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International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

1994

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Performance of Refrigeration Cycle with
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Based on the Reliable Thermodynamic Property Data

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ABSTRACT

A series of test results on refrigeration-cycle-performance is summarized for three different promising HFC refrigerant mixtures which are being considered promising to replace R-22, i.e., R-32/134a (30/70 wt%), R-32/134a (25/75 wt%), and R-32/125 (50/50 wt%) systems. The mass-flow-rate of refrigerants was directly measured. In order to obtain the reliable results on the cycle performance, it is essential to assign the appropriate testing conditions especially in case of employing the refrigerant mixtures. To reach this goal, the reliable thermodynamic property values derived from the virial equations in the vapor phase, the Peng-Robinson equation for the VLE properties and the modified Hankinson-Brobst-Thomson equation for the saturated-liquid densities have been applied as the basis of the present study. The observed cycle-performance is compared with that by R-22. This paper also discusses the characteristics of cycle-performance for each binary refrigerant.

INTRODUCTION

Concerning the cycle-performance of alternative refrigerants to replace R-22, the R-22 Alternative Refrigerants Evaluation Program(AREP) established by the Air-conditioning and Refrigeration Institute (ARI) in USA has reported their results (1, 2) on different alternatives including R-32/134a (30/70 wt%), R-32/134a (25/75 wt%), and R-32/125 (50/50 wt%) which were also tested in this study. There are some discrepancies among those results and the tested conditions and testing equipments are not explained in detail, although the reports provide informative results on various candidates.

For the purpose of tuning the optimum selection of the alternatives, it is needless to emphasize an important role of reliable thermodynamic property data on refrigerant mixtures. The present study, therefore, aims to evaluate the cycle-performance with three different binary HFC-mixtures more exactly on the basis of reliable thermodynamic properties.

EXPERIMENTAL

The testing-facility used is schematically shown in Fig.1 which is a conventional test rig consisted of a compressor, a condenser, an expansion valve, and an evaporator. Since the present facility was originally designed to test compressor performance with R-22, some modifications were needed for testing the refrigerant mixtures. Major

modifications made in the present study include to equip a mass-flow meter, a gas-chromatograph and an oil-concentration analyzer which all installed just before the expansion valve. The condenser was also redesigned into a double-tubing counter-flow type heat exchanger so as to regulate the temperature glides in our future research program.

The pressures and temperatures of the refrigerant are measured at the points 1 through 7 in Fig.1; 1 and 2 at the inlet and outlet of the compressor; 3 and 4 at the inlet and outlet of the condenser; 5 and 6 before and after the expansion valve; and 7 at the evaporator outlet, respectively. The uncertainties of pressure and temperature measurements are estimated to be about ± 50 kPa and ± 0.5 K, although the pressures and temperatures have been controlled within ± 5 kPa and ± 50 mK, respectively.

The mass-flow meter, h, was originally developed in this study which enables to measure the mass-flow rate directly. The mass-flow meter consists of two vessels associated with metallic bellows, A and B, a digital balance, a timer and a 3-way solenoid valve, g. The measurements of mass-flow rate of refrigerant were performed by weighing the refrigerant mass collected in the metallic bellows vessel, B, within a specified time period. Namely, such a measurement was performed by switching the flow-circuit to the mass-flow meter circuit with an aid of the 3-way solenoid valve for three different time-periods; 10 s, 20 s, and 30 s. Having examined the reproducibility, we have found that the maximum difference in mass-flow rate was always smaller than ± 2%. Therefore, we have determined the tested mass-flow rate from the mass difference between 30 s and 10 s measurements and the time-period difference of 20 s.

The uncertainty of refrigerant-composition measurements by means of gas-chromatograph is estimated to be ± 1% due to a precedence calibration of composition-known refrigerant mixtures by weighing the mass of each constituent refrigerant.

