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Numerical Analysis of Plain Fin-and-Oval-Tube Heat Exchanger with Different Inlet Angles

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In industrial applications, oval tubes are often used instead of round tubes in finned tube heat exchanger for improving the heat transfer performance. In this paper, a periodical numerical method based on FLUENT software is carried out to investigate the heat transfer and pressure drop performances of fin-and-oval-tube heat exchanger with 5 different inlet angles (30°, 45°, 60°,75° and 90°). The results show that the overall heat transfer coefficient increases with the decrease of inlet angle. Meanwhile, the pressure drop also increases with the decrease of inlet angle. In addition, the local heat transfer coefficient illustrates the variation of fluid flow between tubes. With the decrease of inlet angle, the average Nusselt number increases by 16.7 %. That is because the fluid is blocked by oval tubes seriously, which also results in the largest increment of the pressure drop by 57.8 %. According to the numerical simulation results, the genetic algorithm is used to find the correlations related the inlet angles.

1. Introduction

In recent years, the energy conservation and efficiency improvement in heat exchanger designs have been paid more attention. Finned tube heat exchangers are commonly used in a variety of applications in air-liquid condensation, process industries and chemistry techniques. Nowadays, researches usually engaged in development of materials and focus on its geometry (Erek, 2005) and patterns (Tang, 2009) of heat exchangers.

The performance of fin-and-tube heat exchangers are related to geometric parameters. Early experiment results achieved by Rich (1973). Lu et al. (2011) illustrated the effects of geometric parameters such as fin pitch, tube pitch, fin thickness and tube diameter in detail. The optimum value for $Q/\Delta P$ was found by numerical simulation. Tang et al. (2009) analyzed the air-side heat transfer and friction characteristics of 5 types of fins. Besides, in order to enhance performance, different kinds of methods are used in finned tube heat exchangers. The effects of the attack angle of delta winglet pair were achieved by Wu et al. (2012) with numerical and experimental methods. It was turned out that the average Nusselt Number of winglets with attack angle of 60° was higher than that of winglets with attack angle of 45° by experiment and computational method. He et al. (2012) used winglet type of vortex generators to enhance air-side heat transfer performance. Another method was carried out by Tao et al. (2007) who used triangular wavy fins to make the performance better. It can be seen that vortex generators and wave fins are always made to enhance the heat transfer on the air-side. The hydrophobic prosperities are very important for chemistry applications. Wang et.al (2002) describes the air-side heat transfer performance of a hydrophilic coating on plain-fin surface.

As is known to all, oval tubes have better performance than that of cylinder ones (Schulenb, 1966). An experiment study was performed to compare the performance between cylinder tube and oval tube by James E.O (2004). The heat transfer and pressure drop characteristics of plate fin-and-oval-tube heat exchanger were analyzed numerically by Erek et al (2005). A three-dimensional numerical study of oval tube and the winglet pairs which can improve the heat transfer significantly was analyzed by Tiwalri et al (2003). The results indicated that the contribution of winglets pairs in heat transfer was more than 43.86 %, undoubtedly. The cooling delta angle on power plant natural draught dry cooling towers was analysed with

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porous media approach by Wang et al (2011).In general, heat exchangers are not placed horizontally but arranged with an angle to the ground in industrial application. In this paper, the inlet angle characteristics of a fin-and-oval-tube heat exchanger have been studied numerically. Because of the non-orthogonal layout of heat exchanger, we much concern about the thermal characteristics of fin-and-oval-tube heat exchanger with different inlet angles. So, 5 different inlet angles of fin-and-oval-tube heat exchanger unite are analyzed by FLUENT in the present study.

2. Model description

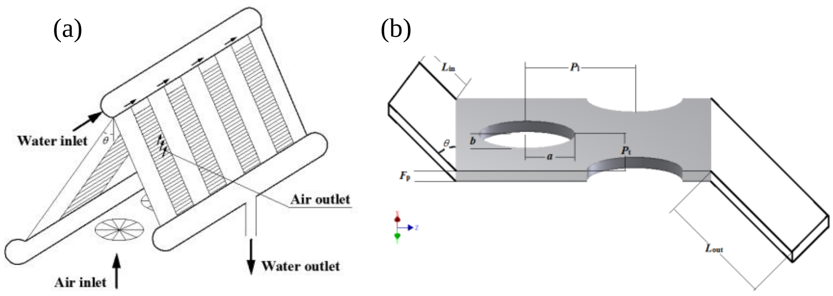

Figure 1: Schematic of the fin-and-oval-tube heat exchanger (a) air condenser system; (b) Schematic of the actual computation domain

