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ABSTRACT

In this paper, characteristics of the evaporator and the capillary tube using R-410A, one of the most promising alternatives to R-22, are measured and realistic models are formulated. In addition a simulation based on these models has been carried out to investigate the system performance with R-410A. Finally the cycle characteristics of both refrigerants are compared. The SEER of systems using R-410A and R-22 can be recognized as almost the same.

INTRODUCTION

CFCs was phased out at the end of 1995 and the regulation of HCFCs starts in 1996. HFC blends, such as R-410A(R-32(50wt%)/R-125(50wt%)) and R-407C(R-32(23wt%)/R-125(25wt%)/R-134a(52wt%)), are regarded as promising candidates to replace R-22, now frequently used as refrigerant in air-conditioners. In particular, R-410A seems to be an interesting alternative to R-22, due to its energy efficiency, but the COP of the former is 5-10% lower than that of the latter at several working pressure levels in a theoretical cycle. (Each compressor efficiency is assumed to be constant.) In Table 1 one example of the COP of R-22 and R-410A, respectively, is shown. On the other hand previous works[8] suggest an enhancement of the performance of R410A in real systems. Therefore it is very important to estimate system performances under practical conditions. Big differences in working pressure between R-410A and R-22 are the points of the estimation. Systems using R-410A have to be operated under high pressure conditions being advantageous as well as disadvantageous in comparison to R-22. High working pressure causes a higher density and therefore low pressure losses in heat exchangers, particularly, in evaporators, but the difference in working pressure makes it (almost) impossible to replace the one refrigerant by the other without redesigning the capillary tube.

This paper describes experimental results of the pressure loss characteristics in the evaporator and the capillary tube characteristics with R-410A for practical systems and the system performance of an air-conditioner in a simulation based on these measurements.

EXPERIMENTS

Evaluation of Pressure Loss in Grooved Evaporators with R-410A

Local pressure loss through the evaporator is investigated to construct a precise model of the evaporator. The experimental setup used in this measurement is shown in Fig. 1. The test section is heated by water to control the evaporating temperature. Table 2 indicates specifications for the evaporator in Fig. 2. The relation between local pressure loss (dP/dZ) and quality x at the evaporating temperature 283.15 K is plotted for the mass flow fluxes G_f=150kg/m^{2}s, 250kg/m^{2}s and 400kg/m^{2}s. The inlet quality x_{in} is fixed to 0.2 and the outlet x_{out} to 1.0 in each measurement. A general formulation to calculate local pressure loss for stratified two-phase flow in a smooth tube has already been proposed and it was estimated using as enhancement factor ϕ_l [1]. The ratio of the pressure loss between a two-phase flow and a liquid ϕ_l can be expressed as function of a parameter Xtt as follows:

$$\phi_l^2 = f(X_{tt}) \qquad (1)$$

where Xtt is defined according to Lockhalt-Martinelli

$$X_{tt}^2 = \left(\frac{1-x}{x}\right)^{1.8} \frac{\rho_g}{\rho_l} \left(\frac{\mu_l}{\mu_g}\right)^{0.2} \qquad (2)$$

where ρ_g, ρ_l denotes the vapor and liquid density, respectively, μ_g, μ_l the vapor and liquid viscosity, respectively.

For the application of this approach to the flow in the grooved tube, the relation between ϕ_l and Xtt is shown in Fig. 3. From this result, the correlation between ϕ_l and Xtt is found to be

$$\phi_l = 4.55 - 0.187 X_{tt} \tag{3}$$

The local pressure loss $(dP/dZ)_{TP}$ for two-phase flow in the grooved tube can be calculated by using ϕ_l, given by the correlation equation (3), and friction factor C_{fl} for rough surfaces:

$$\frac{1}{\sqrt{C_{fl}}} = 2.0 \cdot \log_{10}\left(\frac{D}{\varepsilon}\right) + 1.08 \tag{4}$$

where u_l is the liquid velocity in case of sole liquid flow in the tube, D the inner diameter of grooved tube and ε the groove depth, as follows:

$$\begin{aligned} -\left(\frac{dP}{dZ}\right)_{TP} &= -\phi_l^2 \left(\frac{dP}{dZ}\right)_l \\ &= \phi_l^2 C_{fl} \frac{1}{2D} \rho_l u_l^2 \end{aligned} \tag{5}$$

