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Evaluation of double pipe heat exchanger performance with right angle turbulator

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ABSTRACT

In this paper, two horizontal cross-flow pipes were experimentally investigated. Three discharge rates for hot water were used, namely 1.5, 2.5, and 3 liters per minute, while maintaining a 6 liters per minute cold water discharge rate. Comparing the discharge rates in both instances and relying on the hot water discharge rate that results in the biggest heat exchange was done. The first (typical) type involved the analysis of a heat exchanger with two horizontal tubes of convection flow; in this case, there were no additives in the hot pipe, and the best rate of hot water discharge was 1.5 liters per minute. As a result, the optimal Nusselt number was identified. In the second part, right-angle turbulators with dimensions of 1 mm, 5 mm, and 7 mm were added inside the hot pipe to enhance the heat exchange. It was discovered that the Nusselt number is significantly better than the typical situation (without angles). When adding this kind of turbulator, the Nusselt number is enhanced by 28.1%.

Symbols list

Symbol	Description
Q	the heat transfer rate (in watts or BTUs per hour).
U	the overall heat transfer coefficient (in watts per square meter per Kelvin or BTUs per hour per square foot per Fahrenheit).
A	the heat transfer surface area (in square meters or square feet).
Ŋ	Thermal performance factor
M	Mass flow rate of the fluid (in kilograms per second)
Cp	Specific heat capacity of the fluid (in joules per kilogram per degree Celsius)
Tin	Temperature of the fluid entering the system (in degrees Celsius)
Tout	Temperature of the fluid exiting the system (in degrees Celsius)
ΔT_{lm}	Logarithmic mean temperature difference (in degrees Celsius or Kelvin)
ΔT1	Temperature difference at one end of the heat exchanger (in degrees Celsius or Kelvin)
ΔT2	Temperature difference at the other end of the heat exchanger (in degrees Celsius or Kelvin)
H	Heat transfer rate (in watts or joules per second)
As	Surface area of the heat exchanger (in square meters)
Nui	Nusselt number for internal flow (dimensionless)
Hi	Convective heat transfer coefficient on the inside surface (in watts per square meter per degree Celsius or $W/(m^2 \cdot °C)$)
Di	Characteristic length or hydraulic diameter of the flow (in meters or m)
K	Thermal conductivity of the fluid (in watts per meter per degree Celsius or $W/(m \cdot °C)$)
Re	Reynolds number (dimensionless)
P	Density of the fluid (in kilograms per cubic meter or kg/m^3)
Um	Mean velocity of the fluid (in meters per second or m/s)
M	Dynamic viscosity of the fluid (in pascal-seconds or Pa·s)

