PERFORMANCE OF EVAPORATOR ON AIR SIDE UNDER CONDITIONS OF WET AND PARTIALLY WET SURFACES

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Abstract. *Due to the heat and mass transfer characteristics, the cooling and dehumidifying process of evaporator are complex. In this sense, the present paper details the heat and mass transfer coefficients of moist air over surfaces for different cooling mode: wet, and partially wet. Experimental data was obtained in a conventionl refrigerator operation with R-22 refrigerant. A modeling of the classical evaporator was succintly revewed by using finned surfaces wavy correlations for air-side. The mode of cooling was determined through a latent air–side convective heat transfer coefficient by correlating the experimental data with the model. The heat transfer rate in the evaporator was identified. The wet-bulb temperature of the air inlet and out of evaporator was corrected due to the effect of low velocity of convection and evaporating on moistened bulb. In the literature have values of fouling factor for oil-bearing refrigerant and a fouling factor for air side coefficients for evaporator in operation for five years. The results have shown new values for theses coefficients for evaporator in operation for eight years identifying a reduction of the nominal refrigeration capacity.*

***Keywords**: Refrigeration, modeling, wet surface, evaporator.*

1. INTRODUCTION

Heat exchanger are encountered extensively in the air-conditioning, heating, and refrigeration industries. The heat transfer of finned-tube evaporator was evaluated in a Brazilian household air-conditioning. There are some thermal resistances involved on heat exchangers. The dominance resistance is on the air-side (Wang et al, 1999). To improve the overall heat transfer and reduction the resistance in Air-side, the surface of the tube is finned. The fin patterns of the evaporator are wavy. Correlations were used to predict heat transfer rates according to specific thermal and hydrodynamic operating conditions. In air side, the model accounts the thermal resistance of a wetted coil obtained by the convective heat transfer and mass transfer or phase change of the water vapor. The air is dehumidified and the heat transfer is increased. Many air-conditioning in Brazil have more then five years of used, accordingly the present paper model the performance of evaporator, with the evaluation of thermal resistances and the heat transfer coefficients on evaporators and compare them with experimental data taken from an experimental data of air-conditioning.

The main contribution of the paper is show the effect of air side fouling and crumpled fin on the nominal refrigeration capacity for evaporator with eight years of use.

2. MODEL

The heat transfer from air to refrigerant is the main function of evaporator operates. So, the heat transfer rate may be identified with various thermal resistances. Extensive investigation has been proposed for a modeling for dry coils. Relatively, researches on the cooling and dehumidifying coils are limited due to the complication of simultaneous heat and mass transport of moist air to cooled fin surfaces, (Oskarsson et al; 1990), (Wang, J., Hihara, E., 2003).

The total thermal resistance comprised the resistance of air, air-side fouling, tube, fin-to-tube contact, refrigerant-side fouling and refrigerant. The system model takes into account each one of the thermal resistances. The temperature of refrigerant is constant in the boiling portion and it changes in the superheating. The temperature difference between the refrigerant and the cooling fluid is represented by the LMTD (log mean temperature difference), considering constant the temperature of refrigerant. Although the temperature difference is higher, the result provides has reasonable accuracy. The following equation represents the heat transfer rate.

$$q = \frac{LMTD}{\dfrac{1}{(h_a + h_{lat}).A_t.\eta_s} + \dfrac{1}{h_{f,a}.A_t.\eta_s} + \dfrac{1}{R_{cont}.A_{cont}} + \dfrac{1}{2.k.\pi.L}\ln\dfrac{D_{ext}}{D_{int}} + \dfrac{1}{h_{f,int}.A_{int}} + \dfrac{1}{h_{int}.A_{int}}} \tag{1}$$

Where: h_a is convective heat transfer coefficient [W/m^{2}K]; h_{lat} is latent heat transfer coefficient [W/m^{2}K]; η_s is surface efficiency, A_t is total air side surface area [m^2]; $h_{f,a}$ is air side fouling factor [W/m^{2}K]; R_{cont} is contact conductance [W/m^{2}K]; k is tube heat conductivity [W/mK]; A_{cont} is contact interference area [m^2]; $h_{f,int}$ is refrigerant side fouling factor [W/m^{2}K]; A_{int} is tube inside surface area [m^2] ; h_{int} is boiling heat transfer coefficient [W/m^{2}K].

