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Test Results of Refrigerant R152a in a Mobile Air-Conditioning System

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Abstract

This study presents test results of a mobile air-conditioning system using a potential alternative refrigerant, R152a. A series of performance tests have been carried out and cycle characteristics such as cooling capacity, energy efficiency ratio, suction and discharge pressures, and temperatures are presented, compared to those for the baseline R134a system. Tests were conducted with evaporation temperature of 5℃, condensation temperature of 45℃, subcooling temperature of 5℃, superheating temperature of 5℃, and compressor speed of 500-1500 rpm. The performance of R152a system with readjustment of an expansion valve showed better than those of R134a. The effect of oil on the pressure drop in the evaporator was also addressed.

Key words: Alternative refrigerant, R134a, R152a, air-conditioning system, COP

Nomenclature

COP	coefficient of performance	(-)
L	liquid	(-)
P	pressure	(kPa)
ΔP	pressure difference	(kPa)
T	temperature	(℃)
V	vapour	(-)

1. Introduction

The HFC refrigerants have been widely used as alternatives for CFC/HCFC refrigerants in HVAC & R applications, but the HFCs are listed as global warming gases by the Kyoto Protocol which officially entered in force on February 16, 2005. Furthermore, European Union (EU) regulations require air-conditioned vehicles sold in EU countries to use refrigerants with global warming potentials (GWP) less than 150, starting with new type vehicles in 2011 and expanding to include all new vehicles by 2017. In mobile air-conditioning system, HFC refrigerant, R134a has been predominantly used as a zero-ODP replacement of the CFC refrigerant, R12. Therefore, switching from R134a to low GWP alternatives is critical issue since the GWP of the HFC-134a is 1300.

There are several candidates for alternatives of R134a such as R152a, R744 and R1234yf [1]. R744 (CO_2) is well known natural refrigerant that has been used as a refrigerant for over a century [2]. The main difference between R744 and other common refrigerants is its pressure temperature characteristics as shown in Fig. 1, and in particular a low critical temperature of approximately 31℃. This means that it should be used in a transcritical cycle that differs from conventional compression cycles. R744 also operates at significantly higher pressures and it requires substantial system redesign. R1234yf is less flammable and lower GWP of 4 than R152a, and its thermodynamic properties and cooling performance are similar to R134a. However it is still under testing of toxicity and multiple risk assessments. Refrigerant R152a has GWP of 140 less than 150 and has better thermodynamic and transport properties compared to R134a as shown in Table 1 [3]. Its vapor pressure is similar to R134a and so no significant system design change is required. However, the liquid to vapor density ratio of R152a is around 129% of R134a at 5-10℃ as shown in Fig. 2, and refrigerant distribution characteristics in an evaporator will be different. With more hydrogen atoms in R152a, problems associated

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Table 1. Property Comparison of R134a and R152a.

Property		R134a	R152a	R152a/R134a (Ratio)
Chemical Formula		CF_3CH_2F	CHF_2CH_3	-
Molecular Weight (kg/kmol)		102.0	66.1	0.648
Molecular Diameter (Å)		4.2	3.9	0.93
Normal Boiling Point (℃)		-26.2	-25	-
Critical Temperature (℃)		101.1	113.5	-
Critical Pressure (kPa)		4067	4492	1.10
Density at 5℃ (kg/m^3)	L	1279	947	0.749
	V	17.3	9.9	0.573
Pressure at 5℃ (kPa)		350	315	0.898
Latent Heat at 5℃ (kJ/kg)		195	301	1.54
Thermal Conductivity at 5℃ (W/m-K)	L	0.0971	0.1108	1.141
	V	0.0126	0.0126	1.001
Viscosity at 5℃ (μPa-s)	L	311.9	204.5	0.656
	V	11.18	6.40	0.572
Specific Heat Ratio (C_p/C_v)		1.22	1.28	1.05
Flammable Range (Vol%)		-	5.1-17.1	-
GWP 100 year (CO_2=1)		1300	140	0.108
TLV (ppm)		1000	1000	1.0
Safety Group		A1	A2	-

