfer, particularly in laminar flow. Dhumal GS et al.[4] investigated the effects of twisted tape inserts and exterior helical tape on heat exchanger performance in a counterflow setup with water. The study revealed significant enhancements in heat transfer, with Nusselt numbers increasing by 219%-315% compared to a plain pipe. Friction factors were 4.4 to 8.7 times higher for twist ratios of 6.77, 4.51, and 3.38. An empirical relationship between the Nusselt number and friction factor was established, and the highest thermal performance factor achieved was 3.06. These results provide insights for designing energy-efficient heat exchangers. Zhan F et al.[5] conducted an experimental study to investigate particle deposition characteristics in wavy fin-and-tube heat exchangers. The experimental parameters included fin pitches ranging from 1.6mm to 3.2mm, particle concentrations from 80kg/m³ to 280kg/m³, and air velocities from 1m/s to 3m/s. The results indicated that particles predominantly deposited on the leading edges of the fins and the front sections of the tubes. Weight measurements showed that fin pitch and particle concentration had a monotonic effect on particle deposition. Specifically, the maximum particle deposition per unit area increased by 13.1% as fin pitch decreased and by 6.2% as particle concentration increased. However, the effect of air velocity was more complex; particle deposition initially increased by 6.8% and then decreased by 10.9% as air velocity increased.

Duan F et al.[6] performed a numerical study on the flow and heat transfer characteristics of the fin side of a wavy finned flat tube heat exchanger, comparing their findings with experimental results. They explored the influence of fin spacing, wave spacing, wave amplitude, and Renolds number (*Re*) on flow and heat transfer performance. Correlations for the Nusselt number and friction factor were derived from numerical data. The study revealed that intermittent wavy fins significantly increased heat transfer compared to straight fins. Longitudinal vortices enhanced heat transfer, whereas transverse vortices had a negative effect. Interestingly, the intensities of longitudinal and transverse vortices did not directly determine the Nusselt number across all configurations.

Kim GW et al.[7] conducted a numerical study on heat transfer enhancement using cross-cut flow control in wavy fin heat exchangers. The cross-cut concept involved cutting fins perpendicular to the flow direction. The simulation was performed using non-dimensional governing equations for steady laminar flow, with a parametric study conducted to identify the optimal position and length of the cross-cut. The results demonstrated that the optimized cross-cut wavy fin enhanced heat transfer by up to 23.81% compared to a standard wavy fin. However, the pressure drop also increased by up to 7.04% in the optimized configuration. Vahidifar S and Kahrom M[8] investigated the heat transfer characteristics and pressure drop of a horizontal double pipe heat exchanger equipped with wire coil inserts. Their study examined how wire coils and rings affect heat transfer and pressure drop in the exchanger. Wire coils generate swirl flow, which enhances turbulence and roughness, while rings further promote turbulence and roughness. The experimental data were obtained using wire coils and rings with geometrical variations, including pitches (P/D) of 1, 2, and 4 and wire diameters (d/D) of 0.05, 0.07, and 0.11. For a wire coil with d/D=0.11, P/D=1, and a *Re* of 10,000, the overall enhancement efficiency reached 128%. Several other studies have employed nanofluids to achieve further improvements in heat transfer[9-16]. The aim of this study is to enhance heat transfer for laminar flow of water, reduce the size of the heat exchanger, and achieve a high performance evaluation criteria for a counterflow double - pipe laminar flow heat exchanger.

2 TEST SETUP

The test section consists of a 1 - meter length double - pipe heat exchanger. The inner copper pipe has an inside diameter of 32.5mm and an outside diameter of 35mm. The annulus is a steel pipe with an inside diameter of 76.2mm and is insulated with fiberglass on the outside diameter to minimize heat loss. An electric water heater supplies hot water at an average temperature of 68°C, which flows through the annulus with a manually controlled flow rate. Cold water, at an average temperature of 18°C, flows through the inner pipe in a counterflow arrangement. The length of the inlet cold water pipe is 10 times its diameter to ensure fully developed flow at the heat exchanger inlet. The cold water flow rate is controlled by a hand valve. A float ball flow meter is used to measure the flow rates of both hot and cold water. Temperature data are recorded by sensors inserted at the inlets and outlets of both hot and cold water using a data logger. Two water pumps are used to circulate the water through the heat exchanger. Figure 1 shows the test section.

