International Journal of Heat and Mass Transfer 120 (2018) 534–539

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

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

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ARTICLE INFO

Article history:
Received 16 October 2017
Received in revised form 15 December 2017
Accepted 17 December 2017

Keywords:
Heat exchanger
Wave fin
Tube-by-tube method
Vapor compression cycle
Heat pump
Air-conditioning

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

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https://doi.org/10.1016/j.ijheatmasstransfer.2017.12.088
0017-9310/

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

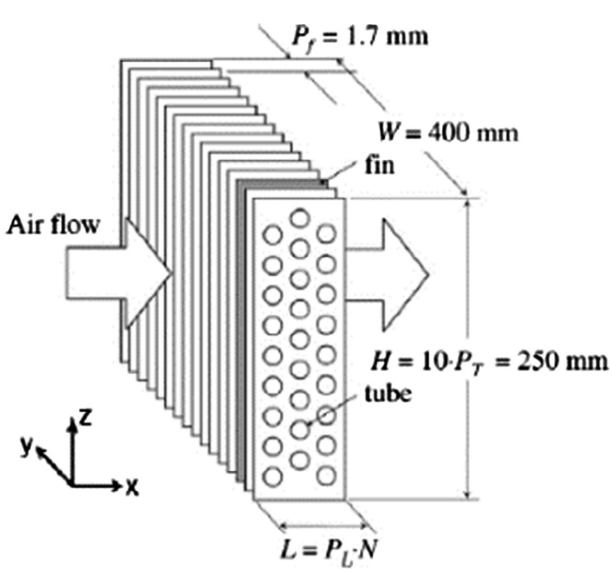


Fig. 1. Schematic of a heat exchanger.

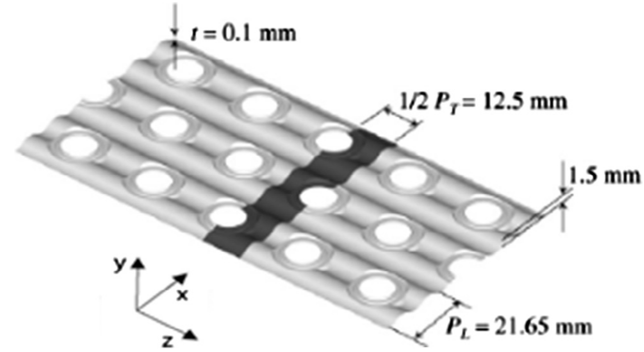


Fig. 2. Wave fin-tube heat exchanger.

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:

