

Optimization of Turbulent Heat Transfer in a Heat Exchanger with Helical Corrugated Tube

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ABSTRACT

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The increase in convective heat shift has been studied in a one-stage non-stable run with the aid of spirally ribbed pipe. Results of the slope-to-breadth proportion $P/DH = "0.27, 0.18, \text{ and } 0.22"$ and "costal-extent to breadth proportion" $e/DH = "0.04, 0.06, \text{ and } 0.02"$ of spirally fluted ducts on the improvement of hotness exchange, thermal presentation portion, and isothermal friction are found and concluded in a coaxial duct hotness evaporator. Large volumes of unstable liquid flow with Reynolds numbers varies from 5500 to 60,000. Compared to corrugated tubes, smooth tubes produce less heat and transfer less heat. Heat conversion rate jumps from 123% to 232% in the test region. As a result, because of the restraint dropping production, the value of the friction factor happens 1.46–1.93 times on a flat tube in curved tubes. Additionally, probing data on the premises of curve fitting of equations are applied to find the "Nusselt number, abrasion part", and thermal appearance section for the fluted duct in terms of slope ratio (P/DH), costal-extent fraction (e/DH), "Reynolds number (Re), and Prandtl number (Pr)".

Keywords: Heat Transfer, Optimization, Heat Exchanger, Nusselt Number, Helical corrugated tub.

INTRODUCTION

In order to raise the hydraulic production of heat exchangers, multiple tactics, such as energy transfer modification approaches, is produced by lowering the heat exchanger's size and operating costs [1]. To put it another way, the HTE may be divided into two categories: the submissive process, which does not need outside potential conception, and the operative process, which does [2]. In the subservient procedure, temperature exchange profitability is improved by reducing the thermal limit layer thickness to increase turbulence in the tube wall layer and by locating an additional fluid mix with enhanced changed tubes, such as the tub with rib, the finned tube, the tube with uneven wall, the fluted tube, the amoral tube, the helical tube, the elliptical axis, etc., shown in Fig. 1. Due to the obvious suitability of the constraint fall accumulation, the hotness exchange improvement in an unstable one-stage run is much aided by the changed ducts, fluted duct, and duct. [3] finds the immerse process in a grooved duct then by fitting and twisting put inside by visible examination and also produced the exploratory relationship for immerse in both perpendicular and almost parallel ducts. A unique method for calculating the hotness shift in stratified mist liquefaction on an upright grooved tube was described by Dengliang *et al.* [4]. The phenomenon of hotness exchange in a "carbon-steel/copper curving grooved duct" was investigated by Wanga *et al.* [5]. Following analysis, it was found that the carbon steel spirally grooved duct provided a higher heat exchange calculated than the carbon steel simple duct by around 17%, while the energy exchange calculated by the copper ordinary duct is less than that of the curved hollowed channel, which is approximately 50%. In several important technical applications, including as heat exchangers, pipeline construction, the manufacturing of heavy metals, the marine, and the chemical sector, corrugated tube is used.

The behavior of convective heat transfer and thermal efficiency in corrugated tubes at different pitch ratios were described by Rainieri and Pagliarini [6]. a summary of their findings, the curved loop considerably stimulated the eddy parts. According to Vicente et al. [7], is thermal friction. The fluted ducts' abrasion portion measure is discovered to be around 250% to 300% more than that of the plane duct in this technique of corrugated tube appearance [8]. The friction factor of plain duct is 2.45 times lower than that of corrugated tube. In their study, Rozzi *et al.* [9] examined how "heat transfer and pressure drop" in a duct heat exchanger with Newtonian and non-Newtonian liquids as the functional liquids occur as a result of wall corrugation in a shell. Vicente together with others [10] Provides an explanation based on inspection that heat transfer takes place in fluted ducts for both "turbulent and laminar" flow, with ethylene glycol and water serving as the functional liquids. Following analysis, it is discovered that the fluted duct has a larger friction factor and heat transmission than the plain duct, ranging from 25% to 30%. The usage of water and oil as the operating fluid causes the hotness exchange and abrasion part characteristics in a spirally fluted duct [11]. Their results are finalized. Following analysis, it was observed that the fluted duct's heat transfer coefficient and friction factor were, in particular, greater than those of the flat duct, ranging from 120% to 160%. Using R-134a as the working liquid, give an investigative task for determining the fluted ducts' extreme warmth interchange determine and limitation decrease [12]. Following analysis, the loop slope provides a significant finding about the constraint fall conducts and the hotness exchange measure. The simple calculation method for a spiraling fluted duct, the heat exchanger and erosion device combine with a deviated stripes explained by Zimparov [13, 14]. The results show that the expected and investigative facts are almost identical.

