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Air-side heat transfer and pressure drop in spiral wire-on-tube condensers

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ABSTRACT

The air-side thermal-hydraulic performance of spiral wire-on-tube condensers is investigated experimentally in this paper. Sixteen prototypes have been manufactured and tested in an open-loop wind tunnel calorimeter. The influence of the following geometric parameters has been evaluated: the number of tube passes, the radial and longitudinal tube spacings and the wire spacing. Measurements of the air-side thermal conductance and pressure drop were carried out for air flow rates ranging from 70 to 220 $m^3\ h^{-1}$. The data were correlated using empirical relationships for the Colburn *j*-factor and the Darcy friction factor. The agreement with the experimental data presented RMS deviations of 0.9% for the air-side heat transfer and 1.3% for the frictional pressure drop dimensionless parameters. A quantitative analysis based on the core volume goodness factors for heat transfer and pressure drop is presented to provide the most viable configuration from the point of view of application in a refrigerator.

Transfert de chaleur côté air et chute de pression dans les condenseurs à serpentin

Mots clés : Condenseur ; Fil ; Tube ; Serpentin ; Convection forcée

1. Introduction

The development of more compact heat exchangers for domestic refrigeration applications has been the focus of several studies thanks to its impact in reducing material costs and energy consumption. It also allows for a more effective usage of the space occupied by the cooling system components, which increases the net storage volume of the refrigerator without changing the external dimensions of the product. Due to its low manufacturing cost, wire-on-tube heat exchangers are still the most advantageous condenser type for household refrigeration applications where natural convection is the dominant heat transfer mode (Hermes and Melo, 2009). However, for higher cooling capacity systems,

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Nomenclature

Roman

a_j	angular coefficient of the *j* correlation [–]
a_f	angular coefficient of the *f* correlation [–]
A	surface area [m^2]
A,B,C,D	normalized parameters in Eq. (24) [–]
b_j	linear coefficient of the *j* correlation [–]
b_f	linear coefficient of the *f* correlation [–]
C	thermal capacity rate [$W\ K^{-1}$]
c_p	specific heat capacity at constant pressure [$J\ kg^{-1}\ K^{-1}$]
C^*	ratio between the minimum and maximum thermal capacity rate [–]
D_t	tube diameter [m]
D_w	wire diameter [m]
e_r	tube radial spacing [m]
er	relative error [–]
f	Darcy friction factor [–]
G	mass flux [$kg\ m^{-2}\ s^{-1}$]
$\bar{h}$	average heat transfer coefficient [$W\ m^{-2}\ K^{-1}$]
j	Colburn *j*-factor [–]
k	thermal conductivity [$W\ m^{-1}\ K^{-1}$]
$\dot{m}$	mass flow rate [$kg\ s^{-1}$]
N_p	number of tube passes [–]
N_w	number of wires [–]
NTU	number of transfer units [–]
P	pressure [Pa]
Pr	Prandtl number [–]
$\dot{Q}$	heat transfer rate [W]
R^2	coefficient of determination [–]
R_s	thermal resistance of the solid surface [$K\ W^{-1}$]
Re	Reynolds number [–]
U	experimental uncertainty of a given variable [–]
UA	overall thermal conductance [$W\ K^{-1}$]
V	velocity [$m\ s^{-1}$]
$\dot{V}$	volume flow rate [$m^3\ s^{-1}$]
t	Student's *t* coefficient [–]
T	temperature [°C]

Greek

α	generalized linear or angular coefficient of the correlations [–]
β	surface area per unit volume [$m^2\ m^{-3}$]
β_k	empirical constants in Eq. (24) [–]
ε	heat exchanger effectiveness of the cross-flow arrangement [–]
$\varepsilon_{cc,2}$	heat exchanger effectiveness of the 2-pass cross-counter flow arrangement [–]
$\varepsilon_{cc,4}$	heat exchanger effectiveness of the 4-pass cross-counter flow arrangement [–]
η_o	overall surface efficiency [–]
μ	dynamic viscosity [$kg\ m^{-1}\ s^{-1}$]
ρ	density [$kg\ m^{-3}$]
σ_f	ratio between the minimum free-flow and face areas [–]
φ	normalized coefficient in Eq. (24) [–]
Ψ_1	thermal conductance per unit volume [$W\ m^{-3}\ K^{-1}$]
Ψ_2	pumping power per unit volume [$W\ m^{-3}$]

Subscripts

2	2-pass heat exchanger
4	4-pass heat exchanger
a	air
cf	internal filling
f	face
fin	fin
i	internal
in	inlet
max	maximum
min	minimum
o	external
out	outlet
w	water

Abbreviations

AAD	absolute average deviation
RMS	root mean square

the benefits of more compact forced convection wire-on-tube condensers seem to overcome the costs and energy consumption involved in pumping the external ambient air through the condenser.

One of the first studies of forced convection in wire-on-tube condensers to be reported in the open literature was that of Hoke et al. (1997). They evaluated experimentally the thermal performance of eight single layer coils made from carbon steel in a wind tunnel, and quantified the effects of the tube and wire diameters, tube and wire spacings, angle of attack, flow velocity and flow orientation (normal to the wires and normal to the tubes). The internal heat transfer fluid was hot water. Correlations were proposed for the wire and tube heat transfer coefficients. The proposed heat transfer calculation method was capable of predicting the air-side experimental data to within ±16.7% error. It was concluded that the angle of attack is an important parameter in the overall thermal performance, with a nearly two-fold increase in the heat transfer coefficient when the angle of attack increased from 0 to 20°.

A relatively small number of wire-on-tube heat exchanger configurations other than single layer coils have been proposed in the open literature for forced convection applications (Dasher, 1996; Petroski and Clausing, 1999; Lee et al., 2000; Ohgaki, 2002; Jenkins, 2003). Petroski and Clausing (1999) evaluated the air-side thermal performance of eight saw-tooth wire-on-tube condensers (also made from carbon steel) under forced convection. Their facility and experimental procedure were similar to those employed by Hoke et al. (1997). They investigated the influence of the following parameters: (i) the flow orientation with respect to the wires (in-line and cross-flow), (ii) wire spacing, (iii) amplitude of the saw-tooth and, (iv) the distance between the test section wall and the heat exchanger. Tests were carried out in an open-loop wind tunnel for air velocities in the range of 0.2–2 m s^{-1}. The air-side thermal conductance was correlated