The uncertainty of oil-concentration measurements in liquid refrigerant just before the expansion valve is estimated to be ± 10% in weight fraction of oil-concentration according to the manufacturer's information.
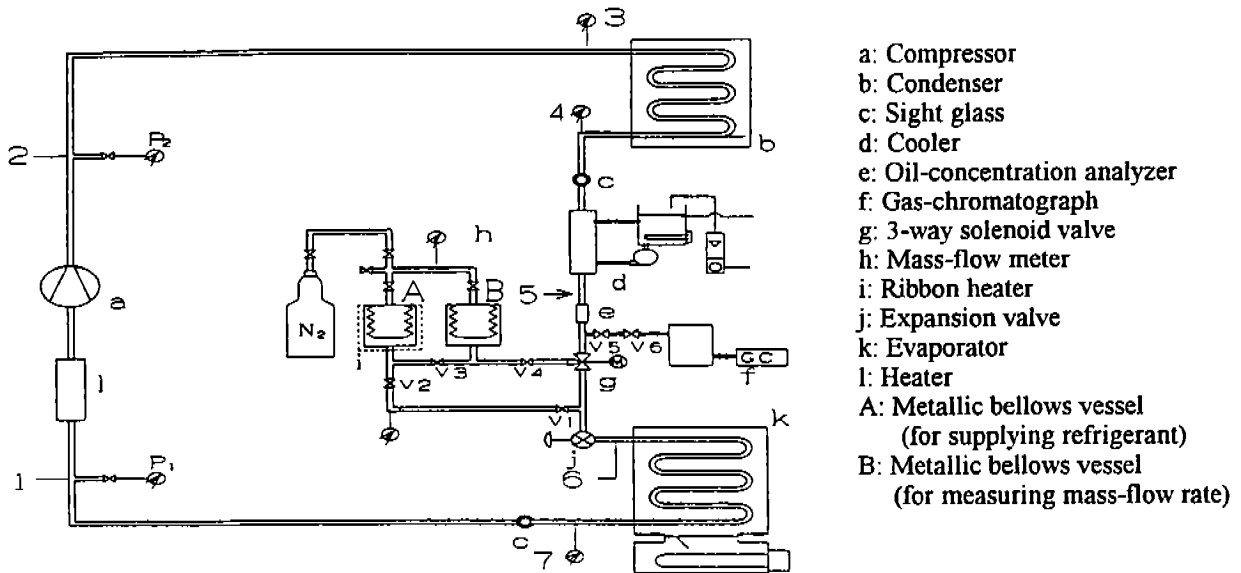

Fig.1 Experimental set-up for testing cycle-performance of refrigerant mixtures

a: Compressor
b: Condenser
c: Sight glass
d: Cooler
e: Oil-concentration analyzer
f: Gas-chromatograph
g: 3-way solenoid valve
h: Mass-flow meter
i: Ribbon heater
j: Expansion valve
k: Evaporator
l: Heater
A: Metallic bellows vessel
  (for supplying refrigerant)
B: Metallic bellows vessel
  (for measuring mass-flow rate)

TESTED CONDITIONS AND THEORETICAL CYCLE-PERFORMANCE

The refrigerant temperatures at the inlet and outlet of the heat exchangers are not the same each other when zeotropic refrigerant is applied to refrigeration cycle. In order to compare the cycle performance with refrigerant mixtures with that by R-22, a unique common test conditions can not be applied to refrigerant mixtures, so the respective test condition has to be prepared for respective composition of refrigerant mixtures which can be derived from the reliable thermodynamic properties. Fortunately a group of the present authors has measured important thermodynamic-property data for the present sets of binary refrigerants; bubble-point pressures and saturated-liquid densities of R-32/134a (3) and R-32/125 mixtures (4); PVTx properties in the vapor phase of R-32/134a (5) and R-

Table 1a Tested conditions and theoretical performance for R-32/134a (30/70 wt%) mixture

