The Performance Analysis of a Hydrocarbon Refrigerant R-600a in a Household Refrigerator/Freezer

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The performance of a hydrocarbon refrigerant, R-600a, as an alternative for R-12 has been evaluated in a 215 ℓ household auto-defrost refrigerator/freezer. A theoretical analysis was performed with NIST REFPROP, based on the ASHRAE refrigeration cycle and a series of tests with R-600a was conducted according to the Korea Standard (KS C 9305). All the tests were performed in the climate chamber of which the temperatures and relative humidities were maintained at 30±1℃ and 75±5%, respectively. The test results showed that the energy efficiencies and the cooling speeds with R-600a were improved by 1~11% and 3~10%, respectively, compared to R-12.

Key Words: Hydrocarbon Refrigerant, Auto-Defrost Refrigerator/Freezer, Cycle Analysis, Pull Down Test, Energy Consumption Test.

1. Introduction

CFCs developed in the 1930's have been most widely used in the field of refrigeration since they almost perfectly satisfied the requirements for good thermodynamic properties, chemical stability, non-flammability, non-toxicity etc. However, CFCs have proven to be harmful to the global environment, and hence regulations against the production and use of CFCs are in progress (Molina and Rowland, 1974 ; UNEP, 1987). These days, researches are actively going on to develop a refrigerator/freezer with CFC alternatives, and to improve the system performance. The refrigerators with alternative refrigerants are being manufactured and sold in most of the developed countries, where R-134a, R-600a and R-22 are mostly used as the substitutes for CFCs. R-22, a HCFC, has an ODP of 0.05 and a GWP of 1600 and is used in some countries, including Japan, as a short term substitute. R-134a, a HFC with an ODP of zero and a GWP of 1200 has lower performance than R-12, and is not an environmentally favorable refrigerant with respect to total equivalent warming impact (TEWI) which is defined as the overall evaluation of the influence on the global environment. In addition, the lubricant for the compressor needs to be changed from mineral oil to synthetic oil.

On the other hand, R-600a, a natural refrigerant with an ODP of zero and a GWP of 3, is one of the most environmentally favorable refrigerants being considered. R-600a has excellent thermodynamic properties and can be used with mineral oil, but it is flammable. However, the refrigerant charge amount of R-600a is about 30~50% of R-12, and there is very little chance of explosion. BOSCH in Germany reported that the explosion possibility of a hydrocarbon refrigerator was one five millionth (Baz, et al., 1995). Calor Gas in England is merchandizing Care series with the hydrocarbon refrigerants as an alternative to CFCs and HCFCs (Ritter, 1995). Hydrocarbon refrigerator/freezers are being produced or developed in Europe, India and China, which have the highest demand for refrigerators in the world. Researches on hydrocarbon refrigerator/freezers are proceeding in other countries including the U. S. A. Kim et al. (1993) showed that the energy efficiency could be im-

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proved by 6~7% using cyclopropane(RC-270) as an alternative for R-12. James et al. (1992) conducted experiments using propane(R-290) on the energy efficiency, flammability etc. Riffe (1994) studied to improve the energy efficiency with isobutane(R-600a). Jurgensen(1994), Gorenflo et al. (1994), Baskin and Perry(1994), and Jung et al. (1996) used propane/isobutane(R-290/600a) as a working fluid for their studies. In addition, Liu et al. (1995) and Liu et al. (1995) used propane/normal butane(R-290/600) as an alternative refrigerant for R-12. They performed the charge optimization tests in a household refrigerator/freezer with different capillary tube lengths and showed that with hydrocarbon mixtures energy savings up to 6% were achieved compared to R-12 baseline test. Chang et al. (1997) investigated the performance and the heat transfer characteristics of the air conditioning system with single component hydrocarbon refrigerants and binary mixtures as working fluids. They reported that some hydrocarbon refrigerants had better characteristics than R-22.

However, most of the commercialized hydrocarbon refrigerator/freezers are of a natural cooling type, and some companies in Germany are in the process of developing a refrigerator/freezer which has two evaporators, one for natural cooling of the refrigerator compartment and the other for fan cooling of the freezer compartment. However, a no-frost hydrocarbon refrigerator/freezer is not yet marketed.

This paper presents the results of the performance evaluation of a hydrocarbon refrigerant R-600a in a household automatic defrost refrigerator/freezer. The theoretical analysis of R-12 and its alternative refrigerants was firstly performed, based on the ASHRAE refrigeration cycle, by using NIST REFPROP. The isobutane refrigerant was evaluated as the best alternative from this analysis. Therefore R-600a was selected as a test refrigerant. A series of the performance tests with R-600a was conducted in a 215 ℓ auto-defrost topmount household refrigerator/freezer and the results were compared to those of the R-12 baseline unit.

