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AIR-SIDE PERFORMANCE OF FINNED TUBE HEAT EXCHANGERS UNDER DEHUMIDIFYING CONDITIONS

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Abstract. *This paper analyses the cooling and dehumidifying of air. Experimental data have been obtained in a testing wind tunnel for the wavy finned tube heat exchanger. The experiments cover a wide range of air velocities. The airside performance, or cooling capacity, dehumidification, pressure drop and heat transfer coefficient of the coil, is shown for different conditions. An enthalpy-based effectiveness model was used for calculating the heat and mass transfer and for predicting the cooling of moist air, considering the entire coil wet.*

Keywords: *Air cooler, Fin-and-tube heat exchanger, Plain wavy fin, Dehumidification*

1. INTRODUCTION

Finned tube heat exchangers are used in a variety of applications of air-conditioning, refrigeration and industrial process. Generally, air coils consist of tubes through which water, oils or refrigerants are forced to flow inside them, while air is directed across the tubes and fins.

The heat exchanger in this study using chilled water and air, is widely used in cooling and dehumidification. The analysis involves some difficult due to simultaneous heat and mass transport of the moist air over the coil surface, causing the exchange surface to be partial or totally wetted. In the cooling process of air, the surface temperature decreases and when it is lower than the dew point temperature, the air becomes saturated with water vapor that begins to condense on the surface.

The heat transfer characteristics of fin-and-tube heat exchangers under dehumidifying conditions have been studied, and the experimental and theoretical of plain fin data in wet condition published (e.g. Kuehn et al., 1998; Elhmahdy and Mitalas, 1977; Braun et al., 1989; Yan and Sheen, 2000; Wang et al., 1997, Pirompugd et al., 2007, Liu et al., 2010).

This paper presents the results of experiments carried out with a wavy fin-and-tube heat exchanger to study the effect of air velocity on the airside performance, like cooling capacity, dehumidification, pressure drop and heat transfer coefficient. Moreover, the enthalpy-based effectiveness model was used to analyze the data considering the coil in wet conditions (Kuehn et al., 1998). The correlations for Colburn factor and friction factor were also considered. In this way, the objective of the experimental and theoretical study is to provide relevant performance data and discuss the heat and mass transfer model.

2. EXPERIMENTAL SET UP AND PROCEDURE

The experimental setup shown in Fig. 1 consists a wind tunnel for the finned tube heat exchanger, which provide controlled temperature, relative humidity and air flow rate, and a chilled water circuit. Air is forced through the tunnel using upstream variable speed centrifugal fan. A chiller supplies the cold water. In the experiments were measure at the air side: dry and wet bulb temperatures, differential pressure in heat exchanger and volumetric flow rate, or air velocity, using two plate nozzles and two pressure transmitters. In the waterside were measured temperatures at inlet and outlet, and flow rate. During the experiments, these operating parameters were monitored in real time by an Agilent 34980A data acquisition system, controlled by a personal computer.

