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자동차 공조시스템용 평행류형 응축기의 모델링

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Modeling of Parallel Flow Type Condenser for Automotive Air Conditioning System

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Abstract

자동차 공조용 시스템에 사용되는 평행류형 응축기에 대하여 실제 운전조건에서 성능을 예측할 수 있는 모델링을 개발하였다. 모델링에 사용된 방법은 유효도-전달단위수법이고, 국소구간을 나누어 해석하는 국소구간법을 사용하였다. 모델링에 사용된 작동유체는 HFC134a이며, 응축기를 흐르면서 방생하는 냉매의 압력손실에 대한 물성변화를 포함시켜 보다 실제에 가깝게 해석하였다. 모델링에는 공기측과 냉매측의 열전달계수와 압력손실계수에 관한 상관식들을 포함하고 있다. 모델링의 결과는 실험값과 비교하여 비교적 잘 일치한다.

Keywords : 평행류형 응축기(Parallel flow type condenser), 유효도-전달단위수법(ε-NTU)

NOMENCLATURE

A	Heat transfer area
A_{Dl}	Area on which profile drag occurs
A_e	End region heat transfer area
A_l	Louvered heat transfer area
A_o	Outside heat transfer area
A_r	End region aspect ratio
A_{S1}	Plain leading and trailing heat transfer area
A_{S2}	Plain turn-around heat transfer area
C	Heat capacity rate
c_p	Specific heat

C_{Dl}	Drag coefficient on the louvers
D_h	Hydraulic diameter
F_d	Fin depth in the air flow direction
F_p	Fin pitch
F_{th}	Fin thickness
f	Friction factor
H	Fin height
G	Mass velocity
h	Heat transfer coefficient or Enthalpy
k	Thermal conductivity
L_l	Louver length
L_p	Louver pitch
m	Mass flow rate
N_l	Number of louvers
NTU	Number of Transfer Unit
Nu	Nusselt number
P	Pressure
Pr	Prandtl number
q	Heat transfer rate
Re	Reynolds number
S_1	Leading and trailing louver length
S_2	Turn-around section length
T_p	Tube pitch
T_{th}	Tube wall thickness
T_w	Tube width
U	Overall heat transfer coefficient
u_c	Minimum area flow velocity
v	Specific volume
V	Velocity
x	Quality of refrigerant

Greek symbols

α	Void fraction
δ	Thickness
Δ	Variation
ε	Effectiveness
η	Surface fin efficiency
θ	Louver angle
μ	Viscosity
ν	Kinematic viscosity
ρ	Density
Ψ	Two-phase multiplier

Subscripts

a	Air
eq	Equivalent
v	Vapor
i	Inlet
l	Liquid
lo	Liquid only
max	Maximum
min	Minimum
o	Outlet
r	Refrigerant
t	Tube

1. Introduction

Heat exchangers like automotive air conditioning condensers are manufactured with a number of designs, usually containing various flow passages with different fin designs. Manufacturers are actively working to improve their designs in order to reduce the size and weight and improve the thermal behavior. In order to achieve these objectives, in the present work a theoretical model has been developed to predict the parallel flow condensers performance under arbitrary operating conditions.

There is a number of studies referred to condensing flow inside horizontal tubes such as those of references[1)2)3)]. Other works relative to heat transfer during condensation are described in references[4)5)]. Cavallini and Zecchin[2)] developed a semi-empirical equation that is simple in form and correlates refrigerant data quit well. Data for several refrigerants, including R-11, R-12, R-21, R-22, R-113, and R-114. So, that has been used in the present study for the prediction of parallel flow condensers. An useful method to calculate pressure drop during condensation are those of references[6)7)]. A more proper method is that of Friedel[8)], that has been used in the present study.