3 EXPERIMENT LAYOUT

The hot and cold water pumps are operated once the hot water reaches the desired temperature. The flow rates of the hot and cold water are manually adjusted to be equal. The flow rate varies within the laminar flow regime ($Re \leq 2{,}100$), with flow rates set at 2L/min, 2.5L/min, and 3L/min for both hot and cold water. After reaching a stable condition,

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Table 1. List of Apparatuses

No.	Equipment's	No. Devices	Specification
1	Flow meter	2	Rang (2-18) l/min ACCURACY ± 1%
2	Data logger	1	DLH 9,097, Supports, vista and Win7 SENSORE: Temperature range: -58 to 257F Cable length: (115cm). Resolution rate :0.0625°C; Accuracy rate : ±0.5°C
3	Electric Water pump	2	0.5HP, 40L/min, 220V, 50Hz
4	Electric water heater	1	3kW

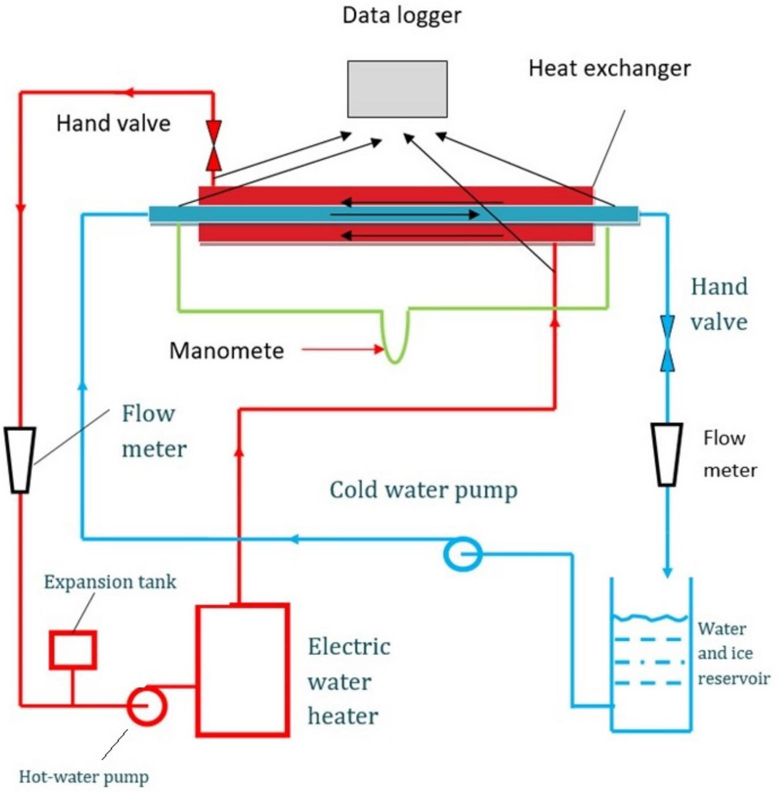

Figure 1. Flow diagram.

the data logger begins recording temperature data, which is then graphed. This procedure is repeated for each case using different spiral airfoil pitch ratios inserted into the cold water pipe, as shown in Figure 2. The spiral airfoil pitch ratios tested are P/D=7, 5, and 3. The performance of these cases is compared with that of a plane pipe heat exchanger in terms of both heat transfer and friction factor.

3.1 Data Reduction

The data obtained from the measurements are substituted into the following formulas to evaluate the thermal performance and friction factor[14].

$$Q_h = \dot{m}\, C_{ph}(T_{hi} - T_{ho}) \ (1)$$
$$Q_C = \dot{m}\ C_{pC}(T_{co} - T_{ci}) \ (2)$$

In which:
Q_h: Heat added joules (J)
$\dot{m}$: Mass flow rate (kg/s)
C_{ph}: Heat capacity at constant pressure (J/(kg\cdotpK)
T_{hi}: Hot water inlet temperature (Ċ)
T_{ho}: Hot water exit temperature(Ċ)
Q_c: Heat rejected joules (J)
C_{pC}: Specific heat capacity of the cold stream at constant pressure (J/(kg\cdotpK)
T_{co}: Cold water exit temperature (Ċ)
T_{ci}: Cold water inlet temperature (Ċ)

Where: $C_h = \dot{m}C_{ph}$, $C_c = \dot{m}C_{pc}$

$$\mathrm{LMTD} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln(T_{hi} - T_{co}) - (T_{ho} - T_{ci})} \quad (3)$$

$$\mathrm{UA} = \frac{Q}{\mathrm{LMTD}} \quad (4)$$

In which:
U: overall heat transfer coefficient
A: area
Q: rate of heat transfer
LMTD: logarithmic mean temperature difference