The convective heat transfer coefficient for the air side of the evaporator may be evaluated in terms of Colburn factor (j) for wavy fin configuration. Wang et al. (1999) construct the correlation for performances of wavy fin-and-tube based from a total of 27 samples of heat exchangers, which is defined as follows:

$$j = \frac{Nu}{\mathrm{Re}.\mathrm{Pr}^{1/3}} = 0{,}324.\mathrm{Re}_{Dc}^{j1}\left(\frac{F_p}{P_l}\right)^{j2}(\tan\theta)^{j3}\left(\frac{P_l}{P_t}\right)^{j4} N^{0{,}428} \tag{2}$$

Where: Re_{DC} is Reynolds number based on tube collar diameter (D_c is fin collar outside diameter D_o+2.δ_f); δ_f is fin thickness [m]; F_P is fin pinch [m]; P_l is longitudinal tube pinch [m]; $\mathbf{P_t}$ is transverse tube pinch [m]; N is row number; $j_{1,2,3,4}$ are constants.

The dehumidification of air is associated with the latent air-side convective heat transfer coefficient. The effect of air dehumidification is to increase in the heat transfer by the change of phase of water vapor. The latent heat transfer can be expressed by the Lewis number (Le) and sensible heat transfer coefficient (h_a) [W/m^{2}K]:

$$h_{lat} = \frac{h_a.i_w.C}{Le.C_{p,da}} \tag{3}$$

$$C = \frac{w_a - w_{s,\min}}{T_a - T_{s,\min}} \tag{4}$$

Where: i_w is condensation heat of water; $C_{p,da}$ is specific heat of dry air. Parameter C is a rate of the difference of specific humidity at air temperature (T_a) and saturation specific humidity at fin condensate surface temperature ($T_{s,min}$) by the difference of temperature respective.

The fouling factor values proposed for Rosenhow et al. (1985), apud (Oskarsson at al., 1990) for the five year old evaporator coil for oil-bearing refrigerant vapors is 2.84 kW/°C.m^2 and for air-side fouling factor operation in industrial air is 2.84 kW/°C.m^2. The fouling factor coefficient for evaporators in operation for eight years was made by comparative analysis with the experimental data.

The Contact resistance appears when the heat is exchanged across an interface where two surfaces are in imperfect contact. The contact resistance between the tube and the fin is determined from the contact interference area and contact conductance. Wang e Chi (2000) comment that the contact conductance is range of 11.0 to 16.0 kW/m^2 K. On the other hand, Wang et al. (2000) show this parameter from range of 10.0 to 15.0 kW/m^2 K. In the present paper was used the mean value of 13.0 kW/m^2 K and contact area was the same that external tube area.

The heat transfer process on the refrigerant side has to be considered in two regions. The first one is the two-phase region and the second one is the superheated region. The heat transfer aspects of two-fase are involved because of a combination of liquid and vapor refrigerant exists in the tube of evaporators and condensers. Altman et al. (1960) presented average coefficients for R-22 evaporating at temperatures from 4.4 to 26.7°C in a tube diameter 8.7 mm on 2,4 m long. Coefficients were determined for approximately 15% vapor quality changes. The range investigated was x= 0.20 to superheat.

$$\frac{\mathrm{h_{int}.D_{int}}}{\mathrm{k}} = 0{,}0225.\left[\mathrm{Re}^2\ .\mathrm{Kf}\right]^{0{,}375} \tag{5}$$

$$\mathrm{Kf} = \frac{\Delta \mathrm{x.i_{lv}}}{\Delta \mathrm{L.g}} \text{ is boiling number} \tag{6}$$

Where: Δx is difference refrigerant quality; i_{lv} is condensation refrigerant enthalpy; ΔL is section length; g is gravity.

The heat transfer average coefficients for refrigerant on superheated process are estimated by Dittus-Bolter correlation applicable to turbulent flow in a circular tube.