2. Theoretical analysis

Table 1 shows some properties and environmental impact of the hydrocarbon refrigerants compared to the synthetic refrigerants, R-12 and R-134a. The hydrocarbons have a lower molecular weight compared to the synthetic refrigerants since they do not have halogens such as chlorine and fluorine in the molecular structure. This characteristic of the hydrocarbon refrigerants makes them more friendly to the environment (Embraco, 1993). Figure 1 shows vapor pressures of R-134a and hydrocarbon refrigerants compared to R-12. The vapor pressure of R-134a is similar to R-12. R-600a and R-600 present lower vapor pressures than R-12 while R-290 has higher pressures.

A theoretical analysis was performed for R-12, R-134a and hydrocarbon refrigerants in the

Table 1 Properties of R-12, R-134a and hydrocarbon refrigerants.

Refrigerant number	R-12	R-134a	R-600a	R-600	R-290
Chemical formula	CCl_2F_2	CH_2FCF_3	$CH(CH_3)_3$	$CH_3(CH_2)_2CH_3$	$CH_3CH_2CH_3$
ODP	1.0	0	0	0	0
GWP(CO_2=1, 100yr)	8500	1200	3	3	3
Molecular weight	121	102	58	58	44
NBP(℃)	−30	−26	−12	−0.54	−42
Critical temperature(℃)	112	101	135	152	97
Critical pressure(kPa)	4100	4067	3631	3796	4248
Flammability(F/N)	N	N	F	F	F

ASHRAE refrigeration cycle (evaporating temperature : −23.3℃, condensing temperature : 54.4℃, liquid and suction gas temperatures : 32.2℃), using the thermodynamic properties from NIST REFPROP (Huber et al., 1996). In the cycle analysis, the same cooling load was imposed for all refrigerants considered. The cooling load of 52.2W was obtained by calculating the heat transfer rate from the environment of 30℃ to the freezer (−18.8℃) and the refrigerator (3℃) compartments for the test unit.

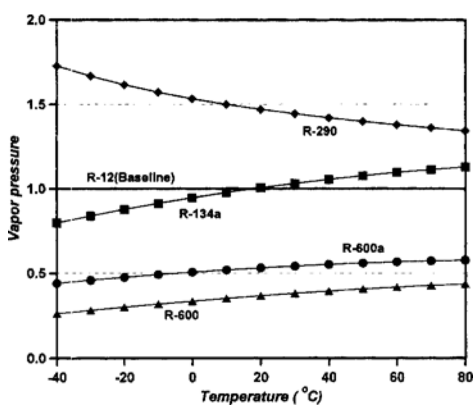

Fig. 1 Vapor pressure curve of different refrigerants.

Table 2 presents the results of the theoretical cycle analysis. The latent heat of evaporation of hydrocarbons is over 2.3 times greater than R-12. This means that the refrigerant mass flow rate should be lower than that of R-12 as shown in the table. The COPs of R-600a and R-600 are 6% and 7% higher than R-12, respectively, and R-134a and R-290 has the same COP as R-12. The volumetric capacity of refrigeration (VCR) for R-290 is higher than R-12, while R-600a and R-600 are lower. Therefore the compressor displacement has to be modified with hydrocarbons to match the refrigeration capacity. It was observed that R-600a and R-600 needed an increase of the volumetric displacement of 91% and 180%, respectively than R-12, while R-290 needed a reduction of 33%. The compressor discharge temperatures of hydrocarbons are 9 to 19% lower than R-12. This is good characteristic from the standpoint of compressor reliability. The refrigerant volume flow rates of R134a and R-600 through an expan-

Table 2 Theoretical characteristics of various refrigerants in the ASHRAE refrigeration cycle.