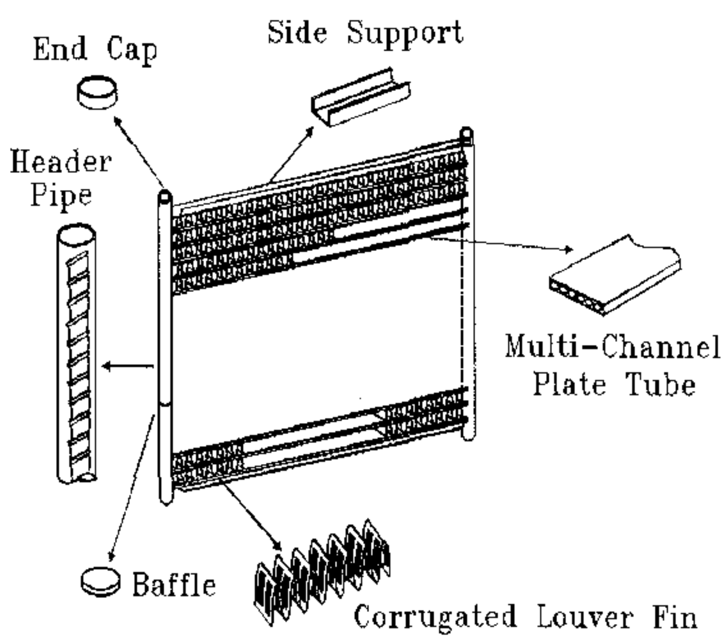


Fig 1. Basic structure of a parallel flow type condenser.

The Louvered fin geometry is widely used in automotive heat exchangers. For this reason, a correlation to predict the heat transfer and friction characteristics is of great interest. There is a number of studies referred to the heat transfer and friction factor of louvered fin such as those of references[9)10)11)12)]. Webb et al.[12)] develops a semi-analytical heat transfer and friction correlations applicable to the louver fin geometry, that has been used in the present study.

The geometry of the condenser of the present study is shown in **figure 1** and **figure 2**. This type of condensers consists of multiple passages for refrigerant flow with different directions and finned cross flow passages for air. Each condenser is considered to have three thermodynamic zones: In the first zone the refrigerant enters in the condenser as superheated vapor and is cooled down to the condensation temperature. In the second zone it condenses loosing heat at a saturation temperature. In the third zone the condensate is cooled down below a saturation temperature (subcooling).

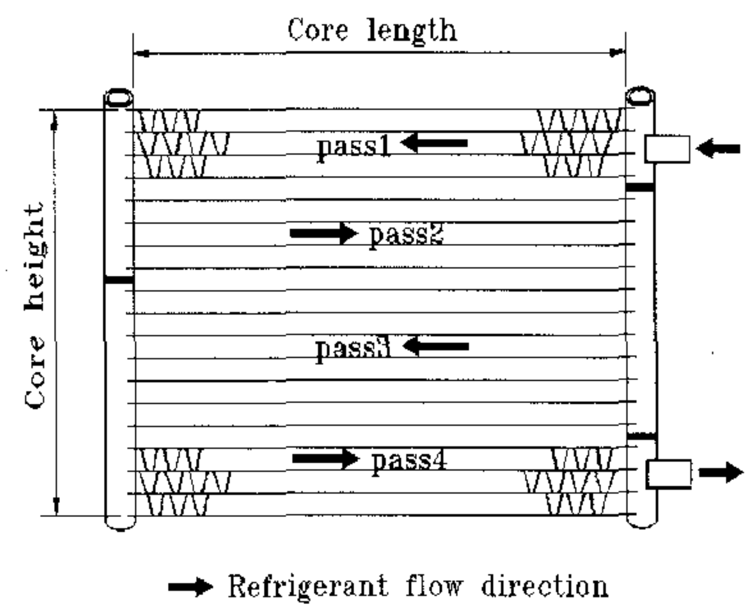


Fig 2. Schematic of a parallel flow type condenser.

2. Basic of the condenser model

The heat in the condenser is transferred from the refrigerant to the wall and through this to the air. Heat transfer coefficients and pressure drop are then calculated as described in the following texts.

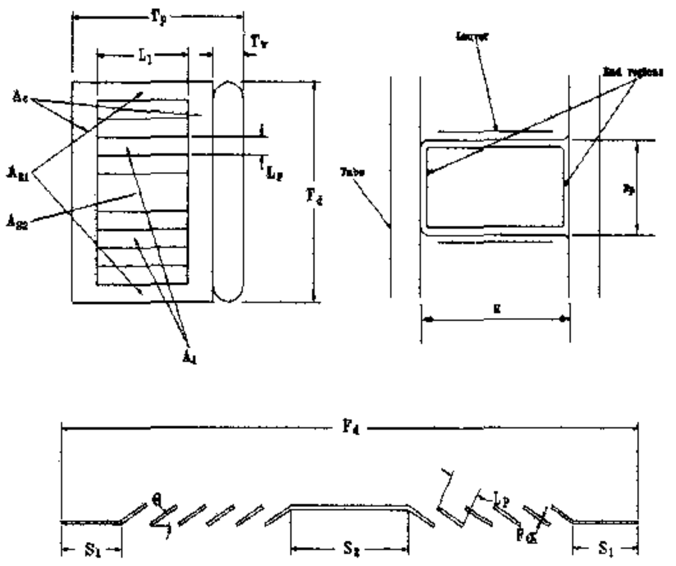


Fig 3. Detailed element of parallel flow type condenser with louver fin.