More hotness exchange measures in hotness evaporator structures have been achieved recently by the use of different augmenting tubes, such as oval centre line ducts, finned ducts, microfin ducts, and fluted ducts, as the non-stable booster for upsetting the thermic surface. The term "conductive hotness exchange" refers to the optional oval centre line duct described by Meng et al. [15]. Multiple longitudinal vortices affected the relevance for heat transfer growth studies in the stagger oval center line ducts. [16] Noted that "heat exchange and pressure fall" characteristics occur in a spirally confused hotness evaporator and joined with a petal-shaped fluted duct for oil cooling. Al-Fahed et al. [17] describe how a slack-stable/dense-stable perverted-strip inset causes the hotness shift measure and constraint decrease in a micro finned tube. Following observation, it is discovered that tight-fit tape has a greater thermal performance value than loose-fit tape.

Friction factor characteristics in tubes with an interior spiral-crack were pointed out by Webb et al. [18]. The effects of the costal extent (0.33 to 0.55 mm), spiral gradient (25 to 45°), and crack start (18 to 45) on the hotness exchange and friction factor was also discussed and shown. The main goal of the current work is to determine the impact of changing the tube's surface on "heat transfer enhancement". Helically twisted tubes are often used to do this [19, 20]. Additionally, displayed are the true relationships between the abrasion element, thermic presentation portion (η), and the Nusselt number (Nu).

EXPERIMENTAL FACILITY

Fig. 2 describes the present work that uses spirally corrugated tubes. The spirally fluted duct's interior and exterior sides are depicted in the diagram. Helically corrugated tubes with pneumatic widths of 25, 25.5, and 24.5 mm, length of 1500 mm, and thickness of 2.5 mm were created using the copper ducts. Nine scroll fluted ducts in a uniaxial duct hotness evaporator are employed for testing. There are three different spirally fluted angles of $P = 5.5, 4.5, \text{ and } 6.5$ mm and three different scroll fluted costal-extents of $e = "1.5, 1.0, \text{ and } 0.5"$ mm ("costal-extent to width proportion", $e/DH = "0.06, 0.02, \text{ and } 0.04"$).