Evaluation of Capillary Tube Characteristics with R-410A

Although there is a theory, based on an one-dimensional adiabatic compressible flow known as Fanno flow, for calculation of a capillary tube used as an expansion device, it is inappropriate in case of capillary tubes with small diameters, because for Fanno flows thermodynamic equilibrium is assumed, but in the real small capillary tube for split type room air-conditioners[3] a delay of evaporation is often observed. This non-equilibrium phenomenon is clearly visible in the measurements shown in Fig. 4. The capillary tube length calculated by Fanno flow doesn't agree with the real length. It depends on the thermodynamic non-equilibrium region. It is very difficult to predict the behavior of two-phase flow in the tube and calculate the state of fluid along the tube from the inlet of the capillary tube to the outlet step by step. Therefore, it is important to find a correlation between critical mass flow rate Gr and other parameters, as condensation pressure P_{cond}, degree of subcooling T_{sc}, tube length L_{cap}, tube diameter D_{cap}, to select the capillary tube for R-410A. The experimental equipment (cf. Fig. 5) was used to test configurations with inner diameters D_{cap} of 1.2, 1.3 and 1.4 mm and the length L_{cap} of 0.8, 1.2 and 1.6m.

The relation between the mass flow rate Gr through the capillary tubes and the degree of subcooling T_{sc} (cf. Fig. 6) reveals a proportional increase of Gr with T_{sc}, but also depends on further parameters. This exact measurement can be correlated with other parameters like D_{cap}, L_{cap}, P_{cond}, via the following equation :

$$Gr = 6280 \times D_{cap}^{\ 1.93} \times L_{cap}^{\ -0.404} \times \left(P_{cond}/P_{crit}\right)^{0.630} \times T_{sc}^{\ 0.112} \tag{6}$$

P_{crit} stands for the critical pressure in this equation. The temperature and pressure range was chose to cover $0\ K < T_{sc} < 10\ K$ and $2700\ kPa < P_{cond} < 3400\ kPa$, respectively. Experimental and data predicted using this correlation agree within the accuracy of $\pm 5\%$.

SIMULATION

System Simulation of an Air-conditioner using R-410A

A sketch of the experimental setup as it was used in the simulation is shown in Fig. 7. This model of the practical system consists of a rotary compressor, two conventional heat exchangers, a capillary tube, an accumulator and some connecting tubes. The compressor efficiency greatly influences the system COP, but adding the real compressor efficiency to the COP means to estimate the compressor characteristics rather than refrigerant properties. Therefore, a theoretical compressor model based on thermodynamic properties is used for this cycle simulation to evaluate exclusively the effect of refrigerant properties. Mechanical losses in the compressor are assumed to be 80%. As for other components, the above results from measurements are applied in a practical cycle. Many other workers examined the heat transfer in the grooved tube [4][6][9]. In addition, R-410A is a near-zeotropic refrigerant and it is possible to apply some the present ideas to predict the characteristics of R-410A. In this study, the equation proposed by Fujii [4] is applied for the condenser and the equation by Yoshida [6] for the evaporator. In order to investigate the heat transfer more detailed, the two heat exchangers are divided into small sections. The solution methods follows closely the algorithms suggested in [2][5]: Some initial conditions are specified and static heat balances of the refrigeration cycle are iteratively calculated until the results converge to closed graphs in the p-h diagram.

In a model including the pressure loss in heat exchangers whose specification is fixed, the COP under standard test conditions (cf. Table 3) of R-410A is almost equal to that of R-22. That is the reason for the pressure loss of R-410A in the evaporator being about 30% smaller than that of R-22. The results of a parametric study on the influence of outdoor temperature on the COP are shown in Fig. 8. For an air temperature of 320 K at the condenser under overload conditions, for example the COP of R-410A turns out to be about 4% lower than the value of R-22 even though the low pressure loss. The COP of the system using R-410A is lower than R-22 because of a working pressure close to the critical pressure. Furthermore, the discharge temperature of the compressor becomes very high. Therefore a system using R-410A might be inadequate where high temperatures are required. On the contrary, the COP of R-410A becomes higher than the value of R-22 under underload conditions

The results of a simulation for a connecting tube with an inner diameter of 8.12 mm and a system working under standard test conditions are shown in Fig. 9. The longer the connecting tube on the suction side is, the better becomes the ratio of the system COP in comparison to R-22. Therefore R-410A is appropriate to systems with long connecting tubes.

