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Original Research

Enhancing the Efficiency of the Double-Tube Heat Exchanger by using a Twisted Inner Tube

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Abstract

This study utilized two double tube-type heat exchangers. The first exchanger employed a smooth inner tube, while the second one utilized a twisted inner tube. The shell was constructed of poly(vinyl chloride) (PVC), while the tube was made of copper with a length of 1000 mm, an outer diameter of 62.24 mm, a smooth tube inner diameter of 14.2 mm, and an equivalent diameter of 11.8 mm for the twisted tube. To minimize heat loss, the shell was insulated externally with a thermal insulator. A flow rate of 3 liters per minute of hot water was passed through a ring-shaped tunnel, with an inlet temperature of 63 °C, to enhance the heat exchanger's performance. The experimental results of the two heat exchangers (smooth and twisted inner tubes) were compared, and the use of water as the primary fluid led to improved performance. The twisted inner tube-type heat exchanger achieved a maximum efficiency of 0.33 at a volumetric flow rate of 5 liters per minute, while the maximum improvement in effectiveness was 65.71% at a volume flow rate of 3 liters per minute in the twisted inner tube-type heat exchanger.

Keywords: heat exchanger efficiency, heat transfer coefficient, double-tube heat exchangers, twisted tube.

1. Introduction

In numerous industrial and practical applications, the transfer of heat is necessary between two flowing fluids through a solid surface (barrier) that separates them. To accomplish this, "heat exchangers" are utilized as devices. Heat exchangers enable the transfer of heat between fluids without mixing them. However, larger heat transfers require more space and higher costs. Therefore, researchers continue to focus not only on performance attributes but also on considering the size, capacity of the designated space, and cost-effectiveness of manufacturing the exchanger. Improving the heat transfer rate can be achieved through two steps: (a) improving heat exchanger design and (b) enhancing the thermal conductivity of the working fluid (Holman, 2008).

Heat exchanger tubes are often made of metals with high heat conductivity, such as copper and aluminum. Various updates have been made to exchanger designs, aiming to improve their thermal performance. Heat exchangers can be classified in several ways, each contributing to enhancing their efficiency in terms of heat transport and practical application. The classifications of heat exchangers are as follows:

- classification based on heat transfers,
- classification based on the number of liquids,
- classification based on surface compactness,
- classification based on building characteristics,
- classification based on flow arrangements,
- classification based on heat transfer mechanism.

A double tube heat exchanger typically consists of two concentric tubes, with a smooth inner tube maximizing the available area. One fluid flows through the inner tube, while the other flows in the opposite direction through the annular space created between the two tubes (Shah, 1986; Pardhi & Baredar, 2012).

If the application requires relatively constant wall temperatures, fluids can flow in parallel flow direction as seen in Fig. 1. This type of exchanger is considered the simplest, with no flow distribution issues and easy disassembly for maintenance cleaning. Twin-tube exchangers are often preferred when one or both fluids are under high pressure, as it is more cost-effective to accommodate narrow corridor tubes compared to large cylindrical structures. For low-volume applications where the total heat exchange area is small, usually a few square feet, two-tube heat exchangers are commonly used. However, they are expensive per unit surface area, typically requiring around 50 square meters or less (Walker, 1982).

The performance of a pulsating heat pipe (PHP) (presumably referring to a heat pipe) is influenced by various factors, including the number of turns, pipe diameter, tube shape, length of the condenser and evaporator sections, overall length of the PHP, heat input, inclination, physical properties of the working fluid, and the volumetric filling ratio (Barrak et al., 2022).

The arrangement of flow direction in a two-tube exchanger generally depends on the desired exchanger efficiency, pressure drop, limitations on maximum and minimum velocities, fluid flow path, allowable thermal pressures, temperature levels, and other design criteria. Cold and hot fluids can flow in opposite or the same direction (Bergman et al., 2011; Bergies, 1999).

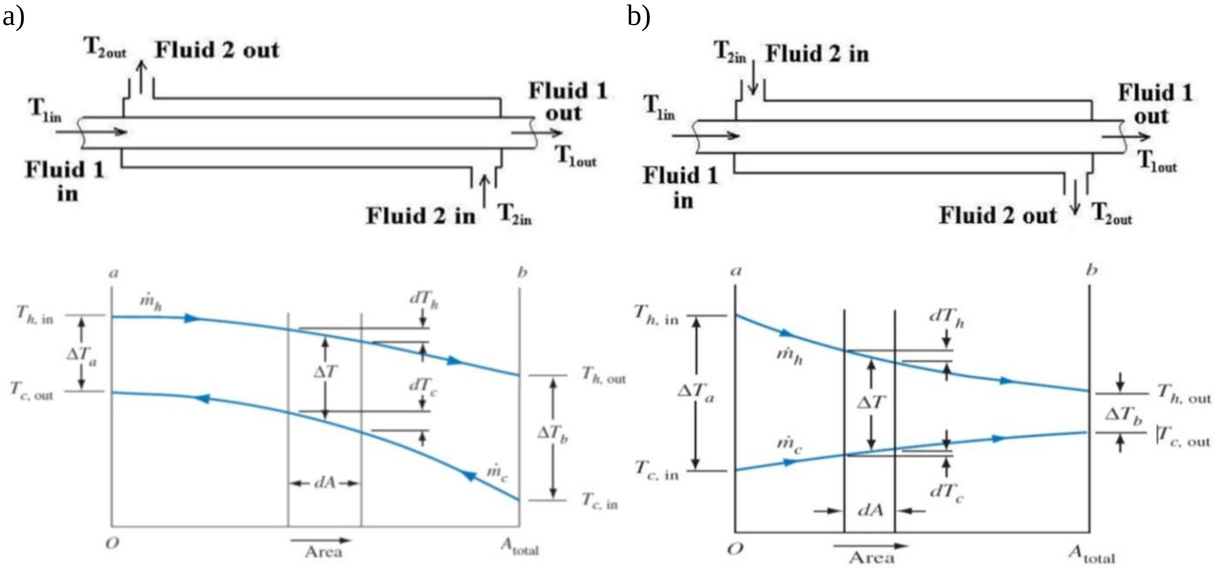

Fig. 1. Single-pass dual-flow heat exchange: a) reverse flow arrangement, b) parallel flow arrangement.