Refrigerant number	R-12	R-134a	R-600a	R-600	R-290
Condensing pressure at 54.4℃ (kPa)	1354	1470 (109%)	761 (56%)	556 (41%)	1883 (140%)
Evaporating pressure at −23.3℃ (kPa)	133	115 (87%)	62 (47%)	39 (29%)	217 (164%)
Pressure difference (kPa)	1221	1355 (111%)	699 (57%)	517 (42%)	1666 (136%)
Pressure ratio	10.2	12.77 (125%)	12.2 (120%)	14.3 (140%)	8.71 (85%)
Refrigeration effect (kJ/kg)	142	186 (130%)	336 (233%)	365 (254%)	354 (246%)
VCR (kJ/m³)	924	878 (95%)	485 (52%)	330 (36%)	1385 (150%)
COP	100	100	106	107	100
Refrigeration capacity[1] (W)	55.2	55.2	55.2	55.2	55.2
Mass flow rate (kg/h)	1.399	1.068 (76%)	0.591 (42%)	0.545 (39%)	0.561 (40%)
Displacement at 3500rpm (cm³/rev)	1.024	1.078 (105%)	1.951 (191%)	2.868 (280%)	0.683 (67%)
Discharge temperature for isentropic process (℃)	127	119 (93%)	103 (81%)	105 (83%)	116 (91%)
Capillary inlet temperature (℃)	32.2	32.2	32.2	32.2	32.2
Specific volume (cm³/kg)	775	845 (109%)	1844 (238%)	1771 (229%)	2080 (268%)
Volume flow rate (cm³/h)	1084	902 (83%)	1090 (101%)	965 (89%)	1167 (108%)

[1] Refrigeration capacity was obtained by calculating cooling loads of the test unit

sion device are 17% and 11% lower than R-12, respectively, while R-290 shows 8% higher volume flow rate. On the other hand, R-600a has almost same volume flow rate as R-12, with indicating that no change of capillary tube seems to be required. However, the capillary length depends on the pressure drop through the capillary and the thermophysical properties such as viscosity and specific volume as well as refrigerant mass flow rate(Gorasia et al., 1991 ; Kuehl and Goldschmidt, 1991). The simple estimation of a capillary based on the reference(Kuehl and Goldschmidt, 1991) shows that the R-600a capillary length should be increased, compared to the R-12 capillary in the ASHRAE refrigeration cycle. R-600a and R-600 seemed to be the best

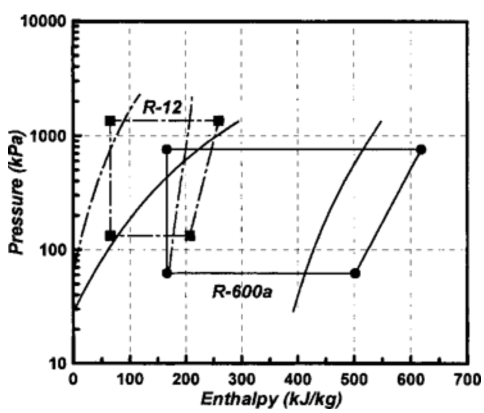

Fig. 2 Pressure-enthalpy diagram for R-12 and R-600a.

alternatives among the refrigerants from the thermodynamic point of view, but R-600a was chosen for the test since the compressor displacement of R-600a to be increased was relatively smaller than R-600. Figure 2 shows the pressure-enthalpy(P-h) diagram for R-600a, compared to R-12.

3. Tests

3.1 Descriptions of test unit

The baseline test unit was a 215 ℓ auto-defrost topmount household refrigerator/freezer which composed of a reciprocating compressor, a forced convection evaporator, a free convection condenser, an expansion device and related attachments. The expansion device was the capillary tube with length 2.8m and an inner diameter of 0.7mm. The capillary tube-suction line heat exchanger was also installed. The refrigerant charge amount of the R-12 baseline unit was 150g. For R-600a tests, only the compressor and the capillary tube were modified. The theoretical analysis indicated that the R-600a compressor should be redesigned to obtain the same refrigeration capacity with the R-12 baseline system. Therefore, the compressor displacement of the R-600a unit was increased by 91%. Mineral oil was used as the compressor lubricant for both refrigerants, with higher purity than 99.5%. An additional capillary tube was also installed between

Table 3 The specifications of the baseline test unit.

Items	Specifications
Unit type	Automatic defrost top-mounted refrigerator/freezer
Internal volume	215ℓ(freezer compartment 54ℓ, refrigerator compartment 161ℓ)
Refrigerant/lubricant	R-12/Mineral oil
Compressor	Reciprocating compressor
Evaporator	Cross-flow fin and tube heat exchanger
Condenser	Natural cooling hot-plate type heat exchanger
Expansion device	Capillary tube(inner diameter 0.7mm, length 2.8m)
Temperature control unit	Thermostat for the freezer compartment Damper for the refrigerator compartment

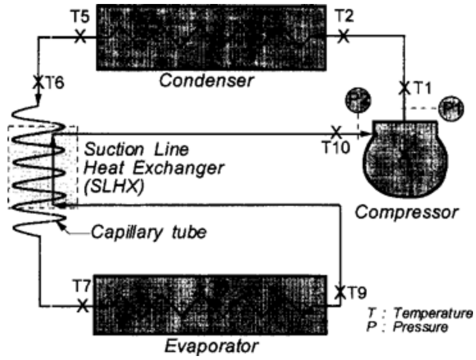

Fig. 3 Schematic diagram of the test unit.

the condenser exit and the inlet of suction-line heat exchanger with R-600a. The refrigerant charge amount with R-600a was optimized.

