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Research Article

Experimental Study of Heat Transfer in Alternating Elliptical Tubes in a Double-Tube Heat Exchanger

Ahmad Reza Sajadi [a*], Mohsen Hosseini [b]

[a] *Department of Mechanical Engineering, Shahrekord University, Shahrekord, Iran*

[b] *Department of Mechanical Engineering, Shahrekord University, Shahrekord, Iran*

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ABSTRACT

In this paper, the thermal and hydrodynamic behavior of alternating elliptical tubes in a double-pipe heat exchanger has been introduced and examined. Additionally, the effect of pitch length and degree of flattening on the rate of forced convective heat transfer and pressure drop in alternating elliptical tubes has been studied and compared with the results of circular tubes. The experiments were conducted in the Reynolds range of 200 to 1200, and the results were calculated using the statistical method of the confidence interval with 95 percent accuracy. The fluid tested was water and oil as the heat transfer base. Experimental results show that the heat transfer rate and pressure drop of alternating elliptical tubes are higher than those of circular tubes, and with a decrease in pitch length and an increase in flattening at a constant tube length, both heat transfer and pressure drop increase. It was also determined that the heat transfer rate of the alternating elliptical tube with the same flatness and a pitch of 120 millimeters is approximately 1.05 and the pressure drop is approximately 1.03 times that of the alternating elliptical tube with a pitch of 280 millimeters. Additionally, the heat transfer rate of the alternating elliptical tube with the same pitch and a flatness of 10 mm was approximately 1.15, and the pressure drop was approximately 1.11 times that of the alternating elliptical tube with a flatness of 12 mm. Finally, it was concluded that the heat transfer rate of the alternating elliptical tube with pitch and flattening of 10-120 mm, 10-280 mm, and 12-120 mm is approximately 2.30, 2.20, and 2 times, respectively, and the pressure drop is 1.33, 1.29, and 1.2 times that of the circular tube. Subsequently, by comparing and examining the criterion for the increase in efficiency ratio for elliptical and circular tubes, it was determined that the performance coefficient of the elliptical tubes is higher than that of the circular tubes. Among them, the elliptical tube with a pitch and flattening of 10-120 mm has the best performance.

1. Introduction

The heating and cooling of a system by fluid is important in many industries. Cooling and heating systems are designed based on various heat transfer methods. Given the world's limited energy resources and the rising energy use costs in recent years, research in energy conservation to reduce its consumption in various processes has gained attention. Numerous studies have been conducted on heat transfer in heat exchangers. Improving heat transfer in heat exchangers leads to a reduction in the size of the exchanger, and the use of more compact and higher-efficiency heat exchangers results in lower energy consumption and operating costs. The use of tubes with specific geometries, due to their structure and high heat transfer coefficient, is considered one of the methods to increase the heat transfer coefficient. Alternating elliptical tubes are a type of tube with a specific geometry and cross-section that have numerous applications in the oil industry, including heating crude oil, heating liquefied natural gas, heating

* **Corresponding author.**
E-mail address: *ahmad.r.sajadi@gmail.com*

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bitumen, air conditioning processes, cooling systems, and food and dairy production processes.

After the fluid passes through the tube, a temperature profile is formed and a boundary layer is created near the surface of the tube. Subsequently, the fluid within the boundary layer approaches the temperature of the tube surface, and this action results in a reduction in heat transfer. In methods where the issues and analyses are based on changing the geometry of the pipe, the goal is to disrupt the boundary layer and enhance fluid mixing, which increases the heat transfer rate.

Andrade et al. [1] experimentally investigated the heat transfer characteristics and pressure drop of internal flow in corrugated tubes. Their experiments were conducted in laminar, transitional, and turbulent flow regimes, and the Reynolds number ranged from 429 to 6212 under adiabatic and diabatic flow conditions. Their results showed that the friction factor of the wavy tubes demonstrated a smoother and more uniform transition compared to the smooth tube, despite the adiabatic and diabatic friction factors for the wavy tubes being similar to the transition from laminar to turbulent flow. It was also determined that the wavy tubes are more efficient and effective under the transitional flow regime. Additionally, the highest increase in heat transfer occurred at a Reynolds number of 2000, particularly with an increase in the Nusselt number for the corrugated tube with a twist pitch of 4.7 to 6 mm and for the corrugated tube with a twist pitch of 3.8 to 12 mm.

