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Research Article

Experimental Assessment of the Alternating Flattened Tubes in a Double-Pipe Heat Exchanger

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ABSTRACT

This study experimentally investigates the thermal-hydraulic performance of alternating flattened tubes in a double-pipe heat exchanger. The primary objective is to evaluate the effects of geometric modifications, including pitch length and degree of flattening, on heat transfer enhancement and pressure drop compared to conventional circular tubes. Four tube configurations were examined: alternating flattened tubes with pitch lengths of 12 cm and 28 cm (flattening of 1 cm), a completely flattened tube (100 cm pitch length, 1 cm flattening), and an alternating flattened tube with a higher flattening degree of 1.2 cm (pitch length 12 cm). Experiments were conducted over a range of Reynolds numbers, and the Nusselt number and pressure drop were measured and analyzedc. The results demonstrate that alternating flattened tubes significantly improve heat transfer compared to circular tubes. The AF_12_1 tube (12 cm pitch, 1 cm flattening) exhibited the highest Nusselt number, achieving 1.9 times that of the circular tube. However, this enhancement was accompanied by an increase in pressure drop, which was 1.22 times higher than the circular tube at higher Reynolds numbers. Increasing the pitch length or reducing the degree of flattening decreased both heat transfer and pressure drop. The completely flattened tube showed a more modest improvement (1.33 times the Nusselt number) with the lowest pressure drop increase. Overall, the alternating flattened tube with shorter pitch length and moderate flattening provides the best thermal performance at the cost of higher flow resistance.

1. Introduction

The topic of increasing the efficiency of heat exchangers is considered one of the new subjects in the field of heat transfer and fluid mechanics, and the research conducted in this area has taken various forms and methods. In general, these studies can be categorized into numerical simulations and experimental forms. The basis for enhancing heat transfer in the conducted studies is the creation of turbulent flow, which is achieved either by adding nanofluid to the desired fluid or by altering the cross-section of the pipes [1-3].

When the fluid flow passes over the surface, the velocity boundary layer begins to form and grow. Similarly, it can be stated that the thermal boundary layer forms when the temperature of the fluid flow passing over the surface differs from the temperature of the surface itself. Fluid particles that come into contact with the surface reach thermal equilibrium with the surface, and their temperature becomes equal to the surface temperature. In this case, the particles at the surface exchange energy with the adjacent particles, and the temperature gradient develops across the fluid. The region of the fluid where the mentioned temperature gradients are present is called the thermal boundary layer. As one moves away from the leading edge, the effect of heat transfer penetrates more into the free stream, resulting in the growth of the thermal boundary layer. As the boundary layer progresses within the fluid, the temperature of the particles within

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the boundary layer approaches the temperature of the pipe surface, resulting in a decrease in the rate of heat transfer. In this method, by changing the geometry of the tube, the balanced form of the boundary layer is disrupted, and particles with a greater temperature difference come into proximity with each other and the wall surface of the tube, increasing the rate of heat transfer [4-6].

Wang et al. [7] conducted a numerical study on the impact of twisted elliptical cross-sections on heat transfer rates and fluid flow patterns in heat exchangers. The Reynolds number range was selected between 50 and 2000, with tube twist rates of 0.17, 0.25, 0.33, 0.5, and a tube surface flattening of 1.2, 1.4, 1.63, 1.8, and 2 centimeters. They found that the heat transfer rate and pressure drop in the twisted oval tube increased compared to the smooth tube. They also stated that at Reynolds 500, the flow transitions from laminar to turbulent. It was also determined that the twisted elliptical tube performs well due to the secondary flow effect compared to the smooth tube. Arad et al. [8] conducted a numerical study on the impact of a new form of tube, titled the three-channel twisted tube, on enhancing heat transfer in shell-and-tube heat exchangers. Their goal in this research was to examine the effect of the strip width-to-tube diameter ratio at values of 0.1, 0.25, 0.34, and 1, as well as the arrangement of the channels in a belly-to-belly (convex parts facing each other) or belly-to-neck (a concave part facing a convex part) configuration on the heat transfer rate. The Reynolds number range examined in this study was between 4000 and 16000. In order to examine the impact of the mentioned parameters, the obtained results were compared with a smooth tube without any modifications. Their results showed that the width of the strips has a direct relationship with the heat transfer rate and the friction coefficient. Chen et al. [9] investigated the effect of helical pipes with hexagonal inlets on heat transfer in the context of CFD simulation. In this study, they examined the parameters of resistance and flow stability in relation to the hexagonal cross-section of the helical tube. In addition, they studied the parameters of geometric features (pitch and depth of the groove), fluid properties, and the Reynolds number. The base heat transfer fluid in their study was water with a density of 977.8 kg/m^3, a specific heat capacity of 4178 J/kg·K, a heat transfer coefficient of 0.667, and a viscosity of 0.4061. Herion et al. [10] conducted a numerical study on the effect of alternating the internal and external cross-sectional areas of two concentric tubes on the heat transfer rate and the flow generated. The inlet fluid temperature to the system is considered to be 321.3 Kelvin, with an inlet velocity of 0.09 meters per second and a Reynolds number of 388. Finally, the results showed that under constant temperature conditions on both the inner and outer walls, the heat transfer performance of the variable cross-section double pipe heat exchanger increases compared to a double-pipe heat exchanger with a smooth circular cross-section geometry. They also stated that using a 0.125 longitudinal phase displacement between the inner and outer walls results in a 43% increase in the thermal performance of the variable cross-section double-pipe heat exchanger.

