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A numerical simulation model for plate-type, roll-bond evaporators

Christian J.L. Hermes[a,*], *Cláudio Melo*[a], *Cezar O.R. Negrão*[b,1]

[a]*POLO Research Laboratories for Emerging Technologies in Cooling and Thermophysics, Department of Mechanical Engineering, Federal University of Santa Catarina, 88040-970 Florianópolis, SC, Brazil*
[b]*Thermal Science Laboratory, Postgraduate Program in Mechanical and Materials Engineering, Federal University of Technology – Paraná, Av. Sete de Setembro 3165, 80230-901 Curitiba, PR, Brazil*

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This study presents a first-principles mathematical model developed to investigate the thermal behavior of a plate-type, roll-bond evaporator. The refrigerated cabinet was also taken into account in order to supply the proper boundary conditions to the evaporator model. The mathematical model was based on the mass, momentum and energy conservation principles applied to each of the following domains: (i) refrigerant flow through the evaporator channels; (ii) heat diffusion in the evaporator plate; and (iii) heat transmission to the refrigerated cabinet. Empirical correlations were also required to estimate the shear stresses, and the internal and external heat transfer rates. The governing partial differential equations were discretized through the finite-volume approach and the resulting set of algebraic equations was solved by successive iterations. Validation of the model against experimental steady-state data showed a reasonable level of agreement: the cabinet air temperature and the evaporator cooling capacity were predicted within error bands of $\pm 1.5\,^{\circ}C$ and $\pm 6\%$, respectively.

Modèle de simulation numérique pour les évaporateurs á plaque et serpentin

Mots clés : Réfrigérateur domestique ; Échangeur de chaleur ; Évaporateur ; Modélisation ; Expérimentation

1. Introduction

Refrigerators and freezers are responsible for approximately 8.5% of energy consumption in Brazil (PROCEL, 1998). As the major part of this energy is wasted by the system components (compressor, condenser, evaporator, and capillary tube) due to irreversible processes, studies to understand such thermodynamic losses may lead to the development of higher efficiency products. Jakobsen (1995) quantified the thermodynamic losses in a 325-l refrigerator and investigated various means of energy optimization. He found that such losses occurred mainly in the hermetic compressor and in the

* *Corresponding author*. Tel.: +55 48 3234 5691; fax: +55 48 3234 5166.
E-mail addresses: hermes@polo.ufsc.br (C.J.L. Hermes), melo@polo.ufsc.br (C. Melo), negrao@utfpr.edu.br (C.O.R. Negrão).
1 Tel.: +55 41 3310 4658; fax: +55 41 3310 4432.

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Nomenclature

Symbols

A	channel cross-sectional area [m^2]
A_k	wall surface area [m^2]
Bi	Biot number ($=\hbar l/k_w$) [dimensionless]
C_a	thermal capacity of dry air [J K^{-1}]
C'''	thermal capacity per unit of volume [J m^{-3} K^{-1}]
d	channel diameter [m]
F	view factor between surfaces [dimensionless]
f	Darcy's friction factor [dimensionless]
G	refrigerant mass flux ($=\rho\vartheta$) [kg s^{-1} m^{-2}]
h	specific enthalpy of the refrigerant [J kg^{-1}]
h_o	overall specific enthalpy of the refrigerant ($=h+\vartheta^2/2$) [J kg^{-1}]
$\hbar_e$	convective heat transfer coefficient between the evaporator surfaces and the cabinet air [W m^{-2} K^{-1}]
$\hbar_i$	convective heat transfer coefficient between the refrigerant and the channel walls [W m^{-2} K^{-1}]
J	radiosity [W m^{-2}]
K_g	overall conductance of the gasket [W m^{-1} K^{-1}]
l	thickness [m]
m	refrigerant mass flow rate [kg h^{-1}]
p	pressure [kPa]
P_d	sealed perimeter of the door [m]
Pe	Péclèt number ($=\vartheta d/\alpha$) [dimensionless]
P_i	inner perimeter of the channel [m]
q_e	convective heat flux between the evaporator surfaces and the cabinet air [$=\hbar_e(T_a-T_w)$] [W m^{-2}]
q_i	convective heat flux between the refrigerant and the channel walls [$=\hbar_i(T_w-T)$] [W m^{-2}]
q_r	radiative heat flux on the surfaces of the evaporator plate [$=(q_{r,7}+q_{r,8})/2$] [W m^{-2}]
t	time [s]
T	refrigerant temperature [K]
T_a	air temperature within the refrigerated compartment [K]
T_e	environment air temperature [K]
T_w	evaporator surface temperature [K]
U	convection heat transfer coefficient between the air and the k-th wall [W K^{-1}]
w	thickness of the evaporator plate [m]
x	dimensionless enthalpy [kJ kg^{-1}]

