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NUMERICAL STUDY ON THE PERFORMANCE OF THE EVAPORATOR FOR CAR AIR-CONDITIONERS

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ABSTRACT

In this study, the method for predicting the performance of multi-pass type evaporators for automobile air conditioners is developed. The present study is based on the previous studies of the flow boiling heat transfer and two phase pressure drop in small diameter tubes. The heat transfer rate, distribution of air/refrigerant temperature, pressure, vapor quality and void fraction are calculated on the various conditions of evaporator's specification, refrigerant flow and air flow. The effect of various calculation conditions on the heat transfer rate and refrigerant pressure drop in evaporator are also examined. It is found that the evaporator size can be reduced keeping almost the same outlet air temperature when the optimum specification of evaporator is selected.

1. INTRODUCTION

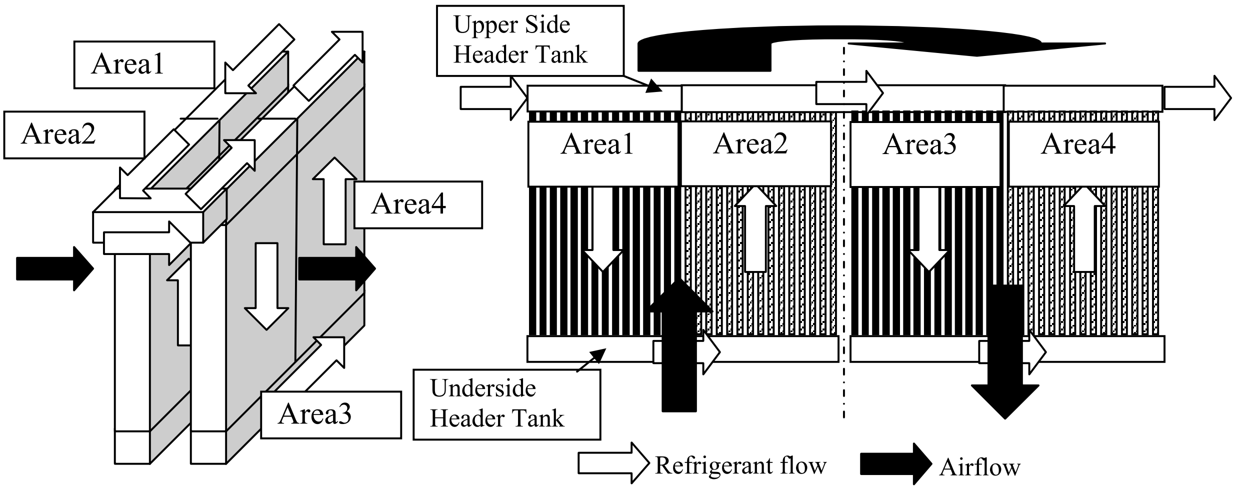
In recent years, from the viewpoint of the protection of the environment, it is highly expected to use natural refrigerants such as carbon dioxide (R744), iso-butane and so on. On the other hand, in order to miniaturize the heat exchangers, there are a lot of studies on heat transfer characteristics of capillary tubes and multi-port extruded tubes. Especially in the field of car air-conditioners, it is highly expected to miniaturize the heat exchangers, to enhance the performance and to use carbon dioxide. Therefore, in order to realize the high-pressure resist, compact and high performance heat exchanger, it is increasing to use mini multi-port extruded tubes as a heat transfer tube for the evaporators, though, plate type heat exchangers are used for car air-conditioners in general. In order to construct the high performance heat exchangers, it is necessary to develop not only the structure, but also the evaluation method of heat exchanger performance.

In this study, it is carried out to calculate the heat transfer rate, distribution of air/refrigerant temperature and pressure and so on based on the previous studies on capillary tubes, multi-port extruded tubes, flow boiling and two-phase pressure drop in smooth tubes. At that time, local heat transfer, pressure drop and air-side heat transfer are taken into account. In this calculation, number of turns (4 and 6), number of tubes (24 and 30: at that time, size of heat transfer area also changed simultaneously), air flow rate (4, 5, 6 and 7 m^3/min.) and refrigerant flow rate (90, 110, 130, 150 and 170 kg/h) are changed as parameters in the case of R134a, and number of tubes (24, 30), air flow rate (6, 7, 8 and 9 m^3/min.) and refrigerant flow rate (60, 80, 100 and 120 kg/h) are changed as parameters in the case of R744. From the calculation results, it is found for some cases, the heat transfer rate and outlet air temperature can be kept almost constant even though the number of tubes and refrigerant flow rate are decreased.