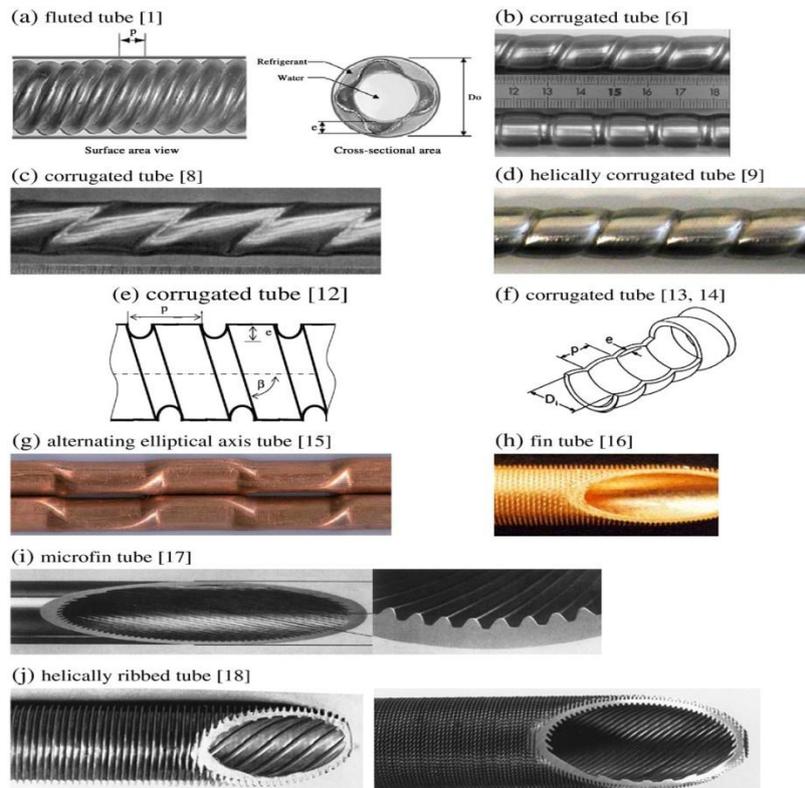


Fig. 1. Surface design of various tubes

Fig. 3 displays a schematic representation for the coordinated duct heat exchanger that is being provided for exploration. The two circular ducts in the coordinated duct hotness evaporator are the cold-water flow that is utilized in the outside tube and the interior duct (also known as a plane duct or “spirally fluted duct”) for the heated water flow. Serially, the simple tube's inner and outer radiuses were 13 mm and 14.3 mm. The exterior duct was constructed out of “stainless steel”, and the inner tube structure was composed of copper [21-23].

The exterior dimensions of the outer duct were appropriately enclosed to delay heat reduction to the closer item. The test involves a continuous run for the chilled water that is employed as the cooler's cooling agent. A cooling pillar rotated cooled water at a temp. of 28°C to control and keep the water's constant temperature in the storage tank. An inverted U-tube manometer was used for the diffusive abrasion testing, taking into account pressure decreases throughout the test portion. Two calibrated rotameters provided the experimental water's titrate run rate.

Since the water tank was filled with water for the purpose of examination, the water was heated and then released into a boiler when the “temperature goes to 70 °C”. The power tap was opened to let warm water flow in before to the “internal duct of a coordinated duct” heat evaporator using a rotameter. With the aid of 2balance pressure gauges mounted on the duct's peak barrier, the temperatures of the exit and the hot waters were noted in steady state conditions, and the pressure drop was recorded. The entry and exit water temperatures were measured using the RTDs approach, which is situated behind the warm water outflow. Measured from 5500 to 60,000, the Reynolds number span was applied to the duct tube diameter



Fig. 2. An illustration of a smooth, helically-corrugated pipe with an indication of morphology

RESULTS AND DISCUSSION

Data acquisition

The internal bore of the experiment duct are also depend on the average abrasion part and Nusselt number. Heat taken by water is-

$$Q_c = \dot{m}_c C_{pw} T_{c,out} - T_{c,in}$$

$$Q_h = \dot{m}_h C_p; w T_{h; out} - T_{h; in}$$

$$Q_{aver} = \frac{Q_c + Q_h}{2}$$

$$Q_{ave} = UA_i \Delta T_{LMTD}$$

where

$$A_i = \pi DL:$$

$$\frac{1}{U} = \frac{1}{h_i} + \frac{A_i \ln \delta D_o - D_i}{2\pi k L} + \frac{1}{A_o h_o} + R$$