2.1 Air heat transfer and pressure drop correlations

The air heat transfer coefficient and pressure drop of the condenser are obtained by using a correlation proposed by Webb et al.[12] for a similar heat exchanger geometry.

The louver fin geometry is shown in figure 3. The fin area is divided into four regions. These areas are A_e, A_{S1}, A_{S2}, and A_l, which represent the non-louvered and end regions, the plain leading and trailing areas, the plain middle areas, and the internal louvered areas respectively. these areas are defined as follows:

$$A_e = 2F_d[(T_p - T_w - L_l) + (F_p - F_{th})] \quad \text{(1a)}$$

$$A_{S1} = 4L_l S_l \quad \text{(1b)}$$

$$A_l = 2L_l L_p (N_l + 1) \quad \text{(1c)}$$

$$A_{S2} = 2L_l S_2 N_{S2} \quad \text{(1d)}$$

Using these areas, the heat transfer correlation is given by

$$\eta h_o A_o = h_e A_e + 0.744\,\eta_f\,\mathrm{k}\,L_l \beta \times Re_{Lp}^{0.580}\left(\frac{2\theta}{\pi}\right)^{0.195}\left(\frac{F_p}{H}\right)^{-0.0522} \quad \text{(2)}$$

where $Re_{Lp} = \frac{u_c L_p}{\nu}$

$$A_r = \frac{T_p - T_w - L_l}{F_p - F_{th}} \le 1$$

$$D_{he} = \frac{4(T_p - T_w - L_l)(F_p - F_{th})}{2(T_p - T_w - L_l + F_p - F_{th})}$$

$$\beta = 2\left(\frac{S_1}{L_p}\right)^{0.5} + (N_l + N_{S2}) + N_{S2}\left(\frac{S_2}{L_p}\right)^{0.5}$$

$$\frac{h_e D_{he}}{\mathrm{k}} = 10.81 + 12.63A_r - 1.61A_r^2 - 18.86A_r^{0.5}$$

The heAe term refers to the unlouvered end regions shown in figure 3.

The friction factor correlations for the two Reynolds number regions are given by

$$fA_o = f_e A_e + C_{Dl} A_{Dl} + 6.242 L_l L_p \beta \times Re_{Lp}^{-0.759}\left(\frac{2\theta}{\pi}\right)^{-0.233}\left(\frac{F_p}{H}\right)^{-0.628} \quad 400 < Re_{Dh} < 1000 \quad \text{(3a)}$$

$$fA_o = f_e A_e + C_{Dl} A_{Dl} + 0.876 L_l L_p \beta \times Re_{Lp}^{-0.455}\left(\frac{2\theta}{\pi}\right)^{0.521}\left(\frac{F_p}{H}\right)^{-0.772} \quad 1000 < Re_{Dh} < 4000 \quad \text{(3b)}$$

where $A_{Dl} = N_l L_l F_{th}$

$$C_{Dl} = 2\pi\sin\theta \qquad 8^\circ < \theta$$

$$C_{Dl} = 0.8 \qquad 8^\circ \le \theta \le 12^\circ$$

$$C_{Dl} = \frac{\cos\theta}{0.222 + \frac{0.283}{\sin\theta}} \qquad \theta > 12^\circ$$

$$f_e = \frac{\nu}{u_c D_{he}}(32.72 + 18.73A_r - 37.04A_r^{0.5} - 0.164A_r^{-1})$$

The profile drag term, $C_{Dl}A_{Dl}$, is caused by pressure drag on the finite thickness of the louvered fins.

2.2 Refrigerant heat transfer and pressure drop correlations

As indicated, three zones are considered: cooling condensation and sub-cooling. This justifies the use of three different heat transfer correlations in the refrigerant side, one for each cooling zone.