Seasonal Energy Efficiency Ratio

In addition to the above evaluation of the system performance, power consumption, depending on the performance of a working air-conditioning system has to be take into account. SEER (Seasonal Energy Efficiency Ratio) is a very simple evaluation method, but can be regarded as one of the more appropriate methods to evaluate the power consumption of the system. Using this approach it is possible to take it into account more, compared to standard conditions, practical operating situations. The SEER ratio between R-410A and R-22, based on a for the U. S. typical CLH (Cooling Load Hours) value and obtained with a Cd value for cyclic operations fixed to 0.25, turns out to be about 1.03, because of high COP of R-410A under underload conditions.

CONCLUSIONS

In this study, new expressions for the characteristics of important components are proposed by some exact experiments for R-410A. From the comparison of the cycle performance between of R-22 and R-410A under various conditions the following conclusions can be drawn: A system using R-410A has different characteristics from one using R-22 under practical conditions. The COP of systems using R-410A under overload conditions becomes worse than one using R-22, but it becomes better under underload conditions. The SEER of practical systems using R-410A and R-22 can be recognized as almost the same.

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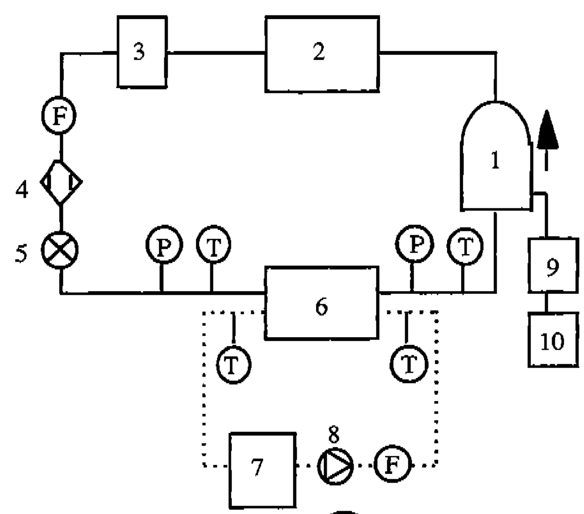
Table 1. Comparison of the theoretical COP between R-22 and R-410A
(Tc = 318.15 K, Te = 283.15 K, Tsc = 5 K, Tsh = 0 K)

	R-22	R-410A	R-410A/R-22
COP	5.5	5.1	0.93

(All thermodynamic properties of R-410A and R-22 by REFPROP version 4 [7])

Table 2. Specifications of the tested evaporator with grooves

Outer diameter (mm)	7.00	Groove depth (mm)	0.18
Inner diameter (mm)	6.50	Number of grooves	50
Wall thickness (mm)	0.25	Lead angle (degree)	18



1. Compressor
2. Condensor
3. Subcooler
4. Drier
5. Expansion valve
6. Evaporator
7. Heater
8. Circulating pump
9. Power meter
10. Inverter

(P) Pressure gauge
(T) Thermo couple
(F) Mass flow meter

Figure 1. Experimental apparatus for evaporators

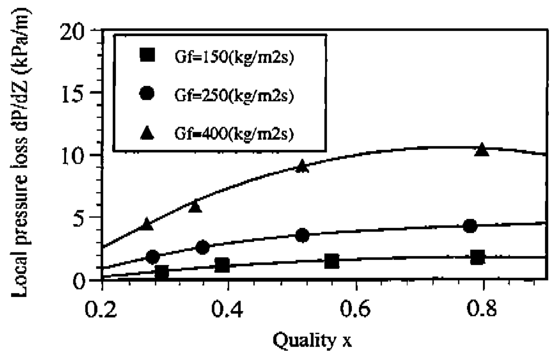


Figure 2. Relation between the local pressure loss and the quality in the evaporator