Figure 1(a) shows the cooling device which is always used in surface condenser and mixing condenser. It is a general problem that the inlet direction is non-orthogonal with the heat exchanger arrangement in industrial applications. The computational domain, which are made with staggered, is shown in Figure 1(b), consists of two fins with half fin thickness, fin tubes, entrance extension and exit extension. Some detailed geometric parameters for numerical simulations are shown in Table 1. The domain is extended 1.0 times for entrance section to ensure the inlet flow uniformity. Meanwhile, the domain is extended 5 times for exit extension to ensure that the exit flow boundary has no flow recirculation. Moreover, the physical properties of air and aluminium are listed in Table 2. The model is created and meshed by ICEM software.
In addition, some assumptions are made to simplify the calculation as follow:

1. The flow is steady and incompressible.
2. The physical properties are constant.
3. The latent heat of phase change is omitted.
4. The thermal radiation and heat dissipation are neglected.
5. The fin surface is smooth.

The fin plates are treated as solid region which are conjugated with air flow. The there-dimensional general governing equation is shown here.

$$\nabla \cdot (\rho \vec{U} \phi) = \nabla \cdot (\Gamma_\phi \nabla \phi) \tag{1}$$

Where U is the velocity vector, ρ is the density and ϕ is the general variable.

Table 1: Geometric details of the computation domain (unit: mm)

Entrance extension	Exit extension	F_p	P_t	P_l	a	b
120	600	2.5	32	60	16	7

Table 2: Properties of the air and fin material

	Density (kg·m^{-3})	Thermal conductivity (W·m^{-1}·K^{-1})	Specific heat (J·kg^{-1}·K^{-1})	Dynamic viscosity (kg·m^{-1}·s^{-1})
Air	1.165	0.0267	1005	1.6
Fin	2710	236	902	-

The boundary conditions for computations are shown as follow:

1. Uniform velocity and temperature are assigned for the inlet boundary.
2. No slip conditions are used to walls at tested section. Periodical conditions are assigned for the lateral surfaces of the test section.
3. The tube wall temperature is fixed at 320 K.
4. The computational domain is extended downstream for fully developed outlet boundary condition. The local one-way condition is used for exit boundary.
5. Symmetrical boundary conditions are assigned to other surfaces.

3. Numerical methods and grid independency validation

The mesh is shown in Figure 2. Considered the accuracy of calculation, a structured grid system is generated by ICEM. There are 5 layers mesh in fin thickness direction, and 20 layers in fluid thickness direction. In order to improve the quality of t grid system, especially for the parts around the oval-tube, the "O" body fitted grid is generated by dividing blocks. The grid independence verification is illustrated in Figure 3. The grid sizes vary from 170,000 to 900,000 and double precision is used in computation.

In the simulations, the Reynolds number is ranging from 2,000 to 10,000 based on the tube outside diameter. So whether the laminar or turbulence model used in simulation should be clarified. In this study, the standard k-ω model has been used for simulation because of the high Reynolds number flow. The model is performed to examine the applicability by comparing with experiment (2004) and the comparisons are shown in Figure 4. Though the laminar model is accurate to predict low Reynolds number flow, it can be seen that k-ω turbulence model is more appropriate to predict the Nusselt number and *f* factor in all range. The deviation of Nusselt number using k-ω method between numerical and experimental results is less than 9% and the deviation of *f* factor is less than 8 %. It is concluded that the calculation is reliable to simulate the heat exchanger problem. In simulation, the central difference and QUICK method are used to discretize the diffusion term and convection term. The residual of velocities, temperature, k and ω have been controlled below 10^{-5}.

In this study, the heat conductivity of the fin must be considered because the fin efficiency which calculated by Eq.(2) should be taken into account. So, some parameters are defined to present the simulation results.

$$\eta = 1 - \frac{A_f}{A}(1 - \eta_f) \tag{2}$$

where η is the overall surface efficiency which can be written in terms of the fin efficiency η_f. The fin surface area A_f and total surface area A are calculated by geometric parameters. In this study, the Reynolds number *Re*, Nusselt number *Nu* and friction factor characteristics are calculated by Eq.(3) to Eq.(7).

$$Re = \frac{\rho u_{max} D_c}{\mu} \tag{3}$$

$$Nu = \frac{hD_c}{\lambda} \tag{4}$$

$$h = \frac{q}{\eta A \Delta T_m} \tag{5}$$

$$j = \frac{Nu}{RePr^{1/3}} \tag{6}$$

$$f = \frac{2 \cdot \Delta p \cdot D_c}{\rho u_{max}^2 \cdot F_1} \tag{7}$$

Where D_c is the hydraulic diameter, q is the heat flux on the surface and u_{max} is the velocity at the minimum flow area $u_{max}=v_{fr}/\sigma$. The term σ is the contraction ratio in flow direction.

