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First-principles modeling of frost accumulation on fan-supplied tube-fin evaporators

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A B S T R A C T

The present paper investigates the frost formation on air-supplied tube-fin evaporator coils under typical operating conditions of light commercial refrigerating appliances. To this end, a first-principles simulation model based on air-side mass, energy and momentum balances was put forward to predict the evaporator frosting over time. Experiments were also carried out to gather key data for the model validation exercise. The numerical results were compared with the experimental air-side pressure drop, air-flow rate, cooling capacity, and accumulated frost mass, with all predictions falling within the experimental uncertainty range. The model was then used to investigate the evaporator thermal-hydraulic performance under frosting conditions accounting for the non-linear effect induced by the combination of the frosted evaporator and the fan-supplied air-flow rate. The effects of progressive frost clogging and low conductivity frost layer on the overall thermal resistance were also assessed. It was found that the former is the main cause of cooling capacity reduction under frosting conditions.

1. Introduction

The frost deposition on tube-fin evaporator coils involves simultaneous heat and mass transfer phenomena that depend on many parameters including heat exchanger geometry, coil surface temperature, and air temperature, humidity, and velocity. In addition to the atmospheric research and the problem of in-flight aircraft icing, the frost growth prediction is also an important issue in refrigeration and air conditioning applications. As demonstrated in previous studies [1,2], frost accumulation on forced air evaporator coils depletes the thermo-hydraulic performance because of the combined increase of the thermal resistance and the air-side pressure drop. As a consequence, the entire refrigeration system is dramatically affected, demanding periodic defrosting actions to recover its original performance.

In an attempt to predict the frost accretion on evaporator coils, different simulation models were put forward in the past decades [3–10]. In general, those models assume the frost growth as a quasi-static phenomenon, apply the Lewis analogy for the mass transfer calculations, and depend on the thermo-physical properties of the frost layer, in particular the frost density, which in turn is affected by both the evaporator surface temperature and the dewpoint at the evaporator inlet. Recently, computational fluid dynamic techniques have also been employed to predict multidimensional frost growth phenomena [11,12], therefore requiring a large computational effort.

Nonetheless, most of the models available in the open literature do not account for the air-flow reduction that actually takes place in fan-supplied evaporator coils operating under frost conditions, which is one of the main causes of the cooling capacity reduction. Chen et al. [13] and Aljuwayhel [14] have both modelled the frost growth on tube-finned evaporators under variable air-flow conditions. The former was validated using experimental data collected under constant air-flow conditions, whereas the latter was focused on industrial refrigeration applications.

However, there is still a need for a validated model of frost accumulation on evaporator coils that accounts for the hydraulic coupling between the fan and the coil. This is, therefore, the main aim of the present study.

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Nomenclature

Roman

A_{face}	evaporator face area [m^2]
A_{min}	evaporator minimum free flow area [m^2]
A_s	evaporator surface area [m^2]
C_f	Fanning friction factor [−]
c_p	specific heat at constant pressure [J kg^{-1} K^{-1}]
D_w	diffusivity of water vapor in air [m^2 s^{-1}]
D_h	hydraulic diameter [m]
D_t	tube diameter [m]
D_f	frosted tube diameter [m]
e	thickness [m]
h_o	convective heat transfer coefficient [W m^{-2}K^{-1}]
i_{sv}	latent heat of desublimation [kJ kg^{-1}]
k_f	thermal conductivity of frost [W m^{-1} K^{-1}]
Le	Lewis number [−]
m	water mass flux [kg$_v$ m^{-2} s^{-1}]
N_S	dimensionless entropy generation rate [−]
Q	heat transfer rate [W]
T	air temperature [K]
T_s	coil surface temperature [K]
T_f	frost surface temperature [K]
V	air-flow rate [m^3 s^{-1}]
x_f	frost thickness [m]

Greek

Δ	variation [−]
α	thermal diffusivity [m^2 s^{-1}]
Φ	relative humidity [%]
ϵ	frost porosity [−]
η_s	the overall surface effectiveness [−]
ρ	density [kg m^{-3}]
τ	frost tortuosity [−]
ω	humidity ratio [kg$_v$ kg$^{-1}{}_a$]