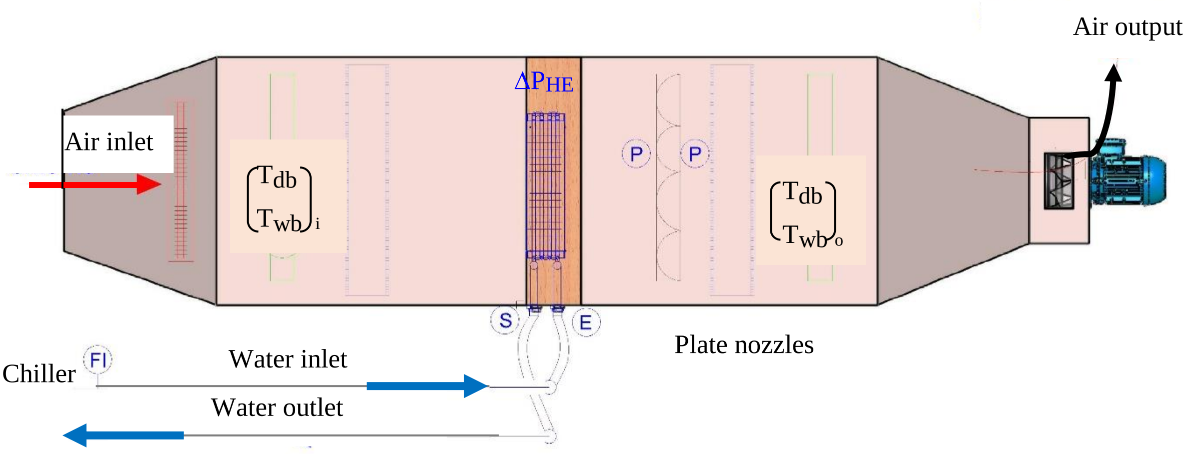
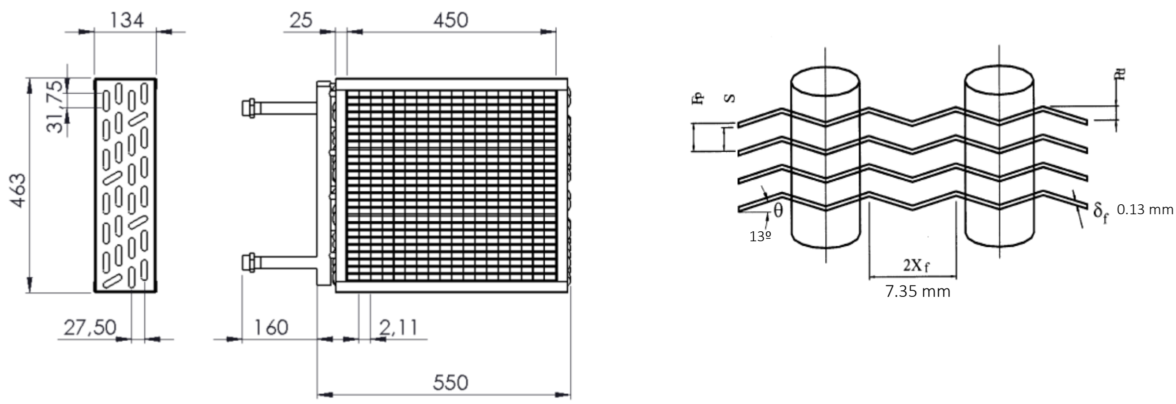


Figure 1. Scheme of the experimental facility and instrumentation.

The heat exchanger was installed at the wind tunnel, and it was insulated to protect from heat loss and air leakage. It has a staggered tube arrangement with 4 rows of copper tubes with 14 tubes per row, totaling 56 tubes with internal diameter of 12.7 mm. The water flow is distributed in 7 circuits. The fins are wavy aluminum plates with spacing of 2.11 mm. The compactness of heat exchanger is 413 m²/m³ and the ratio of minimum free-flow area to frontal area is 0.56. Figure 2a indicate the geometrical configuration of the finned-tube heat exchanger and Fig. 2b the wavy fins geometry on tube array.



(a) (b)

Figure 2. (a) Frontal and lateral representation of finned tube heat exchanger; (b) geometry of wavy fins on a tube array (Huzayyin et al., 2007).

2.1 Test conditions

The tests were performed for air flow rates in the range of 2190 to 2900 m³/h (air face velocities between 3.2 and 4.5 m/s) with water flow rate fixed in 70 L/min. The inlet air temperature of the heat exchangers was 30 ºC, adjusted by electrical resistances (relative humidity of 55%), and the inlet water was maintained at 8 ºC. During the tests, the condensate was collected.

2.2 Data reduction and cooling and dehumidification analysis

For the conditions of the experiment, the air dew point temperature is greater than cold water temperature. Considering the low temperature gradient at fin, the fins tip temperature is lower than the air dew point temperature, and then all fins and tube surfaces are wetted. The coil becomes under a totally wet condition and it is subjected to accumulation of condensate.