Tcond / ℃	Teva /℃	P2 / MPa	P1 / MPa	T1 / ℃	T5 / ℃	COP	Q / kJ·kg⁻¹
45	7	1.72	0.56	17.2	37.5	5.45	195
45	0	1.72	0.44	10.1	37.5	4.40	191
45	-5	1.72	0.37	5.0	37.5	3.86	188
50	7	1.94	0.56	17.0	42.6	4.65	185
50	0	1.94	0.44	10.0	42.6	3.82	182
50	-5	1.94	0.37	4.9	42.6	3.31	179
55	7	2.20	0.56	16.9	47.7	3.94	175
55	0	2.20	0.44	9.8	47.7	3.28	171
55	-5	2.20	0.37	4.8	47.7	2.87	169

Table 1b Tested conditions and theoretical performance for R-32/134a (25/75 wt%) mixture

Tcond / ℃	Teva /℃	P2 / MPa	P1 / MPa	T1 / ℃	T5 / ℃	COP	Q / kJ·kg⁻¹
45	7	1.66	0.55	17.4	37.3	5.55	188
45	0	1.66	0.43	10.2	37.3	4.45	184
45	-5	1.66	0.36	5.2	37.3	3.88	181
50	7	1.88	0.55	17.4	42.5	4.69	178
50	0	1.88	0.43	10.2	42.5	3.83	174
50	-5	1.88	0.36	5.2	42.5	3.37	172
55	7	2.12	0.55	17.4	47.7	3.99	169
55	0	2.12	0.43	10.2	47.7	3.30	165
55	-5	2.12	0.36	5.2	47.7	2.93	162

Table 1c Tested conditions and theoretical performance for R-32/125 (50/50 wt%) mixture

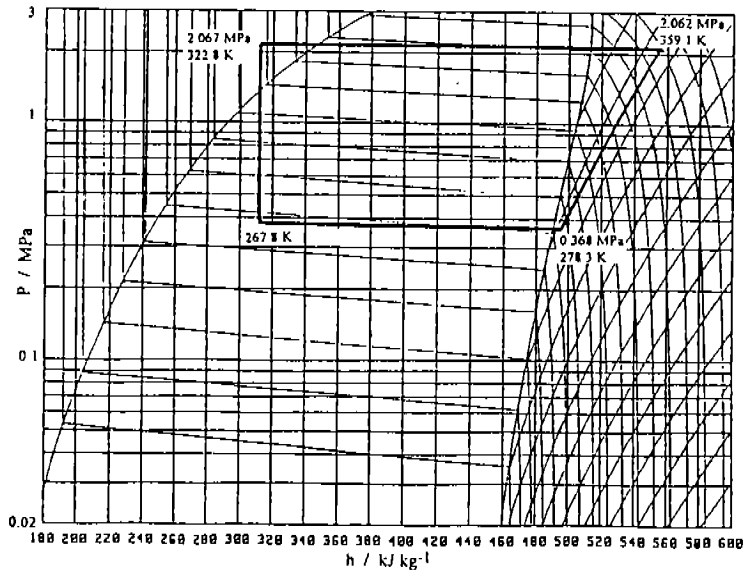| Tcond / ℃	Teva /℃	P2 / MPa	P1 / MPa	T1 / ℃	T5 / ℃	COP	Q / kJ·kg⁻¹
45	7	2.73	1.00	15.0	40.0	6.25	178
45	0	2.73	0.81	8.0	40.0	5.02	176
45	-5	2.73	0.69	3.0	40.0	4.34	175

Fig.2 Pressure-enthalpy diagram with actually tested refrigeration cycle for R-32/134a (25/75 wt%) mixture at the tested condition of 328K (55℃) for Tcond and 268K (-5℃) for Teva.

69

32/125 mixtures (6). Then reliable thermodynamic property values were calculated by fitting virial equations to the measured *PVTx* data in the gaseous phase, Peng-Robinson equation to the bubble-point pressures with a binary interaction parameter being 0.002 both for R-32/134a and R-32/125 mixtures, and modified Hankinson-Brobst-Thomson equation to the saturated-liquid densities, which was originally proposed by Maezawa et al. (7) and applied to the present binary refrigerants (3, 4), respectively.