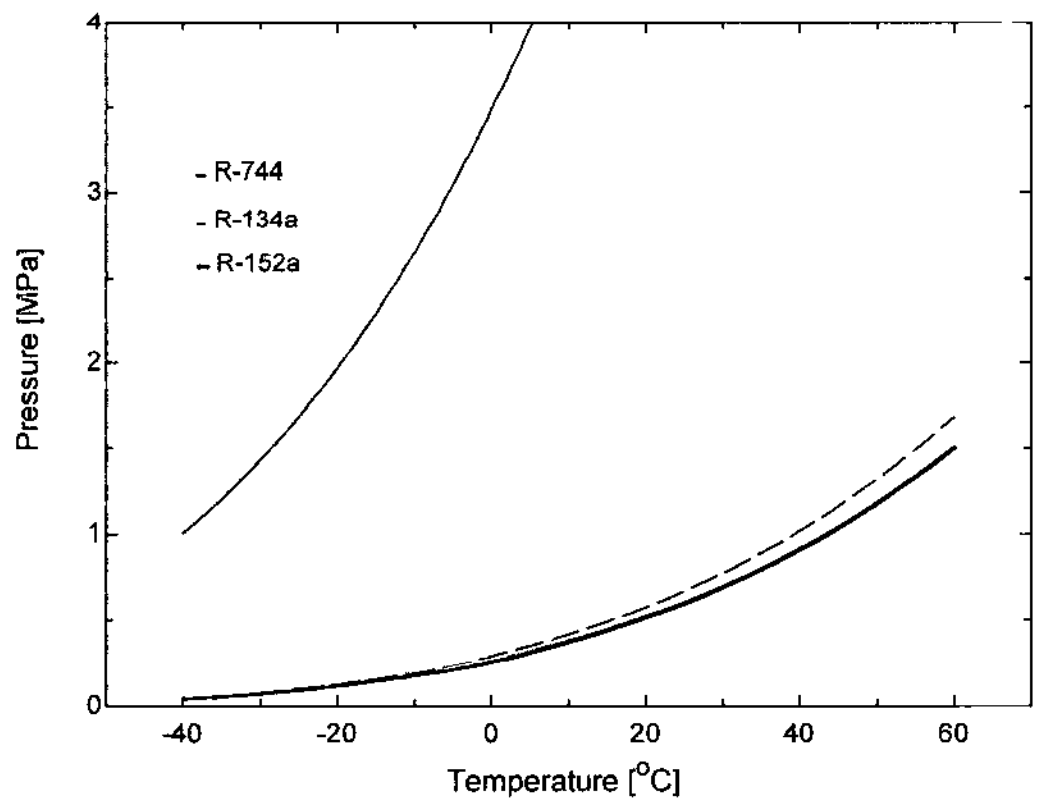


Fig. 1. Vapor pressure curves for R744, R134a and R152a.

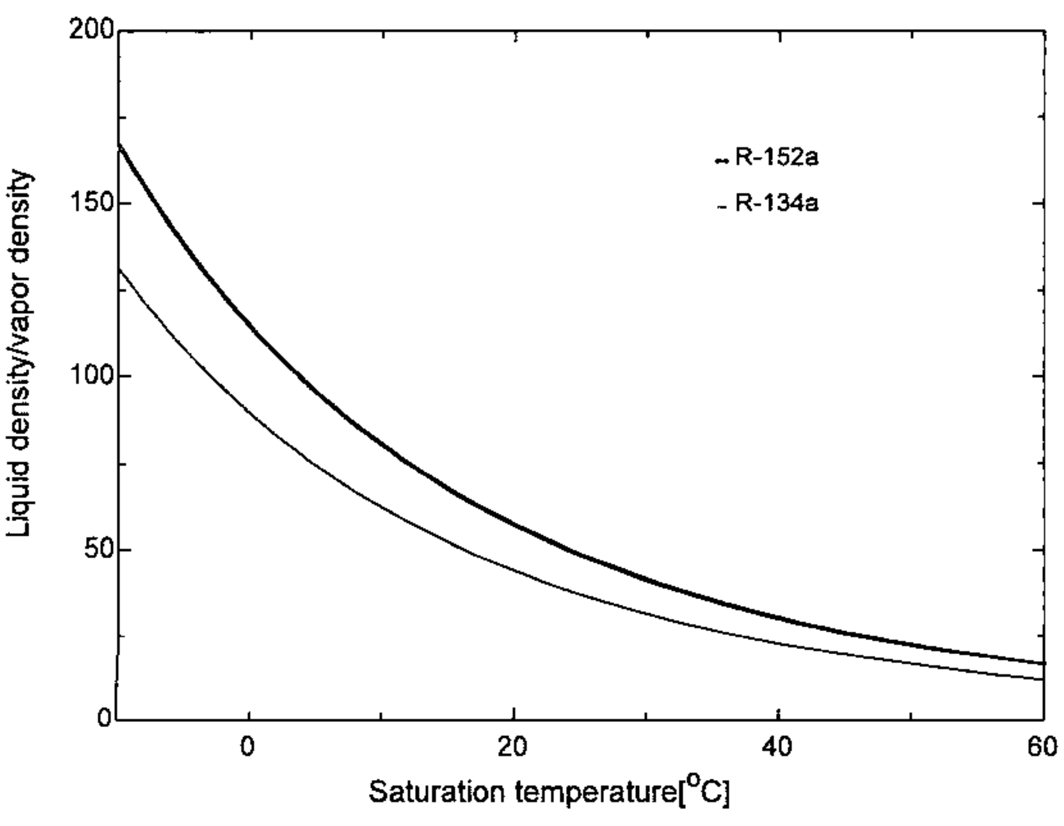


Fig. 2. Liquid to vapor density ratio.

with its flammability are most critical. There are several studies on the system performance evaluation of R152a as a replacement for R134a and R12.

M. Ghodbane [4] presented theoretical evaluation results of R152a and hydrocarbon refrigerants (propane, isobutane, cyclopropane), and their potential as a replacement for R134a in mobile air conditioning systems. He also reported the comparative assessment of a secondary loop air-conditioning system with these alternatives. The paper showed that R152a could be one of the most promising substitutes for R134a from the point of system performance.

J.-I. Lee et al. [5] conducted performance tests of an automotive air-conditioning system using R152a and a mixture of R152a and CF_3I. The tests were carried out with variation of the compressor speed, outside air temperature and flow rate. They found that COPs of R152a were higher than those for the mixed refrigerant at the compressor speeds of 1000 rpm and 1500 rpm, but the COPs of both R152a and mixture were almost similar at a compressor speed of 2000 rpm.

Devotta et al. [6] presented theoretical performance results of R134, R134a, R227, R152a, and R143a as alternatives to R12. Their study was based on the standard refrigeration parameters, including pressure ratio, specific compressor displacement, theoretical Rankine cycle COP and shaft power per ton of refrigeration. They speculated that R152a is the best from the energy point of view.

S. Komatsu et al. (7) compared refrigerant capacity and COP of R152a automobile air-conditioning systems with R134a systems in the running mode tests and estimated fuel consumption based on the climate data in various areas. They found that R152a system showed higher refrigeration capacity and lower fuel consumption compared to R134a system. They also reported that the performance improvement was greater in the areas with higher temperature.