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1. Introduction

There are many different types of heat exchangers divided into groups according to various factors, including structure, flow configuration, heat transfer method, etc. Depending on their intended use, heat exchangers are optimized. To improve heat transfer in heat exchangers and heat transfer within heat exchanger tubes, tubes with fins (precise spiral or zigzag) or spiral wires or angles are used to enhance these devices' functionality. Although heat transfer is significantly increased, pressure is also significantly reduced. Since there are initiatives to create heat exchangers that are more effective at a specific temperature or to enhance the heat exchanger's performance at the lowest temperature used. Regardless of the economic benefits of operating smaller and more efficient systems to reduce environmental damage, at the same time, size should not be reduced at the expense of the Coefficient of Performance (COP) [1]. In order to lower the cost and amount of space needed for installation and operation, numerous studies have been done to improve the thermal performance and heat transmission of heat exchangers. The passive approach is one of several employed [2]. The bulk of researchers concentrate on improving heat transmission in their research. Utilizing a variety of geometries as barriers in the fluid route might enhance heat transfer when turbulence is produced or the fluid stream is altered. Controlling the pressure drop brought on by these disturbances, which raises the thermal performance of heat exchangers and is ultimately impacted by the friction factor, is crucial at the same time. Different researchers have used various geometries with various geometrical and flow parameters for the experimental investigation. Ozceyhan and Bas [3] and Bhuiya et al. [4] they conducted their research using single, double, and triple twisted tapes with various width ratios and twist ratios. They discovered that heat transfer increases as the value of the twist ratio decreases. Jayakumar et al. [5] examined the thermal attitude of fluid flow inside a helical pipe containing bundles of helically coiled tubes using numerical and experimental methods. The numerical and experimental findings of the k-ε turbulence model simulation in FLUENT are compared. The experimental results and the numerical predictions showed a good degree of agreement. Aimvonge and Eiamsa-ard [6] investigated experimentally the impact of helical tapes put in a tube on improving heat transmission. The Reynolds number range for the tube's flow rate is between 8800 and 2300. The experiment was done on a tube that had full-length, consistently spaced helical twisted tape, both with and without a rod. While cold water flows in the annulus, hot air travels through the tube. According to experimental findings, helical twisted tape with a full-length of rod provides the maximum heat transfer rate -equal to 1.1 times that of tape without rod- but it also causes a greater pressure drop. Dewan [7] investigated experimentally the impact of helical tapes put in a tube on improving heat transmission. The Reynolds number range for the tube's flow rate is between 8800 and 2300. The experiment was done on a tube that had full-length, consistently spaced helical twisted tape, both without and with a rod. While cold water flows in the annulus, hot air travels through the tube. According to experimental findings, full-length helical twisted tape with a rod provides the maximum heat transfer rate- equal to 1.1 times that of tape without a rod -but it also causes a greater pressure drop. Ahmad Sabah Al-Hilaly [8] Submit a practical study showing that using aluminum dioxide (Al_2O_3) nanofluid with volume concentrations of 0.5, 1, and 2 percentage increases at the highest flow rate, the heat transfer coefficient (Nusselt number) in a twin pipe heat exchanger decreased by approximately 23.1, 39.1, and 48.93%,

respectively. A study reviews articles on double pipe heat exchangers, a thermal device used in various industries like cooling technology, refrigeration, and sustainable energy applications [13]. It discusses various models, factors affecting heat transfer rate and pressure drop, and their classifications like parallel, counter, and cross flow. Researchers have explored improving the effectiveness of these heat exchangers using turbulators, inserts, rips, channel geometry, and injection fluid methods.
The purpose of analyzing a twin pipe heat exchanger's performance with a right-angle turbulator is to determine how well the turbulator improves heat transfer inside the heat exchanger. The main objective is to ascertain whether adding a right-angle turbulator improves the double pipe heat exchanger's overall efficiency. This entails determining if the turbulator improves the heat exchange efficiency by speeding up the rate of heat transfer between the two fluid streams, which are commonly hot and cold fluids. Analyze how the turbulator affects several aspects of heat transmission, including heat transfer coefficients, thermal efficiency, and overall heat transfer rate. Check to see if the turbulator's design successfully alters the flow patterns, improving heat transfer.

2. Mathematical Model

2.1 Convectional heat transfer coefficient Calculation

One of the many variables that affect how much heat is transmitted between the two fluids circulating in the heat exchanger is the way the fluids flow between one another. A twin-tube heat exchanger can be used with either parallel flow or counterflow. The temperature difference depends on how the heat exchanger is operated, as seen in figure 1, which depicts the distribution of temperature in the heat exchanger with the parallel and counterflow, respectively. In which each fluid flows as well as on the heat exchanger's temperature distribution [9 and 10].

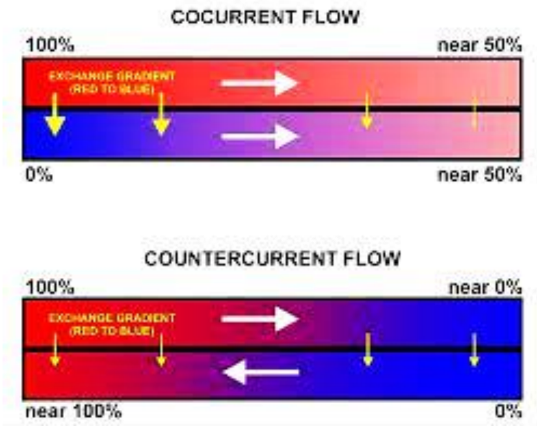

Figure 1. Heat exchange between fluids flowing in a heat exchanger parallel and opposite directions [11].