Zhou et al. [11] conducted an experimental study on the impact of using twisted elliptical tubes on the heat transfer rate and pressure drop in shell-and-tube heat exchangers. Their goal in using this tube form was to increase the heat transfer rate on the tube side and reduce the pressure drop on the shell side. The base heat transfer fluid on the tube side was hot water, and on the shell side, it was cold water, with a Reynolds number range of 2000 to 50000. The diameter of the circular tube in this study was 25 millimeters, the major diameter of the twisted elliptical tube was 29 millimeters, the minor diameter was 19.5 millimeters, and the length of both tubes was 230 millimeters. Their results showed that both the heat transfer rate and pressure drop in the twisted elliptical tube were higher than the corresponding values in the straight tubes. They also reported that the heat exchanger with the twisted elliptical tube would perform better at low flow rates in the tube and high flow rates in the shell.

Kumar et al. [12] conducted a numerical study on the effect of changing the cross-section of the tube from circular to alternating elliptical on the heat transfer rate of shell-and-tube heat exchangers. In this simulation, the inlet temperature values on the tube side were 23.9, 25, and 37.8 degrees Celsius, and on the shell side, they were 33.9, 95, and 199 degrees Celsius. The flow rates were selected in the range of 5.52 to 27.8 for the shell and 18.5 to 69.8 kg/s for the tube. Finally, by comparing the results of heat transfer in the shell-and-tube heat exchanger with a circular cross-section to an elliptical cross-section, it was stated that, firstly, changing the cross-section from a circular form to an elliptical one increases both the heat transfer rate and the pressure drop. Secondly, based on the parameter of the major to minor diameter ratio, when this ratio is less than one, compared to the case when it is greater than one, an increase of 20.2% in heat transfer on the shell side and 14.5% on the tube side has been observed. Vaezi et al. [13] conducted a numerical study on the effect of alternating elliptical cross-sections on the heat transfer rate in shell-and-tube heat exchangers.

They conducted this study assuming laminar and incompressible flow in the double-pipe heat exchanger. The base heat transfer fluid in this simulation was chosen to be water with a Prandtl number of 6.2 and a heat transfer coefficient of 0.6. Their results showed that for all values of the aspect ratio, the shell-and-tube heat exchanger with an alternating elliptical cross-section has a better heat transfer performance compared to the circular cross-section. With the Reynolds number held constant and the aspect ratio increasing, the performance coefficient increases until it reaches its maximum value and then starts to decrease. In the end, they stated that for Reynolds numbers less than 600, the performance improvement factor will be less than one for all values of aspect ratio.

Deepankar et al. [14] examined and designed a new layout for the shell-and-tube heat exchanger. This model includes seven copper tubes, each with an outer diameter of 20 millimeters, an inner diameter of 17 millimeters, and a length of 600 millimeters, placed inside a shell with an inner diameter of 90 millimeters and an outer diameter of 110 millimeters made of stainless steel. The copper tubes are held by six straight and spiral aluminum baffles, and the angle of the spiral baffle varies from 0 to 30 degrees. They stated that helical baffles have a greater impact on improving the performance of shell-and-tube heat exchangers compared to straight baffles, as the heat transfer coefficient and pressure drop increase with the increase in the twist angle of the baffle plates. Jafari Nasr et al. [15] analyzed heat exchangers using twisted tubes in a closed cycle. In fact, they used a numerical algorithm to examine the effects of replacing straight tubes with twisted tubes in heat exchangers in terms of heat transfer coefficients, mass transfer coefficients, and other thermal parameters. They found that using a twisted tube as one of the methods to enhance heat transfer results in a 45% increase in the heat transfer coefficient in the tube. Zhou et al. [16] investigated the effect of perforated baffles on the heat transfer rate of shell-and-tube heat exchangers. They found that the flow changes to a developed form after passing through the first hole, and the liquid velocity gradually increases. Near the baffle plate, a jet flow is created, as well as a secondary flow on the sides of the baffle plate. With the formation of the jet flow and secondary flow, the thickness of the boundary layer decreases, and consequently, the heat transfer rate increases.

