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ANALYSIS OF HEAT EXCHANGER MODELS UNDER DRY, WET, AND FROST CONDITIONS FOR THE EVAPORATORS OF HEAT PUMPS

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1. ABSTRACT

This study presents an analysis of an evaporator model of an air-to-water heat pump under dry, wet, and frost conditions, with more focus on dry and wet conditions in terms of validation. ε-NTU method and Colburn-j factor were used for closure to avoid iterations. The roughly estimated fin and tube external temperature was used to opt for external phases. In the case of condensation, latent heat was used as additional source, and in the case of frost, frost layer was used as a feature causing thermal resistance and local loss to pressure. Despite some underprediction, the model provides valuable candidacy for use in a heat pump system model in dynamic conditions.

2. INTRODUCTION

For many types of commercial domestic heat pumps, there exist a split structure, or any combination that dictates the heat source is ambient air and that there is an external unit. If the heat transfer fluid is a refrigerant which changes the phase in this external unit as a result of the heat transfer, we can call the heat exchanger an evaporator. In many cases, this component is a fin-and-tube type heat exchanger [1].

The heat exchanger models operating under dry, wet, and frost conditions, play a crucial role in predicting the overall performance of a heat pump system because of the challenge to find the heat transfer via the finned surfaces and the phase change inside the tubes. The highly conductive solid materials do not pose a difficulty in modelling the conductive heat transfer. The actual modelling complexity is presented by the air and refrigerant in motion, driven by one or more fans and a compressor, respectively [2].

For the purpose of continuing the system-level simulations without any iterations, closure relations with Colburn-j factors, calculated via geometric parameters and operating conditions, were used to approximate firstly the air-side heat transfer coefficient, and later the fin efficiency. A flag was raised when the fin temperature was below the dew temperature to signify condensation conditions, another one was raised when it was lower than the freezing point. This study delves into the existing options of heat exchanger models for the purpose of finding the best fit for the experimentally found heat transfer. However, the paper shows only a summary of the methodology chosen for reaching the result. For further simplicity, not all theory will be given but instead it will be indicated sparingly via references and the fundamental ideas. Their presentation will be finalized briefly with validation against experimental findings. The study also addresses the challenges posed by frosting.

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3. METHODOLOGY

The methodology involved a comparative analysis of various Colburn-J factor models from the literature for wavy fins to firstly find the air-side convective heat transfer coefficient (h_o) [3]. For that inside the tubes, the same study can be cited. As Equation (1) shows, the coefficient can be found using the density (ρ), the interstitial maximum velocity through the fins and tubes (V_{max}), heat capacity and the Prandtl number of air (Pr).

$$h_o = \frac{j\rho V_{max} C_{p,air}}{\mathrm{Pr}^{2/3}} \quad (1)$$

The Colburn-j factor, using one of the approaches listed in the review paper mentioned above [3] was calculated as seen below [4]. Mostly, the inputs are from the manufacture of the evaporator. The only operational input is Reynolds number based on also the collar diameter (D_c), which is the width of the fin portion wrapped around the tube.

$$j = 0.171 e^{(0.377N)} \mathrm{Re}_{D_c}^{(-0.0142N-0.478)} \left(\frac{P_f}{D_c}\right)^{(0.00412N-0.0217)} \left(\frac{A_o}{A_u}\right)^{(-0.114N+0.44)} \quad (2)$$

Where *N* is the number of tube rows in the flow direction, P_f is the fin pitch, A_o is the total heat transfer area on the air side and A_u is the area around the tubes which are not considered enhanced via fins in terms of heat transfer. This method of calculating the convective heat transfer coefficient helps avoid iterations because the coefficient itself is required for finding the fin efficiency which is also required for finding the convective heat transfer resistance [2].