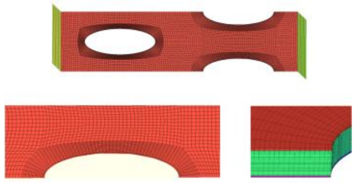

Figure 2: Schematic of grid structure (a) mesh of whole unit (b) mesh surrounding the tube (c) mesh near the fin

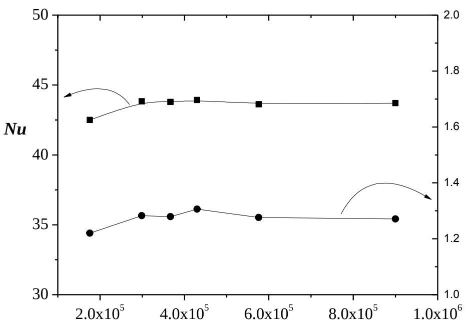

Figure 3: Grid independence test

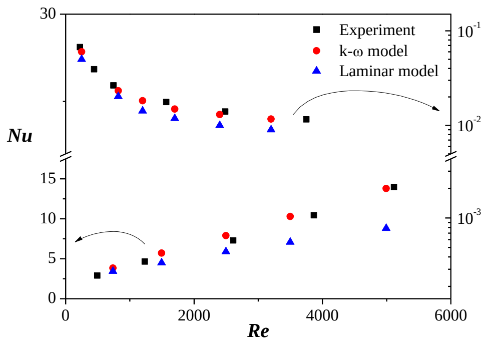

Figure 4: figure of model comparison

4. Results and discussion

Figure 5 shows the streamlines in the mid-plane of the channel of the study area for different inlet angle at *Re* = 7,000. The flow with inlet angle past the oval tube and it can be clearly found out that the wall in the direction of tube arrangement has periodic boundary condition which is plotted in Figure 5. It can be seen that the flow is separated by the oval tubes and clearly exhibits the phenomenon of vortex near the rear of the tube. Meanwhile, the location of stationary point is shifted upward as well as the inlet angle decreases and the velocity gradient is also changed in the channel. Such behaviour can be attributed to the periodical flow condition in vertical direction which results that the downstream flow is affected by upstream flow. Furthermore, it can be seen that the region of vortex enlarges obviously with the decrease of inlet angle. Besides, the sequence of pictures illustrates that the average flow velocity has been increased because of the enhancement of vortex.

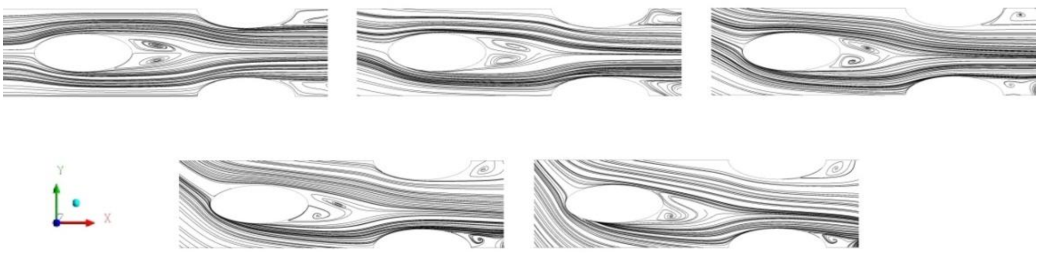

Figure 5: Streamlines on the mid-plane of the channel (a) θ=90°, (b) θ=75°, (c) θ=60°, (d) θ=45°, (e) θ=30°

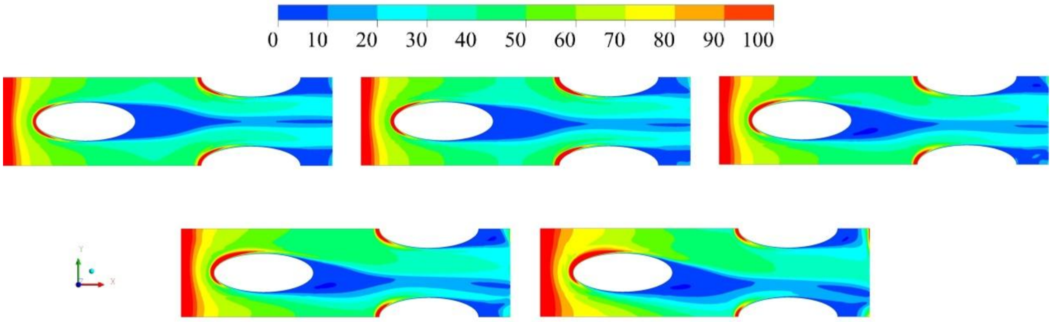

Figure.6 Local Nusellt number over the fin surface for Re=7000 (a) θ=90°, (b) θ=75°, (c) θ=60°, (d) θ=45°, (e) θ=30°