Despite extensive studies on twisted and helically shaped tubes, the experimental investigation of alternating flattened tubes—where circular and flattened sections are arranged periodically—remains limited. Therefore, this study experimentally evaluates the thermal-hydraulic performance of alternating flattened tubes with different pitch lengths (12, 28, and 100 cm) and flattening degrees (1 and 1.2 cm) in a double-pipe heat exchanger. The Nusselt number and pressure drop are measured and compared with a standard circular tube to determine the effectiveness of the proposed geometries

2. Experimental Setup and Mathematical Formulation

The device in question is a two-pipe heat exchanger, including a heat transfer coil between hot oil and cold water, a hot oil boiler, and a circulation circuit, as shown in the system flow diagram in Fig. 1. The oil circulation process in the system can be divided into five distinct sections: the oil circulation driving force, the bypass valve, the double-pipe heat exchanger, the indirect oil heater, and the cooling shell of the heat exchanger.

Fig. 1. Experimental setup.

The pipes used in the experiment are made of copper, with a size of 8.5 inches and a thickness of 0.63 millimeters. On both sides of the collector, two brass tees and a brass valve with a size of 8/5 inches have been installed for the installation of a pressure gauge, thermocouple, and thermometer. The brass fittings are connected on one side to the copper pipe with a nut and washer connection, and on the other side to the pump and tank with a high-pressure transparent hose of size 8/5 inches and gas clamps. Two branches on either side of the collector, at the inlet and outlet of the device, are connected to a mercury manometer via transparent pneumatic hoses and will indicate the pressure difference across the tube (Fig. 2). The length of the manometer tubes is designed in such a way that by increasing the length of the corresponding hose, it allows for the measurement of pressure differences in the range of 0 to 40 centimeters. Two additional branches on the sides of the collector have also been used for installing thermocouples. In this experiment, pipes with circular, flat, and alternately flat cross-sections with different pitch lengths and degrees of flattening have been examined.

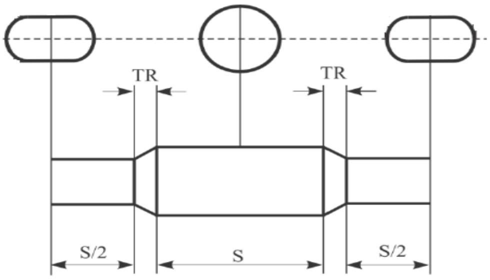

Fig. 2. Geometric parameters of the periodically flat tube.

Table 1. Geometric parameters of the alternating flattened tube.

Tube	S (mm)	TR (mm)	n
C	1370	0	0
AF_12_1	120	40	6
AF_12_1.2	120	40	6
AF_28_1	280	40	3
F	1000	-	1

In order to examine the heat transfer rate in the circular tube, the thermal energy balance on the shell and tube sides of the heat exchanger must be analyzed (Fig. 3).

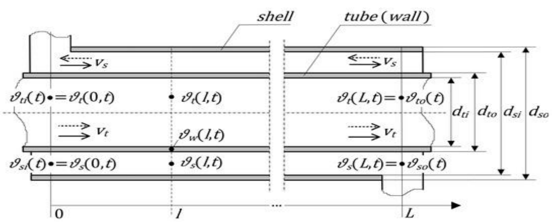

Fig. 3. Schematic of a two-tube heat exchanger based on the differential control volume of the fluid inside the tube.

To examine the thermal energy balance in the shell and tube, the flow rates of the hot and cold fluids must first be calculated according to the following equations:

$$\dot{m}_{Oil} = \frac{\rho V_{oil}}{t_{oil}} \tag{1}$$

$$\dot{m}_{Water} = \frac{\rho V_{Water}}{t_{oil}} \tag{2}$$

In the above equations, V_{oil} and V_{Water} represent the volume of oil and the volume of water in a container with a specified capacity, and t_{oil} and t_{Water} represent the time required to fill a specified volume with oil and water, respectively. Next, to examine the energy balance in the exchanger, based on the efficiency relationship for double-pipe exchangers:

$$\eta = \frac{Q_H - Q_C}{Q_H} = \left[\frac{\dot{m}_O Cp_O (T_{H_{in}} - T_{H_{out}}) - \dot{m}_W Cp_W (T_{C_{out}} - T_{C_{in}})}{\dot{m}_O Cp_O (T_{H_{in}} - T_{H_{out}})}\right] \times 100 \tag{3}$$