As for the condensation formation, a similar way was followed. This time the mass transfer coefficient (h_m) was found via Chilton-Colburn-J factor (j_m) and Schmidt number (Sc) [4].

$$h_m = \frac{j_m \rho V_{max}}{\mathrm{Sc}^{2/3}} \quad (3)$$

$$j_m = 0.315 e^{(-0.441N)} Re_{D_c}^{(0.058N-0.475)} \left(\frac{F_p}{D_c}\right)^{(-0.00471N+0.0216)} \left(\frac{A_o}{A_u}\right)^{(0.00223N+0.223)} \quad (4)$$

The heat transfer in the case of condensation is calculated via finding the mass formation rate of the condensate ($\dot{m}_{cond}$) using the humidity ratios with respect to the ambient conditions (W_o) and fin conditions (W_f).

$$\dot{m}_{cond} = h_m A_o (W_o - W_f) \quad (5)$$

The only frost formation impact of frost formation was considered to be the height of a porous frost layer (x_s), uniformly increasing on the fin surfaces, towards each other in each gap. This was considered a resistance both to air flow and heat transfer from the ambient to the refrigerant. The frost layer growth rate is a function dependent on subsequent ones, such as the frost density (ρ_f), frost mass transfer coefficient ($h_{m,f}$), which can also be approximated via correlations from the literature [5].

$$\dot{x}_s = \frac{h_{m,f}(A - A_s)}{\rho_f \left[1 + b_{\rho_f}(T_s - T_f)\right]} \quad (6)$$

Where *A* is the absolute humidity, s is the subscript for frost surface, where the temperature was assumed 0 °C and b_{ρ_f} is a known coefficient. At a given time, it was assumed that a frost height would constrict the flow so that the system curve would change and fan curve would be followed. In order to have a closure, it was assumed that every frost height would correspond to a flow rate and a pressure loss. Therefore a polynomial fit (

$\Delta P = a_2 v_{air}^2 + a_1 v_{air} + a_0$) using the fan manufacture, also the simplified Gnielinski pressure loss equation, without the changing air density on account of heat transfer [3] were used to generate a look-up table.

$$\left(a_2 - \frac{fA_o}{2\sigma^2 A_c}\right) v_{air}^2 + a_1 v_{air} + a_0 = 0 \tag{7}$$

The refrigerant was R32 with a flowrate from 0.01 to 0.03 kg/s. Air velocity ranged from 0.8 to 1.1 m/s. Total height and width were 0.65 and 0.87 m, tube internal and outer diameters were 6.5 and 7.5 mm, transversal and axial tube pitches were 24.5 and 21.5 mm, Fin thickness and pitch were 0.1 and 1.8 mm, respectively. The row number was 2.

4. RESULTS

The results revealed that the model generally underpredicted the experimental findings. This discrepancy was more pronounced under condensing conditions, highlighting the need for further refinement of the model. In Figure 1, the comparison between experimental and numerical findings for 16 cases has been shown. The rough assumption of assuming the fin temperature to be an average of those of air and refrigerant did not help find a perfect fit with the experimental values. Since there has been no spatially discretized solution or any empirical model adopted for the fin temperature, there is no statement related to the uncertainty posed by the current assumption.

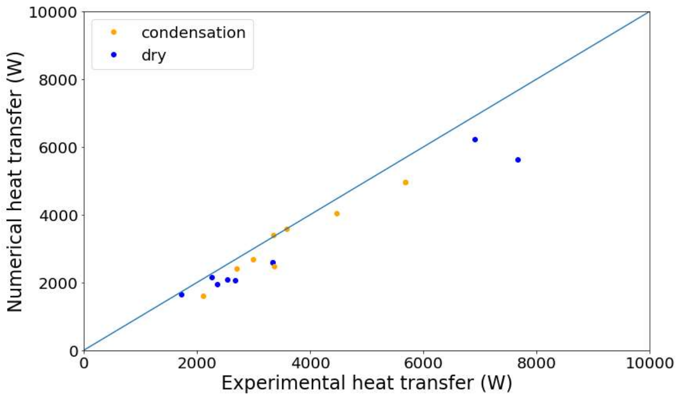

Fig. 1 Comparison between the numerically calculated heat transfer and that calculated experimentally.

5. CONCLUSIONS

Despite the discrepancies, the study provides a valuable foundation for the development of a dynamic evaporator model. The near future work will focus on addressing the underprediction issue and incorporating validated frost conditions into the model, thereby enhancing its applicability to real-world heat pump systems.

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