The conventional ε-NTU relations cannot be directly used to simulate wet coil because of the latent energy release associated with the dehumidification of the air and the simultaneous heat and mass transfer processes that are occurring. An approach was used for data reduction and comparison. The semi-empirical enthalpy-based effectiveness model of cooling, proposed by Braun et al. (1989), assumed that the entire coil to be wet, and therefore the capacitance of the air-side is computed using the saturation' specific heat capacity and the energy transfer due to the heat and mass transfer is characterized using the wet conductance (UA_{wet}). The effectiveness of the coil, ε is computed using the appropriate ε-NTU solution for the heat exchanger configuration and the ε is based on the enthalpy and the humidity ratio of the moist air stream, rather than its temperature, according to:

$$\varepsilon = \frac{i_{a,in} - i_{a,out}}{i_{a,in} - i_{a,out,min}} \tag{1}$$

$$\varepsilon = \frac{\omega_{a,in} - \omega_{a,out}}{\omega_{a,in} - \omega_{a,out,min}} \tag{2}$$

where $i_{a,in}$ and $\omega_{a,in}$ are the enthalpy and humidity ratio, respectively, of the inlet air and the quantities $i_{a,out,min}$ and $\omega_{a,out,min}$ are the minimum possible values of enthalpy and humidity ratio. These minimum values are consistent with the air leaving saturated and the inlet temperature of cold fluid.

The wet conductance is given by:

$$UA_{wet} = \frac{1}{R_{total}} \tag{3}$$

where the thermal resistance, R_{total}, include the water and the tube inner surface, R_{in}, the fouling resistance, $R_{f,in}$, the conduction resistance of the tube, R_{cond}. The air side resistance is function of heat transfer coefficient, h_a, fin efficiency, η_a, total area (primary and secondary fin surface area), A_{total}, and specific heat capacity the air, C_a, and in saturation, $C_{a,sat}$, given by Eq. 5.

$$R_{total} = R_{in} + R_{f,in} + R_{cond} + \frac{C_a}{h_a \eta_a C_{a,sat} A_{total}} \tag{4}$$

$$C_{a,sat} = \frac{[i_a(T_{a,in}, p, RH = 1) - i_a(T_{a,out}, p, RH = 1)}{T_{a,in} - T_{a,out}} \tag{5}$$

Moist air properties are calculated by EES based on ASHRAE Fundamentals Handbook (1997) and it is assumed that air behaves as a non-interacting ideal mixture with water vapor as a dilute component, which introduce small errors at these conditions.

The analysis on the air-side for calculating the heat transfer coefficient, h_a, from the Colburn factor, jh, and the friction factor, f, was performed using correlations proposed by Wang et al. (1997) and Wang (2000), considering the dehumidification conditions, and the fin efficiency by Madi et al. (1998).

3. RESULTS

Figure 3 shows the results of the heat exchanger's cooling capacity as a function of air face velocity. The total capacity, measured and calculated, is very close, with differences of less than 10%. However, the latent capacity is of the same magnitude as the sensible capacity, showing that the latent capacity is oversized because the model assumes that the entire surface of the exchanger is wet. When looking at Fig. 4a, which shows the outlet dry bulb temperatures, it is found that the calculated values are always higher than the measured values, which makes the calculated sensible capacity lower than the measured values. In any case, the trend presented by both data sets is consistent, the outlet temperature increases with increasing air face velocity (or airflow), resulting in a decrease in the air temperature variation (Fig. 4b).

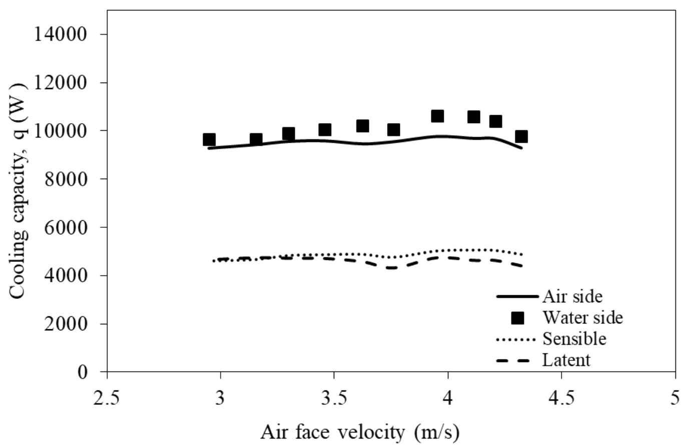


Figure 3. Cooling capacity for different air face velocity.