$$Q = m^{\circ} C_p (T_{in} - T_{out}) \text{ hot} \quad \dots(1)$$

In a twin pipe heat exchanger with a right-angle turbulator, the evaluation of heat transfer and fluid flow characteristics are commonly included in the equation for cold flow. In particular, to figure out the cold fluid's passage through the heat exchanger's pressure drop, temperature profiles, and heat transfer rate. It may be necessary to take into account factors like fluid characteristics, flow rates, shape, and the efficiency of the turbulator in the precise equations and computations involved. Based on the ideas of overall heat transfer coefficient (U) and logarithmic mean temperature difference (LMTD), one typical equation used to get the heat transfer rate (Q) for a double pipe heat exchanger is as follows:

$$Q = U * A * \Delta T_{lm} \quad \dots (2)$$

Where:

- Q is the heat transfer rate (in watts or BTUs per hour).
- U is the overall heat transfer coefficient (in watts per square meter per Kelvin or BTUs per hour per square foot per Fahrenheit).
- A is the heat transfer surface area (in square meters or square feet).

ΔT_{lm} is the logarithmic mean temperature difference (in Kelvin or Fahrenheit).

Logarithmic Mean Temperature Difference

The simple log means temperature difference for counter flow, see figure (2), which is used in equation 2 and 3:

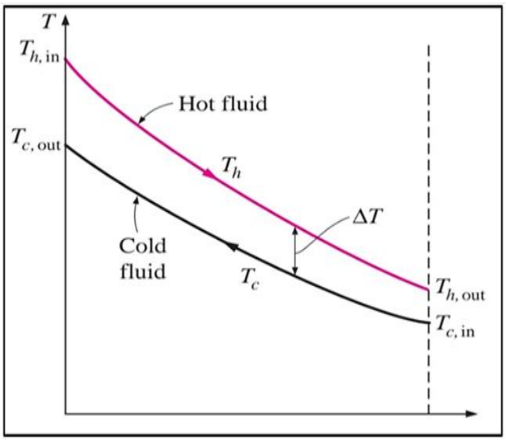

Figure 2. Temperature distribution of the heat exchanger along the heat transfer distance.

$$\Delta Tlm = \frac{(\Delta T1 - \Delta T2)}{ln\ ((\Delta T1)/(\Delta T2))} \quad \text{….(3)}$$

$$H = \frac{Q}{(As\ \ \Delta Tlm\ )} \quad \text{… (4)}$$

$$Nui = \frac{(hi\ \ di)}{K} \quad \text{... (5)}$$

$$Re = \frac{(\rho\ um\ di)}{\mu} \quad \text{… (6)}$$

Friction factor

$$f = \frac{(2 \times \Delta P \times di)}{(\rho\ um)} \quad \text{…(7)}$$

ΔP: is the pressure drop across the test section measured by the manometer.

Thermal performance factor

$$\eta = \frac{\left(\frac{Nu\_(the\ pipe\ within\ the\ addition)}{Nu\_(the\ pipe\ without\ the\ addition)}\right)}{\left(\frac{f\_(the\ pipe\ within\ the\ addition)}{f\_(the\ pipe\ without\ the\ addition)}\right)^{1/3}} \quad \text{… (8)}$$

3. Description of experimental rig

The system shown in figures 3 and 4 is designed to work in two stages. The first stage includes the system work without any addition to the heat exchanger to evaluate the smooth case performance.

Figure 3. Experimental rig photo.

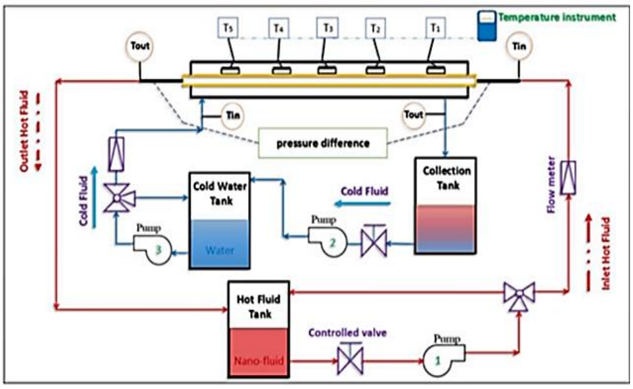

Figure 4. Schematic diagram of the test rig.