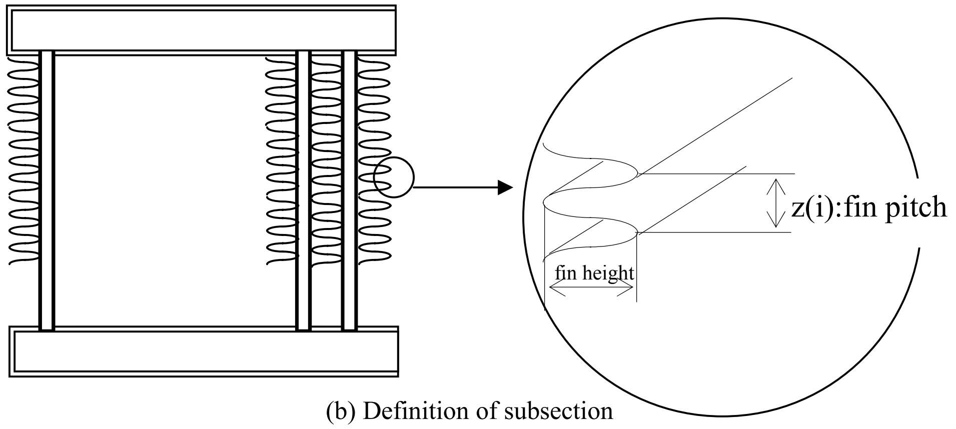
2. PREDICTION METHOD OF LOCAL HEAT TRANSFER

2.1 Calculation Model

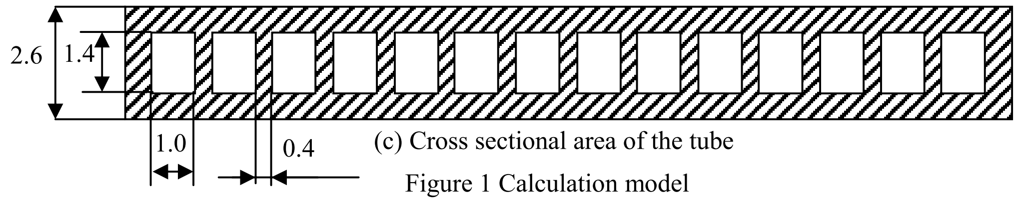
Figure 1(a) shows the calculation model of an evaporator in the case of four turns. Air flows horizontally and refrigerant flows vertically. Multi-port extruded tubes and corrugate fins are set alternately and vertically between the upper side and the underside header tank and arranged in two rows of front and back side for air-flow.



(a) Flow directions of refrigerant and air



(b) Definition of subsection



(c) Cross sectional area of the tube

Figure 1 Calculation model

Refrigerant flows into the upper side header tank of upstream side of air-flow (front-side), and flows out from the upper side header tank of downstream side of air-flow (rear-side). In the case of four turns, refrigerant flows into the upper side header tank of front-side, and flows downward in a half number of tubes of this row. Refrigerant is turned flow direction to upward in the underside header tank, and flows upward in the remaining half number of tubes of this row. After that, refrigerant flows into the upper side header tank of rear-side. Refrigerant then flows downward in a half number of tubes of this row. Refrigerant is turned flow direction to upward in the underside header tank, and flows upward in the remaining half number of tubes of this row and flows out from the upper side header tank of rear-side. In the case of six turns, refrigerant flows into the upper side header tank of front-side, and flows downward in one third numbers of tubes of this row. Refrigerant is turned flow direction to upward in the underside header tank, and flows upward in the center one third numbers of tubes of this row. After that, refrigerant is turned flow direction to downward in the upper side header tank, and flows downward in the other one third numbers of tubes of this row and flows into the underside header tank of rear-side. And, refrigerant flows upward in one third numbers of tubes of this row. Refrigerant is turned flow direction to downward in the upper side header tank, and flows downward in the center one third numbers of tubes of this row. After that, refrigerant is turned flow direction to upward in the underside header tank, and flows upward in the other one third numbers of tubes of this row and flows out from the upper side header tank of rear-side.