2. Features of the twisted tube

Twisted tubes are vortex flow generators utilized to enhance thermal performance. The impact of twisted tubes on heat transfer, pressure drop, and thermodynamic efficiency of thermal exchangers has been studied (Samruaisin et al., 2019). In many thermal applications, twisted tubes are employed to improve heat dissipation by increasing the available surface area (Abbas et al., 2021). Figure 2 illustrates that twisted tubes have gained attention due to their constructable shape, lower cost, and superior thermal performance compared to smooth tubes. The potential improvement mechanisms for this technique include: (1) expanding the surface area and length of the flow path through circuitous routes, (2) increasing fluid velocity due to flow obstacles and specific geometric curvatures, and (3) intensifying the fluid mixing process through centrifugal forces and vortex flows (Yan et al., 2017).

For highly viscous fluids, the heat transfer coefficient is typically low at low Reynolds numbers to prevent a significant increase in pressure drop. Using a convoluted tube is an effective method to enhance heat transport capacity without significantly raising the pressure drop (Yan et al., 2017). Experiments were conducted to observe the changes in pressure drop and heat transmission in a twisted oval tube with varying axis orientations and twist ratios, comparing it to a perfectly circular tube. The pressure drop inside the twisted oval tubes was greater, while the heat transmission rate was higher. Additionally, extensive studies established that as the twist ratios increased, pressure loss and heat transmission decreased, whereas the opposite was true for axis ratios (Wang et al., 2000).

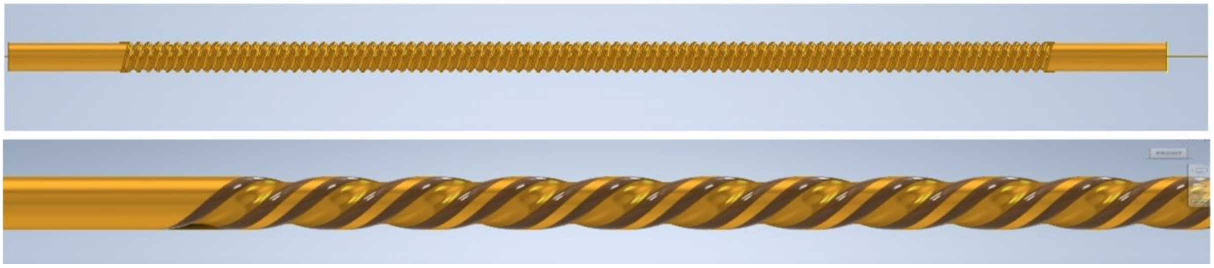

Fig. 2. The twisted tube.

In a study conducted by Samruaisin et al. (2019), a twisted trapezoidal tube and twisted bars with different winding ratios (y/w = 2.0, 3.0, 4.0, and 5.0) were used. Water was employed as the working fluid to simulate a turbulent system with Reynolds numbers ranging from 4,500 to 16,000. The research focused on temperature and velocity profiles as well as variations in local Nusselt numbers to gather information about fluid flow and heat transfer. A comparison was made between the combined devices and the individual use of either a smooth circular tube or a twisted tube. The experimental findings indicated that, for a given Reynolds number (Re), the Nusselt number (Nu), friction factor (f), and thermal performance of the twisted tube reinforced with a twisted bar were consistently higher than those of the twisted tube alone or the simple round tube. Additionally, the thermal performance factor, frictional loss, and heat transfer all improved with a decrease in the torsion ratio (y/w) because a bar with a smaller y/w provided a more regular and robust flow.

In another study conducted by Eltaweel et al. (2020), the performance of an exchanger using a flat plate solar collector was investigated. Two different mediums were used: purified water and multi-walled carbon nanotubes (MWCNTs)/aqueous nanofluids. Both types of heat exchangers were tested in the same location under similar environmental conditions. The use of convoluted tubes in the system improved performance by 12.8% and 12.5% in distilled water and MWCNT/water nanofluid, respectively, compared to employing circular tubes. Furthermore, when using MWCNTs, convoluted tubes resulted in a 34% improvement compared to standard pipes with distilled water.