Tables 1a to 1c show the tested conditions and calculated theoretical performance of R-32/134a (30/70 wt%), R-32/134a (25/75 wt%), and R-32/125 (50/50 wt%) mixtures, respectively. The mean temperature of the inlet and outlet temperatures at condenser and evaporator were fixed at nine different testing conditions, i.e., 45, 50, and 55 °C for the condensing temperature , Tcond, and -5, 0, and 7°C for the evaporating temperature, Teva, with +8°C superheating at the evaporator and -5 °C subcooling at the condenser outlet. It should be noted, however, that the actual tested-conditions were assigned by the pressure corresponding to Tcond or Teva which are the arithmetic mean temperature between the inlet and outlet temperatures at the condenser or the evaporator. For R-32/125 system, the test only at a single condensing temperature was performed because the maximum operation-pressure of the testing-facility is limited up to 3.5 MPa.

In the actual testing, the pressures at the condenser and evaporator were always controlled at the fixed pressure within ± 5 kPa by using an automatic expansion-valve and temperature-controlling systems. An example of actual refrigeration-cycle on a pressure-enthalpy diagram (8) is shown in Fig. 2 in case of R-32/134a (25/75 wt%) mixture.

RESULTS AND DISCUSSION

Mass-flow rate

Depending on the specific-volumes of refrigerant at the compressor inlet, the mass-flow rates, m, are different as shown in Fig. 3. The specific-volume values of R-32/125 (50/50 wt%) mixture are smaller than those of other blends, so the mass-flow rate of R-32/125 mixture is always greater than others by more than 50%. It is clearly observed in Fig.3 that the mass-flow rate increases, whenever Teva becomes higher and Tcond decreases. The mass-flow rate examination for R-32/134a (30/70 wt%) mixture is not completed yet at present.

Cooling capacity

A relative cooling capacity ratio, Q (mixture) / Q (R-22), is shown in Fig. 4. The cooling capacities generally increase as the condensing temperature decreases at a fixed evaporating temperature. Due to the larger mass-flow rate
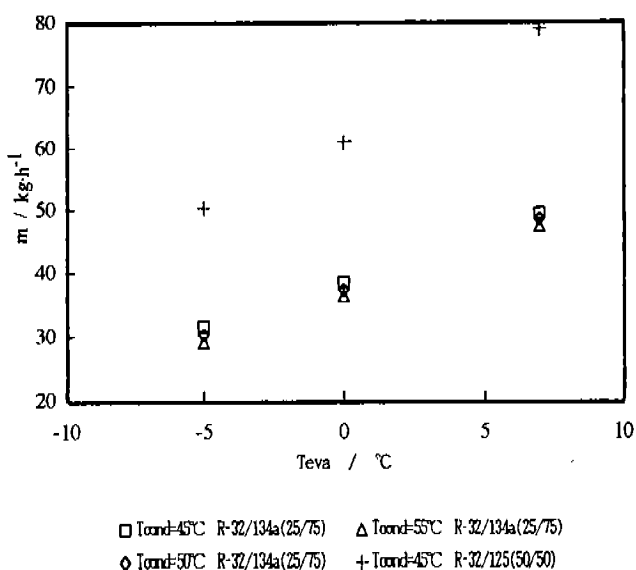

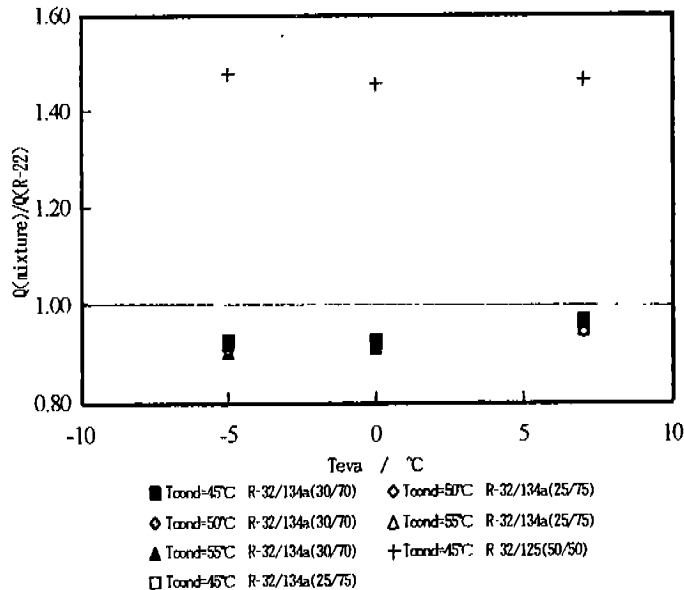Fig.3   Mass-flow rate of binary refrigerant mixtures.

