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Experimental Study on Heat Transfer and Resistance Performance of Elliptical Finned Tube Heat Exchanger with Different Air Entrance Angles

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To save space, sometimes the heat exchanger is necessary to arranged obliquely. In view of the fact that the air entrance flow direction is not orthogonal to the heat exchanger surface in many cases for the direct and in-direct air cooling system, the present study describes the experimental system about the elliptical finned tube heat exchangers inclined towards the air oncoming flow. The inclined angles between the minor axis of the oval tube and the oncoming air flow are 90° and 60° separately. The heat transfer and pressure drop characteristics are investigated. The results show that the overall heat transfer coefficient of 60° is about 9.96 % smaller than that of 90° when the other conditions are the same, while the pressure drop has a significant change when the air velocity is big to some extent. At last the correlations of the convection heat transfer coefficient and resistant coefficient of the air side are acquired. And some conclusions are drawn upon the effects of the air entrance angles on the heat transfer and resistance performance of the elliptical finned tube heat exchanger.

1. Introduction

Sometimes, a heat exchanger is necessary to arranged obliquely to save space. The air oncoming flow direction has a major impact on the heat and resistance characteristics of the air-cooled radiator. It is a widespread problem that the air inlet direction is not orthogonal to the radiator. Circular tubes are used in most of the finned tube heat exchanger. It is proven that using elliptical finned tube can effectively reduce the flow resistance of fluid in the fin side, and lower the energy consumption. Therefore, the research and development of elliptical tube-fin heat exchanger is paid more attention.
The fin-and-tube heat exchangers with circular tubes or oval tubes were studied experimentally and numerically (Jang and Yang, 1998). The results showed that the mean heat transfer coefficient of an elliptic finned tube was 35-50 % of the corresponding circular finned tube with the same tube perimeter, while the pressure drop of the oval finned tube was only 25-30 % of the round finned tube configuration. Erek et al. (2005) had researched the influence of the geometry parameters on the heat transfer of straight fin-and-tube heat exchangers, using the CFD technology. They indicted that the heat transfer would be enhanced while the pressure drop could be reduced through increasing the ellipticity of the elliptical fined tube.

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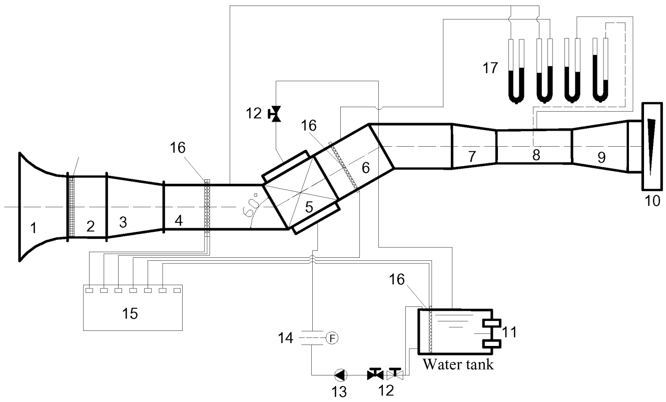
In recent years, there are many studies on circular finned tube heat exchangers (Wang and Chi, 2000; Tang et al., 2009; Xie et al., 2009), while few studies on elliptical finned heat exchangers, especially for the oblique arrange. So the present paper studied the difference of the heat transfer and resistance performance when the air entrance angles are 60 ° and 90 ° of elliptical finned tube heat exchangers.

2. Experimental Apparatus and Procedures

The experiment is conducted in an open wind tunnel. The main features of the experimental apparatus in the system are schematically shown in Figure 1, including two loops: hot water loop and air loop. The hot water loop provides hot fluid at the temperature of 60 °C to the oval tubes of the test core. The air loop is supplied to blow air across the finned tube heat exchanger of the test core, which is positioned in the test section. In the experiment process, the air flow from the laboratory room is induced by the blower, to pass successfully through the pre-duct, test section, after-duct and pitot-tube meter.



1-Entrance; 2-Transition section; 3-Contraction section; 4-Straightening section; 5-Test heat exchanger; 6- Straightening section; 7-Contraction section; 8-Flow metering duct; 9-Expansion section; 10- Blower; 11-Electric heating rod; 12- valve; 13-Water pump; 14-Turbine flowmeter; 15-Data acquisition system; 16-Thermocouples grid; 17-U tube manometer

Figure 1: Schematic of experimental apparatus and system

Air is induced to the wind tunnel by a frequency modulation blower. The air temperature of inlet and outlet across the test core are measured by two sets of multi-point T-type copper-constantan thermocouple grids. Each set contains twelve calibrated thermocouples within the accuracy of 0.2 °C. Meanwhile, the air velocity is measured by a Pitot-tube meter, which is located in the flow metering duct far downstream of the test core, and the wall static pressure of the air before and after the test cores are surveyed by a U-tube water column manometer. The Pitot-tube meter is connected to the microbarometer (at small flow rates) or U-tube water column manometer (at high flow rates).