In a numerical study conducted by researcher Khoshvaght-Aliabadi and Feizabadi (2020), liquid heat transfer properties were investigated using a composite method to compare various pitch ratios and five Reynolds values for laminar flow in tubular heat exchangers. The results showed that both the degree of torsion and the Reynolds number significantly influenced the dependence of transport properties on the optimized models. It was observed that the pressure drop in the twisted tube (TT) with torsion was generally higher than that in the twisted tube alone (TT), which contrasted with the heat transfer performance. However, the TT-minor TT and the TT-major TT exhibited the highest values. The secondary TT-TT configuration achieved the best overall hydrothermal performance, while the straight-tube (ST) equipped with twisted-tape (ST-TT) and the primary TT-TT performed similarly. The findings demonstrated that increasing both the degree of torsion and the Reynolds number led to improved performance across the board. In a numerical investigation conducted by researcher Eiamsa-Ard et al. (2016), the addition of a twisted bar with three channels to a spirally twisted tube was studied to enhance heat transmission. The effects of tube/tape designs (belly-to-belly and belly-to-neck) and the width ratio of the tape (w/D = 0.1, 0.25, 0.34, and 0.5) were examined. The study revealed that the width-to-thickness ratio of the tape has a significant impact on the heat transmission coefficient and friction. Comparing two systems with the same bandwidth, the belly-to-neck configuration was found to promote heat transmission more effectively than the belly-to-abdomen arrangement. At w/D = 0.1, 0.25, and 0.34, the Nusselt numbers for three-indented convoluted tubes with convoluted bands in a belly-to-neck configuration were up to 1.2%, 21%, and 36% higher compared to convoluted tubes without bands. Similarly, the Nusselt numbers of contorted tubes with a three-channel convoluted band in a belly-to-abdominal layout were up to 1.23%, 6.7%, 10%, and 17% higher than those of similar tubes without a band. The increased heat transmission in the belly-to-neck configuration (especially in the large w/D range) can be attributed to the enhanced engagement between the eddy flows induced by tubes and those induced by the tape. Additionally, the friction loss caused by the subsystems in the belly-to-neck layout was lower compared to the belly-to-bottom arrangement. Consequently, thermal efficiency characteristics improved when the systems were arranged with their bellies touching. With a Reynolds number of 5000, the maximum achievable thermal performance was observed to be 1.32 in a twisted tube with a three-channel twisted bar in a belly-to-neck configuration and a tape thickness of 0.34 inches.

In a study conducted by Farnam et al. (2021), it was observed that the wavy walls in a spiral twisted tube constantly alter the orientation and strength of the secondary flows and velocity lines, preventing the growth of thermal layer boundaries in the flow direction and leading to a more uniform temperature distribution. At moderate levels of design parameters, the Nusselt number increased by 14.2% within the investigated Reynolds number range ($600 \leq Re \leq 1200$), while the friction factor increased by 7.7%. The study performed a parameter analysis to understand the interaction of various design variables (helical diameter, helical pitch, and pitch torsion). It was found that the diameter of the spiral had the most significant effect on the thermo-hydraulic properties of the spiral twisted tube, followed by the convexity and step of the spiral. Minimizing both the helical diameter and torsional angle resulted in improved hydrothermal performance. With a Reynolds number of 900 and a spiral diameter of 50 mm, the model achieved a maximum performance index of 1.98.

In a study conducted by Thantharate (2013), four pipe passes were examined, each with a diameter of 0.3 meters, and four different flow rates of liters per minute (LPM) of 1.5 dm^3/min, 1.37 dm^3/min, 0.5 dm^3/min, and 0.24 dm^3/minwere tested, resulting in a Reynolds number range of 625 to 7000. Analytical, experimental, and numerical analyses of turbulent and laminar flow patterns were conducted. The findings showed that the plain tube performed better than the convoluted tube at low flow rates and high inlet temperatures, primarily due to the flow characteristics inside the tubes. The study concluded that when setting up a multi-path design, the use of convoluted tubing is necessary to achieve the desired flow rate. The study aimed to improve the efficiency of a double pipe counterflow heat exchanger using water as the hot and cold fluid, comparing the heat transfer results of twisted and smooth tubes. The experiments were conducted using water to achieve better performance at different flow rates (3, 5, 7, 9, 12 dm^3/min) for the cold fluid and a constant flow rate of 3 dm^3/minfor hot water.

3. Methodology for experiments

The test system depicted in Fig. 3 comprises two double tube-type heat exchangers. It includes two stainless steel basins with a capacity of 10 dm^3 each. One basin is used in the first stage as a reservoir for cold water with a temperature of 18±1°C, which flows through the inner tube of both the smooth tube exchanger and the twisted tube exchanger. This circulation is facilitated by a pump with a horsepower capacity of 0.5 HP and a maximum head of 40 meters. The water exiting the inner tubes (smooth and twisted) at a higher temperature than that of the inner tubes returns to the same basin. The second basin supplies water at ambient temperature to an electric heater through another pump with a capacity of 0.5 HP and a maximum head of 40 meters. Upon entering the heater, the water temperature rises to 63±1°C. The same basin is used to collect the return water from the outer tube of both exchangers, which has a lower temperature than the outer tubes. To maintain temperature consistency throughout the process, the heat exchangers and the plastic inlet and outlet pipes are covered with thermal asbestos.

4. Test configuration

This study aims to investigate improvements in the performance of a circular double-tube exchanger. This section provides a detailed explanation of the experimental procedure conducted for the tests. The experiments were carried out using cold water flowing in the opposite direction to the hot water in the dual pipe exchanger as seen in Fig. 3. Once the device was manufactured, the experiments were initiated to collect the necessary data for the calculations related to the research. The first experiment involved heating the water to a temperature of 63±1°C using an electric heater. The cold water was set at a temperature of 18±1 °C. The flow rate for both hot water (3 dm^3/min) and cold water (3 dm^3/min) was determined. Readings were taken after monitoring the temperature gauges until they reached a steady state to record the temperature values. Following this, the experiment was repeated by maintaining a fixed flow rate for hot water (3 dm^3/min) and varying the flow rate for cold water. Three different flow rates for cold water (3, 4, 5 dm^3/min) were tested. and the temperature values were recorded after reaching to a steady state.