$$\frac{1}{U} = \frac{1}{h_i} + B$$

$$h_i = C Re^m:$$

$$\frac{1}{U} = \frac{1}{C Re^m} + B = A Re^{-m} + B:$$

$$h_i = \frac{1}{\delta} = U - B P:$$

$$Nu = h_i D = k:$$

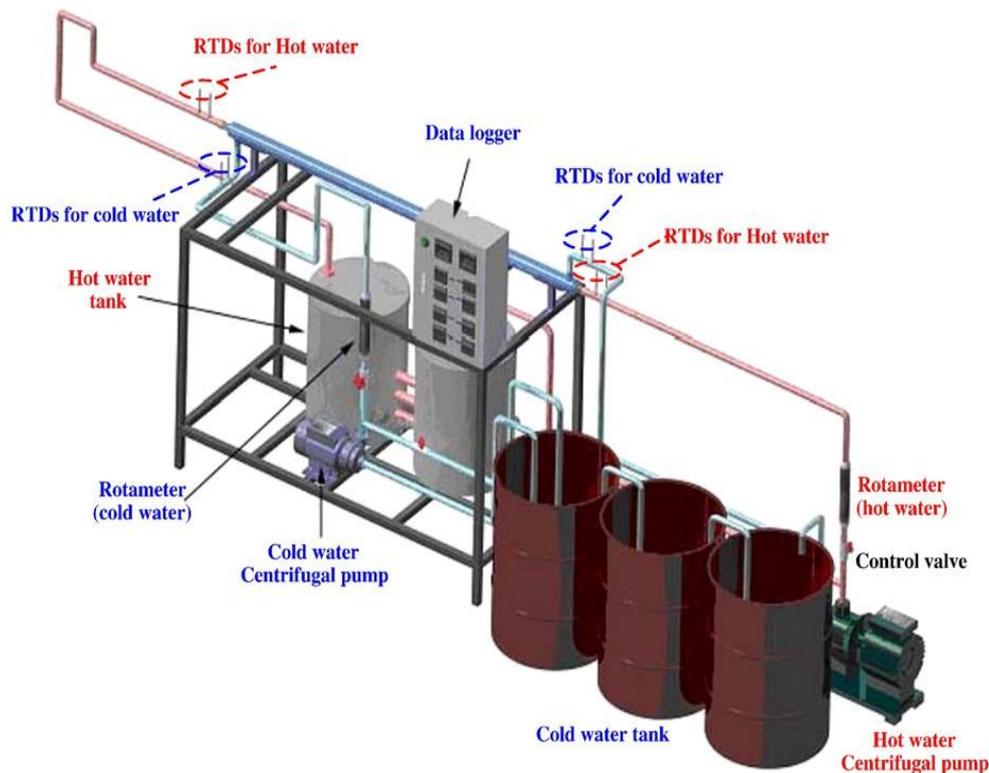


Fig. 3. Simplified figure of exploratory structure

$$Re = \frac{\rho V D}{\mu}$$

$$f = \frac{\Delta P}{L} = \frac{D P}{\rho V^2} = 2$$

$$\eta = \frac{\delta Nu_a}{Nu_s P} = \delta f_a = f_s^{-3}$$

Experimental results

Effects on the “friction factor and the hotness exchange” are displayed in the part for the helically corrugated tube at varied “slope proportions” ($P/DH = “0.22, 0.18, \text{ and } 0.22”$) and “costal-extent to breadth proportions” ($e/DH =$

“0.06, 0.04, and 0.02”). First, the value derived from the quality relations was confirmed by measuring the heat transfer and friction factor using the current plane duct in conjunction with an exploratory potential. The impacts of the slope ratio (P/DH) and “costal-extent proportion” (e/DH) on the hotness evaporators’ “abrasion, thermic presentation”, and “hotness exchange” characteristics are then displayed. Finally, the relationships between the abrasion element, “Nusselt number, and thermal performance” are investigated.

Evaluation of the smooth pipe

The present observed information for the hotness exchange rate and pressure drop in terms of the Nu and friction factor of the plain tube are serially confirmed using Petukhov Dittus - Boelter equations [19], as represents in Fig. 4(a) & 4(b). Lastly, the friction factor has ±8% and the Nu, which has general equations, occurs in ±4% variances to fulfil the current smooth tube consequence.

Result of slope proportion

For corrugated tubes, Fig. 5(a–b) explains how the Nusselt number and Reynolds number are connected through the flat duct and many slope ratios, $P/DH = 0.22, 0.27,$ and 0.18 . Smooth tubes often exhibit lower Nusselt numbers than corrugated ducts for all Reynolds numbers between 113 and 300%, as indicated by the slope angles (P/DH) rate that is measured. The Nusselt number increases abruptly to about 230% above the flat valve in the case of the largest pitch “proportion, $P/DH = 0.27$ ”, which occurs in twisted tube. This is more than $P/DH = 0.22$ and 0.18 , which occur around approximately 5% and 12%, respectively. It has been noted that when the pitch ratio value rises, so does the Nusselt number. “Larger contact sides and corrugated surface components” mix among the “fluid in the tube wall and the core” to produce this result; this is all due to a significant effective turbulence/recirculation flow. Because the surface components are spirally corrugated, the heat exchange value is more effective at lower Reynolds numbers. The main idea is that the thermic limit surface's breadth matters; as the Reynolds number decreases, the surface becomes broader, which leads to a more significant disarray of the thermic limit surface. In particular, as shown in Fig. 6(a), the energy reduction that occurs in the friction factor reaches a large value at lower Reynolds numbers.

