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Thermo-hydraulic analysis of multi-row cross-flow heat exchangers

Cheen Su An [a], Man-Hoe Kim [b,*]

[a] Lasernics Co. Ltd, KAIST ICC, 193 Munji-ro, Daejeon 34051, Republic of Korea
[b] School of Mechanical Engineering & IEDT, Kyungpook National University, Daegu 41566, Republic of Korea

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ABSTRACT

This paper presents thermal hydraulic analysis of the cross-flow finned tube heat exchangers for an outdoor unit in residential air-conditioning and heat pump applications. Performance of heat exchangers affect significantly the system energy efficiency and size of the air-conditioning and heat pumps. The Navier-Stokes equations and the energy equation are solved for the three dimensional computation domain that encompasses multiple rows of the fin-tube heat exchangers. Rather than solving the flow and temperature fields for the outdoor heat exchanger directly, the fin-tube array has been approximated by the porous medium of equivalent permeability, which is estimated from a three dimensional finite volume solution for the periodic fin element. This information is essential and time-effective in carrying out the global flow field calculation which, in turn, provides the face velocity for the microscopic temperature-field calculation of the heat exchanger. The flow field and associated heat transfer for a wide range of face velocity and fin-tube arrangements are examined and the results are presented compared with experimental data. The predicted pressure drop and heat transfer rate for various inlet velocities are in excellent agreement with the measured data.

1. Introduction

The fin and tube heat exchangers are widely used in residential air-conditioning and heat pump applications because of its highly desirable properties such as compactness and manufacturing easiness. The performance of heat exchangers (condenser and evaporator) for vapor compression system affect significantly the efficiency and size of the heat pump system. Many investigators have conducted various experimental and numerical works on heat exchangers to improve system performance [1–12]. Kim et al. [1] conducted a critical review of numerical and experimental studies on the thermal hydraulic performance of louvered fin heat exchangers. Saleem and Kim [2] investigated numerically the effect of louver pitch variation on the air-side thermal hydraulic performance of microchannel heat exchanger in Reynolds number of 50–450. Brignoli et al. [3] evaluated the effect of transport properties of fluid and refrigerant temperatures in heat exchangers. They developed a model which simulates the refrigeration cycle with inlet temperature profiles. The model is also capable of optimizing the refrigerant mass flux in order to improve the system performance. Yashar et al. [4] developed heat exchanger circuit design method with genetic algorithm. Yashar et al. [5] conducted the refrigerant circuit optimization study and they reported that the capacity of heat exchanger and system with optimized circuit can be improved by 7.9% and 2.2%, respectively. Kim and Bullard [6] investigated system performance for a window room air-conditioning system with microchannel condenser and compared the results with the conventional system with fin and tube condenser. An et al. [7], and An and Choi [8] conducted numerical study on thermal hydraulic performance of fin-tube heat exchangers under dry and wet conditions. Effect of heat exchanger configuration on refrigeration cycle performance was also investigated by Saboya et al. [9] and Klein and Reindl [10]. A great deal of efforts have been put into obtain heat-transfer correlations involving, for instance, Reynolds numbers, Colburn j-factor, friction f-factors, etc. This approach of predicting the heat exchanger performance would be economic, when successful, as it avoids time-consuming experiments or computations. However, the empirical correlations, in general, lack generality and cannot readily be modified for changing geometries. Furthermore the air-side flow profile of heat exchanger affects significantly the thermo-hydraulic performance of the heat exchangers in air-conditioning and heat pump applications [5,11,12], however it is very complicated to get the accurate face velocity profile data from experimental or numerical investigations. A more accurate and yet practical approach would be approximate the heat exchanger as a porous medium to obtain the global flow field. The heat-transfer characteristics may then

* Corresponding author.
E-mail address: kimmh@asme.org (M.-H. Kim).

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be calculated by the tube-by-tube method using the flow field already obtained.

The purpose of this paper is to develop such method and apply it to an outdoor unit of a heat-pump system. The details of the analysis procedure and the results are presented in the following sections.

2. Analysis and modeling

The schematic of a heat exchanger under consideration in an air conditioning system is shown in Fig. 1.

The tube arrangement is staggered and the fins are of wave type (see Fig. 2). Rather than solving the flow through the heat exchanger directly with any finite arithmetic, which requires prohibitively high computational efforts, the heat exchanger is modeled as a porous medium. The effective permeability and the inertial resistance factor of the medium, needed to calculate the global flow field for an outdoor unit, are estimated from an accurate three-dimensional finite-volume calculation for a single periodic module of the given heat exchanger depicted in Fig. 3. For the computational domain shown in Fig. 3, the following steady incompressible flow equations are solved for the velocity and temperature fields by using a commercial code, FLUENT [13]:

Continuity equation:

$$\frac{\partial}{\partial x_j}(\rho u_j) = 0 \tag{1}$$

Momentum equations:

$$\frac{\partial}{\partial x_j}(\rho u_j u_i) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}\mu\frac{\partial u}{\partial x}\delta_{ij}\right) + F_i \tag{2}$$