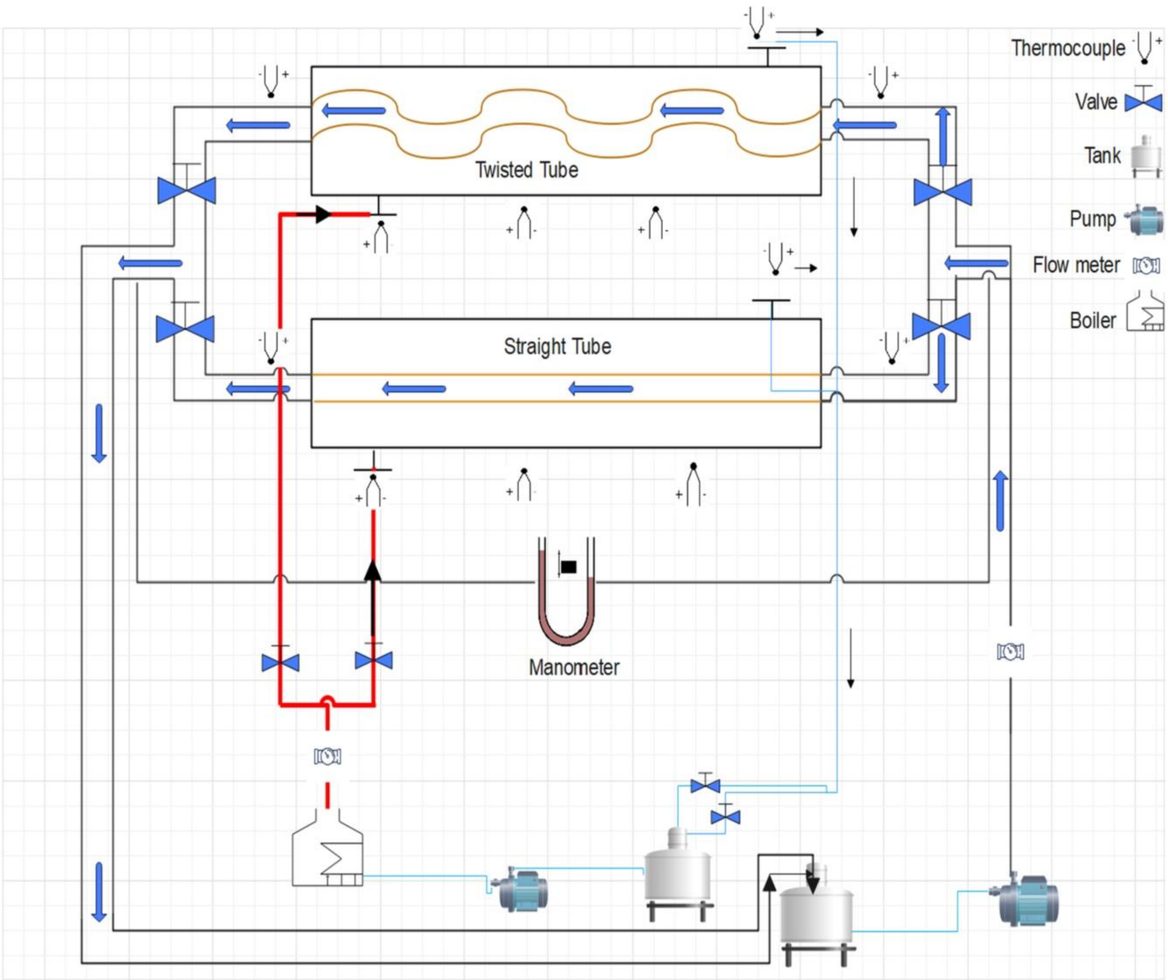

Fig. 3. Test diagram.

The water has been pumped out from the tanks, and all pipe connections have been tested. The pumps are now operating to circulate the working fluid in the exchanger.

The control valve was appropriately set to regulate the flow rate at the required level. The flow rate was monitored using the flowmeters installed in the test device. Cold water is flowing into the inner tube of the exchanger, while the electric heater in the hot tank is turned on to regulate the temperature. The hot water flows through the shell of the heat exchanger. The flow rate for the hot water is fixed at 3 dm^3/min, while three different flow rates (3, 4, 5 dm^3/min) were tested for the cold fluid.

After reaching a steady state, thermocouples were used to monitor the temperatures of the incoming and outgoing flows of the working fluid. The constant measurements were recorded using the Data Loader device. The hot water temperature was maintained at 63±1°C, while the wall temperatures were measured. An electronic pressure gauge was utilized to measure the pressure drop (ΔP) across the inlet and outlet areas of the test section simultaneously.

The total heat transfer coefficient, the amount of heat transferred, and the heat exchanger efficiency were calculated using the following equations (Lee, 2022; Hayder, 2023).
Hot water mass flow rate:

$$\dot{m}h = \frac{V}{60000} \times \rho \tag{1}$$

where $\dot{m}h$ is mass flow rate for hot water, V is volumetric flow rate and ρ is density.
Cold water mass flow rate:

$$\dot{m}c = \frac{V}{60000} \times \rho \tag{2}$$

where $\dot{m}c$ is stand for mass flow rate for cold fluid.

The heat transferred from hot to cold water:

$$q_h = \dot{mh} \times C_{ph} \times (T_{hi} - T_{ho}) \tag{3}$$

where q_h is stand for heat transfer for hot water, C_{ph} is specific heat for hot fluid, T_{hi} is temperature for hot fluid that inter the shell and T_{ho} is temperature for hot fluid at the end of the tube.
The amount of heat gained (qc) by cold water:

$$q_c = \dot{mc} \times C_{pc} \times (T_{c0} - T_{ci}) \tag{4}$$

where C_{pc} is specific heat for cold fluid, T_{ci} is temperature for cold fluid that inter the tube and T_{c0} is temperature for cold fluid that end of the tube.
The average amount of heat transferred (q_{avg}):

$$q_{avg} = \frac{q_h + q_c}{2} \tag{5}$$

The inner portion of a twisted tube has a complex form, thus volume-driven diameters are used instead of the more commonplace metric diameters. It is determined as (Rousseau et al., 2003):

$$Dv_i = \sqrt{\frac{4V_{ol}}{\pi L}} \tag{6}$$

where Dv_i is the twisted tube inner diameter which is volume driven diameter, V_{ol} is volume occupied by the fluid in the tube and L is length of the exchanger.
The inner surface area (A_i) of the inner tube of inner fluid (di,ip):

$$A_i = \pi \times di{,}ip \times L \tag{7}$$

Logarithmic Mean Temperature Difference (LMTD) for counter flow arrangement is determined according to Eq. (8):

$$\text{LMTD} = \frac{(T_{hi} - T_{co}) - (T_{h0} - T_{ci})}{ln\ (T_{hi} - T_{co})/(T_{h0} - T_{ci})} \tag{8}$$