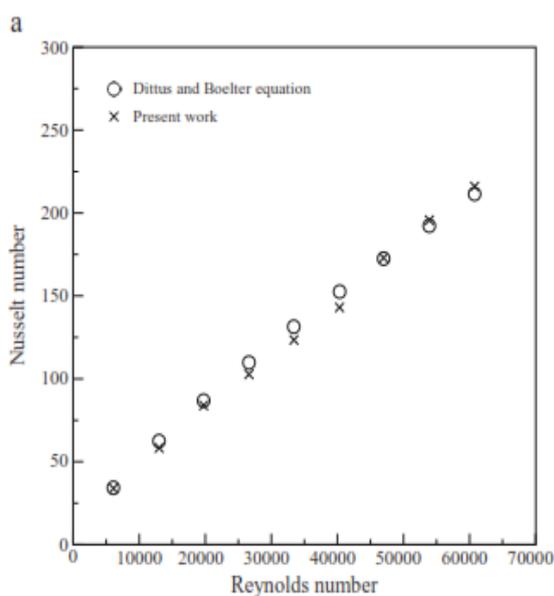


Fig. 4(a)

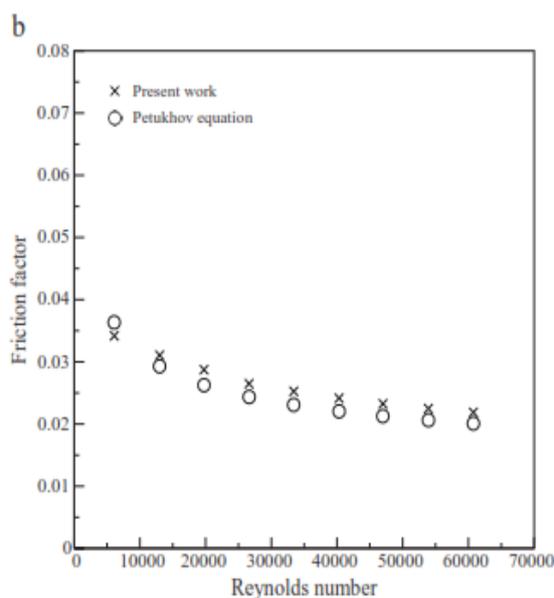


Fig. 4(b)

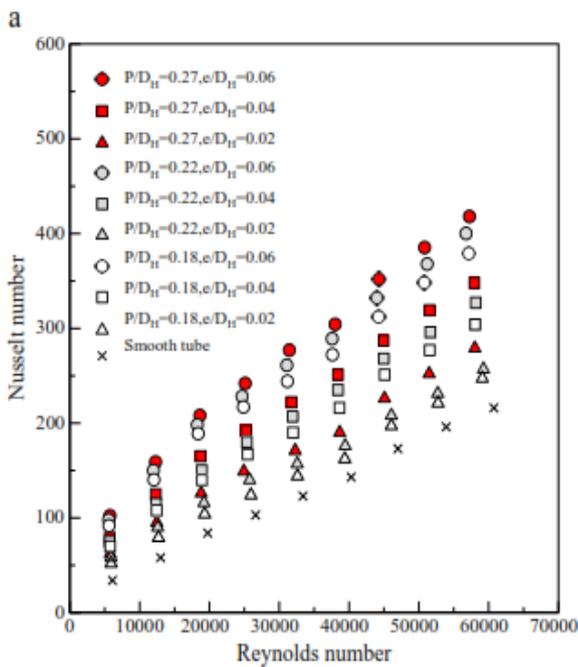


Fig. 5(a)