Chen et al. [2] investigated the heat transfer performance and turbulent flow characteristics in asymmetric wavy tubes both numerically and experimentally. The results showed that the shell section of the asymmetric wavy tubes had a higher Nusselt number, friction factor, and performance evaluation index, respectively 1.7, 1.3, and 2.6, compared to the tube section of the asymmetric wavy tubes. The results also indicate that the turbulent kinetic energy in the shell section is significantly higher. Furthermore, when the Reynolds number is less than 45000, the performance evaluation criteria values are above one, indicating that the proper performance and efficiency of the asymmetric wavy tubes occur only at a specific Reynolds number.

Chen et al. [3] investigated the heat transfer performance in a cross-corrugated heat exchanger for molten salt at Reynolds numbers from 300 to 60000 and Prandtl numbers from 11 to 27. They also evaluated the effects of the Reynolds and Prandtl numbers on the thermohydraulic thermal behavior in laminar, transitional, and turbulent flows. The results showed that the tube with cross grooves significantly increased the molten salt heat transfer performance and reduced the critical Reynolds numbers for the transition from laminar to turbulent flow. It was also determined that for laminar and transitional flow, the increase in heat transfer compared to the smooth tube varies with changes in the Reynolds and Prandtl numbers. While for turbulent flow, the effect of the increase is almost independent of the Reynolds and Prandtl numbers.

Córcoles et al. [4] numerically investigated the effect of groove shape on heat transfer performance in wavy tubes for turbulent flow with a Reynolds number of 25000. Their results showed that considering the same diameter in all cases examined, the pressure drop and Nusselt number decreased linearly with the increase in pitch-to-diameter ratio, and both parameters increased linearly with the increase in height-to-diameter ratio, stiffness index, and groove shape factor. Additionally, the numerical results showed that within the tested range, the Nusselt number and friction factor increased almost linearly with the groove shape factor (compared to the smooth tube and without considering the tube diameter).

Karami et al. [5] experimentally investigated the heat transfer characteristics and pressure drop of multi-walled carbon nanotube nanofluid/heat transfer oil flows in horizontal wavy tubes under uniform wall temperature conditions. The obtained data indicate an increase in the heat transfer rate as a result of using nanofluids compared to the base fluid flow. It was also concluded that the waviness of the tubes reduced the heat transfer rate at low Reynolds numbers. Finally, the greatest increase in the heat transfer rate was observed for the flow regime where the smooth tube is in the transition range and the wavy tube has reached the turbulent flow (i.e., Reynolds 1000 to 3000).

Kaood et al. [6] numerically investigated the thermal and hydraulic characteristics of turbulent water flow in a transverse corrugated pipe, with groove orientations (inner/outer) and various groove shapes (triangular, curved, rectangular, and trapezoidal) at Reynolds numbers from 5000 to 61000 under a constant heat flux boundary condition. Their results showed that the direction and shape of the grooves have significant effects on heat transfer in the form of the Nusselt number and pressure drop in the form of the friction factor. In addition, the average Nusselt number for the groove shapes (internal) of trapezoidal, rectangular, curved, and triangular shapes are 61.52%, 50.12%, 47.82%, and 44.96%, respectively, and are higher than that of the smooth tube.

Navickaitė et al. [7] numerically evaluated the thermal performance of elliptical wavy tubes and also relatively larger elliptical wavy tubes under constant pumping power conditions. They modeled the geometry of interest for incompressible, laminar, and fully developed hydrodynamic flow under the boundary condition of constant wall temperature. Their numerical results predicted that the thermal efficiency increased by up to 400% and maintained less than 2.4 times the volumetric flow rate in the wavy tubes, at the same pressure drop. They also concluded that the global performance evaluation criterion for elliptical wavy tubes increased by up to 14% and for relatively larger elliptical wavy tubes by 11%.

Promthaisong et al. [8] numerically investigated the effect of geometric parameters on the characteristics of turbulent flow and heat transfer in triangular wavy tubes with Reynolds numbers ranging from 5000 to 20000. Their computational results showed that the three-sided corrugated tubes generated a main rotational flow and a spiral rotational flow, which helped reduce the thickness of the thermal boundary layer and increase the heat transfer rate. It was also determined that the friction coefficient uniformly increased with the increase in depth ratio values and the decrease in pitch ratio values. Additionally, it was concluded that the Nusselt number and the friction factor of triangular corrugated tubes in the studied range are 0.8 to 2.31 and 1 to 17.14 times greater, respectively, than those of smooth circular tubes.

