Research Paper

Effect of aspect ratio on heat transfer enhancement in alternating oval double pipe heat exchangers

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HIGHLIGHTS

• The effect of inner pipe aspect ratio as well as Reynolds number on heat transfer characteristics is analyzed numerically.
• Enhancement ratio is defined so as to compare the increase of heat transfer with rise of pressure drop.
• The conditions in which enhancement ratio accepts values higher than one are recognized.

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ABSTRACT

In this paper, a numerical study for laminar, incompressible flow for alternating oval double pipes is carried out in order to analyze the suggested configuration under different conditions. In addition to Reynolds number, the effect of pipe aspect ratio, as one of the significant non-dimensional variables on thermal performance is analyzed quantitatively. The results show that in all aspect ratios, heat transfer rate accepts higher values comparing to circular counterpart. However, it is not considered as oval double pipes are always worth taking the place of circular types, since the pressure loss effects may be dominant over the heat transfer improvement. For this purpose, Enhancement ratio (ER) is investigated as a function of aspect ratio (γ) and Reynolds number (Re). It is revealed that, at a specific Reynolds number, ER increases with aspect ratio until it reaches a max value, then starts declining. Therefore, to determine whether using this configuration is economically justified or not, a map for ER in terms of mentioned parameters is given. It is revealed that the region of ER > 1 becomes more limited at extreme values of aspect ratio resulting in an optimum value for each Reynolds ratio. In addition, for Reynolds values less than 600, ER is always less than one regardless of aspect ratio value. In the end, mathematical expressions for the extreme values of γ that result in higher than one values of ER is suggested.

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1. Introduction

Heat exchangers are one of the most commonly used equipment in chemical processes with wide range of applications including powerhouses, refineries, petrochemicals and air conditioning. Heat exchangers in different types such as steam generator, condenser, evaporator, cooling tower, etc. are frequently used and the performance improvement has been always a matter of interest to engineers. Double pipe heat exchangers have specific role in industry, their applications are mostly considered when the need for heating and cooling of process fluids in small heat transfer area matters. Many researches has been performed on improvement of double tube heat exchangers [1–7]. The main shortcoming of this type is the low rate of heat transfer per unit area. The techniques of heat transfer enhancement are divided into three different methods: active, passive and compound in which both active and passive methods are used to enhance the heat transfer rate. In active method external source of energy such as vibrated plate [8] or electric power for applying electric field across a layer of dielectric liquid which is called electro-hydrodynamic [9,10], is required. On the other hand, passive methods perform without any additional energy like utilization of non-circular geometry [11,12], adding Nano-particle to fluid [13,14] or using fins [15]. Akpinar [16] could increase the heat transfer rate up to 130% by using swirl elements with different diameter and number of circular holes at the entrance section of the inner pipe of a concentric tube heat exchanger. Yang et al. [17] analyzed twisted elliptical tubes with different aspect ratios and twist pitches. They showed
that this type of tubes may enhance the heat transfer rate especially at low Reynolds number, while increase in friction factor in comparison with smooth tubes, was seen.

Guo [18] discussed the uniformity principle of temperature difference field. On the basis of this theory the thermal performance of a heat exchanger may be improved significantly if the uniform temperature difference between hot and cold fluid is achieved. Tao [19] illustrated that to increase heat transfer rate, the angle between temperature and velocity gradient should be decreased. Based on Tao’s theory, Guo [20] presented a novel model which is seen in Figs. 1 and 2. In this model, the pipe is composed of alternating horizontal or vertical elliptical cross sections which lead to secondary flows and reduction of the angle between temperature and velocity gradient which results in heat transfer enhancement. The presented model could enhance the heat transfer rate up to 300% in comparison with circular counterpart, beside the increase in friction factor up to 120%. Chen et al. [20] investigated Guo’s model numerically with k-ε turbulence model. The results show that the area in which an oval cross section is converted from horizontal to vertical or vice versa, has significant effect on heat transfer. However, in this area which is called transition area, a recirculating zone is formed which causes more pressure loss and pumping power accordingly. Chen et al. [21] analyzed Guo’s model parametrically. They considered the effect of variation of transition length and tube length on rate of heat transfer under constant wall temperature condition. They showed that with increase in aspect ratio, both Nusselt number and surface friction factor increase. In addition, in a constant aspect ratio and tube length, decrease in transition length, resulted in Nusselt number and friction factor reduction. Wen-Lih Chen et al. [6] used Guo’s model as inner tube of double tube heat exchanger. They studied this kind of double pipe heat exchanger numerically, and showed that increase in length of heat exchanger leads to reduction in overall heat transfer rate. Hosseinalipour et al. [22] studied a similar configuration numerically under turbulent flow condition with different number of transition modulus. They found that in turbulent flow, the efficiency of this configuration significantly suffers from high pressure drop.