2.1 Cooling mode

The dominance resistance is on the air-side composed for heat transfer sensible and latent coefficients. To evaluate the model with accuracy it is important to distinguish the method of cooling. In the process of the air is transferring simultaneous heat and mass (water) to or from a wetted surface leads the straight-line. The condition of the air drives toward the saturation line at the temperature of the wetted surface (averaged fin surface). Wang, J., Hihara, E., (2003) show three modes of cooling defined for the temperature when moist air is cooled over a finned tube surface. For an inlet state, when:

1. The dew point temperature is equal to, or less than the tube outer surface temperature $T_{adew} \leq T_{bo}$. Process totally dry.
2. The dew temperature is equal to, or greater than the fin tip temperature, $T_{adew} \geq T_{ftíp}$. Process totally wet.
3. The dew temperature is between the fin tip temperature and the fin base temperature, $T_{ftíp} > T_{adew} > T_{bo}$. Process partially wet with two distinct processes:
   a) When $T_{ftíp} > T_{adew} > T_{avf}$, Process with net vapor condensate.
   b) When $T_{avf} \geq T_{adew} > T_{ftíp}$, Process without net vapor condensate.

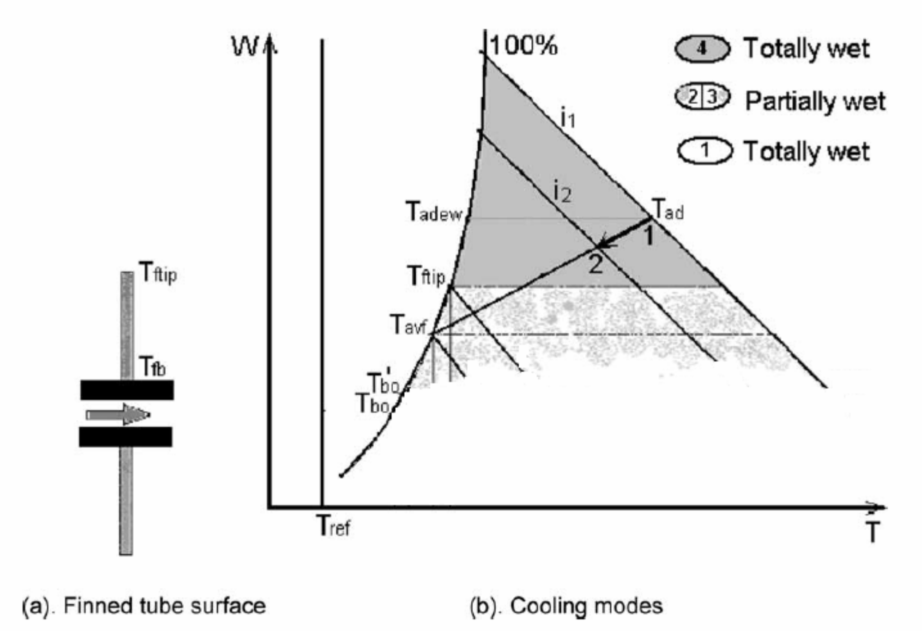

Figure 1. Cooling mode of a moist air over a finned-tube surface. Source (Wang and Hihara, E., 2003).

The average fin surface temperature in model can be expressed as:

$$q = \eta_s.h.A_t.(T_{ad} - T_{bo}) = h.A_t.(T_{avf} - T_{bo}) \tag{7}$$

3. EXPERIMENTAL APPARATUS

In order to illustrate the procedure presented in this paper was employed a test bed composed by an conventional air conditioning system of the mark Springer Carrier with the cooling power of 10.000 Btu/h (2.930 W), supply voltage of 220 V with rated current of 8.5 A and flow the air on evaporator of 495 m^3/h (0,1375 m^3/s), on fig. 2.

In the test bench temperature and pressure of circuit of refrigerant between evaporator have been measuremend. The pressure sensors used are from the brand Gitta properly set in the tubing. Two pressure sensors of low pressure were used. The sensors work in low-pressure gauge range of -206 to 827 kPa (-30 to 120 psig), with a resolution of 7kPa (1 psig). The sensors have been calibrated to pressure through the use of the system of calibration Lamon model MD-04-015 which has an error of 0.15%. The results of the calibration are respectively ± 0.5 psi (3.4 kPa) and ± 0.9 psi (6.2 kPa) for two low-pressure, all calculated with reliability of 90% from t-Student.