Wu et al. [9] numerically studied the flow characteristics and heat transfer within wavy tubes. They examined the effects of groove height and spacing on fluid flow and heat transfer, and conducted a general evaluation of the performance of the corresponding model parameters. In their study, it was determined that for corrugated tubes, the tendency of the friction factor to change with the Reynolds number follows a power law within the range of Reynolds values. Their results showed that the effects of rib height on flow and heat transfer within corrugated tubes are much more significant than the effects of rib spacing.

Li et al. [10] conducted an experimental study on the characteristics of pressure drop and heat transfer in a set of elliptical tubes twisted in an alternating arrangement, at Reynolds numbers from 7500 to 18000 with air as the working fluid in crossflow. Their results showed that the heat transfer coefficient on the air side and the overall heat transfer coefficient have a positive relationship with the Reynolds number, but the air pressure drop coefficient has a negative relationship with the Reynolds number. The overall performance evaluation of the air side also showed that the elliptical-shaped coiled tubes, compared to the straight circular tubes, have better economic performance in cross-flow.

Li et al. [11] experimentally investigated the heat transfer and pressure drop characteristics of the elliptical-shaped coiled tube in crossflow with Reynolds numbers ranging from 7500 to 18000. Their results showed that the elliptical-shaped coiled tube bundle has a greater advantage in terms of convective heat transfer compared to the circular tube bundle with the same arrangement and configuration in crossflow. Additionally, they analyzed the effect of the Reynolds number on the performance of the complex elliptical tube bundle. The results indicate that the overall performance of the complex elliptical tube array is greater than that of the circular tube array by approximately 25.5% to 33.3% over the Reynolds number test range.

Rukruang et al. [12] investigated the convective heat transfer of water flow through a grooved tube with an alternating cross-section, both numerically and experimentally. Their experimental results showed that the Nusselt number increased with the increase in mass flux. Additionally, it was determined that heat transfer is influenced by the flow characteristics, and vortices formed in the bends and curvatures of the wall, with their intensity increasing along the flow direction. Finally, they concluded that the heat transfer and pressure drop in the flat tube with an alternating cross-section are greater than in the circular tube, and this type of tube performs better than the circular tube.

Sajadi et al. [13] investigated the heat transfer, flow resistance, and compression of alternating flat tubes both numerically and experimentally. Their experimental and numerical results showed that the alternating flat tubes have better performance compared to circular and flat tubes. It was also found that the increase in flattening and the number of cross-sectional units of the alternating flattened tube increased the heat transfer rate.

In this paper, a geometry called alternating elliptical tubes is introduced, and its performance is evaluated against circular tubes in a double-pipe heat exchanger. The method of implementing the relevant plan is entirely experimental. In this way, a two-tube heat exchanger has been designed and constructed initially. After conducting initial tests and calibrating the measuring instruments, oil is pumped into the tube by a radial pump, and the temperature of the oil and water is recorded using a thermometer, while the oil pressure difference is recorded at the beginning and end of the path using a manometer. By comparing the device's output data with accepted empirical relationships and ensuring the proper functioning of the testing apparatus, the examination of the desired geometries can be conducted (Fig. 1).

To manufacture and prepare the periodic elliptical tubes, a copper tube with a diameter of 5/8 inch was first purchased. Then, a copper tube with a diameter of 5/8 inch was cut to a length of 137 centimeters and tested to the desired dimensions using clamps and special tools, forming alternating elliptical tubes. Additionally, the alternating elliptical tubes with a pitch length of 120-280 millimeters and flatness levels of 10-12 millimeters were prepared, manufactured, and tested. It is noteworthy that the number of alternating elliptical tubes tested is three, which have step lengths and flattenings of 12-120 mm, 10-280 mm, and 10-120 mm, respectively. Table 1 shows the specifications and parameters related to the manufactured and tested tubes.

Fig. 1. Experimental setup.

Table 1. Specifications and Parameters of Circular and Alternating Elliptical Tubes.