In case of calculation, each calculating subsection has the same length which is divided equally by air- side fin pitch into n pieces of multi-port extruded tube. The fin pitch *z*(i) (where, i is from 1 to n) is defined as the distance between the apexes of two consecutive fins, which is shown at the right side of Figure 1(b). The inlet conditions of air and refrigerant, which are air inlet temperature, air flow rate, refrigerant inlet pressure, inlet quality and refrigerant flow rate, are given. Heat transfer rate of subsection, air outlet temperature, refrigerant outlet pressure and quality are calculated by using given values in each subsection which starts from the first subsection and follow the order. Required thermo physical properties of refrigerants are calculated by REFPROP ver.7 (2002) and the properties of dry air are calculated using reference (Fujii, 1983). Assumptions are as follows.

1. inflow air is dry air
2. there is no temperature distribution in a perpendicular plane to air-flow direction
3. refrigerant flow rate of each pass is the same
4. inlet quality of each pass is the same
5. pressure drop and heat transfer in the header tank are ignored

Figure 1(c) shows the cross sectional area of the tube.

2.2 Correlations of Local Heat Transfer and Pressure Drop

In this study, correlations of refrigerant heat transfer coefficient and pressure drop were adopted based on following assumptions. In the horizontal minichannel, if hydraulic diameter is smaller than Laplace constant, the effect of surface tension becomes large, then, flow pattern approaches to symmetrical about the center axis of the channel. Therefore, horizontal two-phase flow pattern is similar to vertical two-phase flow pattern. The correlation of Kuwahara et al. (2004) is employed for calculating boiling heat transfer of R134a. This correlation is based on the data of a horizontal multi-port extruded tube which was obtained when a hydraulic diameter is smaller than Laplace constant. In the case of R744, correlation of boiling heat transfer is proposed based on the experimental data. These correlations are based on the data of a horizontal multi-port extruded tube which was obtained when a hydraulic diameter is smaller than Laplace constant. Therefore, it is applicable to this calculation condition. In the post dryout region, the empirical correlation, which is made based on the experimental data, is used in order to calculate heat transfer coefficient. Vapor quality at the dryout point, x_c is defined based on the experimental data. In the super heated vapor region, Gnielinski correlation is used. Heat transfer

Table 1 Correlations of heat transfer

Flow boiling heat transfer

R134a

$$\alpha_{\mathrm{R}} = \alpha_{\mathrm{tp}} = \alpha_{cv} + \alpha_{nb}$$

where

$$\alpha_{cv} = 0.023 Re_{tp}^{0.8} \cdot Pr_L^{0.4} \cdot \left(\frac{\lambda_L}{\mathrm{d}_i}\right)$$

$$Re_{tp} = \mathrm{F}^{1.25} \cdot \frac{G \cdot (1-x) \cdot d_i}{\mu_L}$$

$$F = 1 + 1.4\left(\frac{1}{X_{tt}}\right)^{0.88}$$

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \cdot \left(\frac{\rho_V}{\rho_L}\right)^{0.5} \cdot \left(\frac{\mu_L}{\mu_V}\right)^{0.1}$$

$$\alpha_{nb} = K^{0.745} \cdot \mathrm{S} \cdot \alpha_{pb}$$

$$\mathrm{K}^{0.745} = \frac{1}{1 + 0.875\eta + 0.518\eta^2 - 0.159\eta^3 + 0.7907\eta^4}$$

$$\eta = \frac{\alpha_{cv}}{\mathrm{S} \cdot \alpha_{pb}},\ \mathrm{S} = \frac{1}{\zeta} \cdot \left(1 - e^{-\zeta}\right),\ \zeta = \frac{\mathrm{D}_b \cdot \alpha_{cv}}{\lambda_L}$$

$$D_b = C_1 \cdot \left(\frac{\rho_L}{\rho_V} \cdot \frac{Cp_L}{h_{fg}} \cdot T_{sat}\right)^{1.25} \cdot \left(\frac{2\sigma}{g \cdot (\rho_L - \rho_V)}\right)^{0.5}$$

