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Improved thermohydraulic numerical model of a heat exchanger for air-to-refrigerant systems

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ABSTRACT

This paper reports on a numerical model derived to predict the transient thermohydraulic behavior of the air and refrigerant in a heat exchanger for refrigerant systems. The purpose of this study is to improve the numerical model of the fin-and-tube heat exchanger for air conditioners. The equations of mass, energy, and momentum were completely separated for vapor and liquid in a two-phase thermal fluid flow. The model was validated by analyzing experimental data based on the temperature and pressure of the refrigerant. The numerical results of superheating, temperature, and pressure distributions in each zone in the fin-and-tube structure were estimated using the input variables. The maximum errors in temperature difference and pressure were within 5 °C and 4%, respectively.

1. Introduction

The evaporator and condenser are the key devices for the thermal energy exchange in a refrigerator system. The devices are designed such that a fin-and-tube structure with a large number of turns can be arranged in limited space. The tube has a dense arrangement of flat fins. Thus, the heat transfer area per unit volume is considerably large, ensuring excellent installability and enabling the transfer of more thermal energy than can be achieved using other heat exchangers. Because the tube is mainly made of copper and a flat fin of aluminum, these devices are considered to have excellent economic and lightness properties. The transient numerical model for thermohydraulic performance analysis of fin-and-tube heat exchangers (HEXs) employed in the condenser/evaporator of commercial/residential air conditioners have been developed only recently by a few researchers. The mass, energy, and momentum equations for the refrigerant were defined to predict the transient behaviors of temperature and pressure for the refrigerant and air in the condenser/evaporator mode. These studies have considered the two-phase effect in these governing equations by using the relation between the void fraction and dryness.

These governing equations were developed from two perspectives. The first approach is to develop a model by separating the equations for liquid and vapor according to the relation between the void fraction and vapor quality. For the governing equations of mass, energy, and momentum, individually separated models were developed with respect to the liquid and vapor in this approach. This concept has mainly been used for the steady state model of an adiabatic capillary tube [1,2], but Wang et al. first applied it to the unsteady state model of a fin-and-tube HEX for an air conditioner system [3,4]. The second method [5–8] is to develop a model based on equations for mass, energy, and momentum; the liquid and vapor phases are combined according to the void fraction and vapor quality. Although these two methods are identical physically, there are some differences in the solution technique. The first method treats the governing equations as ordinary differential equations, and the solution methods of for the first method vary from those for the second method. Thus, the numerical analysis models of transient behavior for the condenser/evaporator of air conditioners have been developed by the efforts of previous researchers, and numerical results have been validated to be relatively excellent by experimental work. Nevertheless, there is the potential for improvement. For example, in some models, the effect of two-

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Nomenclature

A	area (m^2)
c	specific heat (kJ/kg·K)
d	diameter (m)
f	fin or fractional factor
g	gravity constant (m/s^2)
H	height (m)
HEX	heat exchanger
h	enthalpy (J/kg)
L	length (m)
m''	refrigerant mass flux (kg/m^2·s)
NTU	number of transfer unit
q	heat flux (W/m^2)
P	perimeter (m)
p	pressure (Pa)
Re	Reynolds number
RPM	rad/min
r	radius (m)
S	slip ratio
SC	subcooling
SH	superheating
T	temperature (K)
t	time (s) or thickness (m)
u	velocity (m/s)
v	specific volume (m^3/kg)
W	width (m)
X	tube bank pitch (m)
x	vapor quality fraction
z	coordinate (m)

Greek symbols

α	heat transfer coefficient (W/m^2 K)
ρ	density (kg/m^3)
φ	local volume fraction of fins at the fin and air mix zone
Ψ	void fraction

Subscripts

a	air
co	condenser outlet
eo	evaporator outlet
f	fin
h	hydraulic
in	inlet
l	liquid
o	outer
p	pressure
r	refrigerant
w	wall

phase flow was applied to only the momentum equation, and the mass and energy equations were treated with a single-phase [5]; in some models, refrigerant pressure was obtained by applying an internal energy relation instead of total enthalpy to an energy equation while excluding the momentum equation [3–5]; and in some models, static enthalpy was applied instead of total enthalpy [3–8]. It is more reasonable to express the energy and pressure of a refrigerant by using the total enthalpy and momentum equation, respectively, because the refrigerant, with a constant mass flux, flows in a closed loop.

In this study, a thermohydraulic numerical analysis model was developed for the fin-and-tube HEX widely applied in the evaporator and condenser of refrigerant systems. The numerical model was developed for heat exchange of the refrigerant flowing in the tube and air passing through the tube/fin assembly. The governing equations were developed separately for liquid and vapor, and the purpose was to improve the models of previous researchers. To predict energy and pressure drop, total enthalpy was applied to the energy equation, and the momentum equation was considered in the model. In particular, an energy equation for the mixed zone of fin and air was added according to the physical concept. The components of the refrigerant system are the condenser, evaporator, expansion valve, and compressor, but since the purpose of this work is the improvement of the numerical analysis model for the condenser and evaporator, components other than the HEX were excluded from the analysis considering the scope of this study.

2. The numerical model of the fin-and-tube HEX for the air conditioner system

2.1. Governing equations

The basic shape of fin-and-tube HEX widely used in the air conditioner industry and the concept of control volume are well described in Ref. [9]. The tube used in the heat exchanger of the refrigeration system is considerably longer than the hydraulic diameter; therefore, it has commonly been analyzed with simple, one-dimensional numerical models. Based on the radial direction of the fin/tube assembly, a control volume can be divided into the refrigerant, tube wall, and mixed zones of fin and air. The flow of refrigerant can be mainly divided by considering a single-phase and two-phase, and usually, they are classified by vapor quality and void fraction.

Fig. 1 shows the schematic diagram of the fin-and-tube HEX fabricated in this study. As seen in Fig. 1, the HEX has four inlets and outlets and a distributor. The distributor serves the role of evenly distributing the refrigerant to several inlets, and this reduces pressure loss, enabling the even distribution of pressure in the tube. Hence, it can be assumed that the temperature and pressure of the refrigerant at the four inlets are identical and that the temperature and pressure of the refrigerant at four outlets are identical. Therefore, an analysis of a single tube connected from the inlet to the outlet was sufficient. Meanwhile, the dimensions of the heat exchanger fabricated in this study are summarized in Table 2.

The basic assumptions of the numerical model are introduced in detail in literature [3–8]. Particularly, in the numerical models described in literature, the pressure drop due to the bending of the pipe and the heat transfer by fin conduction have been ignored. Unfortunately, no methods for improving these assumptions are known at present, and these assumptions are valid in this research. In the future, however, these assumptions should be improved to achieve better predictions. In the case of the refrigerant's flow in the single-phase flow, the mass, energy, and momentum equations can be expressed as shown in Eqs. (1)–(3).

$$\frac{\partial m_r''}{\partial z} = -\frac{\partial \rho_r}{\partial t} \tag{1}$$