The efficient overall heat transfer coefficient U_i is calculated using the inner pipe surface area (Shah & Sekulic. 2003):

$$U_i = \frac{q_{avg}}{Ai{,}ip \times LMTD \times F} \tag{9}$$

Heat capacity for cold water:

$$C_c = \dot{m}_C \times Cp_c \tag{10}$$

Heat capacity for hot water:

$$C_h = \dot{m}_h \times Cp_h \tag{11}$$

C_{min} is minimum value out of C_c and C_h, C_{max} is maximum value out of C_c and C_h. The maximum heat transfer q_{max} can be found as below equation:

$$q_{max} = C_{min}(T_{hi} - T_{ci}) \tag{12}$$

The Nusselt number (Nu) in the inner tube is calculated by determining the temperature of the inner surface of the tube, which is assumed to be equal to the outer surface temperature (for small wall thickness). This is done by measuring the temperature along the outer surface of the inner tube and then calculating the internal heat transfer coefficient:

$$T_{w_c} = \frac{T_1 + T_2}{2} \tag{13}$$

T_{w_c} is the temperature of wall of the tube for cold fluid and T_1, T_2 are temperatures that are taken from thermocouples fixed at the wall of the tube.

$$T_{c,avg} = \frac{T_{c_o} + T_{c_i}}{2} \tag{14}$$

$T_{c,avg}$ is the average temperature for cold fluid and T_{co}, T_{ci} are temperatures for cold fluid at entrance and end of the tube, respectively.
Internal heat transfer coefficient (h_i) can be calculated from equation below:

$$h_i = \frac{q_{avg}}{A_i(T_w - T_{c,avg})} \tag{15}$$

$$Nu_c = \frac{h_i d_i}{k_i} \tag{16}$$

where Nu_c is Nusselt number for cold fluid, d_i is tube diameter and K_i is thermal conductivity for inner fluid (cold fluid).

The pressure difference (Δp) was calculated from the digital manometer. The coefficient of friction for the inner tube was calculated from the following equation (Thulikkanam, 2000):

$$f = \frac{2*\Delta P*g_c*d_i}{L*\rho*u^2} \tag{17}$$

where f is coefficient of friction, $\Delta P = (P_2 - P_1)$ is pressure difference, g_c is constant and u^2 is velocity of the cold fluid entering the tube.

The heat exchanger performance evaluation coefficient (PEC) was calculated from the following equation (Ali et al., 2023; Ali & Mohamad, 2022; Berkache et al., 2022; Quader et al., 2023; Shrirao et al., 2013):

$$\mathrm{PEC} = \frac{\left(\frac{Nu_{TT}}{Nu_{ST}}\right)}{\left(\frac{f_{TT}}{f_{ST}}\right)^{\frac{1}{3}}} \tag{18}$$

where Nu_{TT} represents the Nusselt number for twisted tube heat exchangers, Nu_{ST} denotes the Nusselt number for smooth tube heat exchangers, f_{TT} signifies the coefficient of friction for twisted tube heat exchangers and f_{ST} corresponds to the coefficient of friction for smooth tube heat exchangers.

5. Discussion of results

From the results shown in Fig. 4, it is observed that the use of a heat exchanger with a twisted inner tube enhances the temperature difference between the inlet and outlet for both cold and hot water. Additionally, it is noted that the temperature difference decreases with an increase in the volumetric flow rate. This is due to the increased mass flow rate, which leads to an increase in thermal storage. A twisted tube is an effective way to improve heat transfer in heat exchangers. The percentage increase in the temperature difference for the twisted tube compared to the smooth tube is 66.7% at a flow rate of 5 dm^3/min.

Fig. 5 depicts the total internal heat transfer coefficient (Ui), which increases with the flow rate and reaches its highest value at a flow rate of 5 dm^3/min for water in the heat exchanger with a twisted inner tube. The increase in volumetric flow rate leads to an increase in convective heat transfer coefficient.