In measurements of the two points of the refrigerant temperature it was utilized type K thermocouples. Sporket (2001) found that the temperature of the refrigerant is the same of the wall of copper tubing. The temperature sensors were set on the external wall of the tube. The sensors were isolated from the outside environment with the use of mass and overlapping of the plastic tape which acts as a barrier preventing the steam condensing water vapor contained in the air, which measures the temperature change.

The tube evaporator was made of cooper and the fins were made of aluminum. The experimental parameters of evaporator are show on tab. 1.

Table 1. Characteristics of air conditioning

D_{ext} external tube diameter	11.44×10^{-3} m
D_{int} internal tube diameter	8×10^{-3} m
D_c fin collar outside diameter	11.80×10^{-3} m
F_p fin Pitch	1.68×10^{-3} m
L length of evaporator [m]	0,393 m
N number of longitudinal tube row	3
P_l longitudinal tube pitch	14.48×10^{-3} m
P_t transverse tube pitch	24.06×10^{-3} m

The view of the apparatus of the evaporator instrumented is in Figure 2.

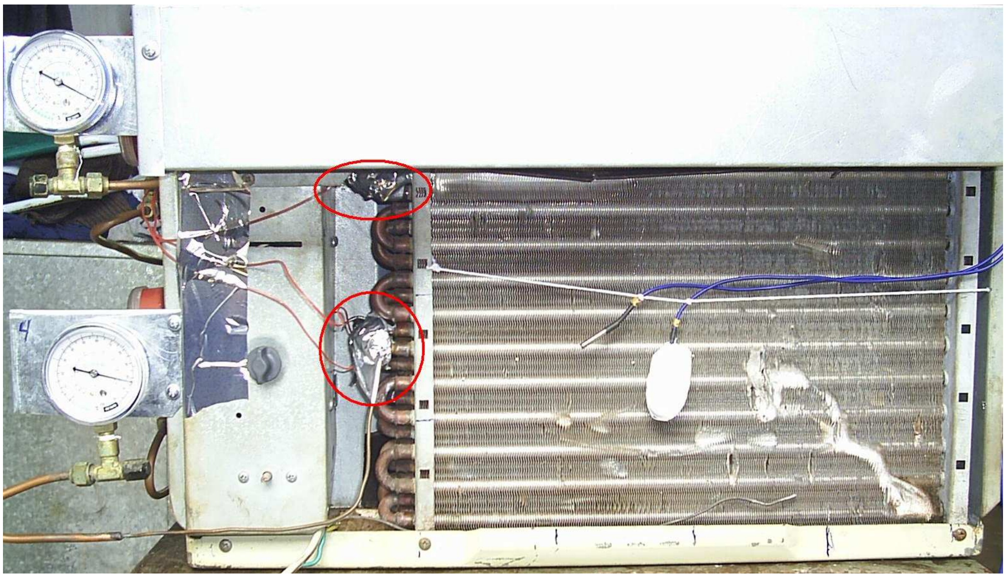

Figure 2. Experiment instrumented, details on the red temperature sensor

The calibration of the thermocouples was accomplished through the use of the oven calibration QuickCal of Isotech which has a range of -12 ° C to 140 C with error of ± 0.4 ° C and the reader of thermocouples 1529 Chub-E4 which obtains an error of ± 0,005 mV in reading. The total error of thermocouple used is ± 0.6 ° C.

The mass flow rate of refrigerant was estimated by the air circuit of evaporator. The air flow is measured by anemometer and the air enthalpy was estimated by the wet bulb and dry bulb temperatures.

4. RESULTS

The tests were driven to evaluate the data. After reaching a steady state condition, a set of data was taken like pressure, temperature of refrigerant, wet bulb and dry bulb temperature of air, the velocity of flow air inlet and outlet evaporator.

The wet-bulb and dry-bulb temperature have been used to measure humidity in air. The process in moistened bulbs is influenced by heat and mass transfer. When the psychrometer is exposed to a moving stream of moist air, evaporation of water and convection heat transfer occurs. This process depends on the velocity of flow air that arrives the steady state for the rate of heat and mass transfer. The wet-bulb temperature of the air inlet and out of evaporator was corrected for the thermodynamic wet-bulb temperature by the correlation of in Threlkeld, (1970). The straight-line law shows that when air is transferring heat and mass (water) to or from a wetted surface, the air condition lead toward with the saturation line at the average fin surface temperature, fig 1. If the wet-bulb temperature is not corrected, some tests do not lead to the temperature of averaged fin surface made by the straight-line in a psychometric chart. The straight-line do not cross the saturation line in a psychometric chart identifying the temperature point.