$$\alpha_{pb} = C_2 \cdot 207 \frac{\lambda_L}{D_{be}} \cdot \left(\frac{q \cdot D_{be}}{\lambda_L \cdot T_{sat}}\right)^{0.745} \cdot \left(\frac{\rho_V}{\rho_L}\right)^{0.581} \cdot \mathrm{Pr}_L^{0.533} \cdot F_r$$

$$F_r = \left(8R_p\right)^{(0.2-0.2P_{red})},\ P_{red} = \frac{P}{P_c},$$

$$D_{be} = 0.51\sqrt{\frac{2\sigma}{g \cdot (\rho_L - \rho_V)}}$$

$C_1=5.0\times10^{-5}$, $C_2=1.25$, $\mathrm{R}_p=1.0$

R744

$$\left(\frac{\alpha_{\mathrm{R}}}{\alpha_{l0}}\right) = 2.1\left(Bo \times 10^4\right)^{0.85} + 0.45\left(\frac{1}{X_{tt}}\right)^{1.1}$$

where

$$\alpha_{l0} = 0.023 Re_L^{0.8} \cdot Pr_L^{0.4} \cdot \left(\frac{\lambda_L}{\mathrm{d}_i}\right)$$

$$Re_L = \frac{\mathrm{G} \cdot (1-\mathrm{x}) \cdot \mathrm{d}_i}{\mu_{\mathrm{L}}},\ Bo = \frac{\mathrm{q}}{\mathrm{G} \cdot \mathrm{h}_{fg}}$$

Post dryout region

$$\alpha_R = \alpha_V \left[1 + \frac{0.08(1-x)\{(\alpha_{tp}/\alpha_V)_{x=x_c} - 1\}}{(x - x_c) + 0.08(1-x)}\right]$$

where, <u>R134a :</u> $x_c = 0.9$

<u>R744 :</u> $x_c = 0.8$

coefficient of air is calculated by an empirical correlation. The correlation of Koyama et al. (2003) is employed for calculating frictional pressure drop. This correlation which is proposed for horizontal multi-port extruded tubes is also applicable, by the same reason as in the case of the correlation of Kuwahara et al. Homogeneous model is employed for acceleration and potential loss. These correlations are shown in Table 1 and Table 2.

2.3 Prediction Calculation Conditions

The effects of the evaporator's specification, refrigerant flow rate and air flow rate on the heat transfer characteristics are examined by the prediction calculation. The calculation conditions are shown in Table 3 and Table 4.

3. RESULTS AND DISCUSSION

3.1 Pressure Drop

Figure 2 shows the relation between air-flow rate, V_{air}, and refrigerant pressure drop ΔP in the case of R134a. Open symbols and half-closed symbols denote the case of 4 turns in the case of 24 tubes per row. Open symbols denote the same conditions of half-closed symbols except the flow rate is 20 kg/h lower than that of half-closed symbols. Closed symbols denote the case of 6 turns in the case of 24 tubes per row. Double open open symbols denote the case of 4 turns in the case of 30 tubes per row. Inverse triangles, circles, triangles, squares and diamonds denote the results of refrigerant flow rate W_R=90, 110, 130, 150 and 170 kg/h, respectively. Comparing half-closed symbols with closed symbols, it is found that pressure drop is extremely large when the number of turns is increased from four to six, in the case of the same refrigerant flow rate and the same number of tubes per row. It is also found that pressure drop, in the case of open symbols, is smaller than the case of half-closed

Table 2 Correlations of pressure drop

Frictional pressure drop

$$dP_f = \phi_V{}^2 \left(\frac{dP}{dz}\right)_{Vo} dz$$

$$\phi_V{}^2 = 1 + CX_{tt} + X_{tt}{}^2$$

$$C = 13.17\left(\frac{\nu_L}{\nu_V}\right)^{0.171}\left(1 - e^{-0.6\sqrt{Bo}}\right)$$

$$Bo = \frac{d^2 g(\rho_L - \rho_V)}{\sigma}, \quad \left(\frac{dP}{dz}\right)_{Vo} = \frac{2 f_V G^2 x^2}{\rho_V d}$$

Table 3 Calculation condition (R134a)