As stated, alternating oval pipes promote the rate of heat transfer as well as pressure loss. Hence, it must be carefully designed so that heat transfer benefits overcome the pressure loss extra costs. In this study, a counter flow double pipe heat exchanger with alternating elliptical tube as inner pipe is investigated. The simulation has been carried out for different aspect ratios ranging from 1.3 to 2 in various Reynolds from 100 to 1600. Enhancement ratio (ER) may be considered as a criteria to distinguish how efficient the enhancement technique really is. A well designed enhancement technique keeps the ER higher than one in different sets of conditions. This ensures that the rise of heat transfer rate overcomes that of pressure loss.

![Fig. 1. Geometry of double pipe heat exchanger with outer circular pipe and inner alternating cross section.](image1)

![Fig. 2. Isometric view from inner pipe configuration; alternating oval cross sections with six transitions.](image2)
affects ER as well as fluid flow characteristics. In this paper, ER is analyzed with respect to pipe aspect ratio, Reynolds number and Reynolds ratio of inner and outer side. A map diagram for ER is developed and the conditions in which heat transfer rise is dominant over the pressure loss (i.e. ER > 1) is specified.

2. Governing equations

Conservation equations for a three dimensional laminar flow with constant properties under steady state condition can be written as:

\[ \frac{\partial u_i}{\partial x_i} = 0 \]  

Continuity eq.:

\[ \rho u \frac{\partial u_i}{\partial x_i} = - \frac{\partial p}{\partial x_i} + \mu \frac{\partial^2 u_i}{\partial x_j \partial x_j} \]  

Momentum eq’s:

\[ \rho c_p u_i \frac{\partial t}{\partial x_i} = \frac{k}{\rho C_1} \frac{\partial^2 t}{\partial x_j \partial x_j} \]  

Energy eq. in fluid:

\[ k_t \frac{\partial^2 t}{\partial x_j \partial x_j} = 0 \]  

Energy eq. in solid:

For inner tube the below boundary conditions are considered. At inlet the temperature and velocity are assumed constant:

\[ u = u_i; \quad t = t_{in} \]  

At the outlet the absolute pressure is equal to atmospheric pressure:

\[ p_{out} = 0 \]  

For outer tube the below conditions are considered. For inlet:

\[ u = u_o; \quad t = t_{in} \]  

For outlet the same condition is defined as inner tube. For inner tube the solid/fluid interface is defined as following:

\[ k_t \frac{\partial t}{\partial n} = k_s \frac{\partial t}{\partial n} \]  

3. Numerical framework

Governing equations are discretized using finite volume approach and implemented on a structured grid shown in Fig. 3. Flow is assumed as incompressible and steady state with constant fluid properties determined at bulk temperature. Equations are simultaneously solved using software package ANSYS Fluent 14.0. SIMPLE algorithm is used for pressure-velocity coupling and second order upwind scheme is implemented for discretization of convection terms.

4. Geometry and grid study

Present study discusses the impact of inner pipe aspect ratio on heat transfer enhancement characteristics in an alternating oval double pipe heat exchanger. For this purpose, variation of heat transfer coefficient and friction factor versus aspect ratio at different inner and outer Reynolds number is investigated. According to Table 1, different parameters are considered as follows:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Inner tube</th>
<th>Outer tube</th>
</tr>
</thead>
<tbody>
<tr>
<td>Working fluid</td>
<td>Water</td>
<td>Water</td>
</tr>
<tr>
<td>Aspect ratio</td>
<td>1.3–2</td>
<td>0.2–1</td>
</tr>
<tr>
<td>Re</td>
<td>100–1600</td>
<td></td>
</tr>
<tr>
<td>Transition length</td>
<td>5 mm</td>
<td>33 mm</td>
</tr>
<tr>
<td>Tube diameter</td>
<td>16.5 mm (circular)</td>
<td>33 mm</td>
</tr>
<tr>
<td>Tube thickness</td>
<td>0.5 mm</td>
<td></td>
</tr>
<tr>
<td>Iron conductivity</td>
<td>77 W/mK</td>
<td></td>
</tr>
</tbody>
</table>