In the first zone, the Nusselt number is calculated from standard expression of Dittus and Boelter[13], and the heat transfer coefficient is then computed by the expression that defines the Nusselt number. In the second zone, the Cavallini-Zecchinc[2] correlation that accounts for the phase change is used:

$$Nu = 0.05\, Re_{eq}^{0.8}\, \Pr^{0.3} \quad \cdots\cdots (4)$$

where Reeq is an equivalent Reynolds number that is a function of vapor Reynolds number Rev and liquid Reynolds number Rel and is computed as:

$$Re_{eq} = Re_v\left(\frac{\mu_v}{\mu_l}\right)\left(\frac{\rho_l}{\rho_v}\right)^{0.5} + Re_l \quad \cdots\cdots (5)$$

In the third zone, the heat transfer coefficient is calculated in the same manner as in the first zone but using liquid refrigerant properties in this case.

The single phase frictional pressure drop is calculated by applying Darcy-Weisbach[14] formula as follows:

$$\frac{dp}{dL} = f\frac{2G^2}{\rho D_h} \quad \cdots\cdots (6)$$

where friction factor f depends on the respective Reynolds number. One such relationship is the Blasius equation:

$$f = 0.079 Re^{-0.25} \quad \cdots\cdots (7)$$

A separated flow model has been considered for the calculation of frictional two-phase pressure change. It is based on the model developed by Lockhart and Martinelli[6] relative to their studies of air-water flows. The concept behind the separated flow model is to calculate the two-phase frictional pressure drop based on a multiplier, ϕ, and the frictional pressure gradient that would result if each phase flowing alone in the tube. Single-phase pressure gradients can be calculated by using the Darcy-Weisbach formula. Either one of the two single-phase pressure gradients liquid or vapor phase can be used to solve the two-phase pressure gradient. The equation used has the form

$$\frac{dp}{dL} = \left(\frac{dp}{dL}\right)_{lo}\phi_{lo}^2 \quad \cdots\cdots (8)$$

The two-phase frictional multiplier lo, was proposed by Friedel[8].

$$\phi^2_{lo} = E + 3.23\, F\, H\, Fr^{0.045}\, We^{0.035} \quad \cdots\cdots (9)$$

where $E = (1-x)^2 + x^2\frac{\rho_l f_{vo}}{\rho_v f_{lo}}$

$$F = x^{0.78}(1-x)^{0.224}$$

$$H=\left(\frac{\rho_l}{\rho_v}\right)^{0.91}\left(\frac{\mu_v}{\mu_l}\right)^{0.19}\left(1-\frac{\mu_v}{\mu_l}\right)^{0.7}$$

$$Fr=\frac{G^2}{gD_h\rho_h^2}$$

$$We=\frac{G^2D_h}{\rho_h\sigma}$$

$$\rho_h=\frac{\rho_v\rho_l}{x\rho_l+(1-x)\rho_v}$$

In two-phase flow the momentum changes of both the liquid and the vapor must be taken into account. The pressure gradient on an arbitrary control volume an be expressed as:

$$\frac{dP}{dL}=G^2\frac{d}{dL}\left[\frac{x^2}{\rho_v\alpha}+\frac{(1-x)^2}{\rho_l(1-\alpha)}\right] \cdots (10)$$

where α is the void fraction and calculated by Zivi[15] correlation.

$$\alpha=\left[1+\left(\frac{1-x}{x}\right)\left(\frac{\rho_v}{\rho_l}\right)^{2/3}\right]^{-1} \cdots\cdots (11)$$

Finally local pressure drops at entry and exit of the condenser tubes are calculated as follows:

$$\Delta P=K\rho\frac{V^2}{2} \cdots\cdots (12)$$

where the coefficient K adopts a value 0.5 for entries and 1.0 for exits.

3. Procedure of calculation

The analysis of condenser is done using the NTU(Number of Transfer Units) method. In this method, the heat exchanger effectiveness is obtained from Kays and London[16].

Depending on whether the hot refrigerant or cold air has the minimum $C=mc_p$, the heat exchanger effectiveness may be either

$$\varepsilon=\frac{T_{r,i}-T_{r,o}}{T_{r,i}-T_{a,o}}\ ;\ \text{for } C_{\min}=C_r \cdots (13a)$$

or

$$\varepsilon=\frac{T_{a,o}-T_{a,i}}{T_{r,i}-T_{a,o}}\ ;\ \text{for } C_{\min}=C_a \cdots (13b)$$

For the refrigerant-to-air condenser with finned tubes, the refrigerant is inside the tubes and the air is on the outside. Both the refrigerant and air inlet temperatures and pressures, air velocity, and refrigerant mass flow rate are known or can be calculated. The heat transfer rate of a heat exchanger segment is

$$q=\varepsilon C_{\min}(T_{r,i}-T_{a,i}) \cdots\cdots (14)$$

where ε value can be obtained by the NTU approach. Expression for several flow configurations are available in Kays and London[16]. For cross flow with both streams unmixed and refrigerant with single-phase,