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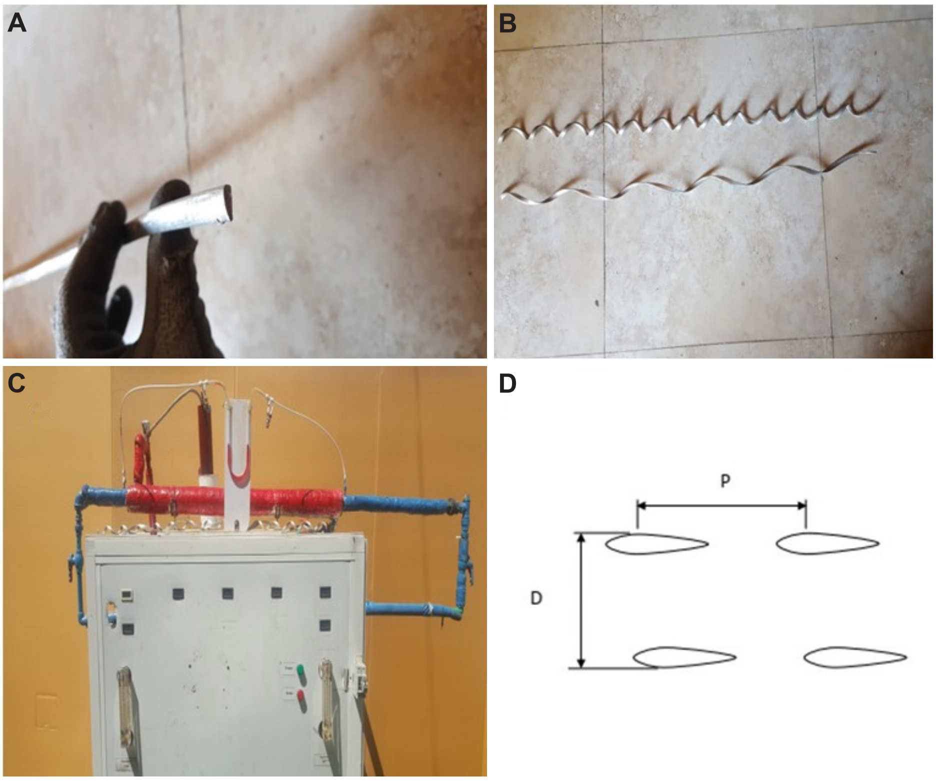

Figure 2. Overview of Airfoil and Experimental Setup: (A) Cross-section of the airfoil, (B) Spiral airfoil wire, (C) Experimental device setup, (D) Pitch ratio configuration.

$$\frac{1}{U_i} = \frac{d_i}{d_o h_o} + \frac{d_i \ln \frac{d_o}{d_i}}{2k} + \frac{1}{h_i} \quad (5)$$

$$NTU = \frac{UA}{C_{min}} \quad (6)$$

$$h_o = \frac{N_u k}{D_h} \quad (7)$$

In which:

N_u: Nusselt number

D_h: Hydraulic diameter (m)

For laminar flow in annulus N_u indicates baseline of Nusselt number and it can be calculated from Shah and London[9] correlation:

$$N_u = 6.2066(1 + 2.3108r^* - 7.7553r^{*2} + 13.2851r^{*3} - 10.5987r^{*4} + 2.6178r^{*5} + 0.4680r^{*6})$$

In which r^* (annulus radius ratio) is a dimensionless parameter that compares the inner and outer radii in annular flow systems.

$$r^* = \frac{r_i}{r_o} \; (0.02 \le r^* \le 1)$$

D_h represent hydraulic diameter and can be found as:

$$D_h = 2(r_o - r_i)$$

$$f = \frac{\Delta p D}{2L\rho V^2} \quad (8)$$

calculates the friction factor (f) in fluid mechanics, which characterizes the resistance to flow in pipes. It relates the pressure drop (ΔP) across the pipe, the pipe diameter (D), and the length of the pipe (L), while also considering the fluid density (ρ) and velocity (V), can be written as below:

$$\Delta P = \rho g h$$

In which:

h: Heat transfer coefficint (W/m^2, K)

The efficiency (ϵ) of a thermal system, can be calculated in Equation (9):

$$\varepsilon = \frac{Q_c}{Q_{max}} \quad (9)$$

Here, Q_c represents the actual heat transfer or energy exchanged in the system, while Q_{max} denotes the maximum possible heat transfer that could occur under ideal conditions.