Fig. 1 \( d_1, \ d_2, \ d_1 \) and \( d_2 \) are minor and major diameters of ellipse, the inlet diameter of inner tube and diameter of outer tube respectively. As depicted in Table 1, \( d_1 \) and \( d_2 \) are constant and equal to 13 mm, 16.5 mm and 33 mm and changes are applied to \( d_2 \) which varies from 16.9 mm to 26 mm. \( L_1 \) and \( L_2 \) are the length of heat exchanger and length of transition area which are equal to 226 mm and 5 mm respectively. The working fluid is water with Prandtl number of 6.2 and thermal conductivity of 0.6 W/m·K at room temperature. Iron is considered as the material of the tubes with 0.5 mm thickness and thermal conductivity of 77 W/m·K. In order to determine the appropriate grid size with grid independent results, a number of runs with three different grid sizes are performed as shown in Table 2. The results for average overall heat transfer coefficient and friction factor are seen in Table 2. According to this table, the difference between results of fine and medium grids are less than 2% which means the medium grid size with 40\( \times \)120\( \times \)310 nodes (in radial, circumferential and axial direction, respectively), is adequate and may be used in order to reduce the computational cost. A cross section view of generated grid is seen in Fig. 3 with denser grids close to inner wall tube to improve the accuracy of results.

In order to check the validity of numerical modeling, the values of overall heat transfer coefficient are compared with that of reported in Ref. [6]. In Fig. 4, \( U \) changes in terms of inner and outer tube velocity, are depicted. The max difference between the results
of present study and data of Ref. [6] is about 6.8% which implies the good agreement between obtained results and literature.

5. Results and Discussion

Temperature distribution for various aspect ratios at the middle of heat exchanger (x = 113 mm far from inner inlet) with Re of 1100 are illustrated in Fig. 5. In addition, The secondary flow formation in vicinity of inner wall in both horizontal and vertical cross sections due to curvature of tube surface, is recognized in Fig. 5. The effect of pipe aspect ratio on induced secondary flows can be clearly observed in Fig. 5, where secondary flows are intensified in higher γ. Furthermore, transition area can disrupt the boundary layer development and consequently prevents the flow to become fully developed which is another contributory factor in heat transfer promotion. Fig. 6 shows the pressure distribution along a transition area in which ellipse orientation turns from vertical into horizontal smoothly. Through the transition, as deeply discussed by Guo et al. [20] flow passage for inner tube at about θ = 0° gets narrow while about θ = 90° gets expanded which leads to pressure increase in θ = 0° and decrease in θ = 90° and vice versa for outer tube. This pressure gradient induces secondary flows, pushing fluid toward θ = 90° at inner tube and toward θ = 0° at outer tube. That is, the flow is accelerating at θ = 90° and decelerating at θ = 0°. Hence, using alternating oval cross section pipe can rise the rate of heat transfer as well as pressure loss. The secondary flow strength is thus significantly under the influence of geometrical constraints such as pipe aspect ratio. As is shown in Figs. 5 and 6, secondary flows seem to be stronger as γ rises.

Streamlines of inner pipe flow corresponding to three different aspect ratios are investigated in Fig. 7. A recirculating zone is observed inside the transition area. The higher the aspect ratio, the larger the separation bubble is. Larger separation bubble causes more energy dissipation and more pumping power is thus required. Hence, with respect to Fig. 7 it can be found that pipe with higher aspect ratio is more affected by the bubble area. For this reason, the designer should pay considerable attention to local pressure loss while deciding on pipe aspect ratio.

The average overall heat transfer coefficient for a double pipe heat exchanger may be determined as follows:

\[ U = \frac{q}{A \cdot \Delta T_{\text{lm}}} \]  \hspace{1cm} (9)

where \( \Delta T_{\text{lm}} \) for a counter flow heat exchanger is defined as:

\[ \Delta T_{\text{lm}} = \frac{\delta t_1 - \delta t_2}{\ln(\delta t_1/\delta t_2)} \]  \hspace{1cm} (10)

where \( \delta t_1 \) and \( \delta t_2 \) are

\[ \delta t_1 = t_{i,1} - t_{c,1} \]  \hspace{1cm} (11)

\[ \delta t_2 = t_{i,2} - t_{c,2} \]  \hspace{1cm} (12)

Friction factor is defined as Eq. (13) where \( \Delta p \) is the pressure loss of inner or outer tube:

\[ f = \frac{\Delta p}{\frac{1}{2} \rho u^2} \frac{d_0}{T} \]  \hspace{1cm} (13)