Evaporator	Number of tubes per row		24, 30
	Number of turns		4, 6
Air side	V_{air}	[m^3/min]	4,5,6,7
	T_{airin}	[°C]	25.0
Refrigerant side	P_{Rin}	[MPa]	0.2
	x_{in}	[-]	0.375
	W_R	[kg/h]	90 - 170

Table 4 Calculation condition (R744)

Evaporator	Number of tubes per row		24, 30
	Number of turns		4
Air side	V_{air}	[m^3/min]	6, 7, 8, 9
	T_{airin}	[°C]	25.0
Refrigerant side	P_{Rin}	[MPa]	2.5
	x_{in}	[-]	0.3
	W_R	[kg/h]	60 - 120

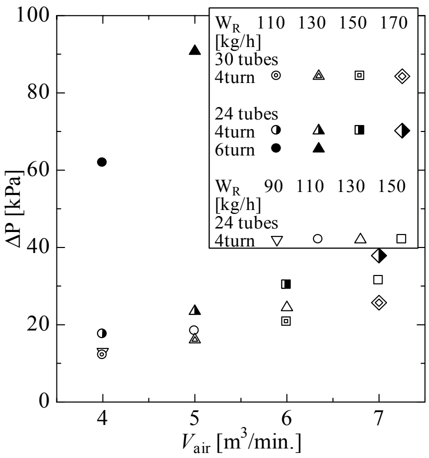


Figure 2 Relations between V_{air} and ΔP (R134a)

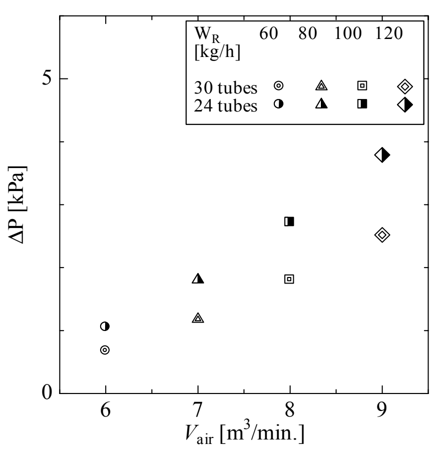


Figure 3 Relations between V_{air} and ΔP (R744)

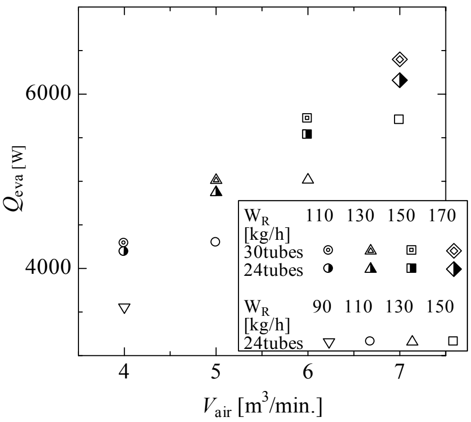


Figure 4 Relations between V_{air} and Q_{eva} (R134a)

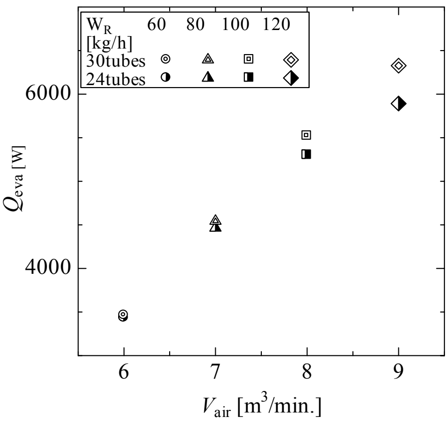


Figure 5 Relations between V_{air} and Q_{eva} (R744)

symbols and almost the same as the case of double open symbols. From these results, it is found that increasing number of turns above 6 makes a large pressure drop, therefore it does not directly affect to improve the performance of systems in the case of using R134a.

Figure 3 shows the relation between air-flow rate V_{air} and refrigerant pressure drop ΔP in the case of R744. Half-closed symbols denote the case of 4 turns in the case of 24 tubes per row. Double open symbols denote the case of 4 turns in the case of 30 tubes per row. Circles, triangles, squares and diamonds denote the results of refrigerant flow rate W_R=60, 80, 100 and 120 kg/h, respectively. In the case of the same refrigerant flow rate, it is found that pressure drop is large when the number of tubes is decreased from 30 to 24, however the value of pressure drop is very small. From these results, it is found that pressure drop does not become a large problem for R744.