$$Q_{max} = C_{min}(T_{hi} - T_{ci})$$

Performance evaluation criterion (PEC) formula is Equation (10)

$$PEC = \frac{\frac{N_{ue}}{N_u}}{\sqrt[3]{\frac{fe}{f}}} \quad (10)$$

In the equation, N_{ue} represents the enhanced Nusselt number, which measures the convective heat transfer after applying enhancements such as turbulators, fins, or modified fluids. This compares to N_u, the baseline Nusselt number, indicating the heat transfer in an unmodified system. In terms of friction, fe refers to the enhanced friction factor, accounting for the increased flow resistance

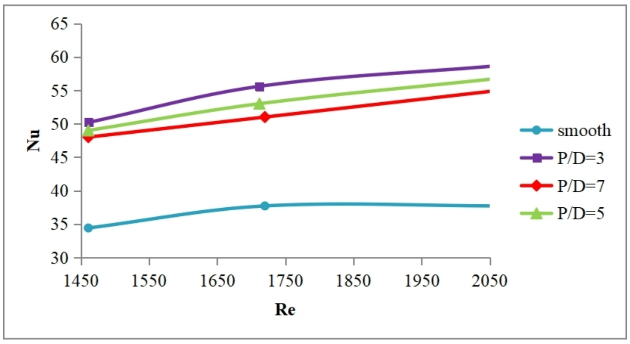

Figure 3. Relation between N_u - Re.

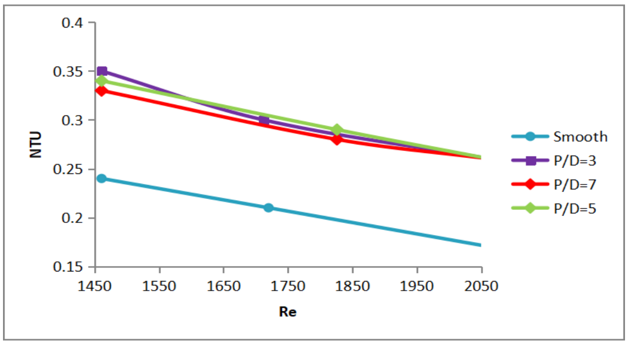

Figure 4. Relation between NTU - Re.

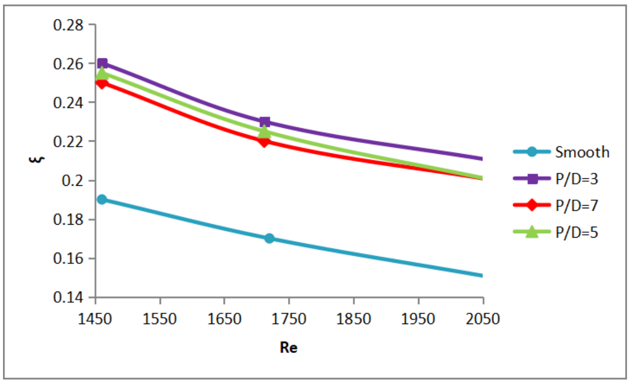

Figure 5. Relation between effectiveness - Re.

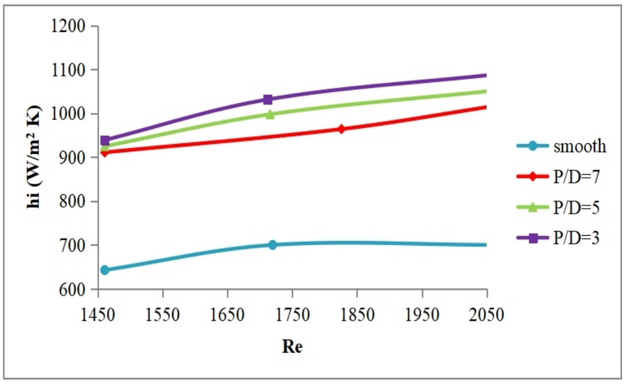

Figure 6. Relation between heat transfer coefficient - Re.