Recently, M. Ghodbane et al. (8) reported the environmental benefits of and engineering safety strategies for automobile air conditioning systems that use R152a as a refrigerant. The conclusion is that R152a can be safely used in secondary loop air-conditioning systems that keep refrigerant out of the passenger compartment, and that R152a used in secondary loop can offer fuel savings, cost savings, and environmental benefits.

The investigation of cycle characteristics of new refrigerants is quite relevant to the optimization of the inherent system. The oil concentration in the refrigerant circulating in the refrigeration cycle affects the pressure drop and the heat transfer in heat exchangers, and the large oil concentration is attributed to deterioration of system performance. This paper presents the cycle characteristics of the refrigerant R152a in an automobile air-conditioning system. The effect of oil on the pressure drop characteristics in the evaporator is also reported to get some insight into oil effect on the system performance.

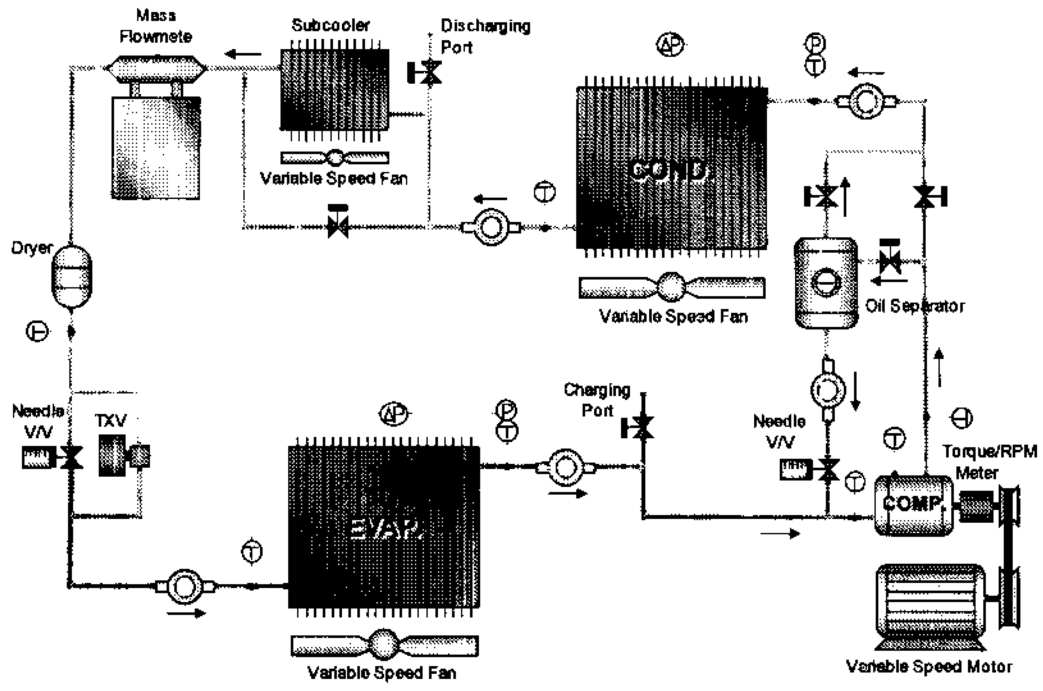


Fig. 3. Schematic diagram of test apparatus.