To examine the above relationship, the water and oil with different flow rates inside the heat exchanger are assessed, and after the system reaches stability, the inlet and outlet temperatures of the oil and water are recorded.
By examining and comparing the results, it was determined that the efficiency of the tested double-tube heat exchanger is less than 11 percent.
Next, to calculate the heat transfer rate, we will use the following equation:

$$Q = \dot{m} Cp (T_{H_{in}} - T_{H_{out}}) \tag{4}$$

On the other hand, the heat transfer rate can also be calculated using the following equation:

$$Q = hA\Delta T_{LMTD} \tag{5}$$

In this relation, the value of ΔT_{LMTD} is as follows:

$$\Delta T_{LMTD} = \frac{[T_{H_{in}} - T_{C_{out}}] - [T_{H_{out}} - T_{C_{in}}]}{ln \frac{[T_{H_{in}} - T_{C_{out}}]}{[T_{H_{out}} - T_{C_{in}}]}} \tag{6}$$

3. Results

In order to examine the effect of surface geometry changes on the heat transfer rate in the heat exchanger, the Nusselt number versus Reynolds number graph can be used. The Nusselt number versus Reynolds number graph for the mentioned geometry is shown in Fig. 4.

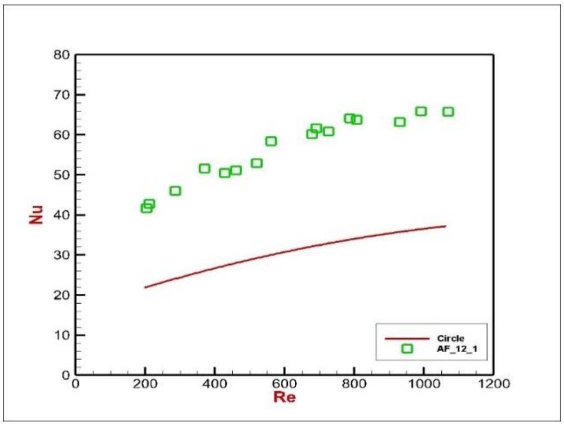

Fig. 4. Nusselt number versus Reynolds number for the AF_12_1 tube.

Considering that the goal of creating flat sections on the tubes is to increase the turbulence in the fluid flow base of heat transfer at low Reynolds numbers and ultimately enhance the heat transfer rate, by comparing the results obtained for the circular tube and the mentioned geometry, it is observed that the Nusselt number for the AF_12_1 tube is 1.9 times the results obtained for the circular tube. In other words, by creating consecutive flat sections in the tube geometry, the chances of particles being near the tube wall increase on one hand, and the turbulence of fluid particles passing thru consecutive flat and circular sections and creating secondary flows inside the tube, as well as the water flow on the shell side affecting the geometric fluctuations of the tube's outer surface and increasing its turbulence on the other hand, will lead to an increase in heat transfer.

The pressure drop chart as a function of the Reynolds number for the mentioned geometry is shown in Fig. 5.

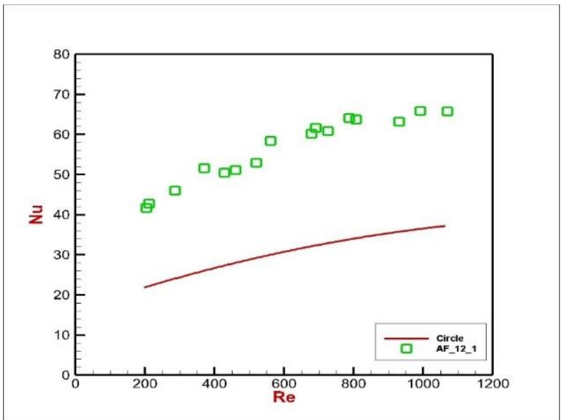

Fig. 5. Pressure drop versus Reynolds number for AF_12_1 tube.

The pressure drop in the mentioned pipe can be examined from two perspectives: Firstly, in the AF_12_1 pipe, considering the reduction in cross-sectional area along the flow path, the velocity of the heat transfer fluid particles increases along the length of the pipe. According to the Darcy-Weisbach equation, the pressure drop is directly proportional to the square of the velocity, and with the increase in the velocity of the fluid particles, the pressure drop of the heat transfer base fluid will naturally increase. On the other hand, when the fluid encounters geometric fluctuations and passes thru them, the turbulence and agitation of the fluid particles increase, resulting in an increase in pressure drop. By comparing the results obtained from the AF_12_1 tube with the circular tube, we will observe that the pressure drop in the AF_12_1 tube in the Reynolds range of 200 to 400 is 1.2 times and in the Reynolds range of 400 to 1000 is 1.22 times the pressure drop in the circular tube. The reason for this is the increase in velocity at higher Reynolds numbers, which in turn increases the pressure drop.