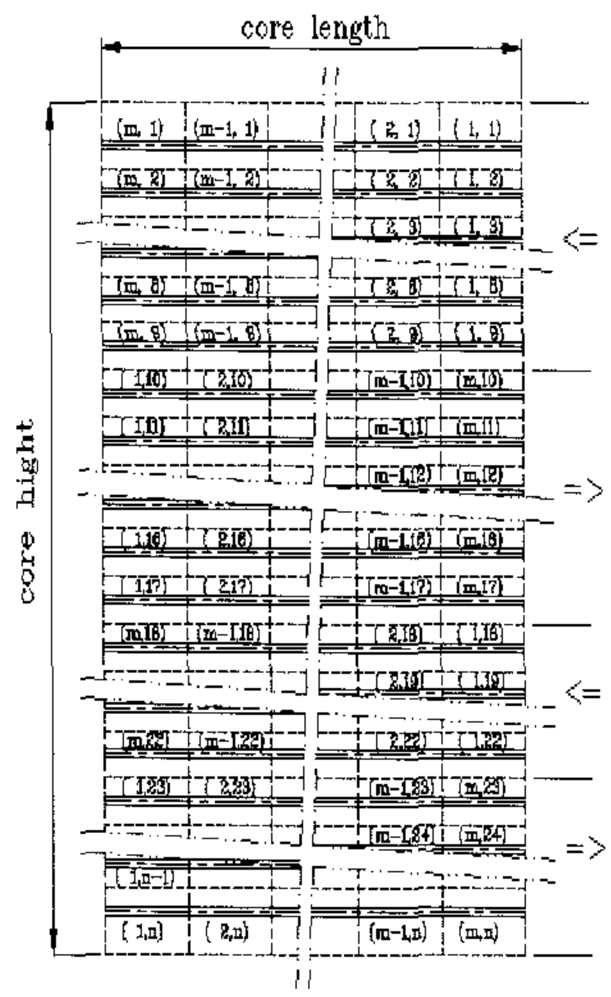
$$\varepsilon=1-\exp\{C_r^{-1}NTU^{0.22}[\exp(-C_rNTU^{0.78})-1]\} \cdots\cdots (15a)$$

For refrigerant with two-phase,

$$\varepsilon=1-\exp(-NTU) \cdots\cdots (15b)$$

where $C_r=C_{min}/C_{max}$

The NTU parameter is defined as UA/C_{min} and may be thought of as a heat transfer a heat transfer sizing factor. Because the tube length is subdivided, the heat transfer area A of each segment can be calculated. The overall heat transfer coefficient, U, for a finned tube



<= Refrigerant Flow Direction

Fig 4. Arrangement of segment.

condenser, assuming no fouling, is given by

$$\frac{1}{UA} = \frac{1}{\eta h_a A_a} + \frac{1}{h_r A_r} + \frac{\delta_t}{k_t A_t} \qquad (16)$$

where η is the surface efficiency.

If state of refrigerant is single-phase, the outlet temperatures for air and refrigerant of a heat exchanger segment is calculated by the heat transfer rate of that. If that is two-phase, quality of the refrigerant is calculated.

For calculated outlet condition of segment, the condenser is subdivided as figure 4 and ε-NTU method is applied as figure 5 The simulation program consists of a main program with several subroutines that are called sequentially. The flow diagram of the simulation program is shown in figure 6

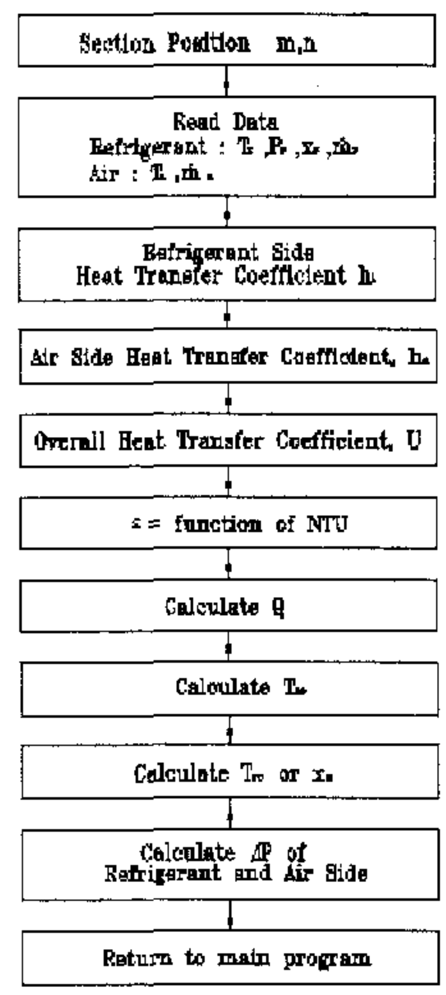


Fig 5. Flowchart of section calculation.