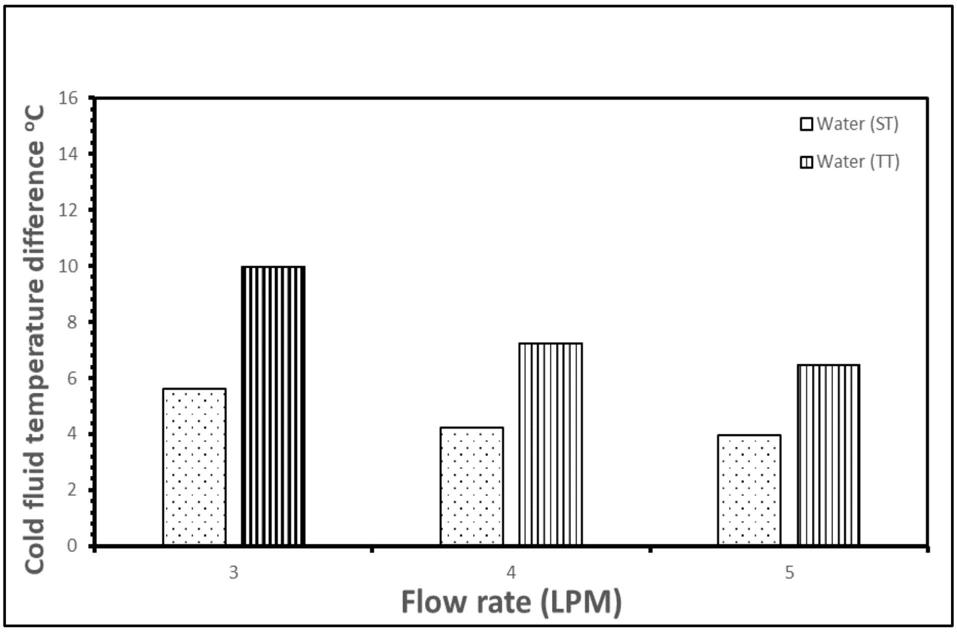

Fig. 4. Difference between the entry and exit temperatures of the fluid passing through the (smooth) and (twisted) inner tube.

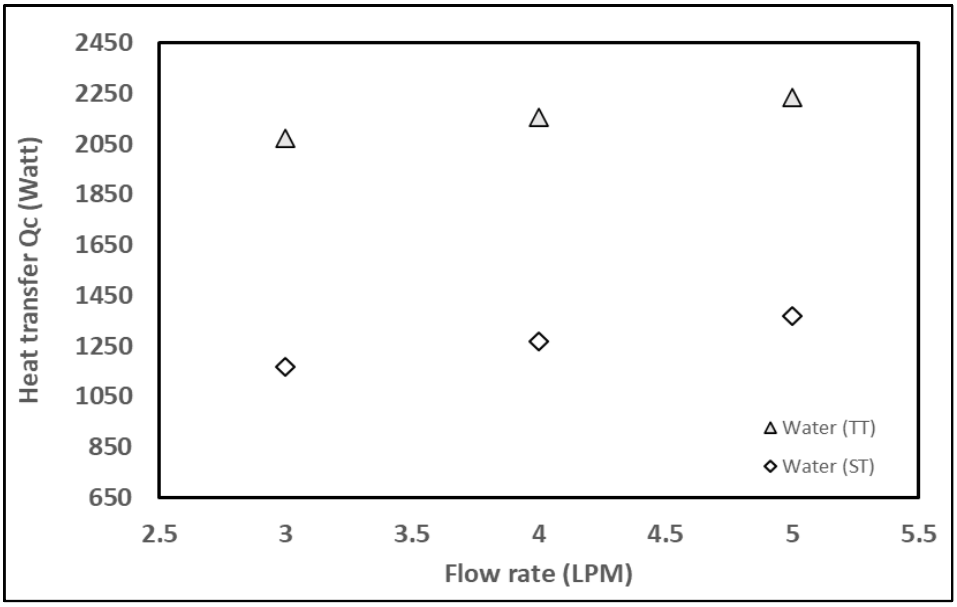

Fig. 5. The effect of changing the three flow rates on the heat transfer rate of the two exchangers.

In Fig. 6, it is observed that there is a direct relationship between the heat transfer rate and the flow rate, as well as an increase in heat transfer when using the twisted inner tube. The highest heat transfer rate (Q_c) is achieved with the flow model that reduces the thickness of the boundary layer. This reduction is due to the random movement and swirling motion generated by the twisted tube, along with the random movement of water, resulting in increased heat transfer.

Fig. 7 illustrates the change in the effectiveness of the heat exchanger using water in the inner tube for both exchangers and at the three flow rates. The highest exchanger efficiency is obtained at a flow rate of 5 dm^3/min in the heat exchanger with the twisted inner tube. According to the Eq. (11), when the heat capacity of the hot water is smaller, the temperature difference is greater, leading to an increase in exchanger effectiveness. The highest improvement in effectiveness (65.71%) is achieved at a flow rate of 3 dm^3/min in the heat exchanger with the twisted inner tube.

Fig. 8 presents the Nusselt number, which increases with the flow rate and reaches its highest value in the heat exchanger with the twisted inner tube at a flow rate of 5 dm^3/min when using water. The Nusselt number, as per Eq. (18), demonstrates the change in the Nusselt number with an increasing flow rate.

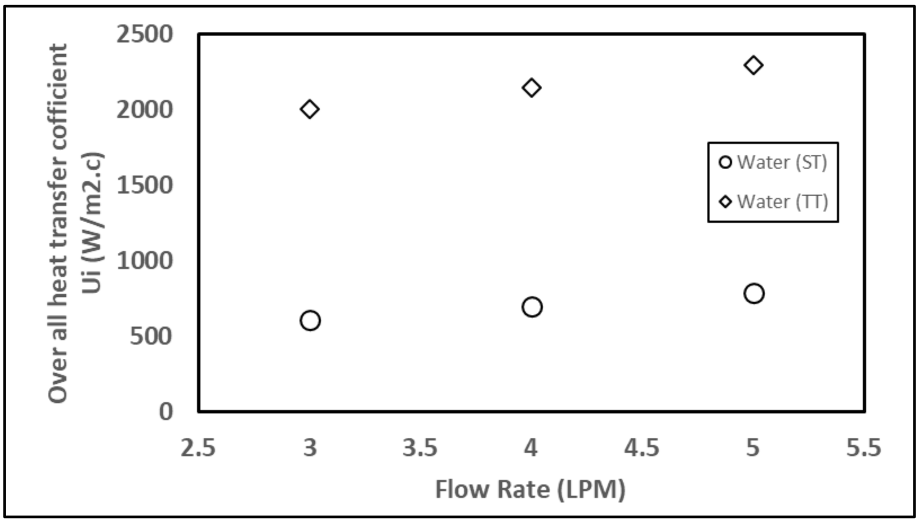

Fig. 6. Changing the coefficient of heat transfer and the rate of flow for the smooth (ST) and twisted (TT) inner tube exchange.