The first experiments were performed for the air-conditioning having air-side fouling and some fin evaporator crumped. Ten tests were made for different air temperature. The experimental measurements of the air temperature in and out of the evaporator allowed to estimate the average fin surface temperature. Comparing the data of the temperature of the average fin surface with the model, it was noticed that there is a larger thermal resistance between the refrigerant and the fin surface. The coefficient fouling factor ($h_{f,int}$) values for oil-bearing refrigerant vapors were made equalizing the temperature of the theoretical model and experimental measurements. The range of values were 2530 to 722 W/°C.m^2 with an average of 1500 W/°C.m^2 for coil in operation for eight years. The value of coefficient fouling factor ($h_{f,int}$) used was 1500 W/°C.m^2 for model of coil.

The cooling mode influences the evaluation of the coefficient of convection. The cooling can occur totally dry, and with dehumidification of air in process totally wet or partially wet. To determine the cooling mode it is important the correct evaluation of the tube outer surface temperature (T_{bo}), the fin tip temperature (T_{ftip}) and the average fin surface temperature (T_{avf}) to compare with the dew point temperature of air inlet of evaporator. In the evaluation of latent heat transfer coefficient at Eq.(4) is used the parameter C at Eq.(5) that depends on minimum condensate surfaces temperature. Fig. 3 shows that the experimental and model runs considerable agreement for temperature.

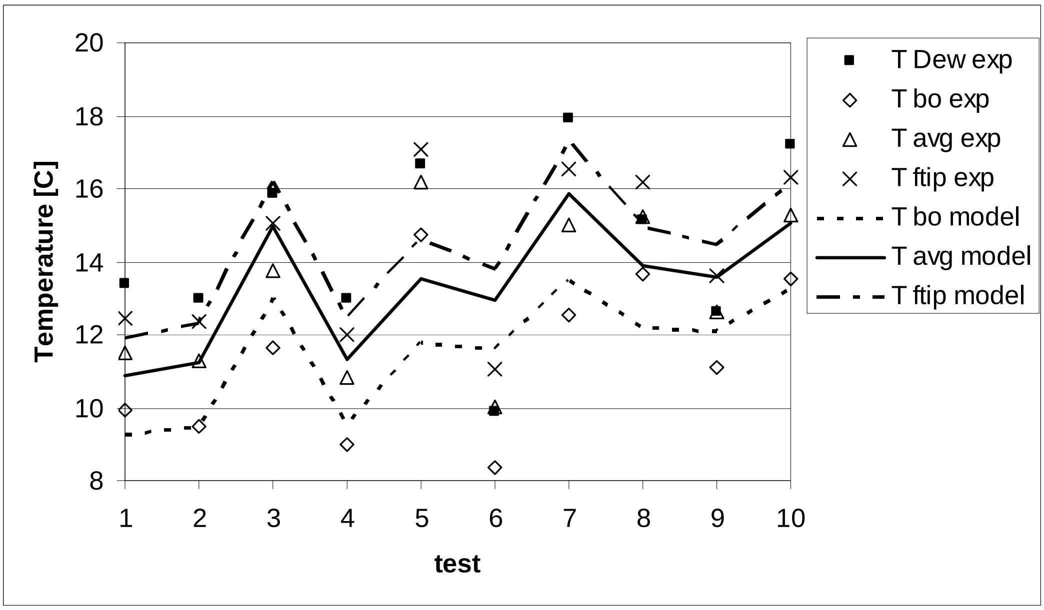

Figure 3. Diagram of fin temperature of model versus experimental

In all tests, the dew point temperature is greater than the tube outer surface temperature; it indicates that the cooling mode is not totally dry. Only in test 6 of model the result, it indicates cooling totally dry.