Hot water is brought to the heat exchanger by a pump from a hot water tank, which can heat the water to 60 °C with electric heating rods. The E-type armoured thermocouples are used to measure the water temperature, and there are four thermocouples placed in the water inlet and outlet on each side respectively. Furthermore, the flow rate of water is tested by a turbine flow meter. At the same time, the static pressure of water side is acquired by Rosemount pressure transmitters.

The experimental samples are covered by the foam insulation of about 10 mm thick to reduce the heat loss to surroundings. More attention is paid to the heat balance to ensure the steady state of heat exchange before writing down the data in the experiments. During the data-acquisition procedure, each measured value is read at least five times, starting reading data no less than forty minutes later when a working condition is changed. And the arithmetic mean of each recorded data is used for checking the heat balance between the energy gain of the air and the energy loss of the water. Then the data with the smallest thermal equilibrium error between air side and water side will be saved. In all the remaining data of the tests, the error is less than 7 %.

Two air entrance angles (90°, 60°) are tested in present study. The test sample is an elliptical finned tube heat exchanger showed in Figure 2, which is only one small part of the air cooling tower used in power plant or chemical factory. All tubes and fins are made of carbon steel. Due to the parameters of the air side having an important effect on the heat transfer and resistance performance of the heat exchanger with different air entrance angles, so the present paper only describe the air side in detail.

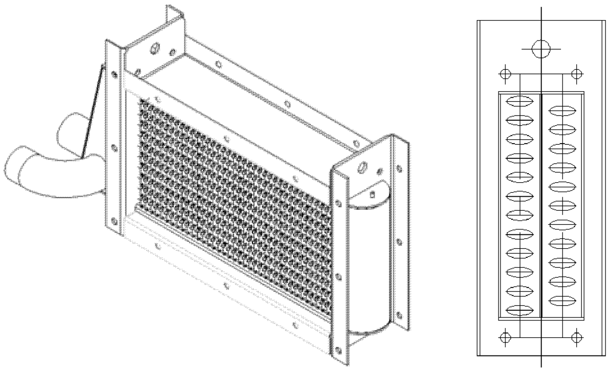


Figure 2: Double tube pass elliptical tube finned heat exchanger

3. Data Reduction

To acquire the air side heat transfer and friction characteristics, Nusselt number (*Nu*) and friction factor (*f*) of the heat exchangers, and find out corresponding power-law correlations of *Nu* vs. *Re*, and *f* vs. *Re* for each air entrance angle, the experimental data should be reduced. The thermal properties of the air are determined by the average value of the inlet and outlet temperatures of the air side and the thermal properties of the water are evaluated by the inlet and outlet temperatures of the water side. The data reduction process is illustrated in detail as follows. The heat transfer rate Q_{w} of water side is given as

$$Q_{\mathrm{w}} = \rho_w \cdot q_{\mathrm{w}} \cdot c_{\mathrm{pw}} \cdot \Delta T_{\mathrm{w}} \tag{1}$$

where q_{w} is the volume flow rate of water.

The heat transfer rate Q_a of air side is calculated by

$$Q_a = m_a \cdot c_{pa} \cdot \Delta T_a \quad (2)$$

where m_a is the air mass flow rate.

The average heat transfer rate of the air and the water side is regarded as the total heat transfer rate, which is computed by

$$Q_m = (Q_w + Q_a)/2 \quad (3)$$

The overall heat transfer coefficient based on the heat exchange area of the air side is given as

$$k = \frac{Q_m}{\Delta T_m \cdot A_o} \quad (4)$$

where ΔT_m is the logarithmic mean temperature difference, and A_o is the heat transfer area of the air side, including the fin area and the base tube area.

The water side heat transfer coefficient h_i, is calculated from the Gnielinski formula (Yang and Tao, 1998)

$$Nu = \frac{(f/8)(Re-1000)Pr}{1+12.7\sqrt{f/8}\left(Pr_f^{2/3}-1\right)}\left[1+\left(\frac{d}{l}\right)^{2/3}\right]c_t \quad (5)$$

where the parameters can be found in reference (Yang and Tao, 1998).

To get the convection heat transfer coefficient of the air side, thermal resistance separation is adopted. The overall heat transfer resistance can be defined as

$$\frac{1}{kA_o} = \frac{1}{h_i A_i} + \frac{1}{2\pi\lambda l}\ln\frac{d_o}{d_i} + \frac{1}{h_o \eta_o A_o} \quad (6)$$

In Equation (6), h_o is the air side convection heat transfer coefficient, and η_o is the overall finned surface efficiency, which can be acquired from the formulas in reference (Wang, 2001). Only h_o and η_o are unknown parameters. So they will be got through some iteration processes by programming with Borland C++ Builder 6.0.

The heat transfer and friction characteristics of the heat exchanger are presented in the following dimensionless forms

$$Nu = h_o D_c / \lambda \quad (7)$$

$$Re_{D_c} = \rho v_{max} D_c / \mu \quad (8)$$

$$f = \frac{2\Delta p}{\rho v_{max}^2}\frac{D_c}{L} \quad (9)$$

where v_{max} is the velocity at the minimum free flow area of the air side.