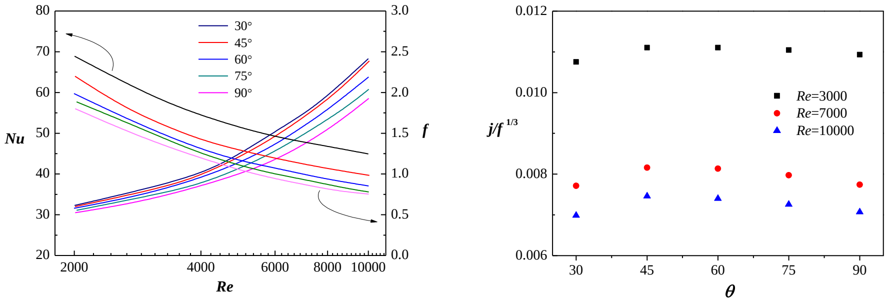

Figure 7: (a) Comparison of average Nu and f factor for different inlet angles (b) Comparison of $j/f^{1/3}$ in different Reynolds number.

Figure 6 illustrates the distribution of local Nusselt number on the fin surface of the heat exchanger with different inlet angle. The pattern is similar to Tao et al. (2007). The maximum Nu presents at the leading edge of the plate fin because the thermal boundary layer there is thin. Then the *Nu* gradually decreases away from the leading edge due to the growth of boundary layer on the walls. It can be seen clearly in Figure 6 that there are some differences in these series of pictures. As the inlet angle decreases, the peak of *Nu* moves upward. Meanwhile, the distribution of local Nusselt number has been no longer symmetrical about horizontal axis. As shown in Figure 6, the gradient distribution of local Nusselt number near the rear of tube in first row changes certainly. However, the changes of gradient distribution near the rear of tube in second row are not noticeable. This is because the oval tubes in first row deflect the flow direction as a baffle. Thus the influence of inlet angle gets smaller with the increase of tube rows.

The Nusselt number and *f* factor which can correctly reflect the performance of heat exchanger which is depicted in Figure 7(a). It turned out that the *Nu* steadily increases with the increase of Reynolds number and it also increases with the decrease of inlet angle. In addition, it is not quite different that the heat transfer performance causing by inlet angle at low Reynolds number. However, for large *Re*, the Nusselt number increases 16.7 % averagely at most comparing θ=30° with θ=90°. Besides, the Nusselt number averagely increases 15.6 %, 10.1 % and 3.9 % when θ equal to 45°, 60° and 75°. Meanwhile, the pressure drop increase about 57.8 %, 31.2 %, 9.0 % and 3.2 % averagely at the same time. This is because making inlet angle decreases inevitably increase the pressure drop and the disturbance on the surface of oval tube becomes strong. Therefore, the heat transfer has been enhanced as well as increasing the drag coefficient. The heat transfer performance, which reflected by the ratio of *j* and *f* factor, is shown in Figure 7(b). It can be seen that the integrated performance is affected by inlet angle more significantly for large *Re* and, especially, the ratio achieved the best optimization at θ=45°.

5. Conclusions

In this paper, a two-row heat exchanger unit model has been established and the inlet angle effects of plain fin-and-oval-tube heat exchanger have been investigated by FLUENT software. Some major conclusions are drawn as follows:

1. With the variation of inlet angle, the streamline has being changed remarkably. The layout of vortex effected by inlet angle obviously and the hydrodynamics determine the distribution of the local Nusselt number.
2. The Nusselt number increases 16.7 % averagely at most for large *Re* comparing θ=30° with θ=90°. Meanwhile, the pressure drop increases about 57.8 % at the same time. Because of the excellent capability of heat transfer, the arrangement of θ=30° is frequently used for industrial applications as the pressure loss is acceptable.
3. Comparing the overall performance of different θ reflected by the ratio of *j* factor and *f* factor, the trend shows that 45° have an excellent performance. Meanwhile, the advantages are more obvious with the increase of *Re*.

Acknowledgements

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