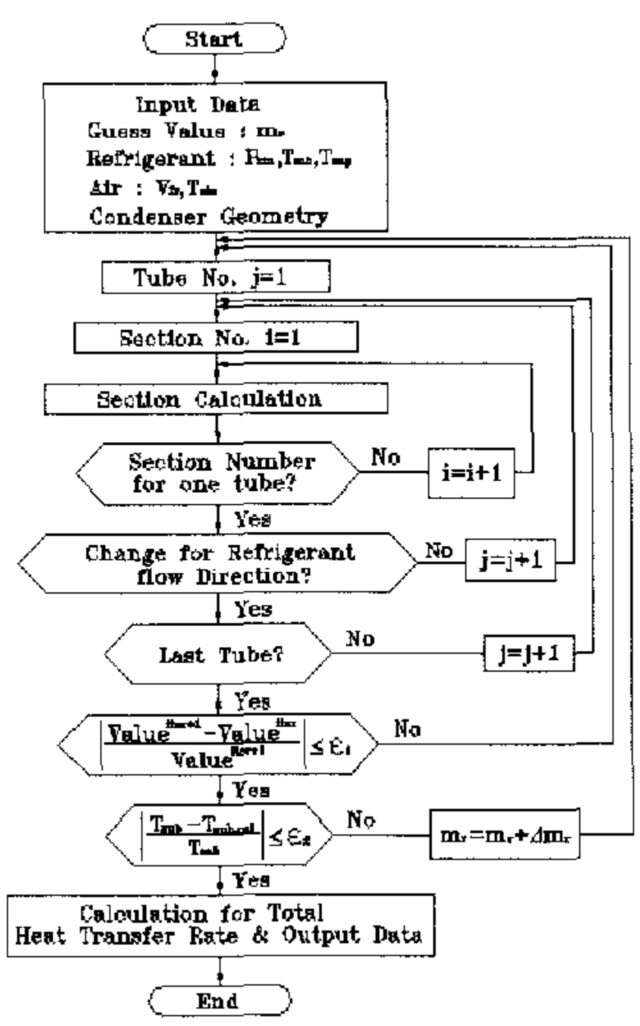


Fig 6. Flowchart of condenser simulation.

heat exchanger geometry, and thermo-physical properties of the tube and fin are The inlet conditions of the working fluids, needed as input data for the computer program. Internal changes to this program can be very easily made to suit a particular application. For example, to change heat transfer coefficient correlation only one subroutine requires rewriting. The program is also provided with subroutines to print and plot the results.