Tube No.	Tube diameter (mm)	Tube length (cm)	Transition zone length (mm)	Pitch length (mm)	Flattenin (mm)g
1	15.88	137	-	-	-
2	15.88	137	40	120	12
3	15.88	137	40	280	10
4	15.88	137	40	120	10

2. Experimental mesurements

By turning on the gas and bringing the water to a boil, then turning on the pump and allowing the heat transfer oil to flow in the circuit using the installed valves, the desired flow rate for circulation in the circuit is adjusted. After completing the above steps, the system must operate for a while to reach a steady state. Steady-state is established when there are no changes in the average fluid temperatures and the measured pressure difference. The average time it took for the system to reach equilibrium during the tests was approximately 30 minutes. The temperatures displayed on the screens, along with the pressure difference in the manometer, are recorded simultaneously, and then the flow rate is measured by calculating the time required for a specified amount of flow (e.g., 2 liters) to pass through the flow meter. To increase the accuracy of the results at each flow rate, the experimental data are recorded five times and calculated based on statistical relationships and the confidence interval method. All results are displayed within the measured confidence interval. The acceptable error of these experiments in this range is shown in the graphs.

The logarithmic mean temperature difference in heat transfer and in calculations related to heat exchangers represents the thermal driving force in the transfer of heat from the hot fluid to the cold fluid within the exchangers, and therefore it is used in determining the amount of heat exchanged by the exchanger. A higher numerical value of this quantity indicates greater heat transfer within the exchanger. The logarithmic temperature difference in various counterflow heat exchangers is obtained from the following relation [14]:

$$\Delta T_{LMTD} = \frac{(T_{h,out} - T_{c,in}) - (T_{c,out} - T_{h,in})}{ln(\frac{T_{h,out} - T_{c,in}}{T_{c,out} - T_{h,in}})} \quad (1)$$

where $T_{c,in}$ and $T_{c,out}$ are the cold inlet and outlet temperatures, respectively, and $T_{h,in}$ and $T_{h,out}$ are the hot inlet and outlet temperatures, respectively.

In calculating the logarithmic temperature difference of the exchanger, with the input and output temperature specifications of the hot and cold fluids of both co-current and counter-current exchangers, the logarithmic temperature difference is calculated for both types of exchangers.

Additionally, the following equation is used to calculate the convective heat transfer coefficient

$$\dot{m}_h C_{p,h}\left(T_{h,i} - T_{h,o}\right) = hA\Delta T_{LMTD} \quad (2)$$

Hence:

$$h = \frac{\dot{m}_h C_{p,h}\left(T_{h,i} - T_{h,o}\right)}{A\Delta T_{LMTD}} \quad (3)$$

In addition, the Reynolds number and the Nusselt number are considered dimensionless parameters of the problem, which are calculated from equations 4 and 5, respectively:

$$Re = \frac{4\dot{m}}{\pi D\mu} \quad (4)$$

$$Nu = \frac{hD_h}{k} \quad (5)$$

where ṁ and μ are the mass flow rate and viscosity of the heat transfer oil, respectively. k is the thermal conductivity coefficient of the heat transfer oil and D_h is hydraulic diameter.

3. Results

Fig. 2 shows the experimental results related to heat transfer in a circular tube against the theoretical results. The middle line indicates perfect agreement between the theoretical and experimental results. The deviation from this line is considered an experimental error. As seen in Fig. 2, most points are close to or on the midline, indicating the appropriate accuracy of the testing apparatus. The maximum error of 10.5% and the minimum of 3.4% compared to the theoretical results are reported by this graph, which is an acceptable amount for experimental work.

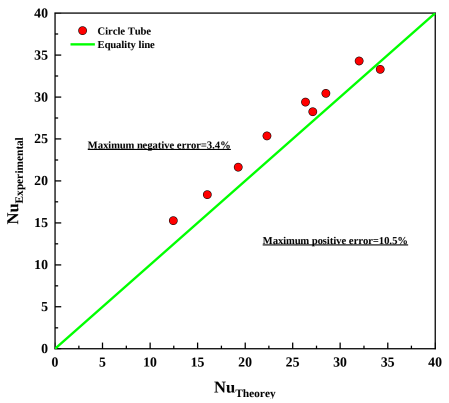

Fig. 2. Comparison chart of heat transfer error between theoretical and experimental relations in a circular pipe.

3.1. Effect of pitch length

In this section, the effect of pitch length on the heat transfer rate of two alternating elliptical tubes with pitch lengths of 120 mm and 280 mm has been addressed. It should be noted that in this section of the experiment, the flattening of both elliptical tubes is 10 millimeters and the length of the tube is 137 centimeters. The results of comparing the effect of pitch length on the heat transfer of alternating elliptical tubes are shown in Fig. 3.