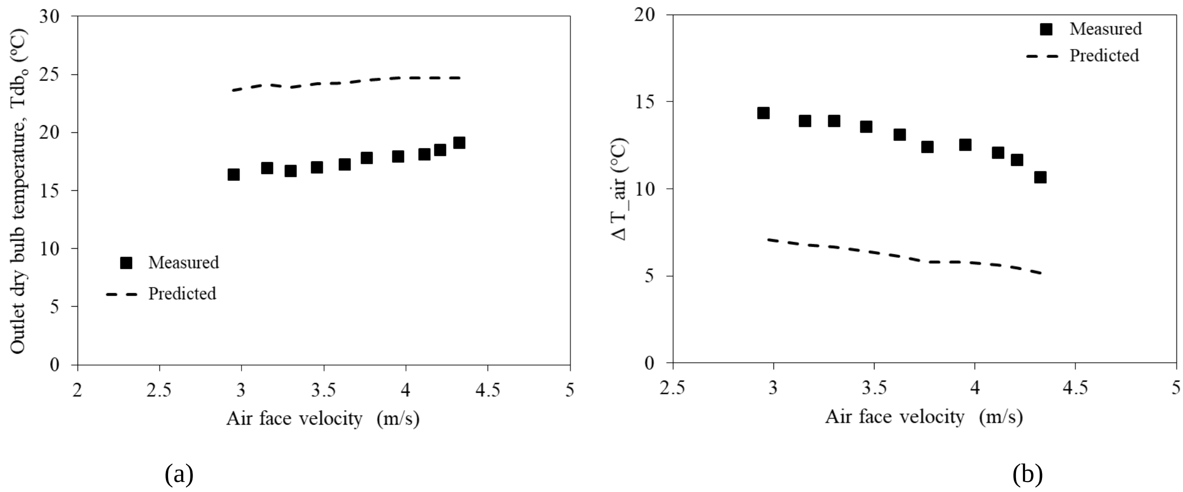


Figure 4. (a) Dry-bulb temperature at outlet; (b) Air different temperature.

Figure 5 shows the variation of absolute humidity (inlet and outlet) as a function of face velocity. During testing, the absolute air humidity at the heat exchanger inlet remains nearly constant. For the tested velocity range, moisture removal from the air was also almost constant.

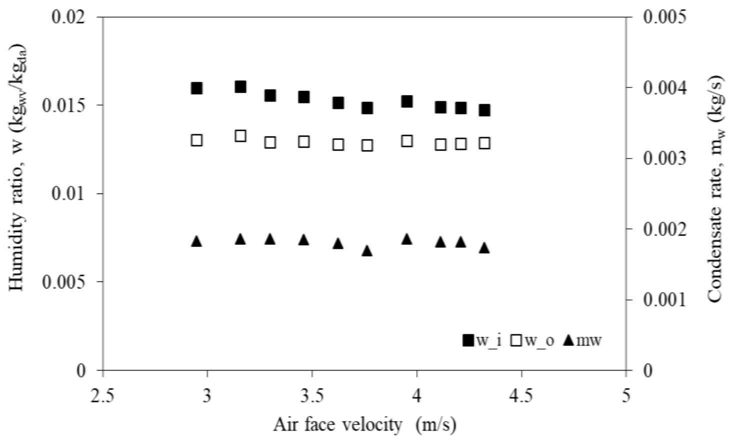


Figure 5. Humidity ratio at inlet and outlet and condensate rate for different air face velocities.