The second stage includes adding a right-angle turbulator, as shown in figures 5 and 6. Leaning at an angle of 90°, the distance between one corner and the other is 15 cm, where these corners were connected to the STD as shown in figure 4. The turbulator material used has thermal conductivity of 80.2 (W/m·K) and thermal capacity of 0.449 (J/g·K). Then three flows of hot water were taken (1.5, 2.5, and 3) L/M, with the water flow remaining constant at 6 L/M. The temperature readings were taken for both cases and compared between them.

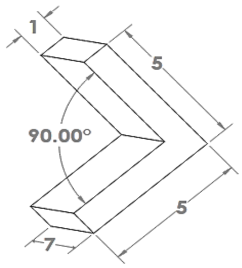

Figure 5. Flow turbulator dimensions in (mm).

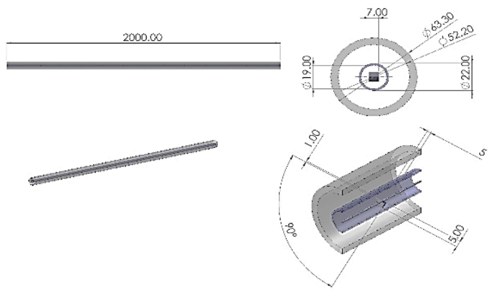

Figure 6. Turbulator arrangements.

The second stage of the heater's operation fixed the temperature at 70 °C and added angles inside the hot water pipe. Temperatures are measured for both the hot and cold sides of fluids; also, the pressure difference along the modified tube is measured.

The device consists of a double-pipe heat exchanger made from concentric copper tubes with a length of 2000 mm as shown in table 1. The copper tube for the inner tube was chosen, having an exterior diameter of 22 mm and an inner diameter of 19 mm. Additionally, the test section's shell side is made up of a polyvinyl chloride (PVC) tube with an outside diameter of 63.3 mm and an inner diameter of 52.2 mm. While the annulus is filled with cooling fluid, the inner tube is filled with heated fluid. The necessary filling was prepared using a sheet of aluminum metal, and the corners were prepared by attaching them to a diameter of 3 mm and notching them on a toothed wire (standard) size (6 M), with a distance between each two corners of 10 cm and a deviation angle between each two corners of 90 degrees.

Table 1. copper properties [12]

Melting temperature	1356 K
Thermal conductivity	391.1 W/(mK)
Coefficient of thermal expansion	17.6×10^{-6} (1/K)
Specific heat	380 J/(kg·K)
Density	8.94 g/cm^3
Electrical resistivity	17.1 nΩ·m
Electrical conductivity	59.1 MS/m
Poisson's ratio	0.33
Young's modulus	120 GPa

4. Tests procedure

The electric heater was turned on and adjusted to a temperature of 70 °C once all the system's components had been fitted. Then, thermocouples were installed and connected to the data recorder so that they could be used to read the temperatures. In order to read the percentage, drop the pressure inside the hot water pipe, and install the electrodes of the pressure difference reading apparatus at two points. The first at the entrance and the second at the pipe's actual outlet, the cold and hot water pumps were then turned on to adjust the cold-water discharge rate and fix it at 6 L/m.

In terms of the hot water, three discharge rates (1.5, 2.5, and 3) L/m were kept, and information on temperature and pressure was kept at each discharge rate. both temperatures.