As shown in Fig. 3, for a fixed length of the heat transfer tube, the heat transfer of the alternating elliptical tube with a pitch of 120 millimeters is greater than that of the alternating elliptical tube with a pitch of 280 millimeters. As we know, changing the structure of heat exchanger tubes is one of the passive methods to increase heat transfer. In this regard, it can be noted that a smaller pitch means a greater number of connecting segments and, in a way, a greater number of transfer areas. Due to the facilitation of secondary flow creation and the increase in turbulence and flow disturbance from one elliptical section to the next, it is expected that the heat transfer rate will increase with the reduction of the pitch from 280 mm to 120 mm. Additionally, by examining the values obtained from the experimental work, it can be observed that the Nusselt number at Reynolds numbers of approximately 200, 550, and 1100 for the alternating elliptical tube with a pitch and flattening of 10-120 mm is approximately 50.45, 68.22, and 85.48, respectively, and for the alternating elliptical tube with a pitch and flattening of 10-280 mm is approximately 48.26, 65.25, and 81.77, respectively. Additionally, by examining the values obtained from the experimental work, it can be observed that the value of the convective heat transfer coefficient at Reynolds numbers of approximately 200, 550, and 1100 for the alternating elliptical tube with a pitch and flattening of 10-120 mm is respectively equivalent to 113.1, 252.7, and 385.7 W/(m²K). For the alternating elliptical tube with a pitch and flattening of 10-280 mm, respectively equivalent to 110.5, 245.10, and 374.10 W/(m²K) have been obtained. By comparing the mentioned values and other data obtained from the experiment, we find that the heat transfer rate of the alternating elliptical tube with a pitch and flattening of 10-120 mm is approximately 1.5 times that of the alternating elliptical tube with a pitch and flattening of 10-280 mm. Moreover, as reported in Fig. 3, with the increase in the Reynolds number, the Nusselt number in the alternating elliptical tube increases, which is expected and evident.

3.2. Effect of flattening

In this section, the effect of the degree of flattening on the heat transfer rate of two alternating elliptical tubes with flattening of 10 mm and 12 mm has been addressed. It should be noted that in this section of the experiment, the pitch length of both elliptical tubes is 120 millimeters and the tube length is 137 centimeters. The results of comparing the effect of flattening on the heat transfer of elliptical tubes are shown in Fig. 3.

As reported in Fig. 3, for a fixed length of the heat transfer tube, the elliptical tube with a flattening of 10 millimeters has a higher heat transfer rate than the elliptical tube with a flattening of 12 millimeters. In this context, it can be noted that the greater the flattening of the tube, the more turbulence and disturbance the fluid experiences within the connecting piece between the two sections of the elliptical tube. With the increase in the degree of flattening from 12 mm to 10 mm, due to the turbulence of the fluid in the transition areas of the alternating elliptical tube and the disruption of the thermal boundary layers, the heat transfer rate increases. Additionally, by examining the values obtained from the experimental work, it can be observed that the Nusselt number at Reynolds numbers of approximately 200, 550, and 1100 for the alternating elliptical tube with a pitch and flatness of 120-12 mm is approximately 43.87, 59.32, and 74.33, respectively. Additionally, by examining the values obtained from the experimental work, it can be observed that the value of the convective heat transfer coefficient at Reynolds numbers of approximately 200, 550, and 1100 for an alternating elliptical tube with a pitch and flattening of 10-120 mm is respectively equivalent to 104.41, 228.12, and 348.01 W/(m^{2}K) have been obtained. By comparing the mentioned values and other data obtained from the experiment, we find that the heat transfer rate of the alternating elliptical tube with a pitch and flattening of 10-120 mm is approximately 1.15 times that of the alternating elliptical tube with a pitch and flattening of 10-120 mm. Also, as shown in Fig. 3, with the increase in the Reynolds number, the heat transfer rate or, in a way, the Nusselt number, has increased in the alternating elliptical tube, which was to be expected.