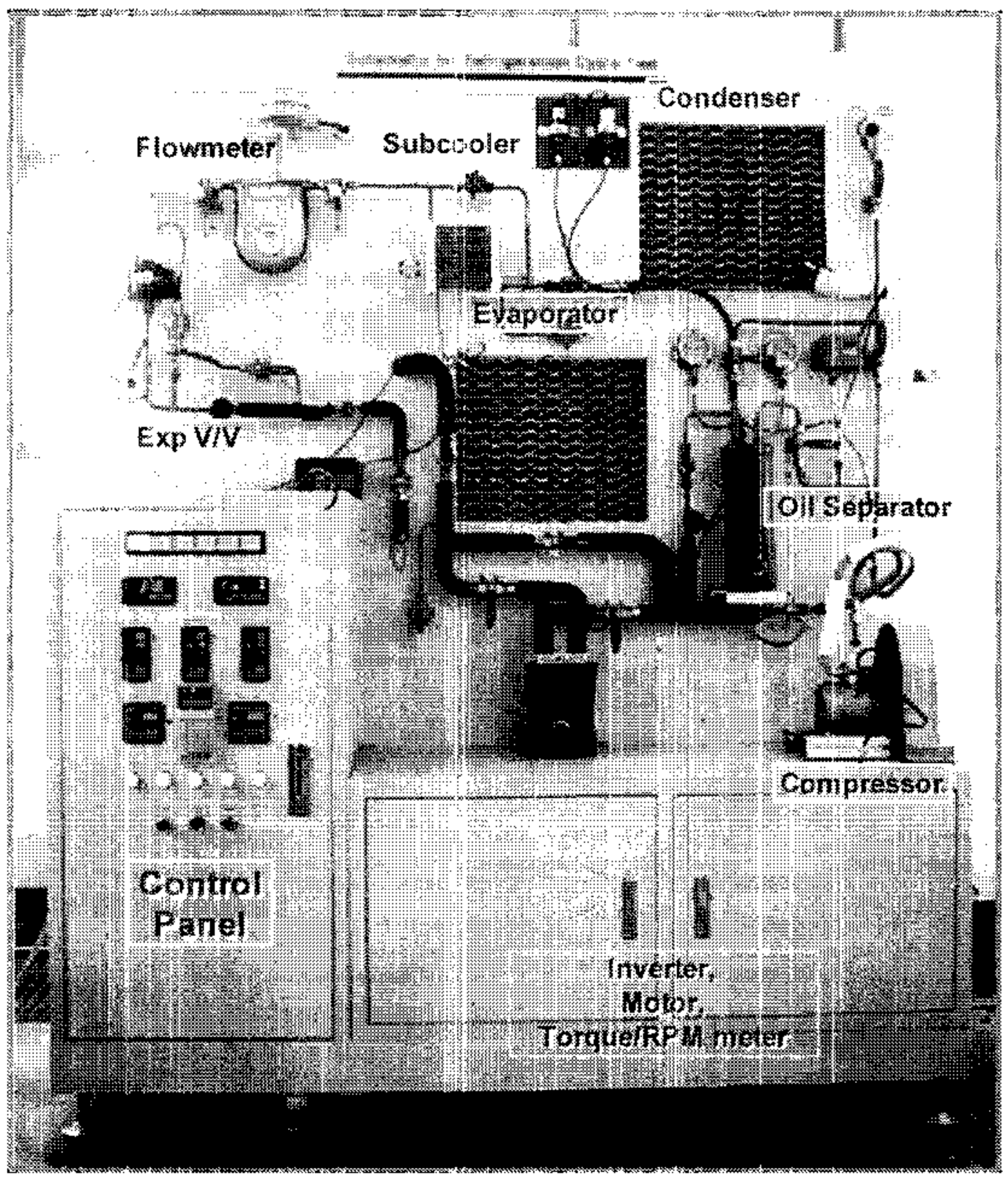


Fig. 4. Photograph of test apparatus.

2. Experiment

The performance tests were carried out in a refrigeration cycle test apparatus. The test apparatus consists of the refrigerant circulation loop and data acquisition system. Figs. 3 and 4 show schematic diagram and photograph of the refrigerant circulation loop. Its components are a compressor, an inverter motor, an oil separator, a condenser, a subcooler, a flow meter, a dryer, expansion devices, and an evaporator.

The refrigerant is circulated by a swash plate compressor of the automobile air-conditioning system. The compressor speed was controlled by the inverter motor. Compressor torque and RPM were measured to calculate compressor shaft power consumption. The refrigerant flow rate was measured using a Coriolis-type mass flow meter with a nominal flow range of 0-200 kg/h and an accuracy of ±0.1%.

The tests were conducted with evaporation temperature of 5℃, condensation temperature of 45℃, subcooling temperature of 5℃, superheating temperature of 5℃, and compressor speed of 500-1500 rpm as summarized in Table 2. The test refrigerants were R134a and R152a. The test data were collected using a data acquisition system consisting of a data logger (Agilent 34970A), a serial communication (RS-232C), and a compatible personal computer. The data were analyzed in real time using a PC and a data reduction program (MS-Excel with a Visual Basic). All the information for the test conditions and test data were displayed on the monitor during the test and the test conditions were regulated based on this information. After the steady state conditions were reached, 20 data points were collected for 120 seconds and aver

Table 2. Test condition.

Item	Condition
Evaporation Temperature (℃)	5
Condensation Temperature (℃)	45
Subcooling Temperature at expansion device inlet (℃)	5
Superheating Temperature at evaporator outlet (℃)	5
Compressor Speed (rpm)	500, 750, 1000, 1250, 1500*

*R152a only.

Table 3. Basic specifications of the test apparatus.

Item	Specification
Refrigerant	R134a, R152a
Compressor	Swash Plate Type, 10 Cylinders, Displacement: 110 cc
Evaporator	Fin and Tube Type, O.D.=9.54 mm, W400×H340×D100 mm
Condenser	Fin and Tube Type, O.D.=9.54 mm, W400×H340×D100 mm
Subcooler	Brazed Parallel Flow Type, W110×H130×D19 mm
Expansion Device	EEV (Electronic Expansion Valve) and Needle Valve
Lubricant	PAG (UCON-244) 150 cc (Initial charge)

aged. The thermodynamic and transport properties of R134a and R152a were obtained from the AHRAE Handbook [3].

The operating conditions such as refrigerant pressures, superheating temperature, subcooling temperature, etc were controlled by the expansion devices and fan motor speed of the heat exchangers. Fig. 5 shows the operating conditions for both R134a and R152a systems at the compressor speed of 500rpm. During the performance test, discharged lubricant at the compressor outlet was returned to the inlet of the compressor by using an oil separator and a needle valve. The refrigerant temperature at the compressor inlet was increased largely due to the discharged hot lubricant from the oil separator. Hence, superheating temperature was estimated at the outlet of the evaporator in the test since it was not suitable to estimate superheating temperature at the compressor inlet.

Same components and lubricant were used for both R134a and R152a system. The baseline tests were firstly performed using R134a, and then, the performance tests for R152a were conducted with the same