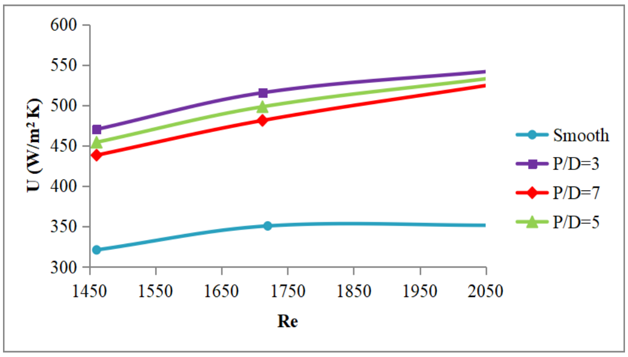

Figure 7. Relation between Over all heat transfer coefficient - Re.

due to the applied modifications, while f is the baseline friction factor, representing the original resistance in the unaltered system. In this formulation, the ratio N_{ue}/N_u shows the level of improvement in heat transfer, whereas the ratio fe/f, with its impact reduced by the cube root, reflects the penalty of increased friction.

4 RESULTS AND DISCUSSION

Heat transfer data were collected for a smooth pipe and for each case using insertion coils with varying pitch ratios. The relationship between the Nusselt number and flow rate is shown in Figure 3. For the smooth pipe, the Nusselt number increased asymptotically with increasing flow rate due to the thermally developing laminar flow. The Nusselt number was further enhanced by the use of inserted coils, as they prevent the development of the thermal boundary layer, reduce resistance by enhancing the convective heat transfer coefficient, and promote fluid mixing. Among the tested configurations, the coil with a pitch ratio of 3 exhibited the highest Nusselt number across all three flow rates.

The number of transfer units (NTU) decreases with increasing flow rate, as shown in Figure 4. Although the increase in flow rate enhances both the overall heat transfer coefficient and the heat capacity rate, the net effect results in a reduction in NTU.

Effectiveness decreases with increasing flow, as shown in Figure 5. At low NTU values, the heat exchanger effectiveness is also low. The behavior of heat exchanger effectiveness varies nonlinearly and asymptotically with the heat transfer area or NTU.

The convection heat transfer coefficient (*h*) is clearly enhanced with flow and insertion modifications, as demonstrated in Figure 6. This improvement is attributed to the prevention of boundary layer development and the maintenance of low resistance, which aligns with findings reported in it[17-20].

Overall heat transfer coefficient (*U*) clearly enhanced with Flow and insertion enhancement as shown in Figure 7. The reason is the preventing of developed boundary layer and keep resistance low as possible.

The behavior of convection heat transfer coefficient (*h*) and over all heat transfer coefficient (*U*) is directly affect heat transfer as in the same manner as shown in Figure 8. the enhancement effect of coils insertion trying to maintain

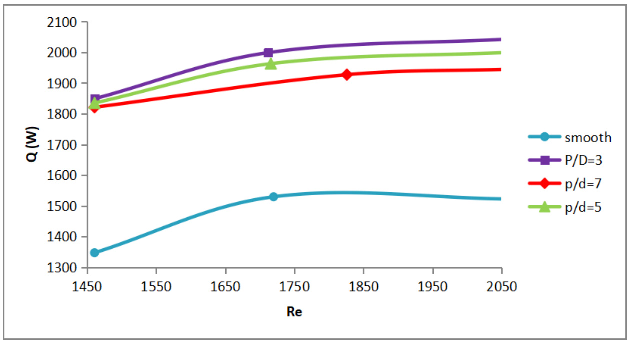

Figure 8. Relation between Heat - Re.

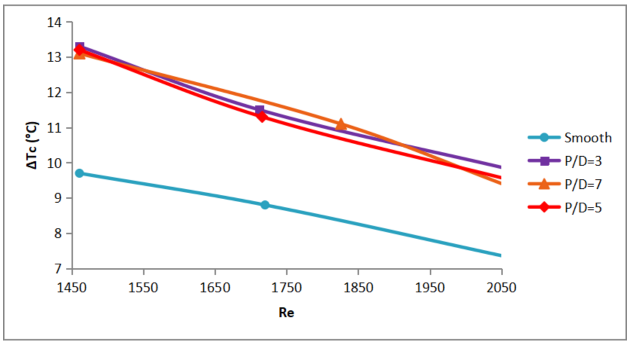

Figure 9. Relation between temp. diff. - Re.

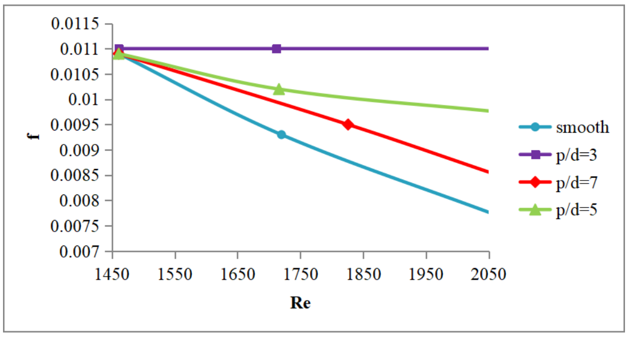

Figure 10. Relation between Friction factor - Re.