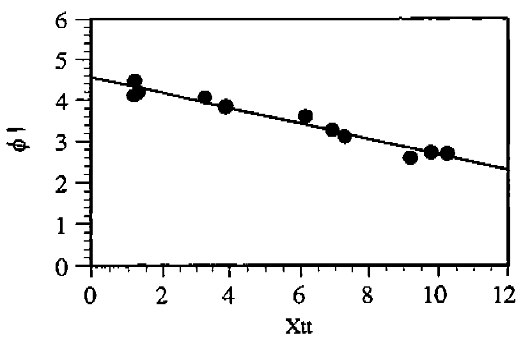


Figure 3. Relation between ϕ_l and Xtt in the evaporator

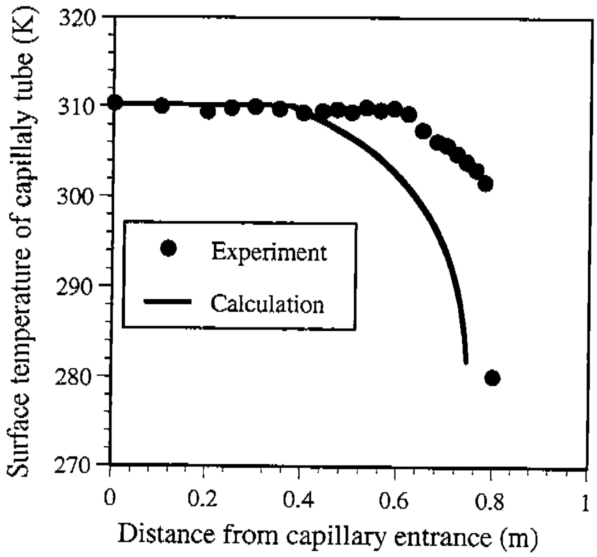


Figure 4. Comparison of the temperature distribution between the experimental result and calculation

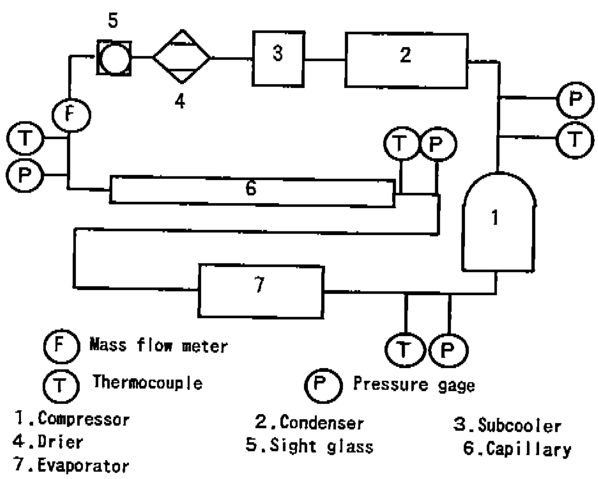


Figure 5. Experimental apparatus for capillary tubes

Table 3. Standard test conditions

Air temp. of indoor	**DB (K)**	**300.15**
	WB (K)	**292.15**
Air temp. of outdoor	**DB (K)**	**308.15**
	WB (K)	**297.15**