4. Results and Discussion

All the measurement instruments had been adjusted to verify the test results and methods before the experiment started to run. Figure 3 shows the correlations of *Nu* with Re_{Dc} and *f* with Re_{Dc} when the air entrance angle is 90°. And Figure 4 gives the heat transfer and resistance performance when the air inlet angle is 60°. From Figure 3 to Figure 4, it can be seen that most of the experiment data are in the range of the fit errors which are less than 8 %. Figure 5 shows the comparison of the overall heat transfer coefficient and pressure drop with different air entrance angles (90°, 60°) when the water velocity *v* in each oval tube is 1.1 m·s^{-1}. It is observed that a marked decrease of the heat transfer while the pressure drop has a slight increase when the air entrance angle changes from 90° to 60°. Comparing the overall heat transfer coefficient of 60° with that of 90°, the former reduces by 9.96 %. For the pressure drop, it can be seen that the pressure drop varies slightly at a small air velocity, while the former becomes bigger and bigger than the latter when the air velocity is more than 1.8 m·s^{-1}, and it increased by 6.66 % when the air velocity reached at 4.4 m·s^{-1}

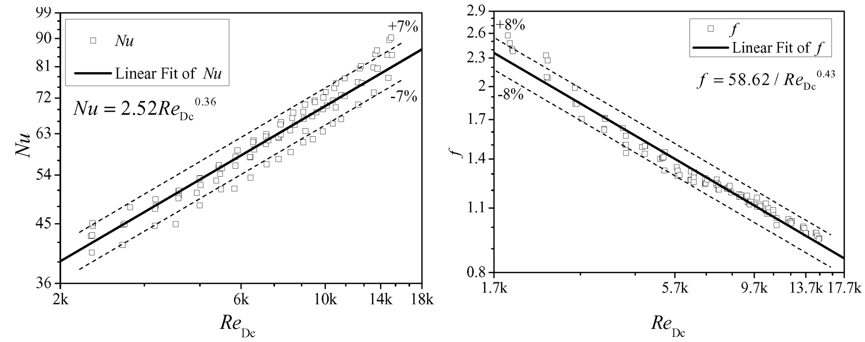


Figure 3: The correlations Nu with ReDc and f with ReDc at air entrance angle=90°

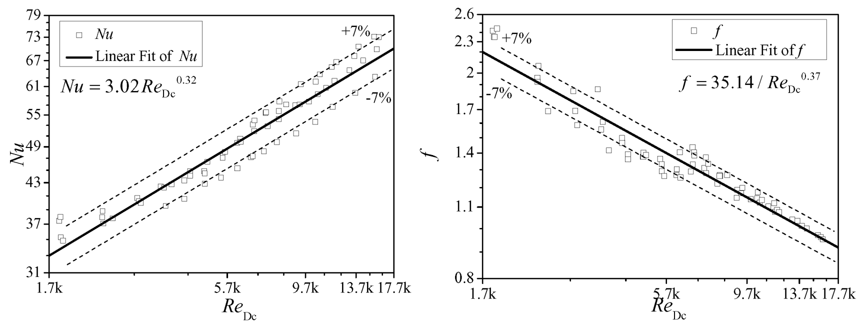


Figure 4: The correlations Nu with Re_{Dc} and f with Re_{Dc} at air entrance angle=60°

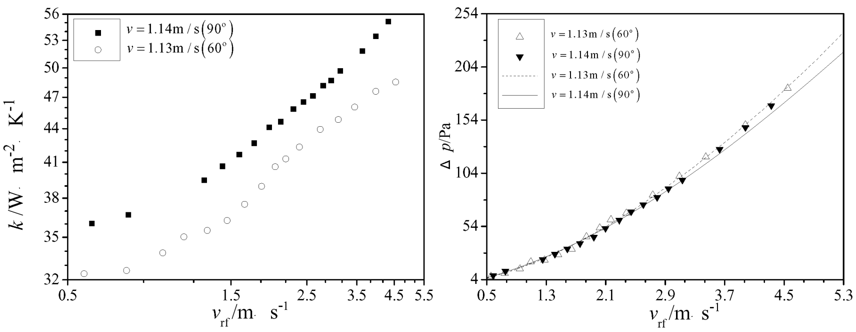


Figure 5: Comparison of overall heat transfer coefficient and pressure drop with different air entrance angle at v=1.1 m·s-1

Because it is not distinct of pressure drop showed in logarithmic coordinates, the trend of pressure drop with the air velocity is shown in Figure 5 in linear coordinate system. The decrease of the overall heat transfer coefficient of 60 % may be caused by the non-uniform air flow when the heat exchanger is placed obliquely.

5. Conclusions

Experimental studies of the air side thermal hydraulic performance of elliptical finned tube heat exchanger with the air entrance angle of 90° and 60° are conducted. The overall heat transfer coefficient has a remarkable decrease when the air entrance angles change from 90° to 60°, while the pressure drop does not change observably. So it is not worth placing the heat exchanger obliquely if only from the consideration of enhancing heat transfer. Otherwise, in order to save space in practical application, the heat exchanger needs to be positioned obliquely. And only two air entrance angles are studied in the present work. So the heat transfer and resistance performance of other air entrance angles of heat exchangers need to be further studied to get the proper angles.

Acknowledgment

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