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

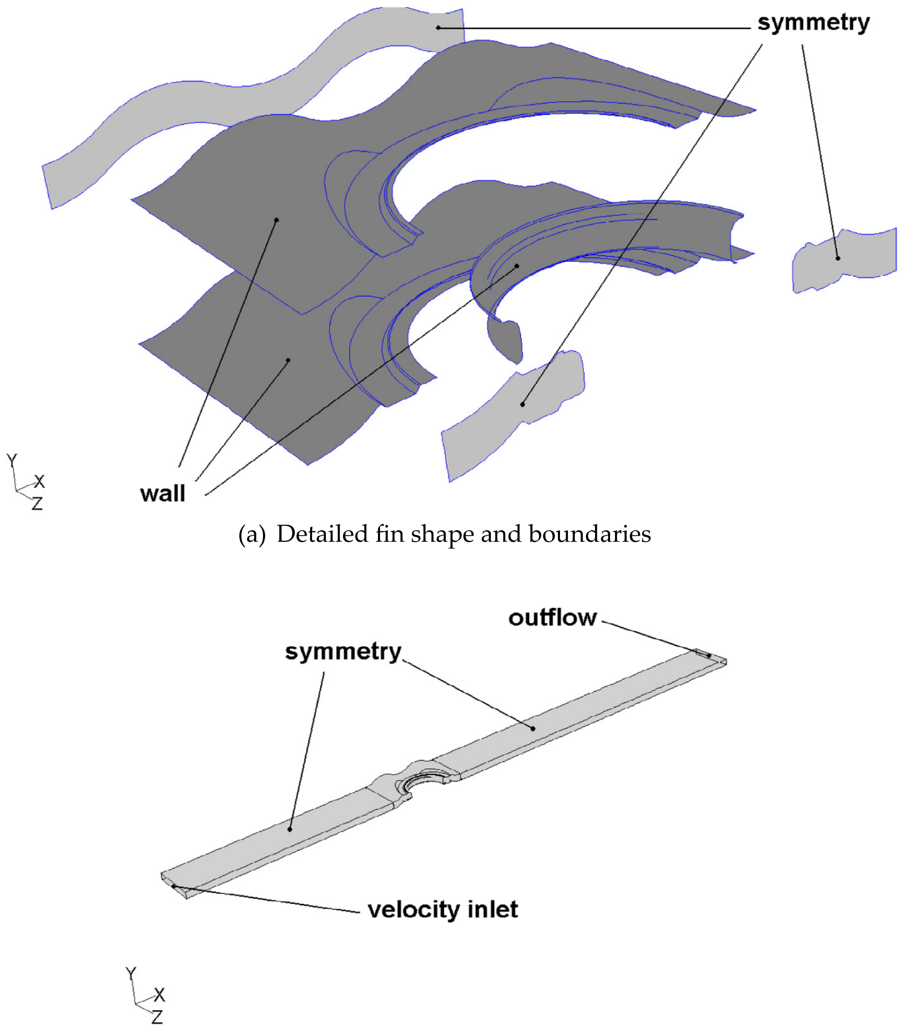
Once the relationship between the pressure drop and the inlet velocity has been established from this calculation, the permeability K and the inertial factor C of the equivalent porous medium can be determined. The Forchheimer formulation [15] is given as:

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

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).



(a) Detailed fin shape and boundaries

(b) Computational domain and boundary conditions

Fig. 3. Schematic of a periodic module.

Using the heat-transfer results for the periodic module, the overall heat-transfer rate for the whole heat exchanger, in which the temperature varies along the tube, may be estimated by the tube-by-tube method. The overall heat transfer coefficient, $\boldsymbol{U}$, of a row of the heat exchanger is defined as

$$Q = U \cdot A \cdot \Delta T_{lm} \tag{11}$$

where ΔT_{lm} is the log-mean temperature difference and A is the heat transfer area.

$$\Delta T_{lm} = \frac{T_{a,in} - \overline{T}_{a,out}}{\ln[(T_w - \overline{T}_{a,out})/(T_w - T_{a,in})]} \tag{12}$$

The net heat-transfer rate Q is estimated by multiplying the fin efficiency to the heat flux calculated for a single periodic module above. The fin efficiency is obtained from the empirical formula of Schmidt [18].

When the UA of the heat exchanger, the inlet temperatures of the air and the water are known, the outlet temperatures can be determined by using Eqs. (13) and (14) which are derived from the energy balance relation of the heat exchanger [19].

$$T_{a,out} = T_{a,in} + \frac{C_r}{C_a}(T_{r,in} - T_{a,in})\left[1 - \exp\left\{\frac{C_a}{C_r}(e^{-UA/C_a} - 1)\right\}\right] \tag{13}$$

$$T_{r,out} = T_{a,in} + (T_{r,in} - T_{a,in})\exp\left(\frac{C_a}{C_r}(e^{-UA/C_a} - 1)\right) \tag{14}$$

Following the procedure described below, the temperature at each point along the heat exchanger can be determined.

1. Assume the air temperature at the exit every tube.
2. Using Eqs. (13) and (14), calculate the water temperature at the exit of each tube for the given air temperature.
3. Calculate the air temperature by using the water temperature given and Eqs. (13) and (14).
4. Repeat Steps 2 and 3 until the air and water temperatures converge.