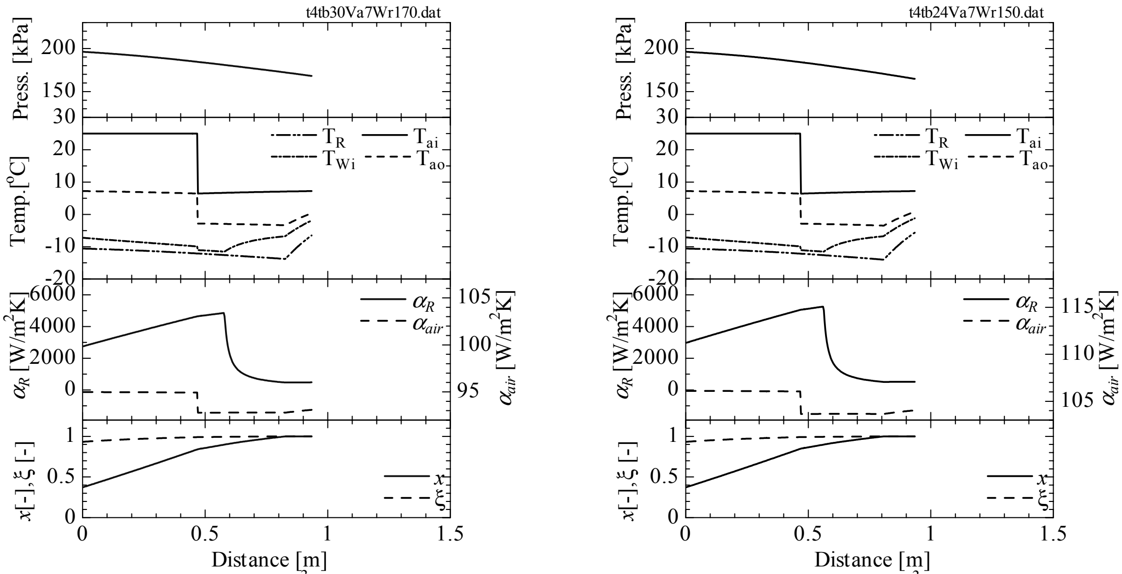
3.2 Heat Transfer Rate

Figure 4 shows the relation between V_{air} and heat transfer rate of evaporator, Q_{eva}, in the case of R134a. Symbols in Figure 4 denote the same as that of in Figure 2. From these results, in the case of the same refrigerant flow rate, it is found that Q_{eva} is almost kept constant when the number of tubes is decreased from 30 to 24. Therefore, the evaporator can be miniaturized from the viewpoint of heat transfer rate. However, in this case, pressure drop is about 5 kPa large as shown in Figure 2. In the case of 20 kg/h lower refrigerant flow rate, Q_{eva} is lower than that of others.

Figure 5 shows the relation between V_{air} and Q_{eva} in the case of R744. Symbols in Figure 5 denote the same as that of in Figure 3. In the case of R744, it is also found that Q_{eva} is almost kept constant when the number of tubes is decreased from 30 to 24. Therefore, the evaporator can be miniaturized from the viewpoint of heat transfer rate. Comparing Figure 5 with Figure 4, if the inlet vapor quality of R744 is around 0.3, there are some conditions that heat transfer rate of R744 becomes almost the same as that of R134a.