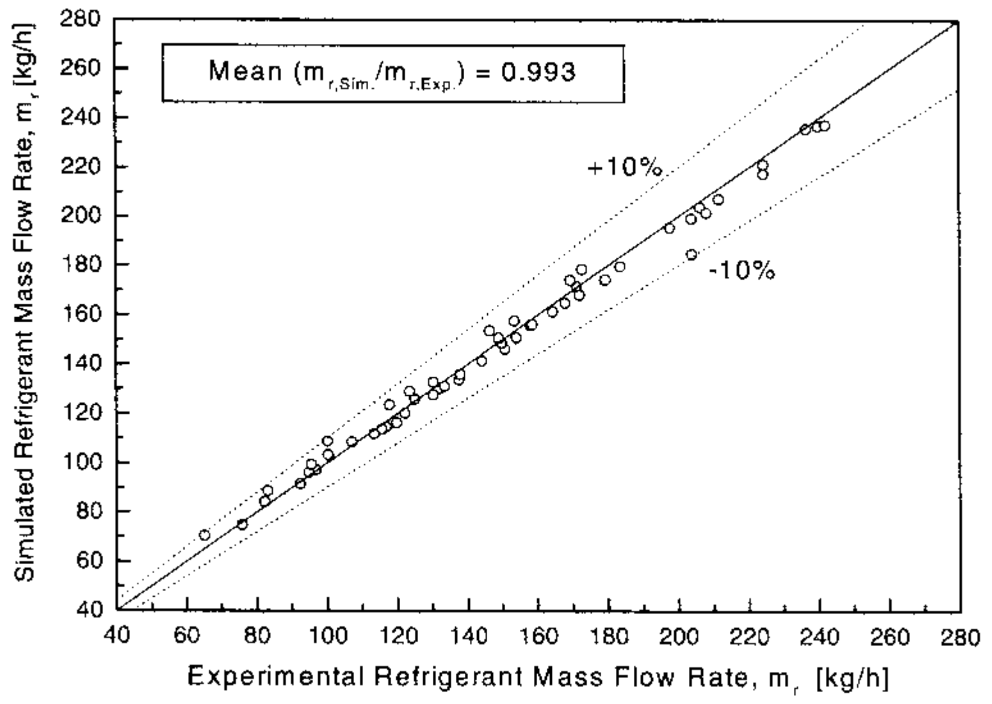


Fig 7. The comparison of the experimented and calculated refrigerant mass flow rate.

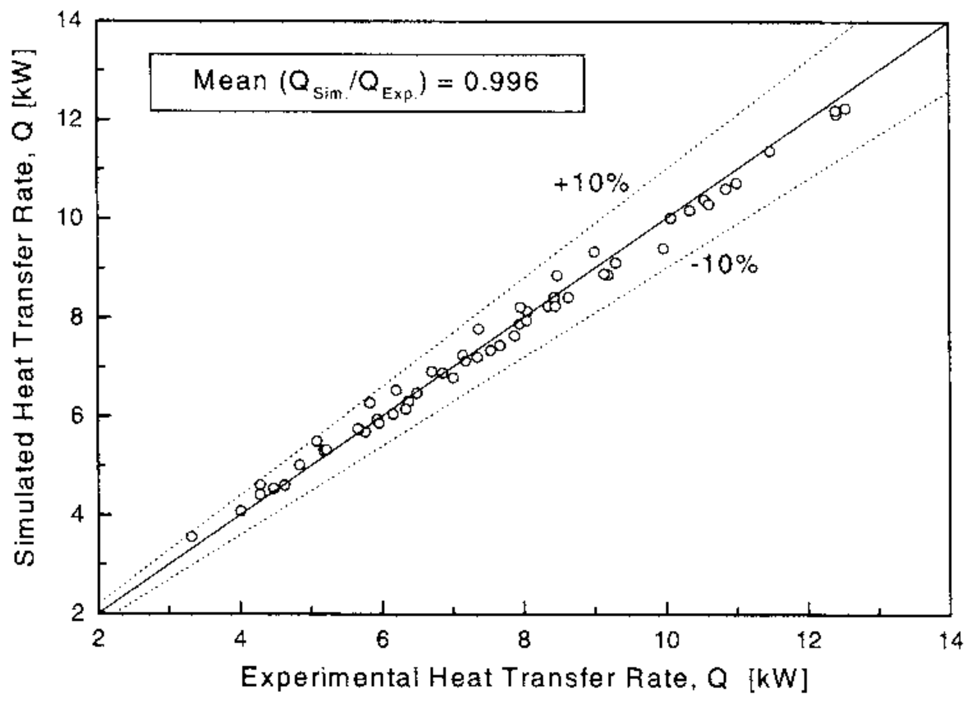


Fig 8. The comparison of the experimented and calculated heat transfer rate.