A major drawback of the above procedure may be that it relies on the fin efficiency which is empirical to determine the actual heat flux. One can avoid this by solving the entire heat-transfer chain simultaneously, from coolant inside the tube to the outside air. The energy equation needs to be solved for a domain that includes the tube and fin element.

The thermal resistance for the system may be broken down to five stages (see Fig. 4): convection from the coolant to the tube ($1/(h_r A_r)$), conduction through the solid ($t_t/(k_t A_r)$), tube and fin collar ($1/(h_c A_r)$), contact resistance between the tube wall and the fin

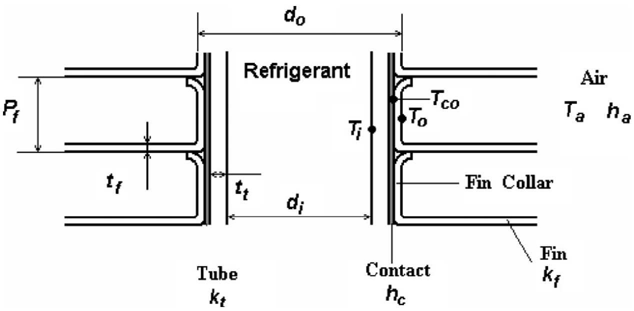


Fig. 4. Detailed view of fin-tube assembly.

Table 1
Relative thermal resistance range for refrigerant air fin-tube heat exchangers.

$1/(h_rA_r)$	$t_t/(k_tA_r)$	$1/(h_cA_r)$	$t_f/(k_fA_r)$	$1/(h_aA_a\eta_s)$
0.08–0.18	0.0008–0.0018	0.12–0.27	0.0005–0.0011	0.55–0.80

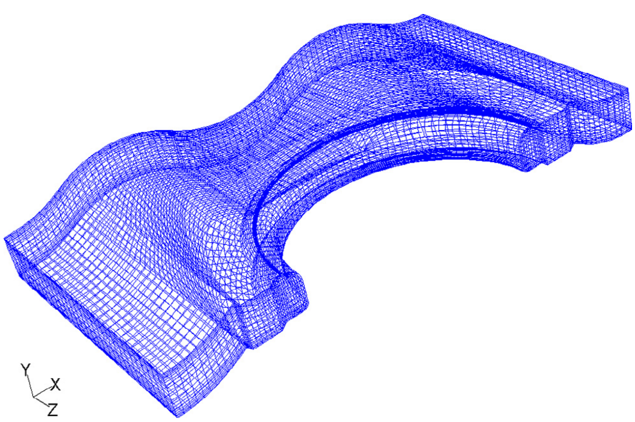


Fig. 5. Sample grid.

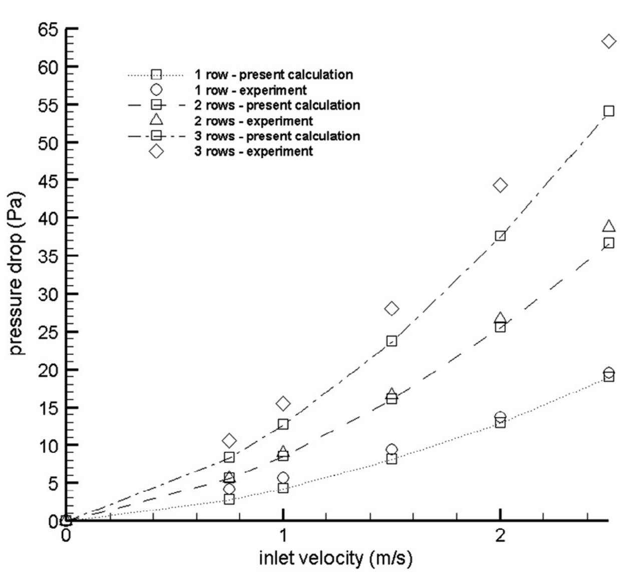


Fig. 6. Pressure drops for various face velocities.

Table 2
Permeability and inertial resistance factor for various row arrangements.