In the experimental data of test 1, 2, 3, 4, 7 and 10 the dew point temperature is greater than the tube outer surface temperature; indicate that the cooling mode is totally wet. In the model result of test 1, 2, 4, 5, 7, 8 and 10, they indicate totally wet cooling. The data of test, 5, 6, 8 and 9, the dew temperature is between the fin tip temperature and the fin base temperature, indicating that the cooling mode is partially wet. In the model result of test, 3 and 9, they indicate partially wet cooling.

The air-conditioning was in operation for eight years and had air-side fouling and some fin evaporators crumpled, and then it was included an air side coefficient fouling factor. This coefficient was determinated equal to the heat transfer rate. The range of values were 77 to 26 W/°C.m^2 with average of 52 W/°C.m^2. The value of air side coefficient fouling factor ($h_{f,a}$) used was 52 W/°C.m^2 for model of coil. The effect of operation time at fouling factor for evaporators coil in use for five and eight year old is show on Tab 2.

Table 2. Comparation of air side fouling coefficient for evaporator.

	$h_{f,a}$ [W/°C.m^2]	Operation time [years]
Rosenhow et al. (1985)	2840	5
Present paper	52	8

The second experimental was conducted for the same air-conditioning washed and uncrumpled fin evaporators. Ten tests were made for different air temperatures. The coefficient fouling factor ($h_{f,int}$) for oil-bearing refrigerant vapors used at in the model was the same 1500 W/°C.m^2 but as the fin was better, the value of the air side coefficient fouling factor ($h_{f,int}$) used was zero.

The effect of air side fouling and crumpled fin is shown in Fig. 4 which also shows data heat transfer rate obtained of first and second experiment by the experimental measurements and the theoretical model. The average heat transfer rate for first and second experimental is 1447,9W and 1880,3 respectably. The air side fouling and crumpled fin reduces the nominal refrigeration capacity (2930 W) at 15%.

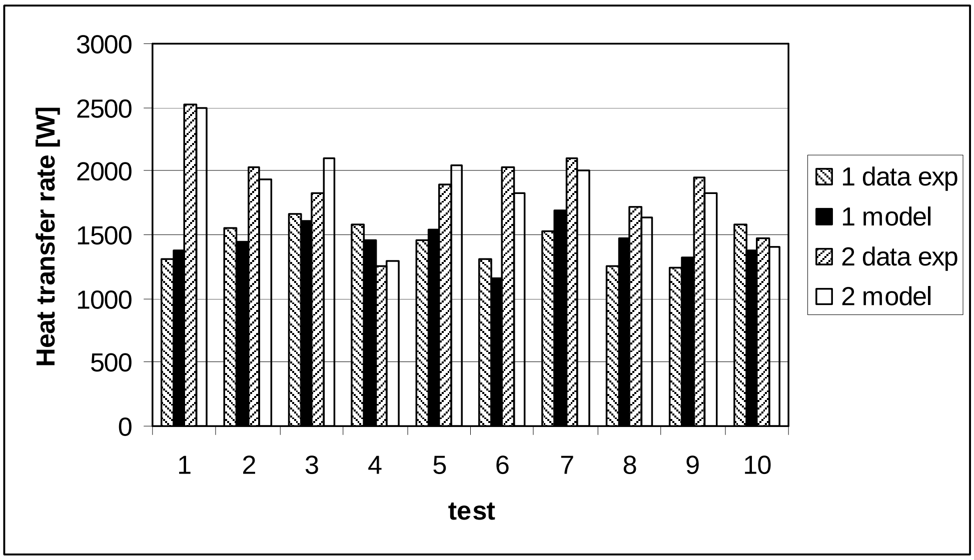

Figure 4. Experimental and modeling data of refrigeration capacity obtained of first and second experiments.

Comparisons of model results with the experimental data for the evaporator capacity between first and second experiment were made. The variation of the evaporator capacity between the experimental data and the models result is between ± 9.05 % for the first experiment and ±12.3% for second experiment with 90% of confiability.

5. CONCLUSION

An experimental and theoretical investigation into effects of the air side and refrigerant fouling factor for evaporator operation for eight years was successfully conducted. The new value of fouling factor for air side coefficient for evaporator in operation for the same time of operation was determinated. A comparison between experimental data and a numerical model was made for the evaporator capacity. A reduction of the nominal refrigeration capacity was identified. This model can become a useful tool for predicting the evaporator capacity.

6. ACKNOWLEDGEMENTS

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