Energy equation:

$$\rho C_p \frac{\partial}{\partial x_j}(u_j T) = k\frac{\partial^2 T}{\partial x_j^2} \tag{3}$$

The flow is assumed laminar as the Reynolds number based on the fin pitch of 1.7 mm is less than 700 [14]. The upstream and downstream boundaries are placed sufficiently far away from the fin, from −3 to 6 times of tube diameter (d_o), so that a uniform velocity distribution and the parabolic condition can be applied at the respective boundaries. The no-slip and the constant temperature conditions are prescribed at the solid wall while the periodic

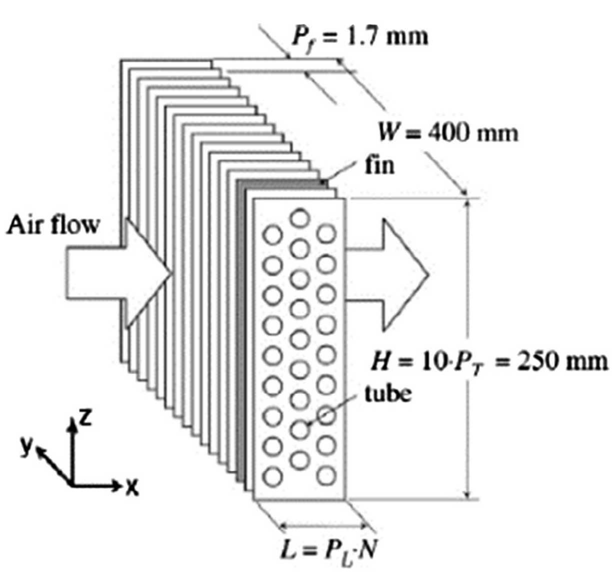

Fig. 1. Schematic of a heat exchanger.

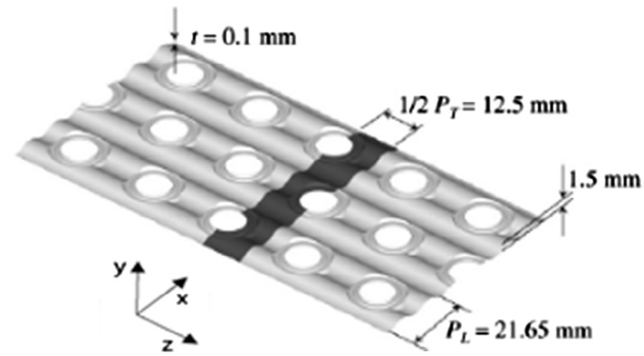

Fig. 2. Wave fin-tube heat exchanger.

and symmetric condition is imposed at the periodic and symmetric boundary. All the boundary conditions are presented as mathematical forms in Eqs. (4)–(9).

Velocity inlet boundary condition is used at the inlet face:

$$u = u_{in} \quad v = w = 0 \quad T = T_{in} \tag{4}$$

No slip and constant temperature conditions at the wall:

$$u = v = w = 0 \quad T = T_w \tag{5}$$

Periodic boundary condition on both sides of the fins:

$$\frac{\partial u}{\partial z} = \frac{\partial v}{\partial z} = 0 \quad w = 0 \quad \frac{\partial T}{\partial z} = 0 \tag{6}$$

Symmetry boundary condition is used on center between fins:

$$\frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0 \quad v = 0 \quad \frac{\partial T}{\partial y} = 0 \tag{7}$$

Pressure outlet boundary condition is applied at the outlet of the fluid domain:

$$\frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial w}{\partial x} = \frac{\partial T}{\partial x} = 0 \tag{8}$$

For fluid-solid interface:

$$u = v = w = 0 \quad T_s = T_f \quad k_s\frac{\partial T_s}{\partial n} = k_f\frac{\partial T_f}{\partial n} \tag{9}$$

Once the relationship between the pressure drop and the inlet velocity has been established from this calculation, the permeability K and the inertial factor C of the equivalent porous medium can be determined. The Forchheimer formulation [15] is given as:

$$\frac{\Delta p}{l} = -\left(\frac{\mu}{K}u + \frac{1}{2}\rho C u^2\right) \tag{10}$$

K: permeability of porous medium
C: inertial factor
U: air velocity.

The first and second terms in Eq. (10) indicate the viscous and inertial characteristics of porous media flow [16], respectively. The heat transfer from the solid of constant temperature to the cooling air is computed from the enthalpy change of the air. Taking the contact resistance between the fin and the tube wall and the fin efficiency into account, the total heat transfer rate of the heat exchanger unit is then calculated from the tube–by-tube method [17] described below for the velocity distribution at the fin inlet obtained in the global flow analysis.

For the global flow calculation, Eqs. (1), (2) and (10) are solved for the outdoor unit (see Fig. 8) of a heat pump system. The heat exchanger segment of the domain is approximated by the porous medium, and Eq. (10) is solved in that region in place of Eq. (2).