One of the strategies to increase the heat transfer rate in heat exchangers is to change the geometry and shape of the tube inside the shell-and-tube heat exchanger and to present the optimal possible configuration. In most of the mentioned cases, the issue of increasing thermal efficiency has been accompanied by an increase in the pressure drop parameter, and these two topics have been examined together. In this research, in order to examine the effects of geometry changes on pressure drop and heat transfer, two parameters, pitch length and flattening degree, are investigated, and finally, the results of the tubes are compared.

The geometry of the pipe in this case has a pitch length of 28 centimeters and a flattening amount of 1 centimeter. Therefore, for a pipe length of 1.37 meters, considering a step length of 28 centimeters, a total of 3 steps will be created, including two flat steps and one circular step in a consecutive and alternating manner. The distance between two consecutive circular and flat sections was considered to be 4 centimeters. The Nusselt number versus Reynolds number graph for the mentioned geometry is illustrated in Fig. 6.

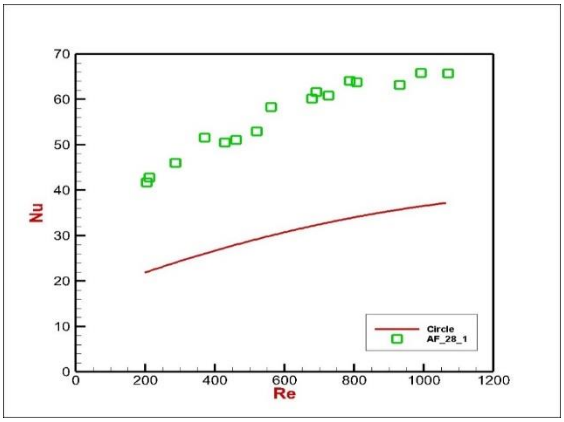

Fig. 6. Nusselt number versus Reynolds number for the AF_28_1 tube.

By comparing the results obtained for the circular pipe and the mentioned geometry, we will observe that the Nusselt number for the AF_28_1 pipe will be 1.8 times that of the circular pipe. The reason for this can be clarified by examining the cross-sectional geometry. Considering that the number of steps in this tube is less compared to the AF_12_1 tube, the fluid particles pass thru fewer geometric oscillations, and consequently, the turbulence and disturbance of the fluid will be less compared to the AF_12_1 tube. This will cause a negligible decrease in the rate of heat transfer increase in this tube. On the other hand, due to the creation of a flat geometry in the tube, the chance of the base fluid particles for heat transfer being positioned near the wall increases, and the heat transfer rate will increase compared to a circular tube.

The pressure drop as a function of the Reynolds number for a pipe with a pitch length of 28 centimeters and a flatness of 1 centimeter is shown in Fig. 7.

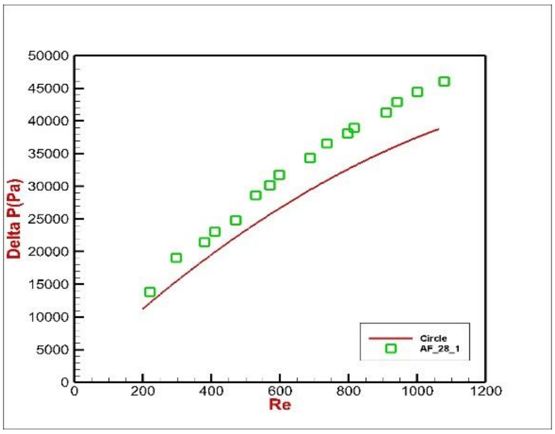

Fig. 7. Pressure drop versus Reynolds number for the AF_28_1 tube.

In this tube, similar to the AF_12_1 tube, the increase in pressure drop along the tube can be attributed to the increase in fluid velocity due to passing through the flat geometry, as well as the increase in turbulence of the fluid particles, which is the basis of heat transfer, due to collisions with geometric fluctuations and passing through them. By comparing the results obtained from the circular tube with the AF_28_1 tube, it can be observed that the pressure drop in the mentioned tube was 1.2 times the pressure drop in the circular tube. Regarding this pipe, it can be noted that since the geometric fluctuations in this pipe decrease compared to the AF_12_1 pipe and the flow travels a longer path at a constant pitch, the increase in pressure drop for the AF_28_1 pipe compared to the AF_12_1 pipe will be reduced. In order to verify the conclusion regarding the AF_28_1 pipe and the effect of increasing the pipe's pitch on heat transfer and pressure drop, the F_100_1 pipe will be examined.