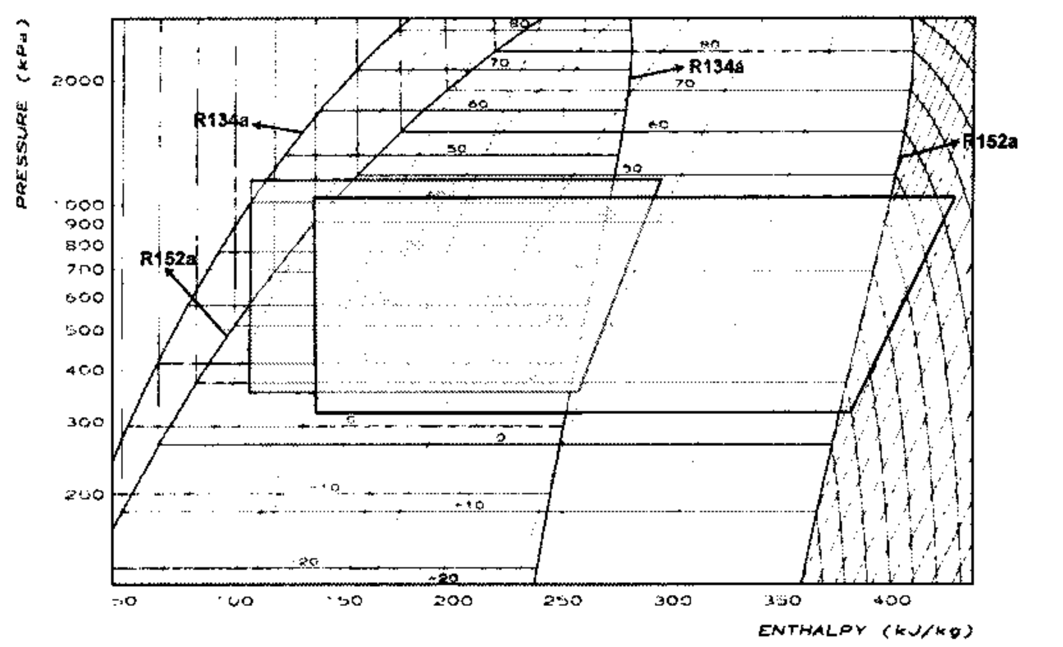


Fig. 5. Test conditions in pressure-enthalpy diagrams for R134a and R152a (compressor speed: 500rpm).

test conditions. The refrigerant charges for both systems are 1.0kg, and the expansion device was adjusted for the systems to have the same test condition for each test. The tests were not conducted in calorimetric chambers, so we could not get the air-side capacity. But, condenser and evaporator fan speeds as well as expansion device were controlled to meet the given test conditions, and so cycle characteristics for both systems were compared at the same test conditions.

The cooling capacity was calculated using the refrigerant mass flow rate and enthalpy difference in the evaporator. It was assumed that the expansion process is isentropic process, and then the enthalpy at the evaporator was calculated from the subcooling refrigerant pressure and temperature at the expansion valve inlet. Uncertainty of the cooling capacity was estimated within ±1.0% based on refrigerant mass flow and enthalpy difference at the evaporator. The coefficient of performance (COP) was estimated with the cooling capacity and the measured shaft power of the compressor. The refrigerant temperatures were measured using RTD sensors and T-type thermocouples with an accuracy of of ±0.1℃. The pressures and the pressure differences were measured by the pressure transducers (with an accuracy of 0.25%) connected to the static pressure taps. In addition to the performance test, the effect of oil on the system performance was also estimated. Oil forming pattern was observed through the sight glass and the pressure drops in the evaporator were measured.

3. Results and discussion

Figs. 6-10 depict the performance test results for R152a system as functions of the compressor speed compared to those for R134a system. Fig. 6 presents the cooling capacity. At the same compressor speed,

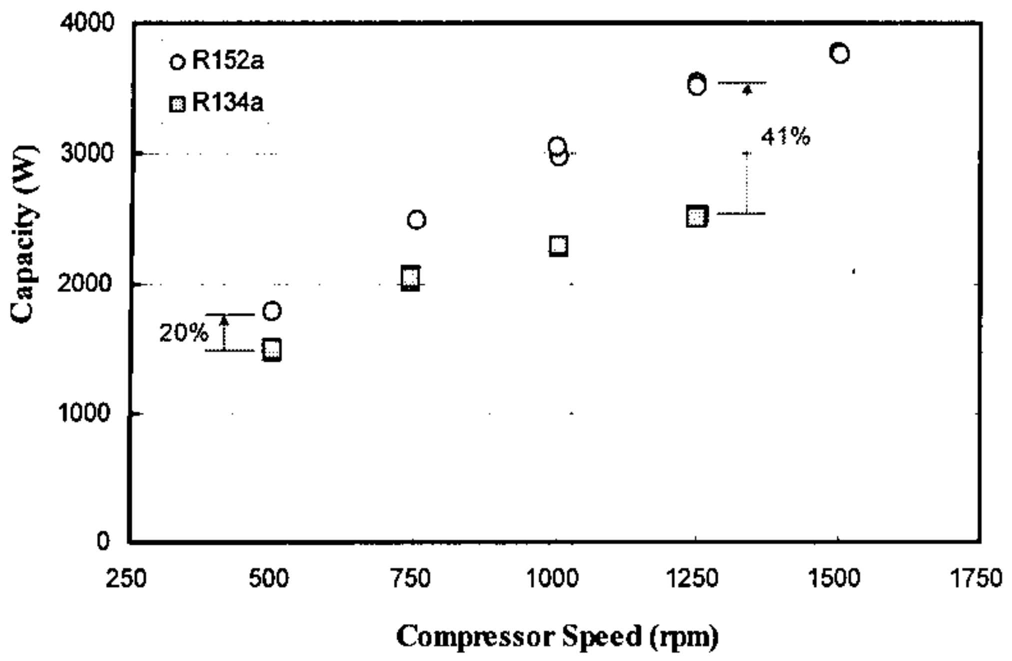


Fig. 6. Cooling capacity.

