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A study of the effective thermal conductivity of frost formed on parallel plate channels

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ABSTRACT

This paper presents an experimental study of the thermal conductivity of frost accumulated on parallel plate channels considering different operating conditions. The experimental work was carried out by means of a purpose-built closed-loop wind-tunnel facility, which provides a strict control of the psychrometric conditions at the entrance of the test section as well as the plate surface temperature. A dataset comprising of 57 experimental data points was gathered in order to come out with a semi-empirical correlation for the thermal conductivity of frost as a function of the frost porosity and the plate surface temperature. The correlation, which is based on the weighted geometric mean of the thermal resistances of moist air and ice crystals, is able to predict ~90% of the experimental data points within the ±15% thresholds.

1. Introduction

Frost is a porous medium comprised of ice crystals and moist air, generally formed through desublimation of the water vapor present in the air stream. Frost often builds-up on evaporators of refrigeration systems, resulting in an increased energy input to accomplish the same refrigerant effect. Many efforts have been done aiming at simulating the evaporator frosting in order to come out not only with robust heat exchanger designs [1,2] but also with smarter defrost strategies [3]. These models have been based on more fundamental studies of the modeling of the frost growing process on flat surfaces [4,5] and parallel plate channels [6].

Most models, however, rely on empirical correlations to compute the thermophysical properties of the frosted media, particularly the density (or porosity) and the effective thermal conductivity, which limits their applications to narrow ranges of frost morphologies. Studies of the frost density have been successfully carried out to come out with a semi-empirical correlation applicable for a wide frost morphology span. For instance, Hermes et al. [7] and Nascimento et al. [8] proposed semi-empirical correlations for the density of frost formed on flat plates and parallel plate channels, respectively, as follows:

$$\frac{\rho_f}{\rho_i} = C\Lambda^{-n}\sqrt{t} \tag{1}$$

where $n = 3/2$ and $C = 0.0024$ for flat surfaces, and $n = 3/4$ and $C = 0.0022$ for parallel plate channels, where ρ_i and ρ_f are the densities of ice and frost in (kg m^{-3}), respectively, and t is time in (s). In addition, Λ is the modified Jakob number introduced by Hermes et al. [7] as follows:

$$\Lambda = \frac{c_p(T_{\text{dew}} - T_w)}{i_{\text{sv}}(\omega_a - \omega_{\text{sat}}(T_w))} \tag{2}$$

where c_p is the specific heat of moist air, T_{dew} is the dew-point temperature, i_{sv} is the latent heat of sublimation, ω_a is the humidity ratio of the air stream, and $\omega_{\text{sat}}(T_w)$ is the humidity ratio of the saturated air at the wall temperature. Recently, Sommers et al. [9] extended Eq. (1) to applications spanning hydrophilic to hydrophobic surfaces based on their own dataset collected for a wide range of contact angles, and natural convection conditions. It is worth of note that Eq. (1) was devised based on the fact that the frost thickness, δ_f, follows the $\sqrt{t}$ scale, so does the frost density, ρ_f, in such a way that the frost mass has a fairly linear dependence with time, as $M \sim \rho_f\delta_f \sim \sqrt{t}\sqrt{t} \sim t$.

A judicious inspection of the open literature carried out in a previous work [10] revealed that there was an opportunity for a physically-based model for the thermal conductivity of frost. Negrelli and Hermes [10] thus came out with a semi-empirical correlation for the thermal conductivity of frost as a function of the frost porosity and the wall temperature. Their correlation was based on 188 experimental data points for flat surfaces collected from Pitman and Zuckerman [11], Brian et al. [12], Auracher [13],

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Nomenclature

Roman

A_s	surface area, m^2
a, b	coefficients of Eqs. (16) and (17)
C	coefficient of Eq. (1)
c_p	specific heat, kJ kg^{-1} K^{-1}
h	heat transfer coefficient, W m^{-2} K^{-1}
h_m	mass transfer coefficient, kg m^{-2} K^{-1}
i_{sv}	latent heat of sublimation, kJ kg^{-1}
k	thermal conductivity, W m^{-1} K^{-1}
Le	Lewis number
M	mass, kg
m	mass flux, kg m^{-2} s^{-1}
q	heat flux, W m^{-2}
R_T	thermal resistance, m^2 K W^{-1}
t	time, s
T	temperature, K
u	air velocity, m s^{-1}

Greek

ρ	density, kg m^{-3}
ε	frost porosity
δ	thickness, m
Λ	modified Jakob number
ϕ	relative humidity
ω	humidity ratio

Subscripts

a	moist air
dew	dew-point
f	frost
g	geometric mean
i	ice
lat	latent
p	parallel association
s	serial association
sat	saturation
sen	sensible
w	wall surface

and Na and Webb [14], showing a good agreement with the experimental data and spanning a wide range of frost morphologies.