The geometry of the pipe in this case is completely flat with a flattening of 1 centimeter. Therefore, for a pipe length of 1.37 meters, a completely flat geometry with a flat length of 1 meter will be placed inside the heat exchanger.

The Nusselt number versus Reynolds number graph for a completely flat tube with a flatness of 1 centimeter is presented in Fig. 8.

In the completely flat tube, unlike the AF_12_1 and AF_28_1 tubes, the number of geometric oscillations in the path of the fluid particles that form the heat transfer base decreases, and the flatness step length increases. Accordingly, by creating a flat section along the length of the tube, the cross-sectional area is reduced, and the likelihood of particles being positioned near the wall increases, which in turn enhances the heat transfer rate compared to the circular tube. On the other hand, with the reduction in the number of geometric oscillations of the tube compared to the two tubes AF_12_1 and AF_28_1, the increase in heat transfer compared to them will be less and 1.33 times that of the circular tube.

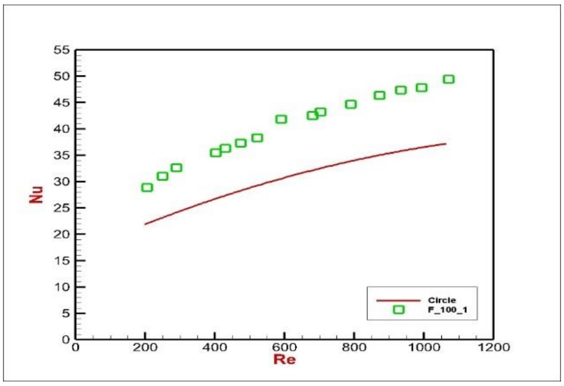

Fig. 8. Nusselt number versus Reynolds number for F_100_1 tube.

The pressure drop chart as a function of Reynolds number for a completely flat tube with a flattening of 1 centimeter is shown in Fig. 9.

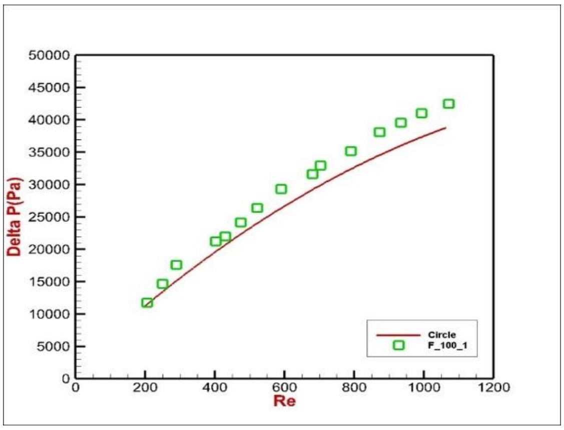

Fig. 9. Pressure drop versus Reynolds number for F_100_1 tube.

In the completely flat tube, such as the AF_12_1 and AF_28_1 tubes, the fluid particles of the heat transfer base will pass through the change in cross-sectional geometry from circular to flat upon entering the tube. As the fluid passes through this geometric fluctuation, the turbulence and particle agitation increase, which in turn raises the pressure drop. On the other hand, with the reduction of the flow cross-section similar to the previous two geometries, the fluid velocity in the pipe path has increased, which, according to the Darcy-Weisbach equation, will lead to an increase in the pressure drop of the fluid on both sides of the heat exchanger. On the other hand, considering that the number of geometric oscillations of this pipe is less compared to the AF_12_1 and AF_28_1 pipes, the turbulence of the fluid particles will be less than in the previous two geometries, resulting in a noticeable reduction in the increase in pressure drop. By comparing the results obtained from the circular tube with the AF_100_1 tube, it can be observed that the pressure drop in the Reynolds range of 200 to 600 is 1.05 times and in the range of 600 to 1000, it is 1.1 times the pressure drop in the circular tube. The reason for this is the increase in speed at higher Reynolds numbers, which consequently increases the pressure drop. Up to this point, the effect of pitch length on pressure drop and heat transfer has been examined, and it was observed that with an increase in the number of pitches and a decrease in their length, both heat transfer and pressure drop increase. Now, in order to examine the effect of the degree of flattening on the aforementioned parameters, the AF_12_1.2 tube is investigated. The geometry of the pipe in this case has a pitch length of 12 centimeters and a flattening amount of 1.2 centimeters. Therefore, with a pipe length of 1.37 meters, a total of 6 steps will be created, including three flat steps and three circular steps, arranged alternately and consecutively. The distance between two consecutive circular and flat sections was considered to be 4 centimeters.