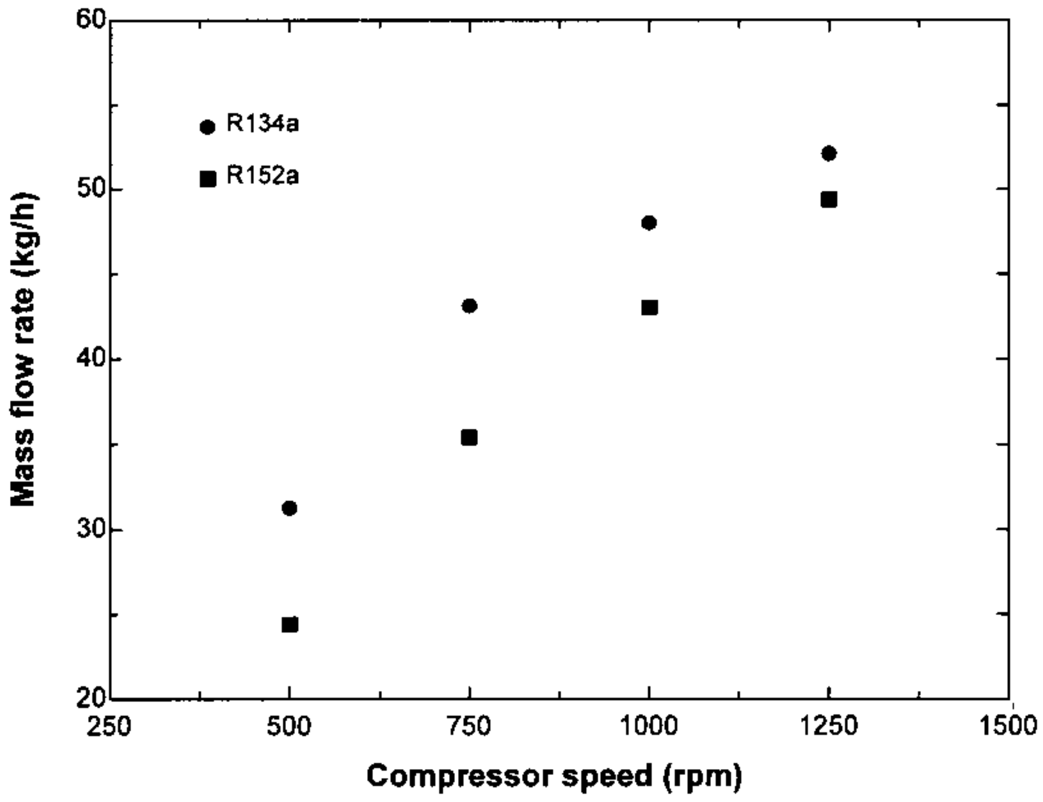


Fig. 7. Refrigerant mass flow rate.

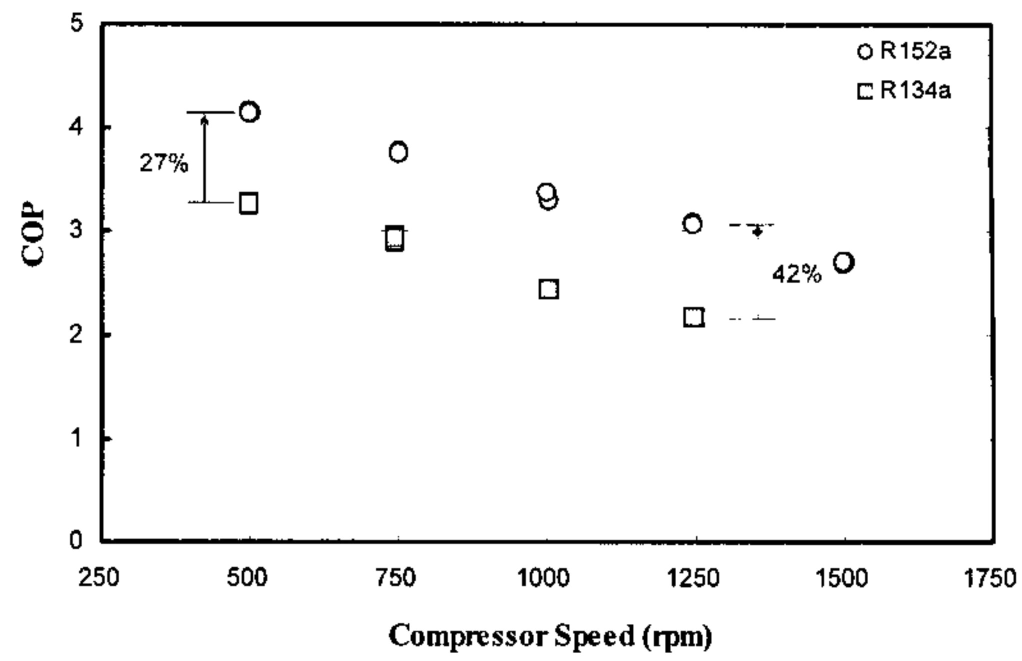


Fig. 8. Coefficient of performance.

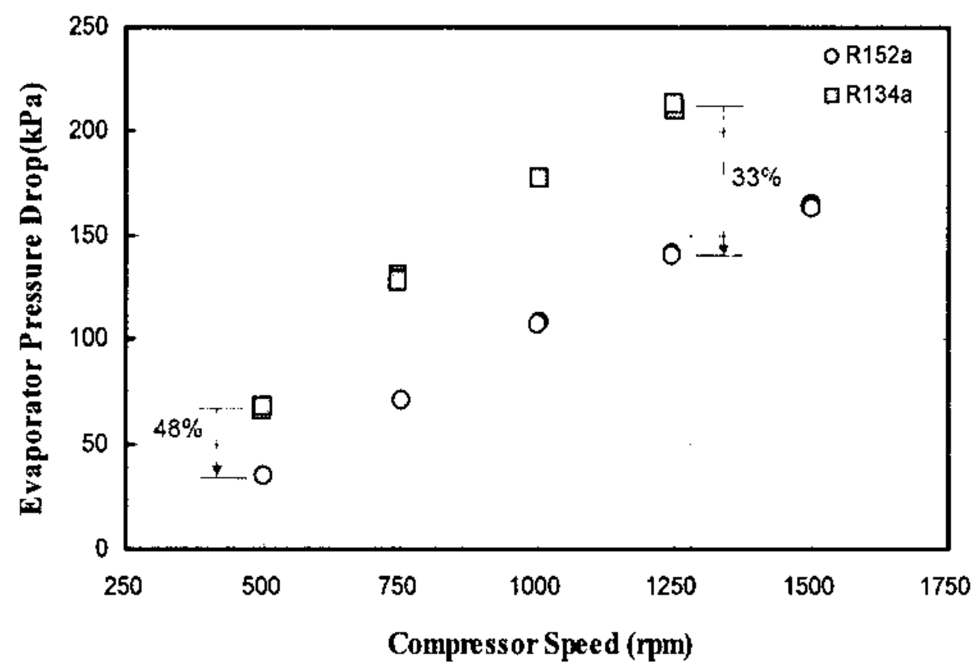


Fig. 9. Evaporator pressure drop.