5. Results and Discussion

Figures (7), (8), and (9) show the variation of temperature along the hot pipe surface for the smooth and modified heat exchanger. In the case of adding the turbulators, it was found that the rate of heat transfer was better than in the smooth case. This is due to the increment in surface area and fluid flow behavior change, which resulted in swirling events that intercepted the liquid, which gave a longer time for heat exchange and energized the boundary layer. Figure (10) shows the downward pressure with the best flow rate for both cases. It was observed that the downward pressure increased as the flow rate increased in the first case, but in the second case, it almost doubled when adding the turbulator. Figure 11 represents the relationship between the friction factor and Reynolds number for the two cases, which is directly related to the pressure drop along the pipe. Figure (12) shows the relationship between the Nusselt number and the Reynolds number. It is clear that the Nusselt number increased with the Reynolds number for both cases at this region of flow, and the modified heat exchanger is the better case due to the enhancement in heat exchange. Figure 13 represents the relationship between the Reynolds number and the performance factor for the smooth and modified heat exchangers. It is concluded that the performance factor for the modified case is the best one due to the heat transfer enhancement that occurred in the pipes.

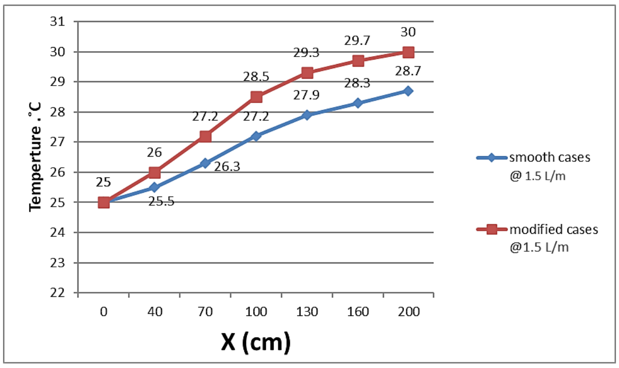

Figure 7. Temperature distributions along the hot pipe surface for a smooth and modified heat exchanger at 1.5 l/m.

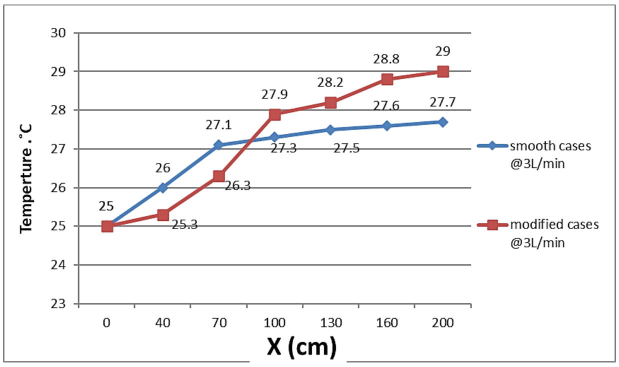

Figure 8. Temperature distributions along the hot pipe surface for a smooth and modified heat exchanger at 3 l/m.

When the temperatures of the hot fluid and the cold fluid cross over one another over the length of the heat exchanger, this phenomenon is referred to as "temperature crossing" in the context of a twin pipe heat exchanger with a right-angle turbulator. This indicates that the temperature of the cold fluid eventually surpasses that of the hot fluid inside the heat exchanger. This behavior is a result of the turbulator's improved heat transfer. A double pipe heat exchanger normally consists of two concentric pipes, the inner pipe carrying the hot fluid (often at a higher temperature) and the outer pipe carrying the cool fluid (typically initially at a lower temperature) in an annular gap between them. In the annular space, the right-angle turbulator is placed to disturb the flow and improve heat transfer. The cold fluid first enters the annular region at a lower temperature than the hot fluid in the inner pipe at the heat exchanger's input. In a conventional two pipe heat exchanger, this temperature differential is what one would anticipate. In the cold fluid, the right-angle turbulator fosters turbulence. The efficiency of heat transfer from the hot fluid to the cool fluid is improved by turbulence, which raises the heat transfer coefficient.

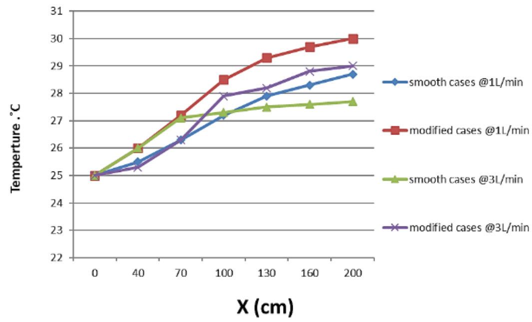

Figure 9. Temperature comparisons for a smooth and modified heat exchanger at 1.5 l/m and 3 l/m.