The freezer compartment temperature was regulated by a thermostat. On the other hand, the refrigerator compartment temperature was adjusted by a damper setting, which controlled the flow rate of cold air from the freezer to the refrigerator compartment. Table 3 shows the simple specifications of the baseline test unit and Figure 3 schematically describes the refrigerator/freezer used in this study, with the locations for temperatures and pressures measurements indicated.

Thirteen T-type thermocouples and two pressure transducers were attached to measure the temperatures and pressures. Also thermal masses (copper weights) of 25mm diameter and 40mm height were installed with thermocouples inserted in them, for measuring the temperature of the refrigerator and the freezer compartments, respectively. One thermal mass was located in the middle of the freezer compartment and the other at 1/3 of the height of the refrigerator compartment from the bottom. Temperatures and pressures were measured with a hybrid recorder, and a power meter was used to measure compressor input power and total power consumption.

3.2 Test conditions and methods

All experiments were performed in a climate chamber, according to the Korea Standard (KS C 9305). The temperatures and the relative humidities inside the chamber were maintained at 30 ±1℃ and 75±5%, respectively, for all tests. The location of the refrigerator/freezer in the chamber was fixed and the distance between the back of the refrigerator/freezer and the chamber wall was maintained at 70mm.

After baseline tests for R-12 were conducted without any system changes, a series of tests was performed for the R-600a unit with different capillary lengths. For R-600a tests, a compressor with an increased displacement was applied, and the charge optimization tests were performed with three different capillary tubes (baseline capillary plus additional capillary length of 1.0, 1.5, and 2.0 m). For each capillary tube, the amount of refrigerant to be charged was determined through the test. In the energy consumption tests, the defrost timer was disconnected, and the freezer and the refrigerator compartments were maintained at −18.0±0.5℃ and 3.0±0.5℃, respectively, by adjusting the temperature controller. Power consumption was measured to make a comparison of energy consumption per day. In no load pull down tests, initial temperatures of the freezer and the refrigerator compartments were set at 30℃. The temperature controller was taken apart to make continuous operation possible, and then pull-down speeds to reach −15℃ for the freezer compartment and 5℃ for the refrigerator compartment were measured, respectively.

4. Test Results and Discussion

The test results with R-600a are presented in Figs. 4∼11, compared to those of R-12. The baselines are the test results of the original R-12 unit. Figure 4 shows the energy consumptions for various charge amounts of R-600a with three different capillary tube lengths. Energy savings were achieved by about 1∼11% compared to the R-12 baseline test, except for the cases with too low or too high refrigerant charge amounts. The best results with R-600a was obtained at 36.7% (55g) of the R-12 charge with an additional capillary tube length of 1.5m. The energy efficiency was improved up to 11.2% in this case. The decreased charge of R-600a is partly due to its lower vapor density. The lower refrigerant charge yields that the cyclic losses can be decreased

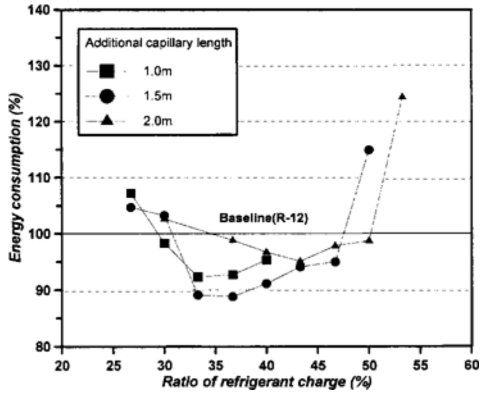

Fig. 4 Energy consumption with various refrigerant charge amounts.

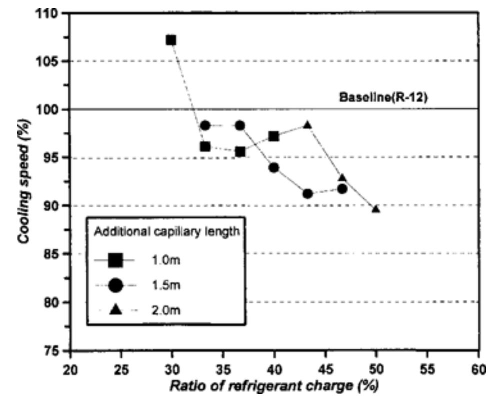

Fig. 6 Cooling speed in freezer compartment with various refrigerant charge amounts.