In Fig. 7a the values of air pressure drop as a function of face velocity are shown. Increasing the air face velocity, increases the shear stress between the air and the coil surface and this significantly increases the pressure drop across the coil. The model used for the friction factor calculation, showed in Fig. 6, leads to a good agreement with the measured results. However, it should be noted that the model does not take into account the increase in pressure drop due to condensate formation. Figure 7b presents the heat transfer coefficient, *ha*, calculated by Colburn Factor, *jh*, with a correlation specific for the wavy fin. According the air velocity increases, consequently *Re*, the coefficient increases, but in the average the value is 64 W/m²K.

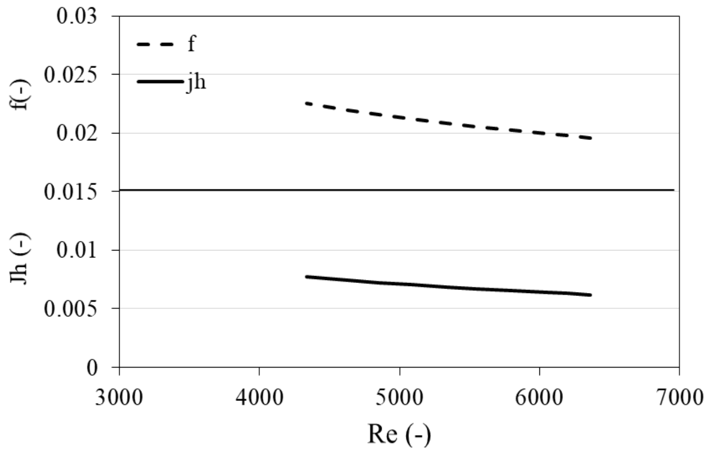
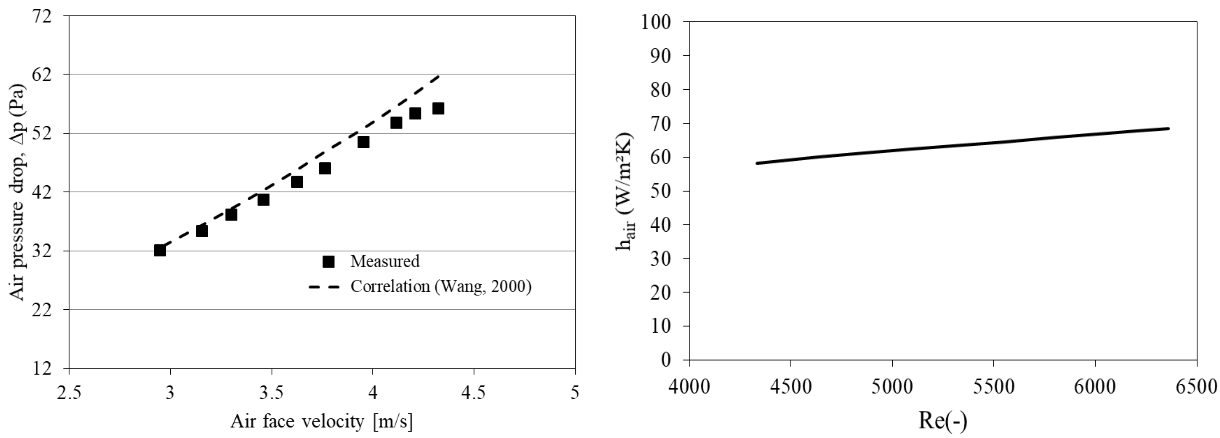


Figure 6. Friction factor and Colburn factor with Reynolds number.



(a) (b)

Figure 7. (a) Air pressure drop and (b) heat transfer coefficient.

4. CONCLUSIONS

The work presented the first results of an experimental study of air cooling and dehumidification in an fin-tube heat exchanger with chilled water. The influence of air velocity variation on outlet conditions, temperatures, humidity ratio, and cooling capacity was verified.

The model used considering heat and mass transfer to a totally wet surface showed results in the range of about 10% compared to the experimental values, however it was found that the model overestimates the latent capacity of the coil. Regarding the pressure loss even disregarding the condensate in the surface of the tubes and fins, it represented the behavior in an appropriate way.

5. ACKNOWLEDGEMENTS

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