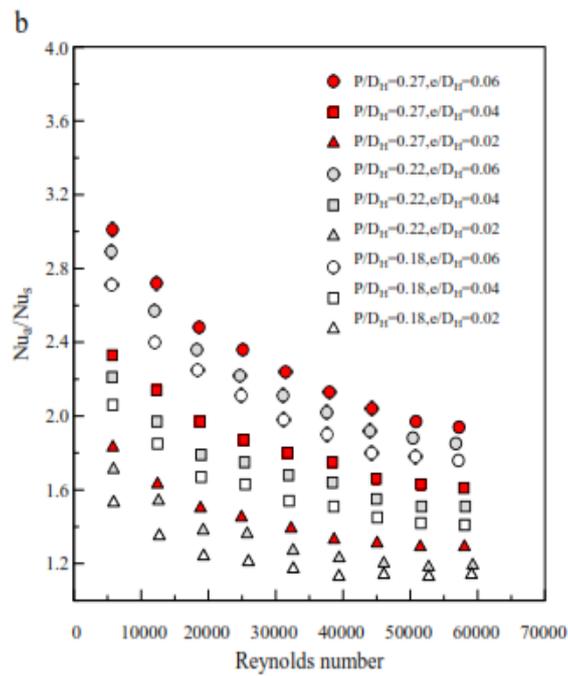


Fig. 5(b)

Fig. 6(a) & 6(b) shows the effect of “pitch proportion through fluted duct” on diffusive abrasion part characteristics at different Reynolds numbers. As the Re rises in the corrugated duct, the friction factor, also known as the pressure fall, decreases. In particular, smooth tubes have lower friction coefficients and Reynolds numbers than any corrugated tubes. Turbulence flows and recalculating flows are caused by the fluid's pressure loss resulting from a significant viscosity drop that occurs alongside the “duct wall from spirally fluted layers”. $P/DH = 0.18, 0.22,$ and $0.27,$ the mean abrasion part of the fluted ducts, are substituted by “1.93, 1.8, and 1.87” times on the plane duct, and $P/DH = 0.27,$ which has the largest slope proportion, results in an average bigger abrasion part than “ $P/DH = 0.18$ and 0.22 ”, which are around 3% and 7%, respectively.

Fig. 7 displays the thermodynamic presentation component (η) of the fluted duct with many pitch proportions. This figure displays two inspections that are visible: First, the presentation component increases as the pitch proportion increases, and second, the description part decreases as the Reynolds number increases. It is recommended that fluted ducts be used for saving power at lower Reynolds numbers. The fluted ducts with “ $P/DH = 0.18, 0.22,$ and 0.27 ” at $e/DH = 0.06$ provide the maximum thermic presentation of 2.26, 2.33, and 2.15, sequentially. The average thermic presentation section of the fluted duct, with a value of $P/DH = 0.27,$ is found to be 4% and 9% greater than that of the less pitched portions, $P/DH = 0.22$ and $0.18,$ based on the current work span.

Effect of rib-height ratio

In Figs. 5(a)& 5(b) and 6(a) & 6(b), the pressure drop and hotness exchange value are serially displayed in place of the abrasion part and Nu as a result of the costal-extent to breadth proportions ($e/DH = 0.04, 0.02,$ and 0.06) experienced in the fluted duct. The Nusselt number and the abrasion section of the figure grow as the ratio of costal-extent to width increases. Owing to helical ribs that encourage turbulence run adjacent to the duct wall and disruption of the thermic limit surface. In between the range of 20–29% and 50–67%, respectively, the value of the magnified duct with $e/DH = 0.06$ the Re value causes the Nusselt number value to become greater than the values with $e/DH = 0.04$ and $0.02.$ Larger rib-heights can cause more peripheral flow, which increases the transmittive hotness exchange value. In instance, the maximum valve thermal performances of 1.55, 1.89, and 2.33 are produced at $P/DH = 0.27$ by the enlarged ducts with $e/DH = 0.06, 0.04,$ and $0.02.$