Fig. 4   Cooling capacity ratio releative to R-22.

of R-32/125 (50/50 wt%) mixture, the cooling capacity is also greater than those of R-32/134a mixtures by more than 50%, while of R-32/134a (30/70 wt%) mixture shows greater capacity than that with 25/75 wt% mixture except at one tested condition.

It also becomes evident that the cooling capacity with R-32/125 (50/50 wt%) mixture is always significantly larger than that with R-22 by more than 45%, while that with R-32/134a mixtures is 3-10% lower than that with R-22.

Compression power

The required compression power in case of R-32/125 (50/50 wt%) mixture is always higher than that with R-22 nearly by 55%, while that of R-32/134a mixtures is 2-8% less than that of R-22, as shown in Fig. 5. In the present study, we have found that the compression power required for R-32/134a (30/70 wt%) mixture is slightly greater than that for R-32/134a (25/75 wt%) mixture at all tested conditions.

Energy efficiency

Due to the greater compression power being required in case of R-32/125 (50/50 wt%) mixture, the energy efficiency becomes worse than that of R-32/134a mixtures, by 1-5%, as shown in Fig. 6. The energy efficiency of R-32/134a (25/75 wt%) mixture is always better than that of R-32/134a (30/70 wt%) mixture. On the other hand, the energy efficiency of R-22 is superior to that obtained by any mixtures except at one tested condition.
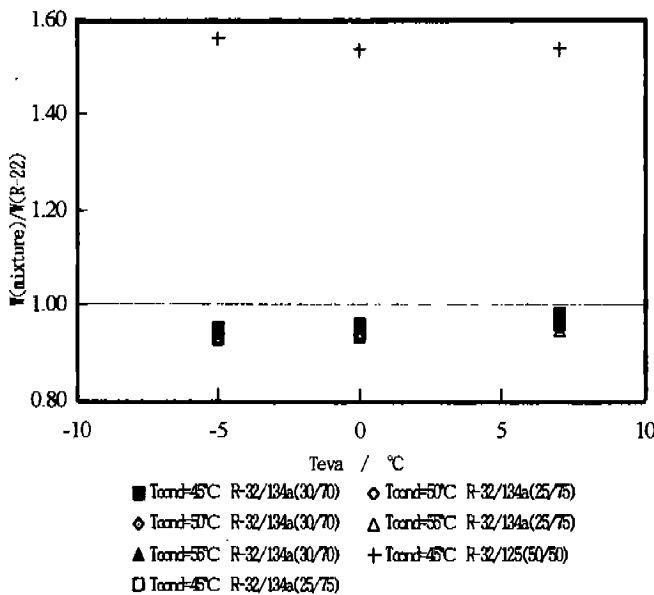