Number of rows		K (m^2)	C (m^{-1})
1	Experiment	9.5725e−8	1.1236e−2
	Calculation	2.2944e−7	1.7794e−2
2	Experiment	1.7118e−7	1.6988e−2
	Calculation	1.9416e−7	1.6181e−2
3	Experiment	1.2964e−7	1.6845e−2
	Calculation	1.9915e−7	1.6210e−2

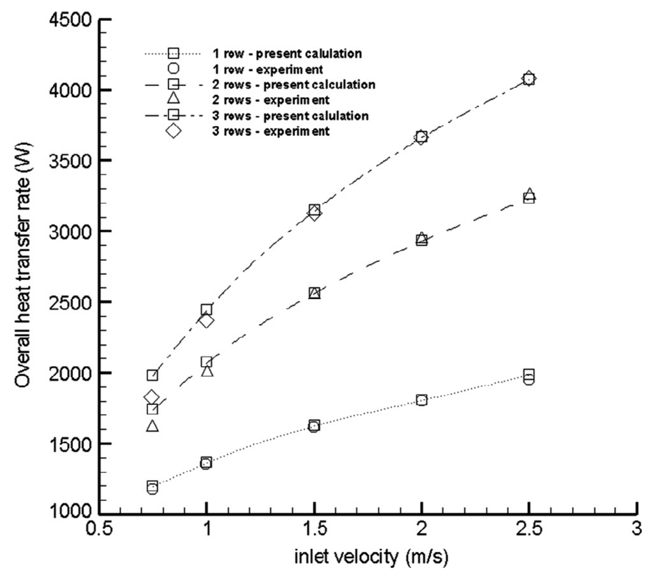


Fig. 7. Overall heat transfer rate versus various face velocities.

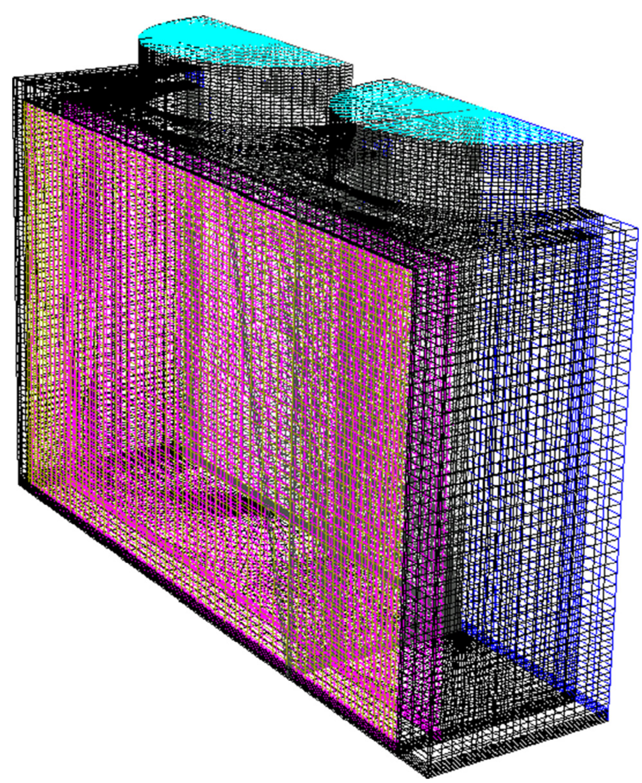
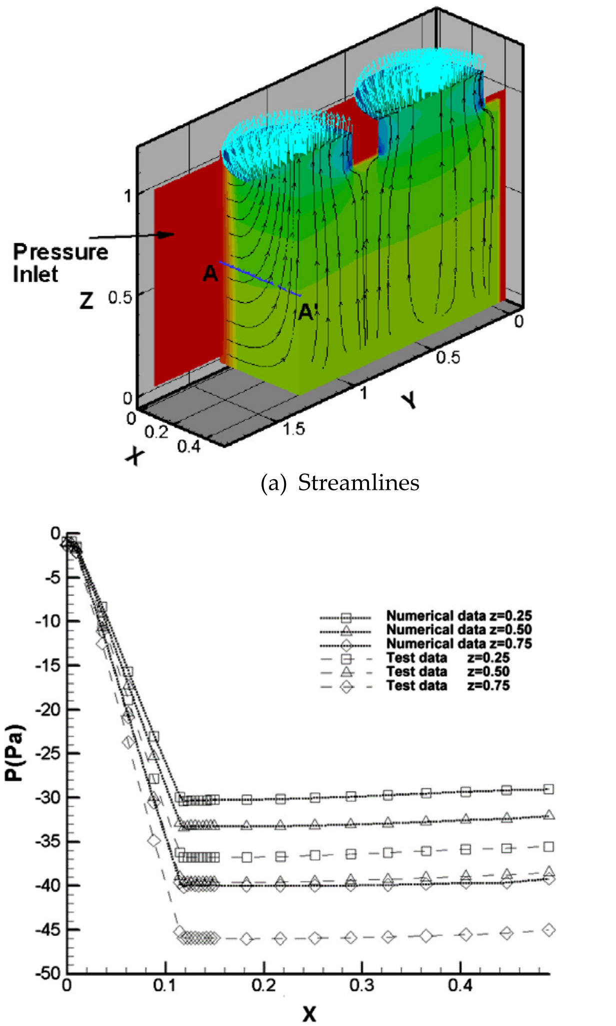