cooling capacity of R152a system is 20-41% higher than R134a system. Although refrigerant vapor density and mass flow rate of R152a is lower than that of R134a (Fig. 7), the cooling capacity of R152a system is higher than R134a system because of its larger latent heat. This capacity difference increases with the compressor speed. However, note that discrepancy in refrigerant mass flow rate is decreased with compressor speed (from 22% at 500rpm to 5% at 1250rpm) as shown in Fig. 7. At the given evaporation temperature, the lower compressor inlet temperature (Fig. 5) is attributed to the larger pressure drop in the evaporator of R134a system. Then, this results in decreasing vapor density at the compressor inlet.

Fig. 8 shows coefficient of performance (COP). The cooling COP is estimated cooling capacity divided by the compressor shaft work. At the same compressor speed, R152a system shows 27-42% higher COP compared to R134a system. As compressor speed increases, the COPs of both R152a and R134a systems decrease and the COP difference between the two systems is increased up to 42% at the compressor speed of 1250rpm. However, note that the COP of R152a system (3.7) is 60% higher than that of R134a system (2.3) at the same cooling capacity of 2.5 kW, which is found at the compressor speed of 750 and 1250 rpm for R152a and R134a systems, respectively.

Fig. 9 presents pressure drop in the evaporator. The evaporator pressure drop of R134a system shows 33-48% larger than that of R152a system. The pressure drop ratio between two refrigerant systems decreases as compressor speed increases. It is partly due to the ratio of refrigerant mass flow rate is decreased with compressor speed as shown in Fig. 7. Larger pressure drop of R134a system in the evaporator is attributed to mainly its larger viscosity and density as shown in Table 1.

Fig. 10 shows discharge temperatures. It is found that discharge temperature of R152a system is about 10℃ (6.4-10.8℃) higher than R134a system. However it may be worthwhile to note that both systems have similar discharge temperatures at the same cooling capacity. The discharge temperatures of R152a and R134a systems are 78 and 79℃ at the same cooling capacity of 2.5kW, as shown in Figs. 6 and 10.

In addition to the performance evaluation test, oil circulation tests were carried out with and without oil

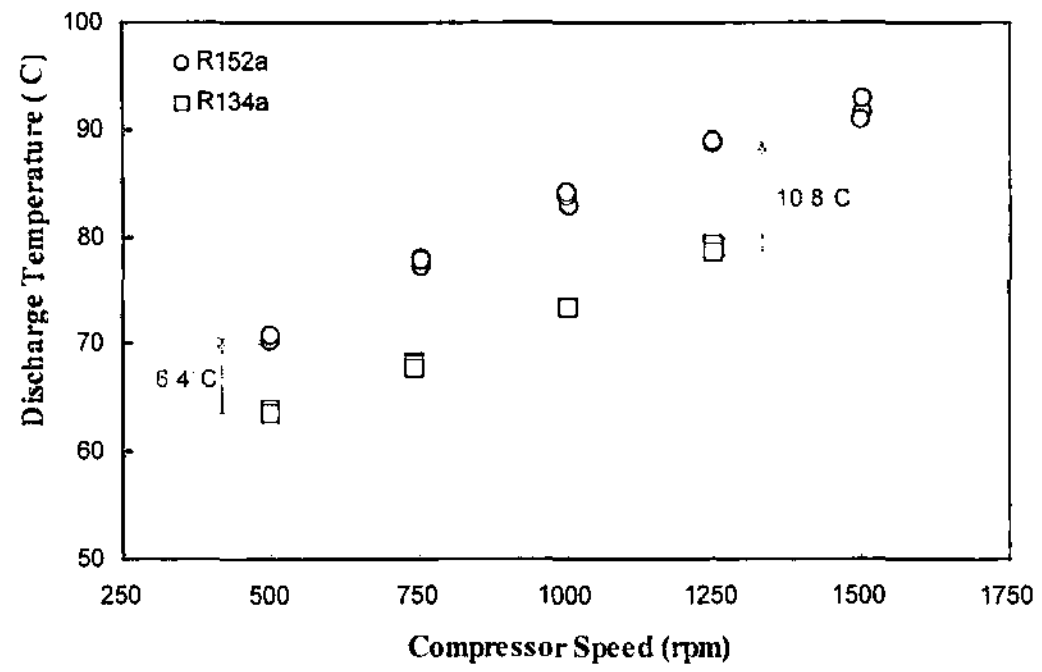
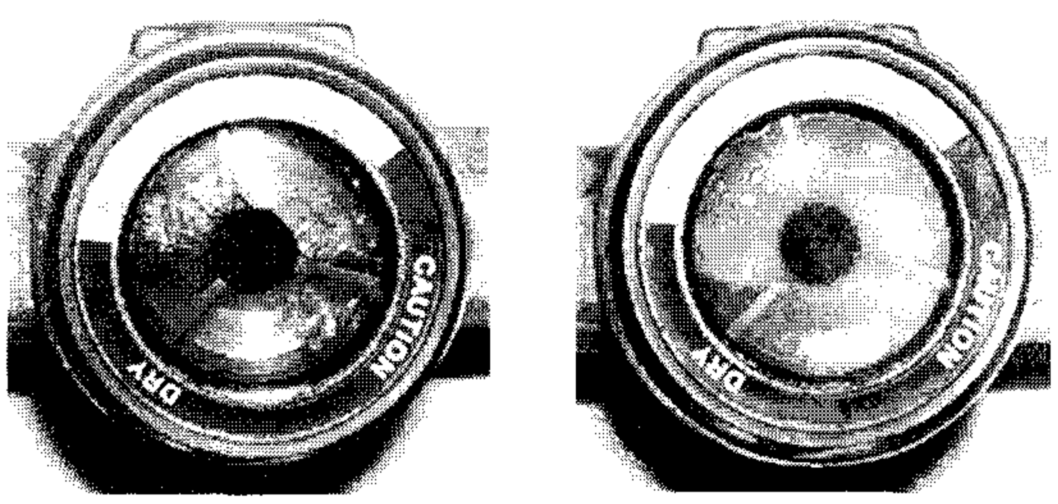