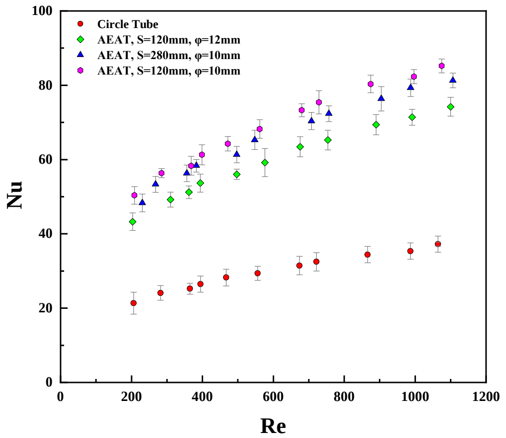

Fig. 3. Comparison chart of heat transfer for an intermittently elliptical tube with different pitch lengths and flattening compared to a circular tube.

3.3. Pressure drop

As shown in Fig. 4, it can be observed that the friction coefficient values at Reynolds numbers of approximately 200, 550, and 1100 for the circular pipe are 0.311, 0.115, and 0.061, respectively. Additionally, analyses of the values obtained from the experimental work indicate that the pressure drop in all three alternating elliptical tubes, with respective pitch lengths and flattening of 12-120 mm, 10-280 mm, and 10-120 mm, is greater than that in the circular tube. Additionally, the pressure drop values at Reynolds numbers of approximately 200, 550, and 1100 for the circular pipe are respectively equivalent to 11607.19, 25349.04, and 38690.64 Pa. By comparing the values of the circular pipe and other experimental data with the alternating elliptical pipes, we find that the pressure drop of the alternating elliptical pipe with a pitch and flattening of 10-120 mm, the alternating elliptical pipe with a pitch and flattening of 10-280 mm, and the alternating elliptical pipe with a pitch and flattening of 10-120 mm are approximately 1.33, 1.29, and 1.2 times that of the circular pipe, respectively. Additionally, Fig. 4 shows that the elliptical tube with a pitch and flattening of 12-120 mm provides a greater pressure drop compared to the elliptical tube with a pitch and flattening of 280-10 mm. Therefore, it is concluded that in the current experiment, the degree of flattening is a more significant factor for increasing pressure drop compared to the pitch length. Additionally, the alternating elliptical tube with a pitch and flattening of 10-280 mm exhibited a greater pressure drop than the alternating elliptical tube with a pitch and flattening of 12-120 mm.

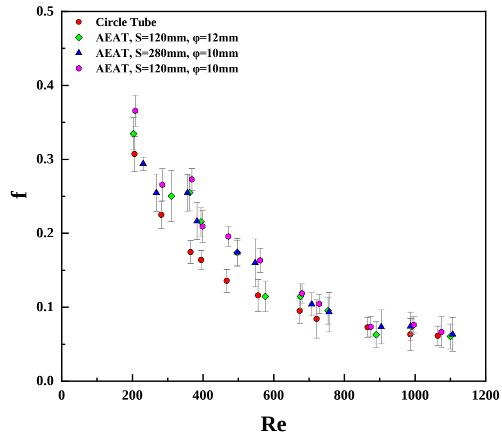

Fig. 4. Comparison chart of friction coefficient for an intermittently elliptical tube with different pitch lengths and flattening compared to a circular tube.

4. Conclusions

The thermal and hydrodynamic behavior of alternating elliptical tubes in a double-pipe heat exchanger has been introduced and examined. Additionally, the effect of pitch length and degree of flattening on the rate of forced convective heat transfer and pressure drop in alternating elliptical tubes has been studied and compared with the results of circular tubes. The most important results obtained from this research are:

- For a fixed length, the heat transfer in an elliptical tube with a pitch of 120 millimeters is greater than an elliptical tube with a pitch of 280 millimeters.

- The heat transfer rate of the elliptical tube with a pitch and flattening of 120-10 mm is approximately 1.5 times that of the elliptical tube with a pitch and flattening of 10-280 mm.
- In a fixed length, the heat transfer of an elliptical tube with a flattening of 10 mm has a higher heat transfer rate than an elliptical tube with a flattening of 12 mm.
- The heat transfer rate of the alternating elliptical tube with a pitch and flattening of 10-120 mm is approximately 1.15 times that of the alternating elliptical tube with a pitch and flattening of 12-120 mm.
- With the increase in the Reynolds number, the Nusselt number in the alternating elliptical tube has increased. In other words, the increase in the Nusselt number is a function of the pitch length and the degree of flattening, and this value increases with a decrease in pitch length and an increase in the degree of flattening.
- The heat transfer of all three elliptical tubes, respectively with a pitch and flattening of 12-120 mm, 280-10 mm, and 10-120 mm, is greater than that of the circular tube.
- The heat transfer rates of the elliptical tube with a pitch and flattening of 10-120 mm, the elliptical tube with a pitch and flattening of 10-280 mm, and the elliptical tube with a pitch and flattening of 12-120 mm are approximately 2.30, 2.20, and 2 times that of the circular tube, respectively.
- The pressure drop of the alternating elliptical tube with a pitch and flattening of 10-120 mm is approximately 1.03 times that of the alternating elliptical tube with a pitch and flattening of 280-10 mm.
- The pressure drop of the alternating elliptical tube with a pitch and flattening of 10-120 mm is approximately 1.11 times that of the alternating elliptical tube with a pitch and flattening of 12-120 mm.