The flow passage in either side is under the influence of aspect ratio which leads to pressure drop to vary with this parameter. Friction factors for both sides are calculated at various Reynolds numbers as shown in Figs. 8 and 9. At a specified Reynolds number the friction factor has higher values than circular counterpart in inner pipe (i.e. \( f_{\text{circ}}/f_{\text{oval}} > 1 \)) while for outer pipe at small values of \( \gamma \) less friction factor may be achieved. The comparison between the friction factor of present and circular heat exchanger shows more pumping power is usually required for present heat exchanger than circular type. Thus, using oval heat exchanger rather than circular double pipe, in spite of heat transfer promotion may not be efficient due to high friction. It is shown that aspect ratio increase causes outer side friction ratio to increase whereas for the inner side, at smaller Reynolds number (i.e. \( Re < 600 \)) friction ratio decreases. It is noticeable that at a particular value of γ for \( Re > 600 \) friction ratio starts growing. This particular value which is specifically important in larger Reynolds decreases with increase in Reynolds. Hence, the pipe aspect ratio must be chosen with respect to Reynolds number to have the least pressure loss.

The average static pressure along the pipe length is shown in Fig. 10. As mentioned, in transition area the recirculating zone leads to pressure loss and heat transfer weakening. Hence, in transition area the pressure gradient is more intense than other regions. Furthermore, with increase in aspect ratio the pressure loss becomes stronger in transition area.

The average overall heat transfer coefficient is a function of inner and outer Reynolds number, fluids Prandtl number, aspect ratio and other geometrical parameters such as transition length, number of transitions and total length. In present study the inner and outer Reynolds number as flow variables and aspect ratio as...
Fig. 5. Temperature distribution and induced secondary flows for Re = 1100, r = 0.8 at x = 113 mm.

Fig. 6. Pressure distribution for 4 cross sections along the transition area.
an important geometrical variable, are chosen so as to analyze their influence on overall heat transfer coefficient and enhancement characteristics. As shown in Figs. 11 and 12, overall heat transfer coefficient is a relatively strong function of aspect ratio. That is, the higher the aspect ratio, the higher heat transfer coefficient is. The thermal performance of an oval double pipe may be compared with the circular counterpart using the ratio of overall heat transfer coefficient (i.e. $U_{oval}/U_{circ}$). The variation of this parameter in terms of Reynolds and aspect ratio are shown in Fig. 11. Since in all cases $\xi > 1$ is provided, using alternating oval tube, supplies more heat transfer rate than circular type regardless of aspect ratio or Reynolds number. Another parameter studied here is Reynolds ratio denoted by $r$. As expected, higher values of $r$ results in higher $\xi$. In case of $\gamma = 1.3$, heat transfer is raised up to 6.8% and 12.7% for Re ratio of 0.4 and 1, respectively. In addition, only in the case of $\gamma = 1.3$, at $Re = 1600$ the slope of the graph is still increasing. While for the lager aspect ratios in $Re < 1600$ an inflection point is observed. As $\gamma$ increases, the inflection point of the graph is located at smaller $Re$. Furthermore, the relative
influence of $r$ at smaller aspect ratio is more apparent. It is noteworthy that for $\gamma = 2$, heat transfer rate is improved as much as 80% at $Re = 1600$ and $r = 1$ relative to circular type. The dependence of $\xi$ on pipe aspect ratio is directly presented in Fig. 12. It illustrates that increase in aspect ratio leads to heat transfer enhancement. Here the minimum value of aspect ratio is 1.3 which is not making so much difference in heat transfer rate (i.e. less than 5%). For $Re < 100$, the heat transfer improvement is not noteworthy, while for $Re > 600$ heat transfer enhancement starts being sensible. As $Re$ increases, the effect of aspect ratio is more evident. Thus, it is at a particular range of aspect ratio and Reynolds number that the merits of oval pipe are revealed and out of this range, not much difference in heat transfer coefficient is made. That is, beside the aspect ratio, efficiency of this configuration strongly depends on Reynolds number.

As stated, oval tube improves heat transfer rate, but the increase in required pumping power is the price must be paid. Therefore, to distinguish that oval heat exchangers are worth utilizing or not, a non dimensional parameter including pumping power as well as heat transfer rate is needed which is called enhancement ratio ($ER$) and defined as Eq. (14) by dividing heat transfer ratio to pressure drop ratio. Obviously, in cases the enhancement ratio has higher values than one ($ER > 1$), the heat transfer enhancement is dominant over the rise of pressure loss [7]. As expected, $ER$ is a function of aspect ratio and Reynolds number.

Fig. 11. Overall heat transfer ratio versus Reynolds number; $x$ $r = 0.4$, $\triangle$ $r = 0.6$, $o$ $r = 0.8$, $\square$ $r = 1$.

Fig. 12. Effect of aspect ratio on overall heat transfer ratio in $r = 1$.