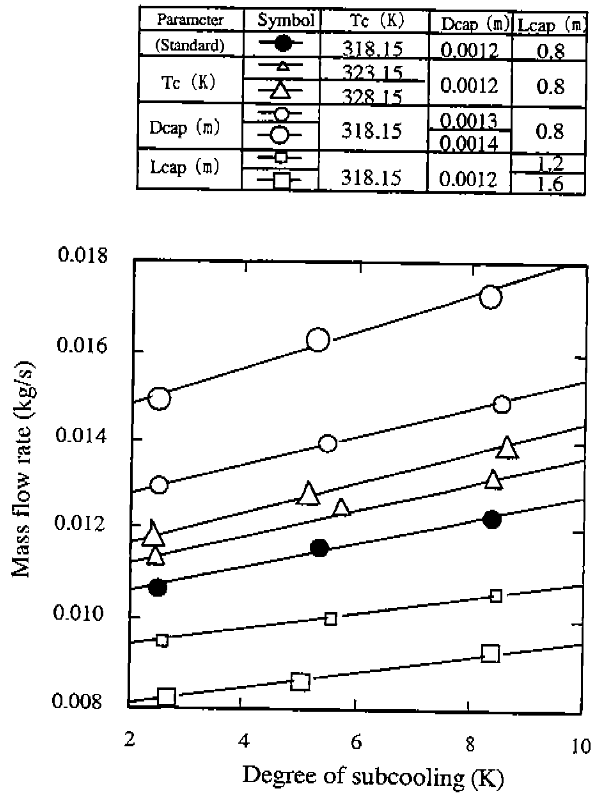


Figure 6. Relation between mass flow rate through the capillary tube and degree of subcooling

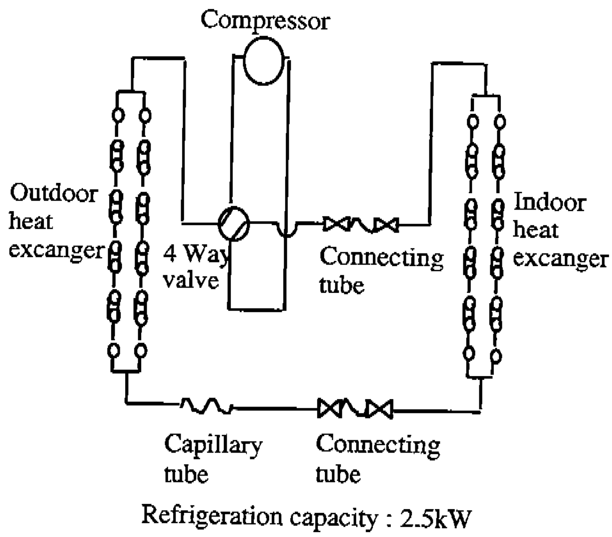


Figure 7. Schematic view of the system

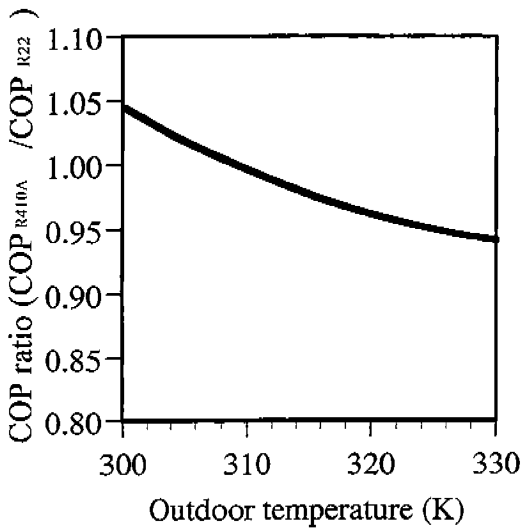


Figure 8. Influence of the outdoor temperature on the COP

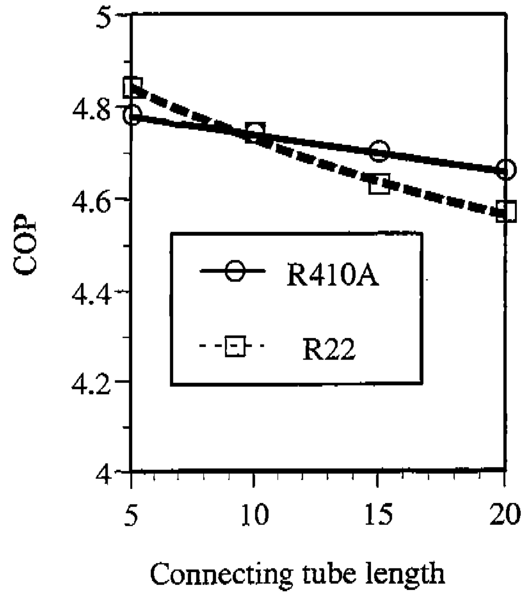


Figure 9. Influence of the length of the connecting tube on the COP