Subscripts

d	densification
f	frost surface
g	growth
i	i-th control surface
lat	latent
s	coil surface
sat	saturation
sen	sensible water vapor
1	inlet
2	outlet

2. Mathematical model

2.1. Heat exchanger model

The mathematical model was developed considering that the evaporation temperature and coil surface temperature are uniform over the entire evaporator. Hence, the air-flow across the heat exchanger was modeled following a one-dimensional approach, according to which the evaporator coil was divided into non-overlapping control volumes, as depicted in Fig. 1. In addition, bearing in mind that the frost formation is a quite complex phenomenon, the following assumptions were taken into account: (i) the heat and mass transfer process were regarded as quasi-static; (ii) the frost layer thickness was assumed to be uniform on both the tubes and fins; (iii) the Lewis analogy was invoked; and (iv) the air properties were considered to be uniform at the control volume inlet and exit sections.

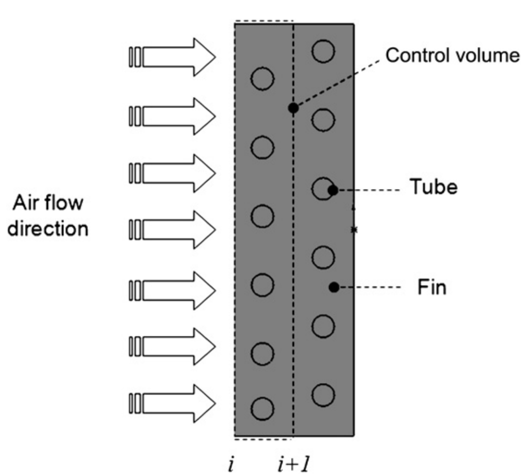

Fig. 1. Schematic representation of the physical model.

Therefore, energy and mass balances over the control volume were invoked to provide one-dimension air temperature (T) and humidity ratio (ω) distributions along the coil, as follows:

$$T_{i+1} = T_{f,cv} - \left(T_{f,cv} - T_i\right) \exp\left(-\left.\frac{\eta_f h_o A_f}{\rho_a c_{p,a} V}\right|_{cv}\right) \tag{1}$$

$$\omega_{i+1} = \omega_{\text{sat}f,cv} - \left(\omega_{\text{sat}f,cv} - \omega_i\right) \exp\left(-\left.\frac{\eta_f h_o A_f}{\rho_a c_{p,a} V Le^{2/3}}\right|_{cv}\right) \tag{2}$$

where the index cv refers to the control volume (see Fig. 1), A_f is the frosted heat exchanger surface area, h_o is the air-side convective heat transfer coefficient, η_f is the frosted surface effectiveness, $c_{p,a}$ and ρ_a are the specific heat and the density of moist air, respectively, V is the air-flow rate, T_f and $\omega_{\text{sat}f}$ are the temperature and humidity ratio (assumed to be saturated) at the frost surface, respectively, and Le is the Lewis number.

Mass and energy balances on and within the frost layer were applied to each control volume to provide the frost surface temperature (T_f) as a function of the coil surface temperature (T_s), yielding [15]

$$T_{f,cv} = T_s + \frac{Q_{\text{sen},cv} + Q_{\text{lat},cv}}{A_{s,cv}} \frac{x_{f,cv}}{k_f} - \frac{\rho_a \omega_{\text{sat},s} i_{sv}}{k_f} \frac{D_w}{\tau} \times \left[\cos h\left(\frac{\omega_{\text{sat}f,cv}}{\omega_{\text{sat},s}}\right) - 1\right] \tag{3}$$

where k_f is the thermal conductivity of frost obtained from Lee et al. [16], i_{sv} is the latent heat of sublimation, τ is the tortuosity of the frost layer obtained from the correlation of Zehnder [17], ϵ is the porosity of the frost layer calculated from an empirical frost density, D is the diffusivity of water vapor in air, and x_f is the frost layer thickness.

In addition, the sensible and latent heat transfer rates, and the air-side pressure drop at each control volume are calculated respectively from