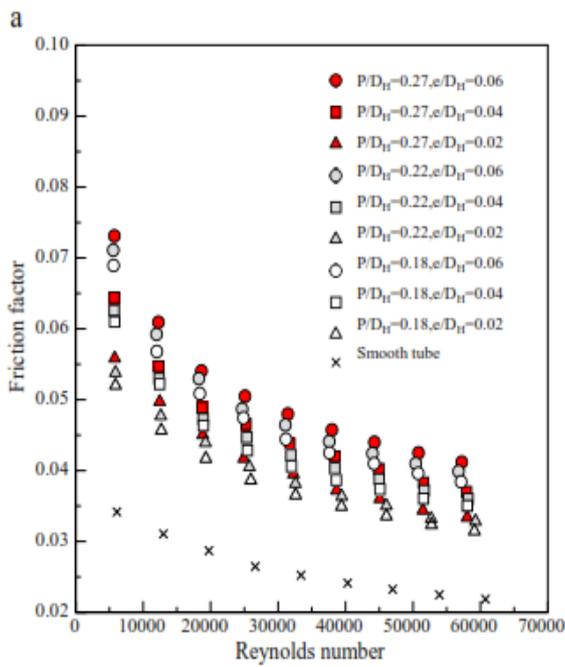


Fig. 6(a)

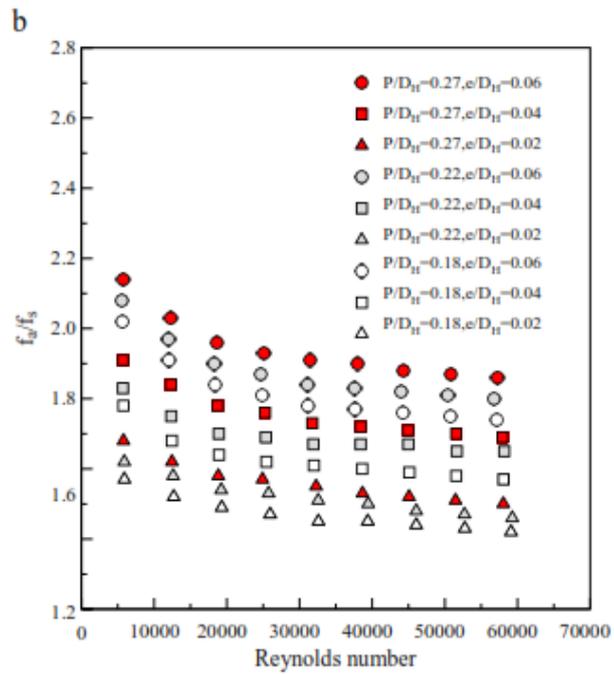


Fig. 6(b)

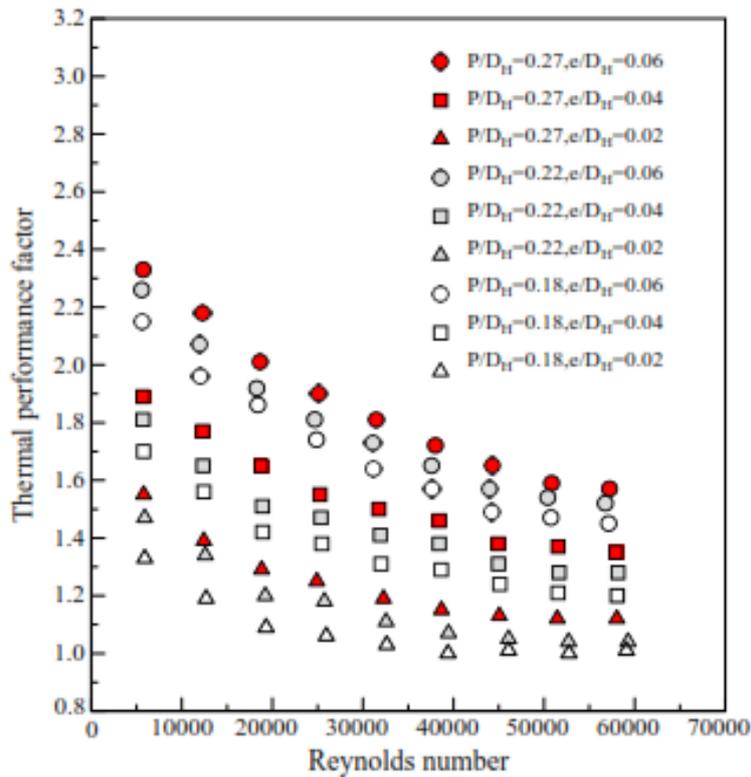


Fig. 7

Correlations

The experimental heat transmission data importance, the “friction factor for single phase” turbulent flow in scroll folded tubes, and the thermal stability for many slope-to-breadth measurements “P/DH = 0.27, 0.18, and 0.22” and “costal-to-breadth” measurements (e/DH = 0.04, 0.06, and 0.02) are all included in the following instructions:

