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Numerical analysis of an air condenser working with the refrigerant fluid R407C

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Abstract

As CFC (clorofluorocarbon) and HCFC (hydrochlorofluorocarbon) refrigerants which have been used as refrigerants in a vapour compression refrigeration system were know to provide a principal cause to ozone depletion and global warming, production and use of these refrigerants have been restricted. Therefore, new alternative refrigerants should be searched for, which fit to the requirements in an air conditioner or a heat pump, and refrigerant mixtures which are composed of HFC (hydrofluorocarbon) refrigerants having zero ODP (ozone depletion potential) are now being suggested as drop-in or mid-term replacement. However also these refrigerants, as the CFC and HCFC refrigerants, present a greenhouse effect.

The zeotropic mixture designated as R407C (R32/R125/R134a 23/25/52% in mass) represents a substitute of the HCFC22 for high evaporation temperature applications as the air-conditioning.

Aim of the paper is a numerical–experimental analysis for an air condenser working with the non azeotropic mixture R407C in steady-state conditions. A homogeneous model for the condensing refrigerant is considered to forecast the performances of the condenser; this model is capable of predicting the distributions of the refrigerant temperature, the velocity, the void fraction, the tube wall temperature and the air temperature along the test condenser. Obviously in the refrigerant de-superheating phase the numerical analysis becomes very simple. A comparison with the measurements on an air condenser mounted in an air channel linked to a vapour compression plant is discussed. The results show that the simplified model provides a reasonable estimation of the steady-state response and that this model is useful to design purposes.

Keywords: Vapour compression plant; Condensation; Homogeneous model; Zeotropic mixture

1. Introduction

In this paper a simulation model for the refrigerant and the air flows, related to an air condenser is presented, together with its experimental validation. Usually, two models for the refrigerant are used: either the condensing refrigerant is considered as a homogeneous mixture of gas and liquid, or these two phases are separated [1]. With reference to the first model the condensation of vapour irrespective of the type of flow regime in the condenser tube can be treated as a homogeneous model under diabatic conditions with heat removal. Using this model the velocities of the vapour and of the liquid phase are identical and then no shear stress between the phases is allowed.

For the non-homogeneous model, the hypothesis of a constant condensation pressure leads to a considerable simplification in the model and then in the numerical solution as well. If the purpose is the study of the mass and of the energy balance only, this assumption gives small errors. If the study has design purposes it is necessary to know the changes in the temperature of the condensation according to the pressure drop, while it is less important to reach a high accuracy in the computation of the velocities [2]. This accuracy could be reached, for example, by considering the homogeneous model. For this reason the homogeneous model is examined contemplating the refrigerant pressure

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Nomenclature

Symbols

A	cross sectional area of the tube (m^2)
c_p	specific heat (kJ/kg K)
d	distance between the row of the tubes toward the air flux (m)
D	diameter (m)
f	friction coefficient
F	pressure loss (kPa/m)
j	enthalpy (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
M	mass per metre (kg/m)
p	pressure (kPa)
$\dot{q}$	heat flux (kW/m^2)
Q	heat power
Re	Reynolds number
s	fin thickness (m)
T	temperature (K)
U	heat transfer coefficient (kW/m^2 K)
x	vapour quality
X_{tt}	dimensionless constant
y	length in air flow direction (m)
w	refrigerant velocity (m/s)
z	length in refrig. flow direction (m)

Greek symbols

α	void fraction
Δ	difference
η	efficiency
θ	time
λ	heat conductivity (kW/mK)
μ	dynamic viscosity (kg/ms)
ρ	density (kg/m^3)
Φ_L	friction multiplier

Subscripts

air	air
c	condensation
co	condenser
exp	experimental
fin	fins
G.P.F.	gas phase friction
i	inside
in	inlet condition
L	saturated liquid
L.P.F.	liquid phase friction
num	numerical
o	outside
out	outlet condition
r	refrigerant
t	tube
T.P.	two-phase
T.P.F.	two-phase friction
V	saturated vapour
w	tube wall

drop inside the tubes. The model presented is for steady-state and is based on physical equations. With an appropriate division of the condenser coil, the finite difference equations are solved by means of the Newton–Raphson iteration algorithm.

In the simulation of the de-superheating of the refrigerant R407C the equations representing the model are very simple treating a single-phase refrigerant.

2. Homogeneous refrigerant mathematical model

According to the hypothesis of the homogeneous model, the velocities of the vapour and of the liquid phase are equal not considering the slip velocity between liquid and vapour during the condensation; besides the refrigerant flow is considered one-dimensional, pure (no oil contamination), the refrigerant vapour and liquid are incompressible and with the two phases in thermal equilibrium conditions.

The two-phase flow behaves like a single phase having fluid properties whose values are mean value for the flow. For the refrigerant the variations of the kinetic and potential energies are negligible. The air exchanges heat power with a finned surface and a fin efficiency is considered too; the axial conduction within the pipe wall is ignored. The air heat transfer coefficient is uniform and no mass and energy accumulation occurs treating the air as incompressible.

2.1. Equations for the refrigerant

Continuity equation in the two-phase area:

$$\frac{\partial \rho_{T.P.}}{\partial \theta} + \frac{\partial(\rho_{T.P.} w_{T.P.})}{\partial z} = 0 \tag{1}$$

Momentum equation in the two-phase area:

$$\frac{\partial(\rho_{T.P.} w_{T.P.})}{\partial \theta} + \frac{\partial(\rho_{T.P.} w_{T.P.}^2)}{\partial z} = -\frac{\partial p_{T.P.}}{\partial z} + F_z \tag{2}$$

where

$$\rho_{T.P.} = \alpha\rho_V + (1-\alpha)\rho_L$$

the void fraction equal to

$$\alpha = \frac{\rho_{T.P.} - \rho_L}{\rho_V - \rho_L} = \frac{1}{1 + \frac{\rho_V(1-x)}{\rho_L x}}$$

Treating the two-phase flow as an equivalent single-phase flow, the frictional contribution to the overall pressure gradient can be determined using a conventional friction factor $f_{T.P.F.}$ [3]