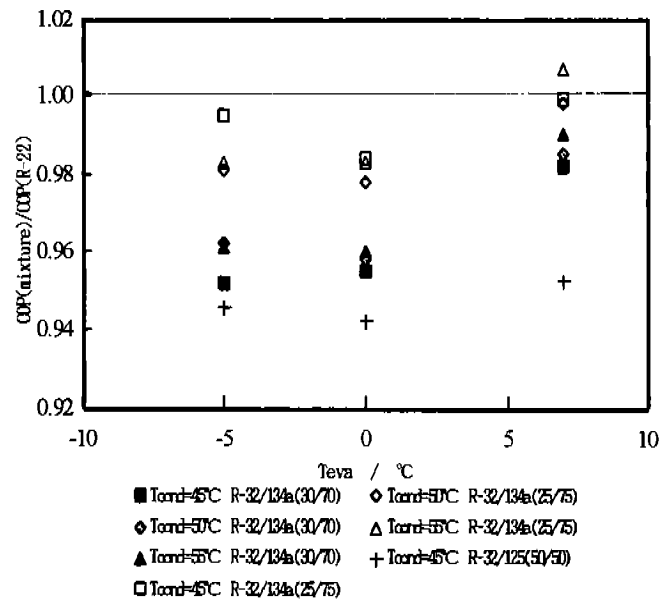Fig. 5 Compression power releative to R-22

Fig. 6 Energy efficiency ratio releative to R-22

Refrigerant composition and oil-concentration

The composition and oil-concentration in liquid refrigerant just before the expansion valve was measured by using a gas-chromatograph. To reach the equilibrium condition, it takes about three hours. The largest composition-change was observed in case of R-32/134a (30/70 wt%) mixture throughout the present study. The reason is considered due to a minor leakage of refrigerant from the testing-facility, so we add R-32 so as to maintain the composition being 30 wt% R-32 within the testing-facility. Actual refrigerant compositions of R-32/134a mixtures are summarized in Table 2 for nine different tested conditions, while the composition of R-32/125 mixture remains unchanged almost exactly at 50/50 wt%, since this binary refrigerant mixture is recognized as a zeotropic refrigerant. The oil-concentration of R-32/134a (30/70 wt%) mixture was measured in the liquid phase at the position just before the expansion valve. We have found that 0.17 to 0.69 wt% ester-oil was always solved in R-32/134a (30/70 wt%) mixture in the liquid phase. The oil-concentration increases with increasing condensing and evaporating temperatures.

Table 2. Actual composition of R-32/134a mixtures within the cycle
and the oil-concentration in the liquid refrigerants

(Teva / ℃) / (Tcond / ℃)	R-32 (30 wt%) / R-134a (70 wt%)	oil-concentration / wt%	R-32 (25wt%) / R-134a (75 wt%)
45 / 7	32.5 / 67.5	0.32	23.9 / 76.1
45 / 0	31.2 / 68.8	0.22	24.3 / 75.7
45 / -5	30.7 / 69.3	0.17	23.6 / 76.4
50 / 7	31.5 / 68.5	0.47	23.7 / 76.3
50 / 0	32.3 / 67.7	0.30	22.9 / 77.1
50 / -5	31.8 / 68.2	0.26	23.4 / 76.6
55 / 7	31.8 / 68.2	0.69	23.3 / 76.7
55 / 0	31.9 / 68.1	0.49	22.5 / 77.5
55 / -5	31.4 / 68.6	0.40	23.1 / 76.9

CONCLUSION

A series of refrigeration-cycle-performance tests has been performed for three different binary refrigerant mixtures on the basis of our reliable thermodynamic property information.

Because of dense gaseous condition, the mass-flow rate and the cooling capacity by R-32/125 (50/50 wt%) mixture are quite higher than those by R-32/134a mixtures by more than 50%. The higher pressure level, on the other hand, is required for R-32/125 mixture which causes the reduction in efficiency of compressors. From an energy-saving viewpoint, improvement would be necessary for the conventional compressors designed for R-22.

Based on the conventional compressors, the cycle performance with R-32/134a mixture is superior to that with R-32/125 mixture. The energy efficiency in case of using 25/75 wt% is better than that in case of 30/70 wt% of R-32/134a mixtures

The thermodynamic cycle performance of R-32/134a (30/70 and 25/75 wt%) and R-32/125 (50/50 wt%) mixtures have to be studied more extensively by taking into the consideration not only the global environmental and safety issues but also the heat transfer enhancement in heat exchangers for the final selection of the R-22 alternative in realistic air-conditioning systems.

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