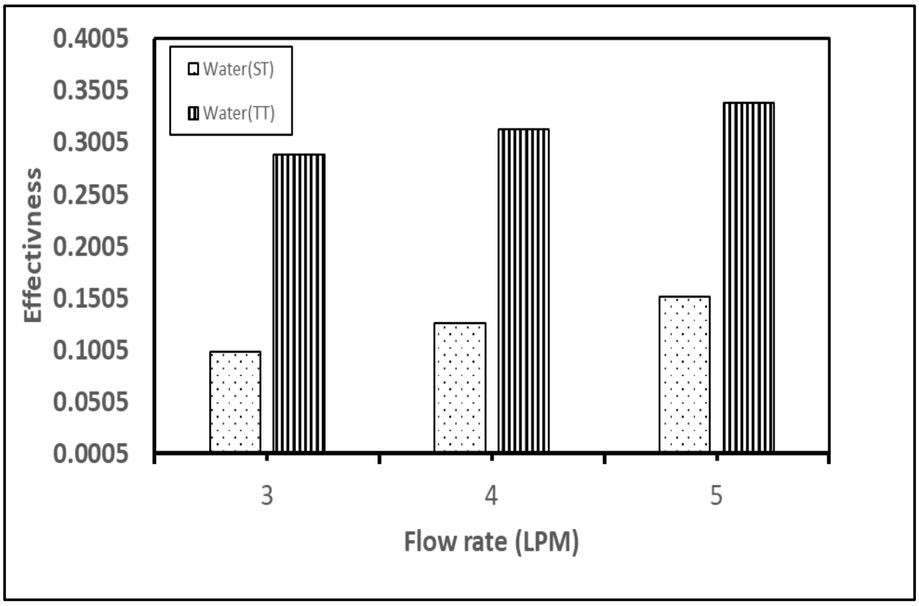

Fig. 7. Changing the effectiveness of the exchanger and the three flow rates for the two exchangers.

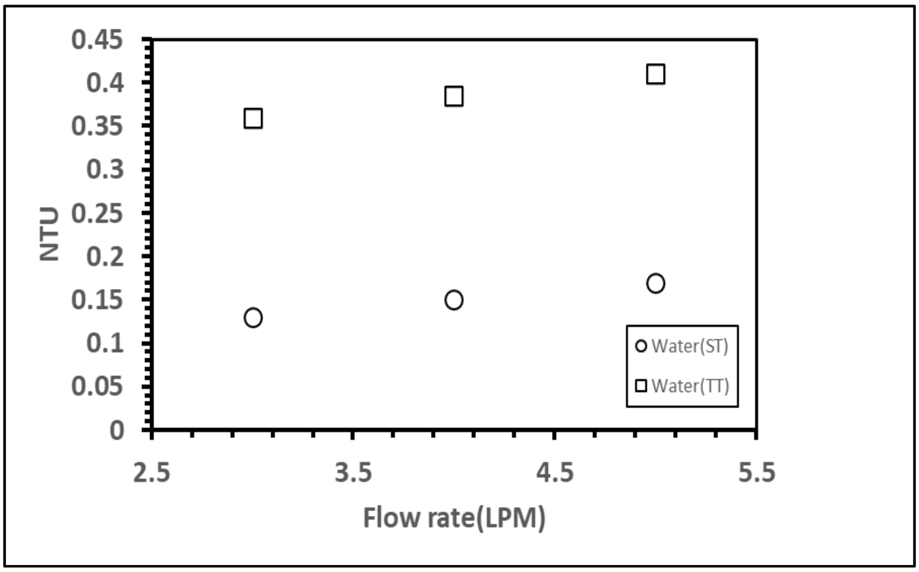

Fig. 8. Changing the Nusselt number with changing the flow rate.

Fig. 9 demonstrates the effect of the Reynolds number (*Re*) on the friction coefficient of water at the three flow rates (3, 4, 5 dm^3/min). It is observed that the friction coefficient decreases with an increase in the Reynolds number. For water, the highest value of the friction coefficient is obtained in the heat exchanger with the twisted inner tube at a flow rate of 3 dm^3/min.

Fig. 10 shows the performance evaluation coefficient at the three flow rates, as we notice that the performance evaluation coefficient decreases with the increase in the volumetric flow rate, as it reached its lowest value at the flow rate (5 dm^3/min). The performance evaluation coefficient decreases after the flow rate (3 dm^3/min) and that is based on the values of the Nusselt number and the friction coefficient, where the highest value of the performance evaluation coefficient is at the flow rate (3 dm^3/min), where the value of the transferred heat and the internal heat transfer coefficient are the highest possible, and therefore the Nusselt number increases according to Eq. 18, and thus we get the highest value of the performance evaluation coefficient (3.28).