Fig. 8. Computational grid for an outdoor unit.



(a) Streamlines

(b) Pressure distribution

Fig. 9. Streamlines (a) and pressure distribution (b) for the outdoor unit of an air-conditioning system.

collar ($t_f/(k_f A_r)$), and convection from the fin surface to the air ($1/(h_a A_a \eta_s)$). A general order of relative thermal resistance for each phase is given in Table 1 [20].

It can be seen from the table that the resistance due to conduction through the tube wall or fin collar is negligible. The thermal resistance between the fin surface and the cooling air is reflected through the use of fin efficiency. The effects of the convection inside the tube and the contact resistance are determined empirically for this fin shape and given in Youn [20]. Corresponding values for h_r and h_c are 1.2340×10^4 W/m^2 K and 6.791×10^3 W/m^2 K, respectively, and are used in the present analysis.

3. Results and discussion

The fin pitch of the heat exchanger unit under consideration is 1.7 mm and the number of rows is 1, 2 or 3. Fig. 5 shows a sample grid about a single tube, which is non-uniformly distributed to place more grids near the solid surface. Only half of the unit fin module has been considered using periodicity and symmetry boundary conditions. Approximately 200,000 grid cells are used to fit the domain of each row after grid independence study. Additional 10,200 and 17,000 grid cells are distributed for the upstream and downstream regions of the fin, respectively [8]. Fig. 6 shows the pressure drop versus the inlet flow velocity. Comparing calculation pressure drop with the test data has the largest error of 18.2% for 3-row heat exchanger partly due to the laminar flow assumptions. Using 2nd order polynomial curve fitting, K and C are determined and are presented in Table 2. The overall heat transfer rates calculated by the tube by tube method is shown in Fig. 7, and calculated values are good agreement with test data in the error range of 0.5–2.5%.

Fig. 8 shows the computational grid for the outdoor unit of the heat-pump system. Using the model coefficients in Table 2 as inputs for the analysis of the heat exchanger, the flow field for a given pressure drop is calculated. Fig. 9 shows the streamlines and pressure distribution in the outdoor unit while the resulting inlet velocity profile for 2- or 3-row tube array is depicted in Fig. 10.