Fig. 13. Enhancement ratio versus aspect ratio and Reynolds number.
numbers. Fig. 13 shows how the pipe aspect ratio affects enhancement ratio where \( ER^2 \) is plotted versus \( \gamma \) and Reynolds number to give better visualization and easier comparison. \( ER \) reaches its max value at a unique value of \( \gamma \), here called \( \gamma_{\text{max}} \), while for higher values (i.e. \( \gamma > \gamma_{\text{max}} \)), \( ER \) begins to decline. \( \gamma_{\text{max}} \) is itself a function of Reynolds number. That with higher Reynolds number, higher aspect ratio is recognized as optimum (i.e. As \( Re \) increases, \( \gamma_{\text{max}} \) is increased, too). Clearly, at higher Reynolds number, due to more pressure loss \( ER \) decreases. At \( \gamma = 1.5 \), for all Reynolds numbers \( ER \) is below one, while at \( \gamma = 1.75 \), \( Re > 600 \), \( ER \) has the largest values among the other aspect ratios. In the case of this paper, \( \gamma = 1.75 \) and \( Re = 1100 \) results in the highest enhancement ratio which is the most desirable condition.

\[
ER = \min \left( \frac{q_{\text{oval}}/q_{\text{circ}}}{\Delta p_{\text{oval}}/\Delta p_{\text{circ}}}, \frac{q_{\text{oval}}/q_{\text{circ}}}{\Delta p_{\text{oval}}/\Delta p_{\text{circ},a}} \right) \quad (14)
\]

Fig. 14 indicates the conditions in which enhancement ratio equals to one (\( ER = 1 \)). The surrounded region illustrates design area (\( ER > 1 \)) which is the most desirable because heat transfer enhancement is achieved with reasonable amount of pressure loss, whereas outside this region is \( ER < 1 \). The region \( ER > 1 \), is strongly affected by Reynolds number, aspect ratio and Reynolds ratio. As it is shown, it is slenderized with reduction of Reynolds ratio and increase in Reynolds number. As it is expected, with increase in Reynolds number, pressure drop grows rapidly until it overcomes heat transfer enhancement. Hence, the region of \( ER > 1 \) should be thinner as Reynolds increases which is well reflected in Fig. 14. Furthermore, as the difference in Reynolds number between inner and outer sides grows, the region \( ER > 1 \) becomes thinner. To facilitate the appropriate choice of aspect ratio, mathematical representation of Fig. 14 for \( \gamma_{\text{min}} \) and \( \gamma_{\text{max}} \) are given as Eqs. (15) and (16) with maximum error of 0.48% and 0.90%, respectively.

\[
\gamma_{\text{min}} = a_0 + a_1 Re + a_2 Re + a_3 Re \cdot Re \quad (15)
\]

\[
\gamma_{\text{max}} = a_0 + a_1 Re + a_2 Re + a_3 Re + a_4 Re \cdot Re \quad (16)
\]

where \( \gamma_{\text{max}} \) and \( \gamma_{\text{min}} \) denote the extreme values of aspect ratio that provide \( ER > 1 \) condition and \( a_i \) coefficients are constants and given in Table 3. Therefore, aspect ratio must be chosen such that the condition \( \gamma_{\text{min}} < \gamma < \gamma_{\text{max}} \) is satisfied. In this manner, \( ER \) has values higher than one.

6. Conclusion

In this paper, a numerical study for incompressible, laminar flow in double pipe heat exchangers with alternating oval tube as inner tube, is carried out. The influence of aspect ratio on heat transfer enhancement characteristics is analyzed in detail. For this purpose, variation of heat transfer rate and friction factor versus aspect ratio at different inner and outer Reynolds number is investigated. A mathematical expression is also presented to determine overall heat transfer coefficient. The results of the research can be concluded as:

1. Oval alternating tube as inner tube of double pipe heat exchanger, regardless of aspect ratio and Reynolds number promotes the rate of heat transfer comparing to circular type.
2. Heat transfer rate is enhanced with increase in aspect ratio. This phenomenon is more sensible for Reynolds numbers larger than 600 while for values lower than 100 no noteworthy improvement is achieved.
3. Unlike heat transfer, enhancement ratio does not vary monotonically with aspect ratio, it accepts its max value at a unique value of \( \gamma \) while for higher values, \( ER \) declines. The optimum value of aspect ratio is a function of Reynolds number. That is, with higher Reynolds number, higher aspect ratio is recognized as optimum.
4. There are certain conditions in terms of Reynolds number, aspect ratio and Reynolds ratio in which the overall enhancement ratio is higher than one. In this area, heat transfer enhancement is achieved with reasonable rise of pressure loss. The region of \( ER > 1 \) becomes thinner with increase in Reynolds number and Reynolds ratio reduction.

References