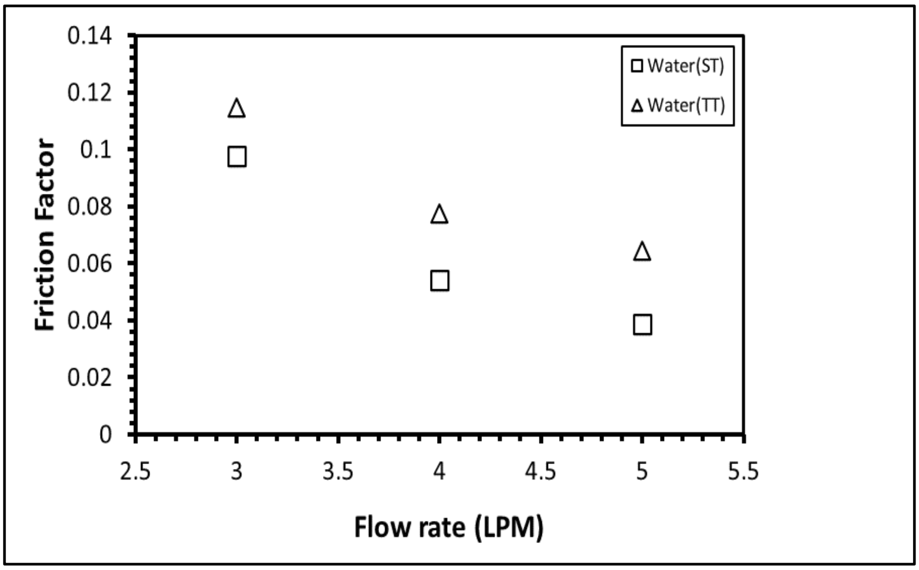

Fig. 9. Comparing the friction coefficient of water at the three flow rates.

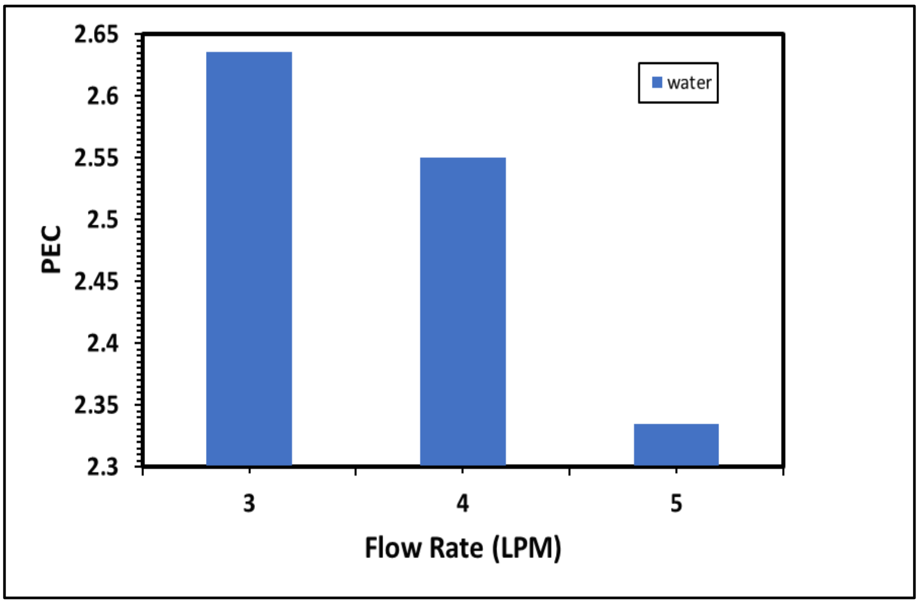

Fig. 10. Performance evaluation coefficient (PEC) using water at the three flow rates.

6. Conclusions

The overall heat transfer coefficient increased with an increase in the average amount of heat transferred in the heat exchanger with a twisted inner tube. However, the value decreased with increasing flow rates since there is not enough time for efficient heat exchange to occur in both exchangers. The highest efficiency of the exchanger was achieved at a flow rate of 5 dm^3/min in the heat exchanger with the twisted inner tube. According to Eq. 11, this is due to the smallest heat capacity of the hot water resulting in a greater temperature difference, thus increasing the exchanger efficiency. It was observed that the friction coefficient decreased with an increase in the flow rate, reaching its highest value at a flow rate of 3 dm^3/min in the heat exchanger with the twisted inner tube. The highest value of the performance evaluation coefficient (PEC) for the heat exchanger was observed at a flow rate of 3 dm^3/min.

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Zwiększanie Wydajności Dwururowego Wymiennika Ciepła oprzez Zastosowanie Skręconej Rury Wewnętrznej

Streszczenie

W badaniach wykorzystano dwa dwururowe wymienniki ciepła. W pierwszym wymienniku zastosowano gładką rurę wewnętrzną, natomiast w drugim zastosowano rurę skręconą. Płaszcz wykonano z poli(chlorku winylu) (PVC), natomiast rura wykonana z miedzi charakteryzowała się następującymi wymiarami: długość 1000 mm, średnica zewnętrzna 62,24 mm, średnica wewnętrzna 14,2 mm i średnica zastępcza dla rury skręconej 11,8 mm. Aby zminimalizować straty ciepła, rura została zaizolowana od zewnątrz izolatorem termicznym. Aby poprawić wydajność wymiennika ciepła, przez tunel w kształcie pierścienia przepuszczano gorącą wodę z szybkością 3 dm^3/min., a temperatura na wlocie wynosiła 63°C. Porównano wyniki eksperymentalne dwóch wymienników ciepła (gładkiego i o skręconej rurze wewnętrznej) i stwierdzono, że zastosowanie wody jako płynu roboczego doprowadziło do poprawy wydajności wymiennika. Wymiennik ciepła wykonany ze skręconej rury wewnętrznej osiągnął maksymalną wydajność 0,33 przy objętościowym natężeniu przepływu 5 dm^3/min., natomiast maksymalna poprawa efektywności wyniosła 65,71% przy objętościowym natężeniu przepływu 3 dm^3/min.

Słowa kluczowe: sprawność wymiennika ciepła, współczynnik przenikania ciepła, dwururowe wymienniki ciepła, skręcona rura.