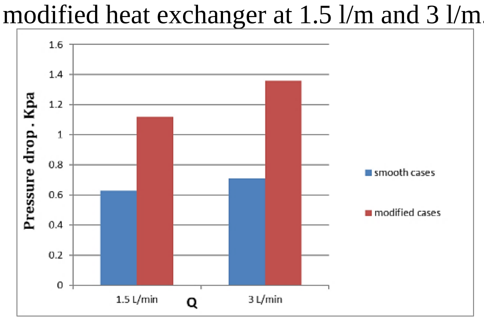

Figure 10. Pressure drop comparisons for a smooth and modified heat exchanger at 1.5 l/m and 3 l/m.

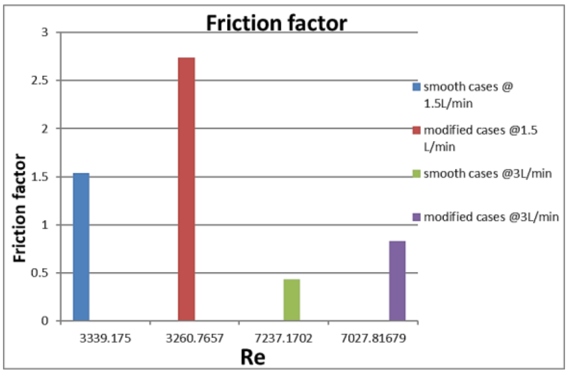

Figure 11. Relationship between the friction factor and Reynolds number for the two cases.

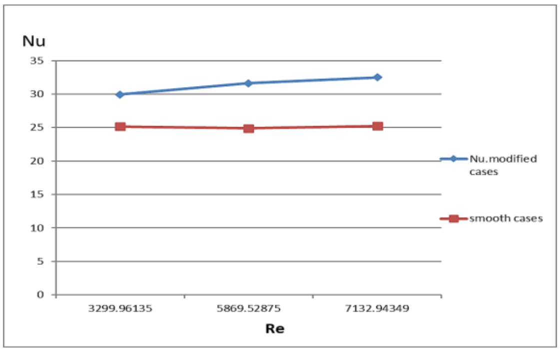

Figure 12. Relevance of the Reynolds number and Nusselt number for smooth and modified heat exchanger.

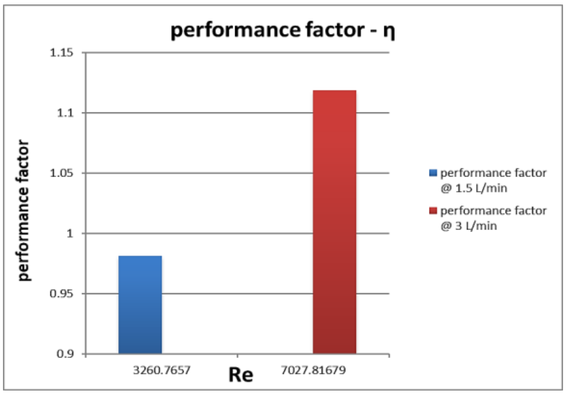

Figure 13. The relevance between the Performance factor and Reynolds number for two flow rates is the best heat transmission.

The results obtained were compared with previous research [14], which represents heat transfer improvement in a double pipe heat exchanger by means of perforated turbulators. Through this work, it is observed that the amount of Nusselt number increases with the increase in Reynolds number, as the value of the Reynolds number, as found in the current work, reached 5869. Nusselt number 25 for the smooth case and 31 for the modified case, while in the previous research, it reached 27, and this indicates the accuracy of the solution compared to previous research.

6. Conclusions

1. The best value of the water flow rate for the smooth heat exchanger was 1.5 lt/min, which introduces a Nusselt number of 25.1558.
2. For the modified heat exchanger, the best volume flow rate of the hot fluid was 3 lt/min, which gives a Nusselt number of 32.48.
3. It is concluded that the improvement process by adding a right-angle turbulator has a significant impact on the Nusselt number.

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