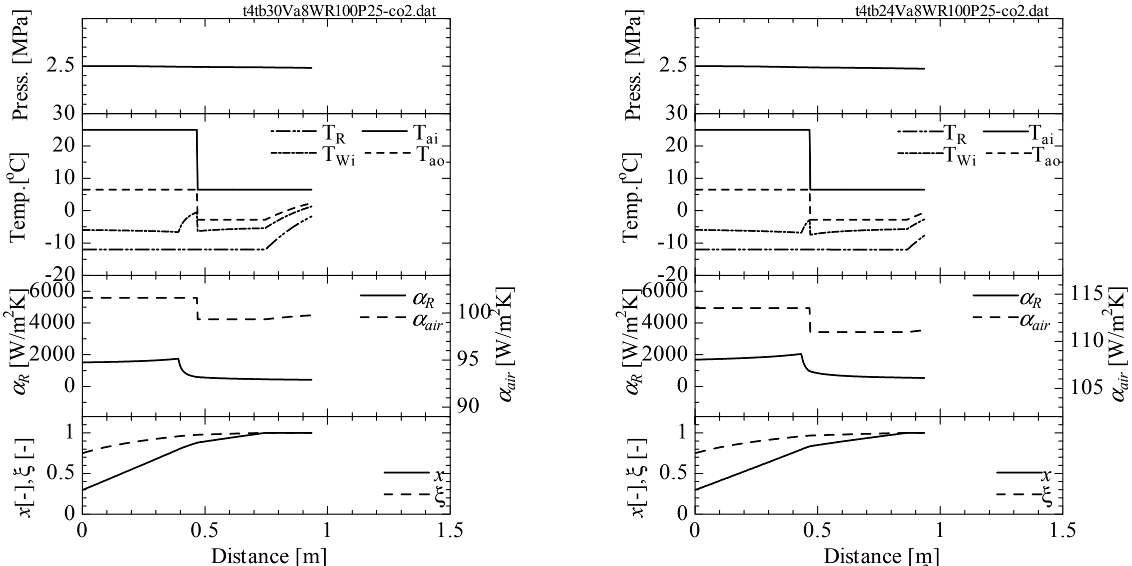
3.3 Variations of Pressure, Temperature, Heat Transfer Coefficient, Vapor Quality and Void Fraction in Evaporator

Figure 6 (a) and (b) show variations of refrigerant pressure, refrigerant temperature, T_R, air temperature at the inlet of each row, T_{ai}, air temperature at the outlet of each row, T_{ao}, inner wall temperature of the tube, T_{Wi}, refrigerant-side heat transfer coefficient, α_R, air-side heat transfer coefficient, α_{air}, vapor quality, x and void fraction, ξ, along the distance from the inlet of the evaporator, in the case of R134a. T_{ai} and T_{ao} change stepwise at the position of about 0.47 m, it is caused that the row of evaporator changes at this position from front-side to rear-side, and air temperature at the outlet of front-side row becomes that at the inlet of rear-side row. In the case of 30 tubes per row, V_{air}=7 m^3/min. and W_R=170 kg/h as shown in Figure 6 (a), pressure drop is about 25 kPa, air temperature at the outlet of front-side row is about 7 °C, and that at the outlet of rear-side row is about -3 °C. Refrigerant flow becomes superheated vapor single-phase flow around the position of 0.85 m, after that air temperature increase until around 0 °C. In the case of 24 tubes per row, V_{air}=7 m^3/min. and W_R=150 kg/h as shown in Figure 6 (b), pressure drop is about 31 kPa, air temperature at the outlet of front-side row is from about 6 to 7 °C, and that at the outlet of rear-side row is about -4 °C. Refrigerant flow becomes superheated vapor single-phase flow around the position of 0.8 m, after that air temperature increase until around 0 °C. From these figures, it is found that there is no large difference between the two for all parameters. As shown in Figure 4, heat transfer rate of evaporator in the case of Figure 6 (b) is about 700 W lower than the case of Figure 6(a), however, air temperature at the outlet of rear-side row is almost the same. Therefore, the evaporator can be miniaturized from the viewpoint of outlet air temperature. Figure 7 (a) and (b) show variations of refrigerant pressure, refrigerant temperature, T_R, air temperature at the inlet of each row, T_{ai}, air temperature at the outlet of each row, T_{ao}, inner wall temperature of the tube, T_{Wi}, refrigerant-side heat transfer coefficient, α_R, air-side heat transfer coefficient, α_{air}, vapor quality, x and void fraction, ξ, along the distance from the inlet of the evaporator, in the case of R744. In the case of 30 tubes per row, V_{air}=8 m^3/min. and W_R=100 kg/h as shown in Figure 7 (a), pressure drop is about 1.8 kPa, air temperature at the outlet of front-side row is about 6.5 $°C$, and that at the outlet of rear-side row is about -3 $°C$. Refrigerant flow becomes superheated vapor single-phase flow around the position of 0.75 m, after that air temperature increase until around 2 $°C$. In the case of 24 tubes per row, V_{air}=8 m^3/min. and W_R=100 kg/h as shown in Figure 7 (b), pressure drop is