References

[1] Andrade, F., Moita, A., Nikulin, A., Moreira, A., Santos, H., 2019. Experimental investigation on heat transfer and pressure drop of internal flow in corrugated tubes, International Journal of Heat and Mass Transfer, 140, pp. 940-955.

[2] Chen, X., Han, H., Lee, K.-S., Li, B., Zhang, Y., 2019. Turbulent heat transfer enhancement in a heat exchanger using asymmetrical outward convex corrugated tubes, Nuclear Engineering and Design, 350, pp. 78-89.

[3] Chen, Y., Tian, J., Fu, Y., Tang, Z., Zhu, H., Wang, N., 2019. Experimental study of heat transfer enhancement for molten salt with transversely grooved tube heat exchanger in laminar-transition-turbulent regimes, Applied Thermal Engineering, 132, pp. 95-101.

[4] Córcoles, J., Belmonte, J., Molina, A., Almendros-Ibáñez, J., 2018. Influence of corrugation shape on heat transfer performance in corrugated tubes using numerical simulations, International Journal of Thermal Sciences, 137, pp. 275-262.

[5] Karami, M., Akhavan-Behabadi, M.A., Fakoor-Pakdaman, M., 2016. Heat transfer and pressure drop characteristics of nanofluid flows inside corrugated tubes, Heat Transfer Engineering, 37, pp. 106-114.

[6] Kaood, A., Abou-Deif, T., Eltahan, H., Yehia, M., Khalil, E., 2019. Numerical investigation of heat transfer and friction characteristics for turbulent flow in various corrugated tubes, Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, 233, pp. 457-475.

[7] Navickaitė, K., Cattani, L., Bahl, C.R., Engelbrecht, K., 2019. Elliptical double corrugated tubes for enhanced heat transfer, International Journal of Heat and Mass Transfer, 128, pp. 363-377.

[8] Promthaisong, P., Jedsadaratanachai, W., Eiamsa-Ard, S., 2018. Effect of geometrical parameters on turbulent flow and heat transfer behaviors in triple-start corrugated tubes, Journal of Thermal Science and Technology, 13, pp. JTST0008-JTST0008.

[9] Wu, F., Zhou, W., 2012. Numerical simulation and optimization of convective heat transfer and pressure drop in corrugated tubes, Heat Transfer Research, 43, 2012.

[10] Li, X., Zhu, D., Sun, J., Mo, X., Liu, S., 2019. Heat transfer and pressure drop for twisted elliptical tube bundles with staggered layout in crossflow of air, Applied Thermal Engineering, 148, pp. 1092-1098.

[11] Li, X., Zhu, D., Yin, Y., Liu, S., Mo, X., 2018. Experimental study on heat transfer and pressure drop of twisted elliptical tube

bundle in cross flow, Experimental Thermal and Fluid Science, 99, pp. 251-258.

[12] Rukruang, A., Chimres, N., Kaew-On, J., Wongwises, S., 2019. Experimental and numerical study on heat transfer and flow characteristics in an alternating cross-section flattened tube, Heat Transfer—Asian Research, 48, pp. 817-834.

[13] Sajadi, A.R., Kowsary, F., Bijarchi, M.A., Sorkhabi, S.Y.D., 2016. Experimental and numerical study on heat transfer, flow resistance, and compactness of alternating flattened tubes, Applied Thermal Engineering, 108, pp. 740-750.

[14] Sajadi, A.R., Sorkhabi, S.Y.D., Ashtiani, D., Kowsari, F., 2014. Experimental and numerical study on heat transfer and flow resistance of oil flow in alternating elliptical axis tubes, International Journal of Heat and Mass Transfer, 77, pp. 124-130.