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Liquid side heat transfer and pressure drop in finned-tube cooling-coils operated with secondary refrigerants

Caroline Haglund Stignor[a,*], Bengt Sundén[b], Per Fahlén[c]

[a]SP Technical Research Institute of Sweden, Energy Technology, P.O. Box 857, S-501 15 Borås, Sweden
[b]Lund Institute of Technology, Division of Heat Transfer, P.O. Box 118, S-221 00 Lund, Sweden
[c]Chalmers University of Technology, Building Services Engineering, S-412 96 Gothenburg, Sweden

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Abstract

In this study full-scale experiments with two different conventional cooling-coils aimed for display cabinets were performed. Heat transfer and pressure drop on the liquid side for three different single phase secondary refrigerants were studied and compared to predictions by existing correlations. Predominantly, the laminar flow regime was studied. The results show that when predicting the heat transfer performance on the liquid side of a cooling-coil the Gnielinski correlation for thermally developing flow and uniform wall temperature boundary conditions (T) leads to good agreement for $0.0014 < x^* < 0.017$ if $50 < Re < 1700$, assuming a new entrance length is formed after each U-bend. In addition, these entrance lengths must also be accounted for, when predicting the pressure drop on the liquid side of the cooling-coil. The uncertainty of measurement can be a problem in this type of investigations and this has been taken into consideration when analysing the results.

Keywords: Refrigeration; Display cabinet; Heat exchanger; Finned tube; Experiment; Heat transfer; Pressure drop; Secondary refrigerant; Propylene glycol; Laminar flow

Transfert de chaleur et chute de pression côté liquide à l'intérieur des serpentins de refroidissement fonctionnant à l'aide de frigoporteurs

Mots clés : Réfrigération ; Meuble de vente ; Échangeur de chaleur ; Tube aileté ; Expérimentation ; Transfert de chaleur ; Chute de pression ; Frigoporteur ; Propylène glycol ; Écoulement laminaire

1. Introduction

Finned-tube cooling-coils are used in many applications. One of particular interest is for cooling of the air in display cabinets in the commercial sector, since refrigeration of merchandise in supermarkets is responsible for a significant

* Corresponding author. Tel.: +46 10 5165545; fax: +46 33 131979.

E-mail address: caroline.haglund.stignor@sp.se (C. Haglund Stignor).

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Nomenclature

A	area (m^2)
c_p	specific heat ($J\,kg^{-1}\,K^{-1}$)
D	outer diameter of tubes (m)
d	inner diameter of tubes (m)
d_h	hydraulic diameter $d_h = 4A_f/P$, where P = perimeter (m)
F	correction factor (−)
f	Darcy friction factor, $f = \Delta p d(2/(\rho L_{tube} u^2))$ (−)
h	specific enthalpy ($J\,kg^{-1}$)
k	number of U-bends along a liquid circuit (−)
L	characteristic length, e.g. in the definition of Reynolds number (m)
L_{tube}	heated tube length (m)
$L_{tube,tot}$	total length of a liquid loop or circuit (m)
$\dot{m}$	mass flux ($kg\,s^{-1}$)
N	number of parallel loops or circuits (−)
n	number of tube rows (−)
Nu	Nusselt number, $Nu = (\alpha d_h/\lambda)$ (−)
P_d	wave height of fins (m)
p	pitch (m)
Δp	pressure difference (Pa, kPa or bar)
Pr	Prandtl number, $Pr = (c_p \mu/\lambda)$ (−)
$\dot{Q}$	cooling capacity (W)
Re	Reynolds number, $Re = (\rho u L/\mu)$, in this paper based on empty tube diameter (−)
t	temperature (°C)
U	expanded (total) uncertainty of measurements
U	overall heat transfer coefficient, Eqs. (5) and (6) ($W\,m^{-2}\,K^{-1}$)
u	velocity ($m\,s^{-1}$)
$\dot{V}$	Volume flow ($m^3\,s^{-1}$)
X_f	projected wave length (mm)
Y	factor adjusting for viscosity variations in the liquid flow (−)

Greek letters

α	heat transfer coefficient ($W\,m^{-2}\,K^{-1}$)
δ	thickness (mm)
λ	thermal conductivity ($W\,m^{-1}\,K^{-1}$)
η	fin efficiency (−)
μ	dynamic viscosity (Pa s)
ρ	density ($kg\,m^{-3}$)
$\zeta_{U\text{-bend}}$	resistance coefficient for a U-bend (−)

Subscripts

a	air
b	liquid secondary refrigerant (brine)
B	bulk, liquid secondary refrigerant at bulk temperature
fin	fin
H	uniform heat flux boundary condition
i	based on maximum inside (envelope) diameter
in	in to cooling-coil, at inlet conditions
l	longitudinal (direction of air flow)
lm	logarithmic mean
m	mean
out	out from cooling-coil
t	transverse (direction of air flow)
T	uniform temperature boundary condition
w	wall, liquid secondary refrigerant at wall temperature
w	by weight (if preceded by %)

amount of the energy use in the this sector [1]. Lately, indirect cooling by means of a liquid secondary refrigerant has become frequently used, especially in the Nordic countries, due to regulations for the use of synthetic refrigerants. The tube-coil heat exchanger was originally designed for evaporation of a refrigerant and then the heat transfer coefficient on the tube side is very high. However, using a liquid secondary refrigerant as heat transfer medium, heat transfer on the tube side will be much lower, especially for laminar flows, which is often the case for many secondary refrigerants, due to relatively high viscosity at low temperatures.

The heat transfer and pressure drop performance on the air side of finned-tube-coils have been thoroughly analysed by many researchers, see, e.g. the ref. [2]. Regarding the liquid side, Mao et al. [3] and Hrnjak [4] investigated heat transfer in a display cabinet cooling-coil with secondary refrigerants. They reported unexpectedly high heat transfer coefficients on the secondary refrigerant side at low Reynolds numbers. This is explained by the occurrence of thermally developing regions after the U-bends. Hong and Hrnjak [5] investigated the U-bend effect further and correlations for different fluids were developed (i.e. curve fits from experiments with a coaxial heat exchanger). The researchers concluded that the thermal development and heat transfer following a U-bend are very similar to that after the inlet of the tube. Apart from these studies, there are little experimental data regarding heat transfer and pressure drop characteristics for cooling-coils cooled by secondary refrigerants available in the literature.

Several papers presented during the last years deal with selection of secondary refrigerant for secondary loop refrigeration systems and they point out that for some liquids the flow regime could be laminar in the cooling-coils, e.g. the refs. [6,7]. There are several different heat transfer correlations for laminar flow forced convection in ducts presented in the literature [5,8–12]. According to Melinder [6] the correlation presented by Sieder and Tate [10] can be used for laminar flow of liquid secondary refrigerants in a cooling-coil and the tube length used in the correlation might represent one straight tube length,