Note that in order to create flat sections on the tube, the mentioned tube is placed between two press jaws and compressed to the size of a metal piece with a cubic geometry of a specified width. Therefore, for pipes with a flattening of 1 centimeter, the mentioned piece has a width of 1 centimeter, and for pipes with a flattening of 1.2 centimeters, the mentioned piece has a width of 1.2 centimeters, and its flattening will be less compared to the previous pipes. The Nusselt number versus Reynolds number graph for a flat-plate tube with a pitch of 12 centimeters and a flattening of 1.2 centimeters is shown in Fig. 10.

Similar to the AF_12_1 tube, this tube also has three circular steps and three flat steps arranged alternately, with the difference being that its flatness is less. Therefore, one of the reasons for the increased heat transfer rate compared to the circular tube is the reduction in the cross-sectional area through which the heat transfer fluid particles pass. In other words, by creating a flat geometry, the likelihood of heat transfer fluid particles being near the wall will increase, and consequently, the rate of heat exchange will also increase. On the other hand, by creating geometric fluctuations along the pipe, the turbulence and agitation of the fluid flow, which is the basis of heat transfer, will increase on both the shell and tube sides, resulting in an increase in the heat exchange rate. However, in this tube, due to the lower flattening rate compared to the AF_12_1 tube, which is 1.2 centimeters, the reduction in the cross-sectional area through which the fluid flows and the chance of fluid particles being near the wall of the tube is less than in the AF_12_1 tube, and ultimately, the increase in heat transfer compared to the mentioned tube will be less.

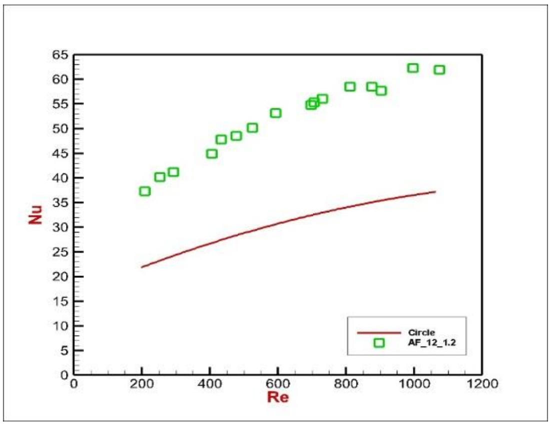

Fig. 10. Nusselt number versus Reynolds number for the AF_12_1.2 tube.

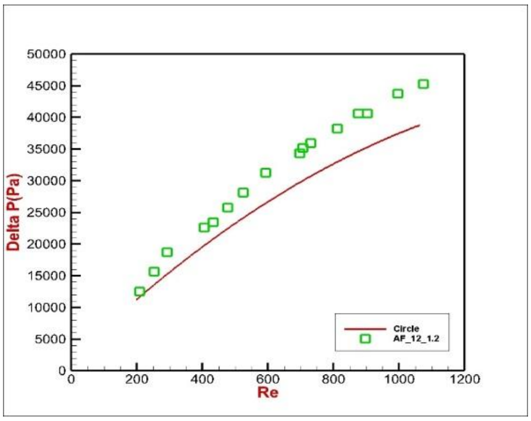

Fig. 11. Pressure drop versus Reynolds number for the AF_12_1.2 tube.

On the other hand, due to the reduction in the degree of flattening, the intensity of the geometric oscillation from the flat cross-section to the circular cross-section, compared to the AF_12_1 tube, has decreased, and consequently, the turbulence of the fluid particles due to passing through the sections will also decrease compared to the AF_12_1 tube. By comparing the results obtained from this tube with the circular tube, it can be observed that the Nusselt number of this tube was 1.7 times that of the circular tube. The pressure drop chart as a function of the Reynolds number for an alternating flat tube with a pitch length of 12 centimeters and a flattening of 1.2 centimeters is illustrated in Fig. 11.

In this pipe, similar to the AF_12_1 pipe, the increase in pressure drop can be attributed to two reasons: Firstly, in the AF_12_1.2 pipe, due to the reduction in cross-sectional area along the flow path, the velocity of the heat transfer fluid particles increases along the length of the pipe. According to the Darcy-Weisbach equation, the pressure drop is directly proportional to the square of the velocity, and with the increase in the velocity of the fluid particles, the pressure drop of the heat transfer base fluid will naturally increase. On the other hand, when the fluid encounters geometric fluctuations and passes through them, the turbulence and agitation of the fluid particles increase, resulting in an increase in pressure drop. As mentioned, in the AF_12_1.2 tube, the degree of flattening decreases compared to the AF_12_1 tube. With the reduction in the degree of flattening, the increase in the velocity of the heat transfer fluid particles due to passing through the flat geometry compared to the AF_12_1 tube will decrease, and consequently, the increase in pressure drop will also significantly decrease. On the other hand, with the reduction in the degree of flattening of the cross-section, the intensity of geometric fluctuations in the path of the heat transfer fluid particles decreases, and consequently, the turbulence resulting from the fluid passing through the mentioned sections decreases compared to the AF_12_1 tube, leading to a reduction in the increase in pressure drop for this reason. By examining and comparing the results obtained from the pressure drop graph for the AF_12_1.2 tube and comparing it with the circular tube, it can be observed that the pressure drop in the periodically flat tube with a pitch length of 12 centimeters and a flattening of 1.2 centimeters will be 1.18 times the pressure drop in the circular tube.