It is seen from Fig. 10 that the cooling air flows more readily through the upper part of the heat exchanger than the lower part, especially for the single-row case, since the length of the flow passage is shorter there. However, the velocity profile becomes more

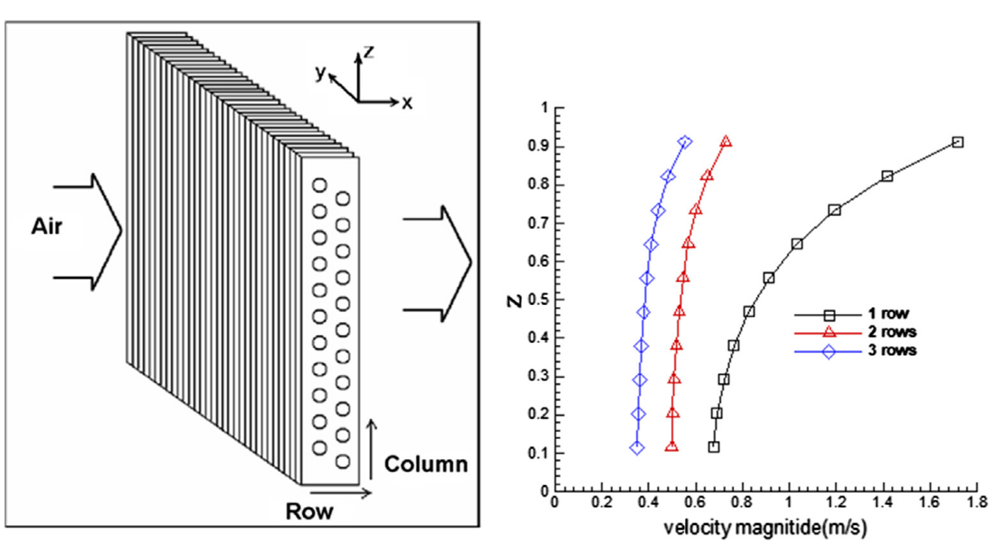


Fig. 10. Velocity distribution at the inlet for a given pressure drop.

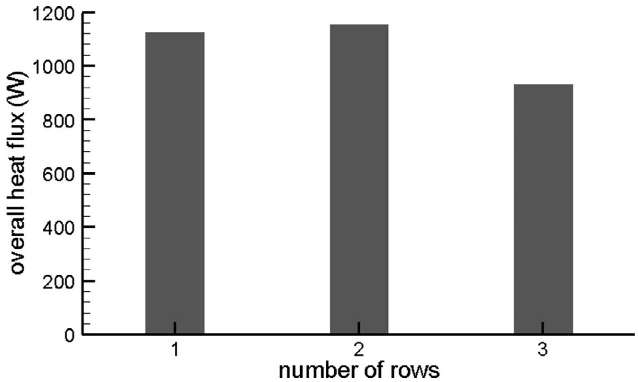


Fig. 11. Overall heat transfer rate for number of rows.

uniform as the number of row increases. This is because the relative variation in the total pressure loss from the inlet to the exit becomes less as the pressure drop across the heat exchanger increases. The velocity profiles at the inlet plane can be used to repeat the tube-by-tube analysis for more realistic heat-transfer analysis; the results are presented in Fig. 11. For a given pressure loss through the outdoor unit, the two-row heat exchange appears to perform best in the study.

4. Conclusions

An efficient and accurate heat-transfer analysis procedure for a multi-row cross-flow heat exchanger of an air-conditioning system has been developed. The steady state 3-dimensional continuity, momentum, and energy equations are solved by using the finite volume method to obtain the thermal-hydraulic characteristics of multi-row fin and tube heat exchangers. Approximating the heat exchanger by a porous medium, of which the effective permeability and the inertial resistance factor are estimated by a separate three-dimensional finite-volume calculation, the overall flow field can be calculated fairly efficiently. By coupling the three-dimensional heat-transfer results for a periodic domain with the tube-by-tube method, the total heat transfer characteristics for the whole heat exchanger can be obtained. If the appropriate turbulent model is adapted in the study, the results will be better than laminar model.

Conflict of interest

The authors declare that there is no conflict of interests.

Acknowledgement

This work is supported by the Ministry of Trade, Industry & Energy (MOTIE, Korea) under Industrial Technology Innovation Program (No. 10070117).

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