Fig. 10. Discharge temperature.



(a) with oil separator (b) without oil separator

Fig. 11. Photographs of evaporator inlet sight glass with and without oil separator (R152a).

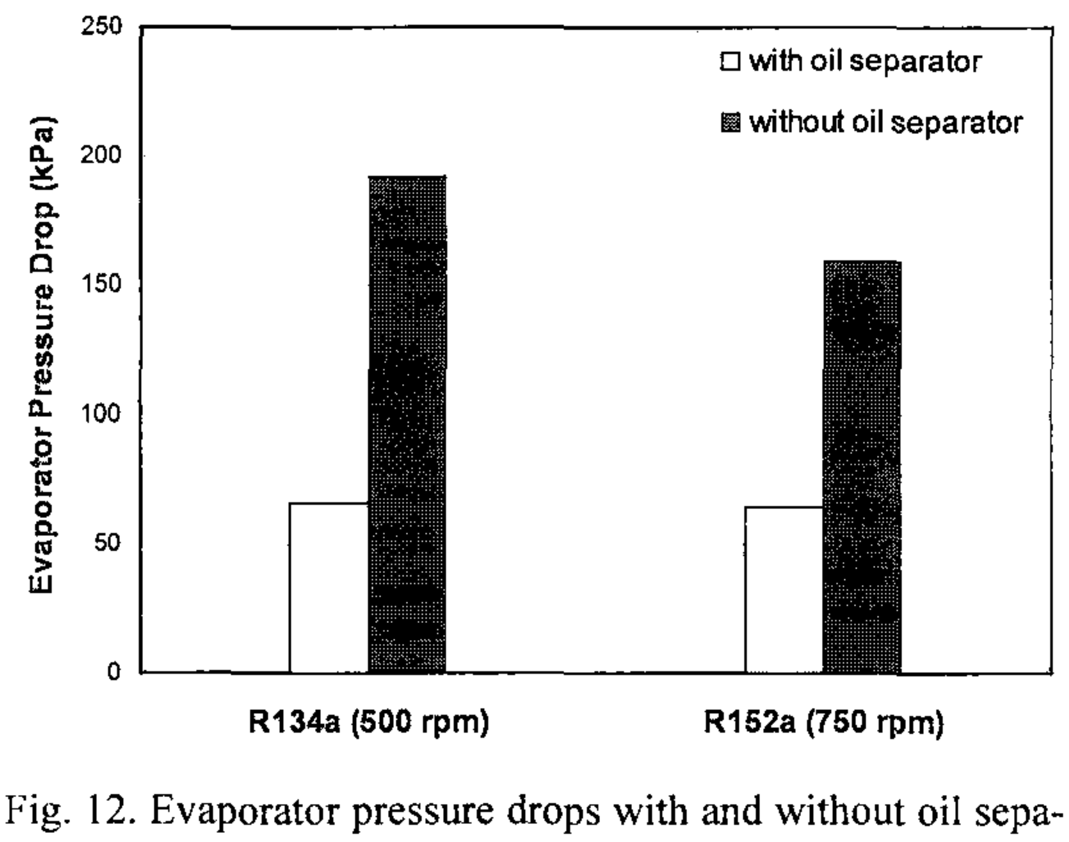


Fig. 12. Evaporator pressure drops with and without oil separator.

separator to get insight into the effects of oil on the system performance. Oil forming flow was observed through sight glass and pressure drops in the evaporator were measured. Fig. 11 shows photographs of evaporator inlet sight glass with and without oil separator. When the test was performed without oil separator, severe oil forming was observed, however the oil less flow was observed in case with oil separator. This oil forming resulted in increasing pressure drop significantly in the evaporator. Similar results were found with R134a system. Fig. 12 shows pressure drops in the evaporator with and without oil separator. Substantial pressure drop increment in the evaporator was found in the test without oil separator. Since the pressure drop increment in the evaporator due to oil concentration affects principally to the system performance and efficiency, further detailed work should be needed.

4. Conclusions

The performance evaluation of R152a as a replacement for R134a was conducted with evaporation/condensation temperatures of 5/45℃, subcooling/superheating temperatures of 5℃, and compressor speed of 500-1500 rpm in an automobile air-conditioning system. The system performance of R-152a with readjustment of an expansion valve showed better than that of R134a. The conclusions are as follows:

1. At the same compressor speed, R152a system showed 20-41% higher cooling capacity, 27-42% higher coefficient of performance, 33-48% lower pressure drop in the evaporator, and 6.4-10.8℃ higher discharge temperature than those of R134a system. However at the same cooling capacity of 2.5 kW both systems have similar discharge temperature, and R152a system shows 60% higher COP compared to R134a system.
2. Severe oil forming phenomena at the inlet of the evaporator were observed for the test of both R152a and 134a systems without oil separator. This oil forming phenomena increased significantly pressure drop in the evaporator.

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