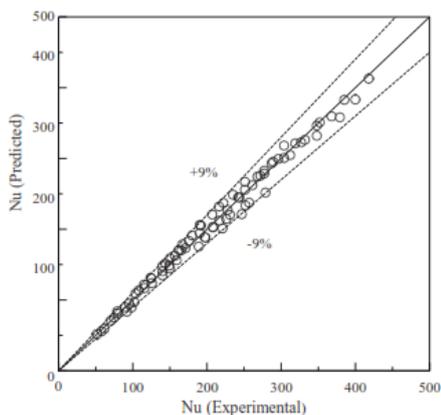


Fig. 8

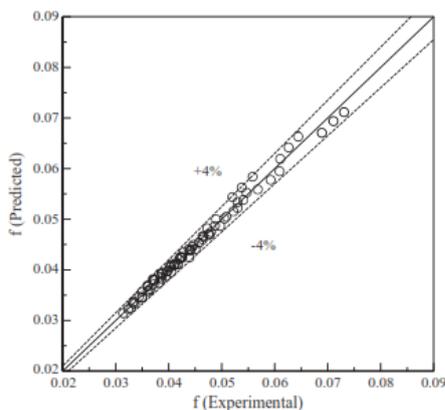


Fig. 9

It demonstrates that the pitch proportion (P/DH) and the Nusselt number (N) are only found in the abrasion section, and that the costal-extent proportion (e/DH) is a major contributor to the necessary space of thermal performance factor characteristics. It could have happened because the pitch ratio is easier to determine than the costal-extent proportion on several. Figs. 8, 9, and 10 are categorized in a sequential manner based on the division between the friction factor, thermal performance factor, and the Nusselt number discovered by the present study with values determined by the current relationship. The graphs illustrate that the mean deviations of the several regressions for thermal performance, Nusselt number, and abrasion portion fall within about ±7%, ±9%, and ±4% ranges.

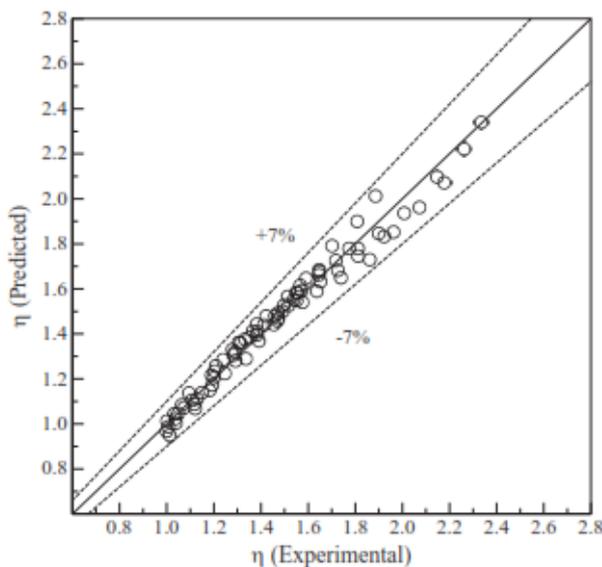


Fig. 10

CONCLUSION

This study examines and monitors the thermal potential, erosion element characteristics, and heat exchange enhancement of a swirly fluted duct. Nine spirally corrugated ducts that are processed have three different costal-to-breadth proportions ($P/DH = "0.27, 0.18, \text{ and } 0.22"$) as well as three different costal-extent to breadth proportions ($e/DH = "0.06, 0.02, \text{ and } 0.04"$). This unique outcome indicates that when the slope angles (P/DH) and the costal-extent ratio (e/DH) increase, so does the value of the abrasion element in the thermo graphic presentation section and the Nusselt number. The boosted duct with slope ratio $P/DH = 0.27$ and "costal-extent proportion $e/DH = 0.06$ " at lower Re , where "friction factor and Nu are 2.14 and 3.01" times higher than the empty tube, is found to have the highest thermal presentation part value in the current work, close to 2.33. Based on this, there is a consistent empirical link between the friction factor, Nu and "thermal performance factor".

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