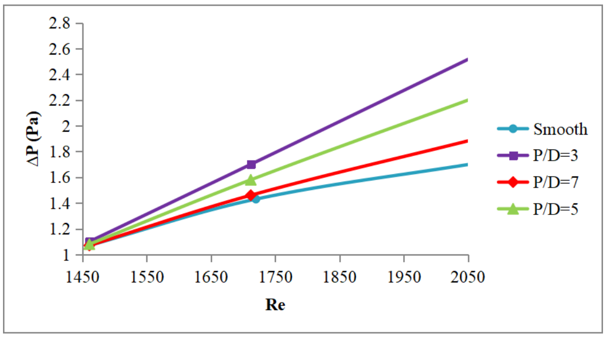

Figure 11. Relation between pressure drop - Re.a

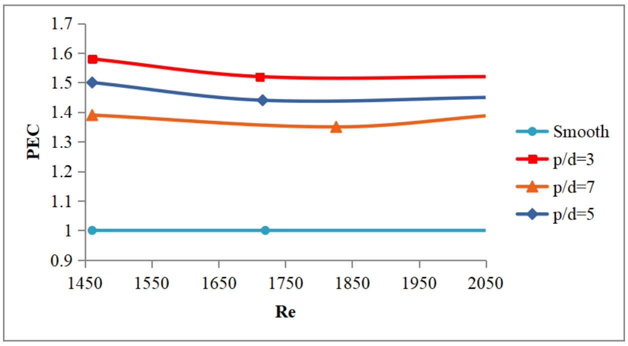

Figure 12. Relation between PEC - Re.

the Flow in the form of thermally developing to maintain the heat Flow as high as possible.

Cold water temperature difference reduced with increasing Flow rate because heat capacity rate (*C*) increased as the Flow rate increased and little time a valuable for water to absorb heat through the length of heat exchanger, as shown in Figure 9.

The Fanning friction factor increased with the use of the insertion enhancement and decreased with flow, except for the coil pitch of 3, as shown in Figure 10. The Fanning friction factor is strictly based on the true wall shear stress and represents the skin friction that is related to convective heat transfer. In general, according to Reynolds' analogy, skin friction and heat transfer are interconnected, meaning that an increase in heat transfer is accompanied by an increase in the skin friction factor for an enhanced heat transfer surface. For the coil pitch of 3, the enhanced heat transfer led to an increase in the friction factor. The increase in friction factor due to heat transfer equaled the decrease in friction factor due to flow, resulting in no net variation in the friction factor with changes in flow.

Pressure drop slightly increased with larger pitch ratio due to more resistance against Flow as shown in Figure 11. The turbulent effect caused by coil insertion increased the pressure drop with Flow.

Performance of evaluation criteria larger than one for all three coil insertion pitch, especially for coil pitch (3), that mean a good enhancement obtain for the amount of heat transfer as compared with increasing in pumping power, the details shown in Figure 12.

5 CONCLUSION

The experimental results show that the highest heat transfer enhancement, reaching 134%, was achieved with an airfoil insertion pitch of 3, with a corresponding friction factor increase of 142%. The PEC reached 152% at a flow rate of 3L/min, compared to other pitch ratios and the smooth pipe. At a flow rate of 2L/min, the heat transfer enhancement reached 137%, with nearly no increase in friction factor, and the PEC was 158% for the same pitch ratio (3), as compared to other pitch ratios and the smooth pipe.

Acknowledgements
Not applicable.

Conflicts of Interest
The authors declared no conflict of interest.

Author Contribution

In the article, the first author Hussein Hayder Mohammed Ali made significant contributions by designing the structure of the article and developing the scientific plan for the experiments. This includes organizing the overall flow of the research and ensuring that the experiments were carefully planned and aligned with the study's objectives. Khaled Yahya Shimer was responsible for the writing and conducting the final edition of the article. He played a key role in refining the text, improving clarity, and ensuring the manuscript met publication standards. Together, they ensured the research was presented effectively, combining strong experimental planning with clear and polished writing.

Abbreviation List

NTU: Number of transfer units
PEC, performance evaluation criterion
Re: Reynolds number

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