Greek letters

α_w	thermal diffusivity of the aluminum [m^2 s^{-1}]
α_k	thermal diffusivity of the insulation [m^2 s^{-1}]
ε	surface emissivity [dimensionless]
σ	Stefan–Boltzman constant [W m^{-2} K^{-4}]
ρ	specific mass of the refrigerant [kg m^{-3}]
θ	generic temperature [K]
ϑ	average flow velocity [m s^{-1}]
ϕ	generic variable per unit of mass
τ_w	shear stress on the channel wall ($=fG\vartheta/8$) [Pa]
τ	time constant [s]

Subscripts

k	each of the nine cabinet internal surfaces (see Fig. 4)
d	advective flux through the downstream control surface
u	advective flux through the upstream control surface
w	evaporator wall

Superscripts

*	value from the previous iteration
0	value from the previous time-step

evaporator, and also that the latter showed the best system/component performance ratio, i.e., the highest system performance improvement with the lowest component-level investment.

In Brazil, the most widely used evaporator for household refrigerators is known as the plate-type, roll-bond evaporator. Basically, it consists of a plate formed by two powder-coated aluminum sheets, with channels in which the refrigerant evaporation takes place, while a buoyancy-driven air circulation occurs at the outer side. The combination of low cost and reasonable performance – compared to plate-and-tube heat exchangers – has led to a steady increase of its application.

In order to better understand this component performance, a research program has been conducted at the Federal University of Santa Catarina. Firstly, an experimental study on the heat and fluid flow in a roll-bond evaporator was carried out by Silva et al. (1999), who developed an in situ calorimeter test facility. Melo et al. (1998), based on the expertise gained by Silva et al. (1999), investigated alternative channel geometries, such as that shown in Fig. 1. Although this component was investigated experimentally, there is still a need for mathematical models for predicting its thermal behavior. Numerical simulation models for vapor compression systems have been developed since the early 1980s (Chi and Didion, 1982; Yasuda et al., 1983; MacArthur, 1984; Murphy and Goldschmidt, 1985, 1986; Sami et al., 1987; Jansen et al., 1988; Melo et al., 1988). At the beginning both heat exchangers (condenser and evaporator) were treated as even lumps. More recently (Gruhle and Isermann, 1985; Nyers and Stoyan, 1994; Jia et al., 1995; Judge and Radermacher, 1997; García-Valladares et al., 1998; Stevanvic and Jovanovic, 2000; Aprea and Reno, 2002) new modeling approaches were introduced that allowed a distributed simulation of the heat exchangers, but none of them are applicable to roll-bond evaporators.

The aim of the present study is to report on a numerical modeling approach for roll-bond evaporators similar to that shown in Fig. 1, which can be used for both steady-state and transient simulations. In order to supply the proper boundary conditions to the evaporator model, the refrigerated cabinet illustrated in Fig. 2 was also taken into account.

2. Mathematical modeling

The mathematical model proposed in this study was divided into three sub-models: (i) the evaporator channels, (ii) the