Nevertheless, only one record was found in the open literature with regard to the thermal conductivity of frost formed on parallel plate channels [15]. Their study, however, correlated the thermal conductivity as an empirical function of the frost density only, in the form of a quadratic polynomial fit, which showed errors up to 50% when compared against their own experimental data. Therefore, the review of the state-of-the-art points out that the open literature lacks of a physically-consistent model for the thermal conductivity of frost accumulated on parallel plate channels which is also applicable to a wide range of frost morphologies. The present paper is aimed at accomplishing this task.

2. Experimental work

2.1. Test rig

The test rig is a closed-loop wind-tunnel facility, as depicted in Fig. 1. The apparatus allows a strict control of the psychrometric conditions of the air, the air velocity and the plate temperature. The wind-tunnel has a 200 × 200 mm cross section and its walls are made of 50-mm thick EPS plates sandwiched between 20-mm thick plastic liners so as to provide structural resistance. The air-loop is comprised of two straight sections and two return bends. One branch contains the 1/2″ and 1″ nozzles for air flow measurements, an evaporator coil for cooling and dehumidifying the air stream, and two PID-driven electric heaters, a finned 350-W one for regulating the air temperature and a coiled 180-W one immersed in a humidifying tray for controlling the air humidity. A DC fan is also used to set the air velocity at the test section. The air flow rate was measured using a differential pressure transducer ranging from 0 to 125 Pa with an uncertainty of 0.25% (full-scale).

On the other branch, there is an EPS-insulated 1.0-m long straight channel, which ensures the proper thermal-fluid-dynamic development, and the test section itself, which is comprised of two PID controlled thermoelectric devices, as illustrated in Fig. 2. The test section is comprised of two T-shaped aluminum blocks, whose bigger ends (L = 120 mm × W = 60 mm) are connected to the thermoelectric cells located at the outer side of the wind-tunnel walls to promote the heat dissipation from the hot ends. The smaller ends (L = 40 mm × W = 60 mm) form the parallel plate channel where frost builds-up.

The test section itself, i.e. the channel, is formed by two parallel 3-mm thick copper plates (contact angle of 67° ± 3°) placed at the top of the aluminum blocks. Two thermocouples and a heat flux sensor (±5 W/m^2 uncertainty) were embedded into each plate, as shown in Fig. 2, for the sake of temperature and heat flux measurements, respectively. Four T-type thermocouples (±0.2 K uncertainty) were located at the test section entrance (2) and exit (2) ports, being the upstream ones responsible for controlling the 350-W heater. A square 22-mm sided triple-layer glass window is located in front of the test section to allow visualization. A stereoscopic device, with a 3 megapixel 10× ocular lens and 0.5× photographic lens camera, was used to take pictures of the frost layer during the test. The illumination was provided by optical fibers. The frost thickness was measured from the images with a ±50 μm uncertainty.

A homogenizer was placed at the channel inlet to promote flow mixture. A capacitive relative humidity transducer (±2% uncertainty) was installed at the channel inlet to control the inflow air humidity by means of the 180-W heater. Additional thermocouples were installed at the nozzle and at the surrounding air, which was indicated by • outside the rig in Fig. 1, and kept at 20 °C ± 2 °C by an on–off controlled air conditioner.

2.2. Test procedure and data processing

Before starting a particular test, all variables must achieve steady-state conditions. To avoid frost formation during the transient regime, a by-pass system was adopted, which consists of a manual damper that diverts the air flow from the test section, so that the cold surfaces are not exposed to the moist air stream. During the test, pictures of the test section are taken at every minute. The images were then processed to come out with the time-evolution of the frost thickness, δ_f. The frost mass, M, was measured afterwards by a high-precision scale with ±0.01 g uncertainty. The frost density was then calculated from:

$$\rho_f = \frac{M}{A_s(\delta_{f,\mathrm{top}} + \delta_{f,\mathrm{bottom}})} \tag{3}$$