It can be observed that for all pipes, whether circular or with alternating flat and completely flat sections with varying pitch lengths and degrees of flattening, the heat transfer rate decreases with an increase in the Reynolds number. In simpler terms, in heat transfer systems, with an increase in fluid flow rate, the intensity of the increase in heat transfer rate decreases. On the other hand, as mentioned in Chapter Two, the main goal of changing the geometry of the tubes in shell-and-tube heat exchangers is to increase the turbulence in the flow within the system and ultimately enhance the heat transfer rate. In fact, by making these changes, the fluid flow, which is the basis of heat transfer within the system, is removed from a uniform and steady state, and the mixing of the flow particles increases. As a result, the likelihood of fluid particles moving from the center of the tube toward the tube walls increases, ultimately enhancing heat transfer. On the other hand, on the shell side, as water fluid particles pass over the outer surface of the tube and interact with the geometric fluctuations of the tube surface, the turbulence of the flow increases, and consequently, the heat exchange rate increases. In this context, based on the graph, it can be observed that the highest Nusselt number corresponds to the AF_12_1 pipe, while the lowest value corresponds to the circular pipe.

On the other hand, considering the degree of flattening as constant, with an increase in the pitch length, the Nusselt number decreases, and this decrease will be to the extent that in a completely flat tube, the increase in the Nusselt number has a noticeable drop. The reason for this can be attributed to the reduction of geometric fluctuations present in the path of the heat transfer fluid. Accordingly, in the AF_12_1 tube, due to the numerous changes in the cross-sectional area through which the flow passes, the fluid particles experience more turbulence and mixing, resulting in a significant increase in the Nusselt number. On the other hand, in the completely flat tube, the increase in the mixing of the heat transfer fluid particles is due to passing through the geometric fluctuations related to the inlet and outlet of the fluid into the exchanger, and the amount of heat transfer increase due to this reason is reduced. But in a completely flat tube, the main reason for the increase in heat transfer is the reduction in the cross-sectional area through which the flow passes, which increases the likelihood of particles being near the wall and thus increases the heat transfer rate.

On the other hand, considering a constant pitch length, the amount of heat transfer increases with the increase in the degree of flattening. This phenomenon can be explained as follows: with an increase in the degree of flattening, the cross-sectional area through which the flow passes decreases, and the turbulence and disturbance created in the fluid particles that facilitate heat transfer increase as they pass through these sections, ultimately leading to an increase in heat exchange. On the other hand, with the reduction of the flow cross-section, the likelihood of particles being near the wall of the tube increases, and consequently, the heat transfer rate will also increase.

4. Conclusions

In this research, the effect of alternating flattened tubes on heat transfer and pressure drop in a double-pipe heat exchanger was experimentally evaluated. Four geometries with varying pitch lengths and degrees of flattening were tested and compared against a standard circular tube. The results clearly indicate that introducing periodic flattened sections disrupts the thermal boundary layer, increases flow turbulence, and enhances the proximity of fluid particles to the tube wall, all of which contribute to improved heat transfer. Among all configurations, the AF_12_1 tube demonstrated the highest heat transfer enhancement, achieving a Nusselt number 1.9 times that of the circular tube, due to the combined effects of reduced cross-sectional area and increased geometric oscillations along the flow path. Regarding the effects of pitch length and flattening degree, it was observed that reducing the pitch length (increasing the number of flattened sections) increases both heat transfer and pressure drop. Conversely, increasing the pitch length or decreasing the flattening degree reduces these parameters. The completely flattened tube (F_100_1) showed the lowest enhancement, confirming that alternating geometries are more effective than a single continuous flattened section. Although the pressure drop increased in all modified tubes, ranging from 1.05 to 1.22 times that of the circular tube, the overall thermal performance suggests that alternating flattened tubes are a viable passive technique for heat transfer enhancement. Future work should focus on optimizing the pitch length and flattening ratio to maximize thermal efficiency while minimizing pressure drop penalties.

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