(a) 30tubes, 4turns, V_{air}=7m^3/min, W_R=170kg/h (b) 24tubes, 4turns, V_{air}=7m^3/min, W_R=150kg/h

Fig.6 Variation of Pressure, Temperature, Heat Transfer Coefficient, quality and Void Fraction along Refrigerant Flow Direction



(a) 30tubes, 4turns, V_{air}=8m^3/min, W_R=100kg/h (b) 24tubes, 4turns, V_{air}=8m^3/min, W_R=100kg/h

Fig.7 Variation of Pressure, Temperature, Heat Transfer Coefficient, quality and Void Fraction along Refrigerant Flow Direction about 2.7 kPa, air temperature at the outlet of front-side row is from about 6.5 $°C$, and that at the outlet of rear-side row is about -3 $°C$. Refrigerant flow becomes superheated vapor single-phase flow around the position of 0.85 m, after that air temperature increase until around 0 $°C$. From these figures, it is found that there is no large difference between the two for all parameters. As shown in Figure 5, heat transfer rate of evaporator in the case of Figure 7 (b) is almost the same as the case of Figure 7 (a), and air temperature at the outlet of rear-side row is also almost the same. Therefore, the evaporator can be miniaturized from the viewpoint of outlet air temperature.

4. CONCLUSIONS

In this study, it is carried out to calculate the various design parameters about the performance of evaporator, and the main conclusions are,

- Pressure drop is extremely large when the number of turns is increased from four to six, in the case of the same refrigerant flow rate and the same number of tubes per row.
- Increasing the number of turns above 6 makes a large pressure drop, therefore it does not directly affect to heighten the performance of systems in the case of using R134a.
- In the case of using R744, it is found that pressure drop does not become a large problem.
- Heat transfer rate is almost kept constant when the number of tubes is decreased from 30 to 24. Therefore, the evaporator can be miniaturized from the viewpoint of heat transfer rate. However, in this case, pressure drop become large for R134a.
- Heat transfer rate is almost kept constant when the number of tubes is decreased from 30 to 24. Therefore, the evaporator can be miniaturized from the viewpoint of heat transfer rate.
- If the inlet vapor quality of R744 is around 0.3, there are some conditions that heat transfer rate of R744 becomes almost the same as that of R134a.
- Decreasing evaporator size and refrigerant flow rate, there are some conditions that air temperature at the outlet of evaporator is almost kept constant. Therefore, the evaporator can be miniaturized from the viewpoint of outlet air temperature for R134a.

NOMENCLATURE

Bo	Bond number	(–)
C	correction factor	(–)
C_1	constant	(–)
C_2	constant	(–)
C_p	isobaric specific heat	(J/kg K)
d	diameter	(m)
f	friction factor	(–)
G	mass velocity	(kg/m^2 s)
g	gravitational acceleration	(m/s^2)
h_{fg}	latent heat	(J/kg)
P	pressure	(Pa or kPa or MPa)
Pr	Prandtl number	(–)
Q	heat transfer rate	(W)
q	heat flux	(W/m^2)
Re	Reynolds number	(–)
R_p	Roughness factor	(–)
T	temperature	($°C$)
V	volumetric flow rate	(m^3/min.)
W	mass flow rate	(kg/h)
x	vapor quality	(–)
z	subsection length	(m)
α	heat transfer coefficient	(W/m^2 K)
ΔP	pressure drop	(kPa)
σ	surface tension	(N/m)

Subscripts

ai	air at the inlet
air	air
ao	air at the outlet
c	critical
cv	convection
eva	evaporator
i	inner
in	inlet
L	liquid
nb	nucleate boiling
R	refrigerant
red	reduced
pb	pool boiling
tp	two-phase
V	vapor
W	wall

λ	thermal conductivity	(W/m K)
μ	viscosity	(Pa s)
ν	kinematic viscosity	(m^2/s)
ρ	density	(kg/m^3)
ξ	void fraction	(–)
X_{tt}	Lockhart-Martinelli's parameter (–)	
ϕ_V	Lockhart-Martinelli's parameter (–)	

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