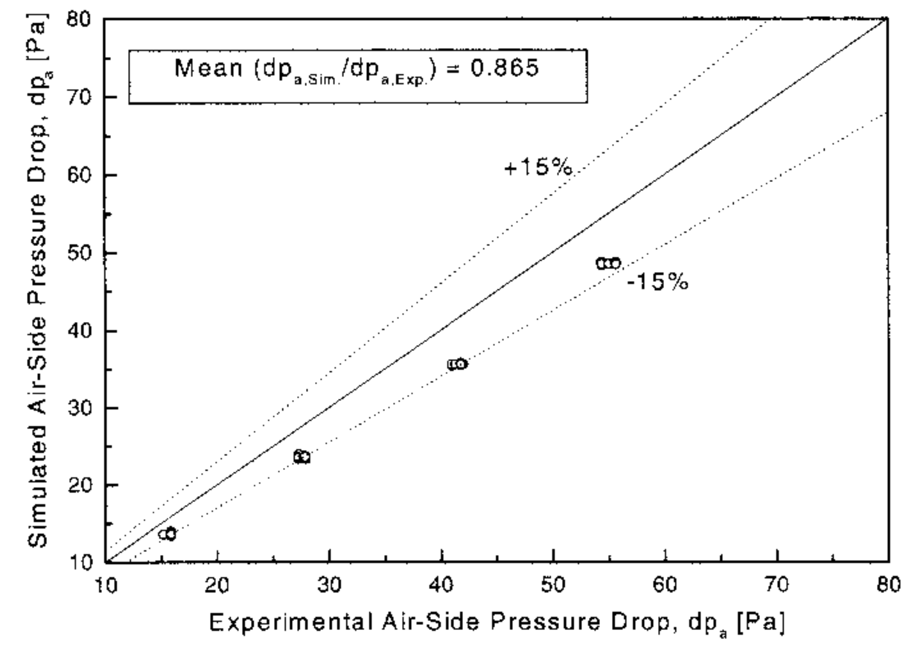


Fig 9. The comparison of the experimented and calculated air-side pressure drop.

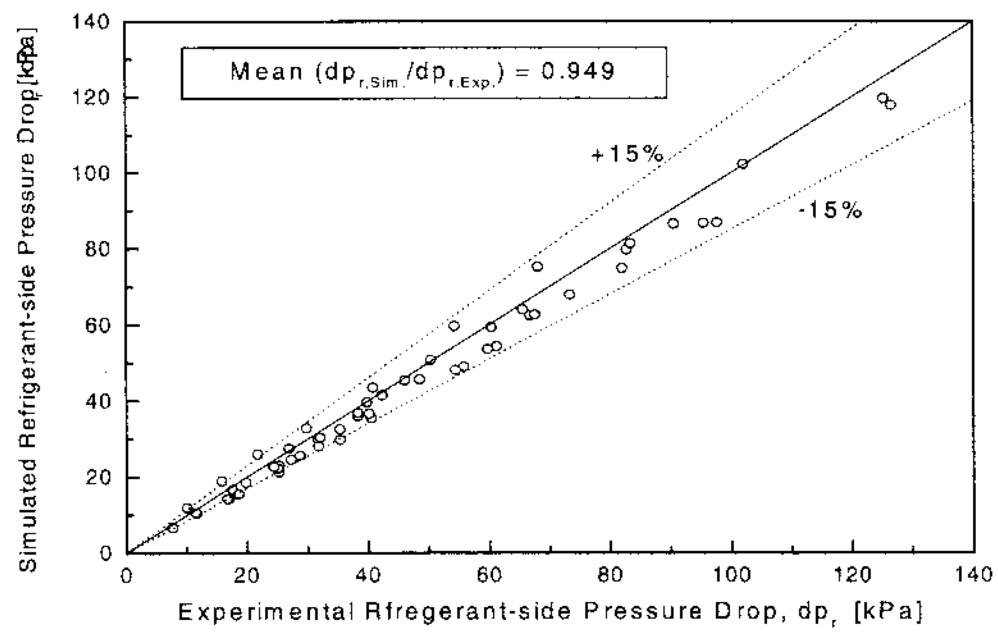


Fig 10. The comparison of the experimented and calculated refrigerant-side pressure drop.

4. Discussion

The model formulation presented above has been implemented in a computer program. The operating conditions are: input temperatures, air velocity, and design parameters(geometrical dimensions).

The results of the model have been compared with experimental data provided by the condenser manufacturer. Figure 7 to 10 show the comparison of model results with experimental data for the refrigerant mass flow rate, heat transfer rate, air-side pressure drop, and refrigerant-side pressure drop. The deviation between the results of the model results and experimental data for the refrigerant mass flow rate and the heat transfer rate is about 10%. For air-side pressure drop and a refrigerant-side pressure drop, it is about 15%. The agreement between model results and experimental data is good for the operating conditions.

5. Conclusions

A prediction model for the performance of automotive parallel flow condensers has been developed. The real geometry of the condensers has been modelled. Proper heat transfer correlations are used for calculating Nusselt number or heat transfer coefficient in air and refrigerant sides. Expressions for the Nusselt number that accounts for the phase change inside the condenser have been used successfully. Single-phase and two-phase frictional, local pressure drops and momentum exchanges correlations have been used properly. The model shows a good prediction capabilities in the rage of operating conditions covered by test procedures.

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