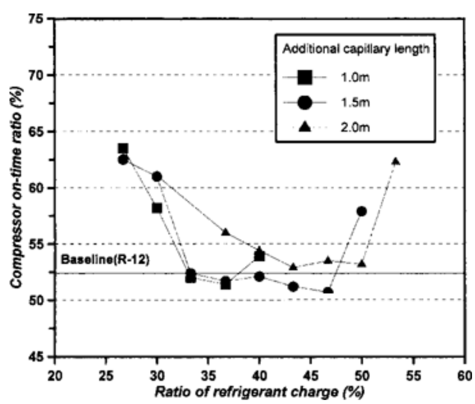

Fig. 5 Compressor on-time ratio with various refrigerant charge amounts.

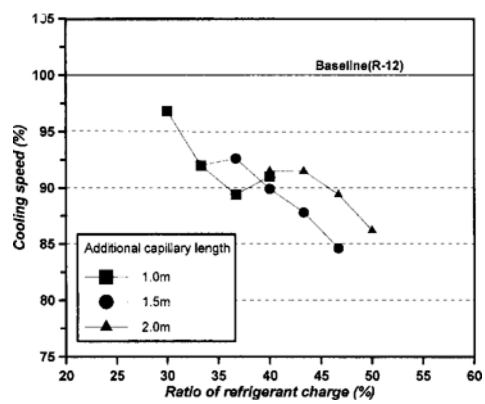

Fig. 7 Cooling speed in refrigerator compartment with various refrigerant charge amounts.

during the shut-down and start-up of a compressor.

Figure 5 shows the trends of the compressor on-time ratio versus the charge amount of R-600a. The compressor on-time ratio for R-600a was equal to or lower than that for R-12.

Figures 6 and 7 show cooling speeds in the freezer and the refrigerator compartments, respectively, versus the charge amount of R-600a. The cooling speed is defined as the time for the temperatures of the refrigerator and freezer compartments to reach 5℃ and -15℃, respectively, when the temperatures of the freezer and refrigerator compartments are initially set at 30℃. It is improved with increase of the refrigerant charge amount for a given capillary tube length due to the higher refrigeration capacity, within the scope of this study. And it tends to be improved with decrease of the capillary tube length for the same charge amount since the refrigerant flow rate increases. The improved cooling speeds with R-600a, even with less charge amount compared to R-12, are due to its large latent heat of evaporation as shown in Fig. 2.

Figures 8 to 11 show the test results for the no load pull down test. Figure 8 shows the behaviors of the compressor input power with various charge amounts of R-600a. The input power of the compressor with R-600a is lower than that with R-12 by 10% in spite of the larger displacement compressor. Figure 9 shows the discharge temperature of the compressor for various charge amounts of R-600a. The discharge temperature for R-600a is lower than R-12 by more than 5℃,

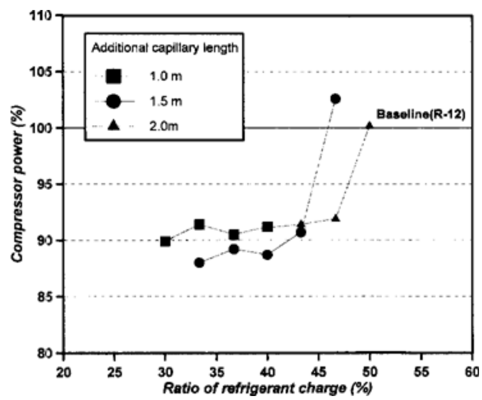

Fig. 8 Compressor input power with various refrigerant charge amounts.

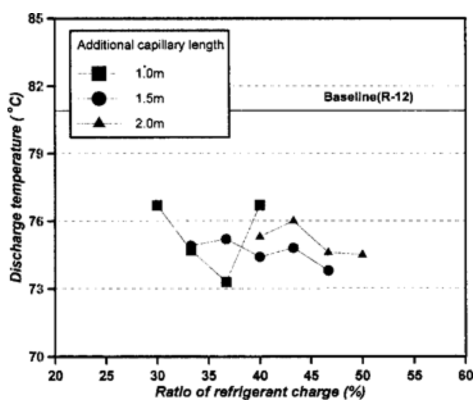

Fig. 9 Compressor discharge temperature with various refrigerant charge amounts.

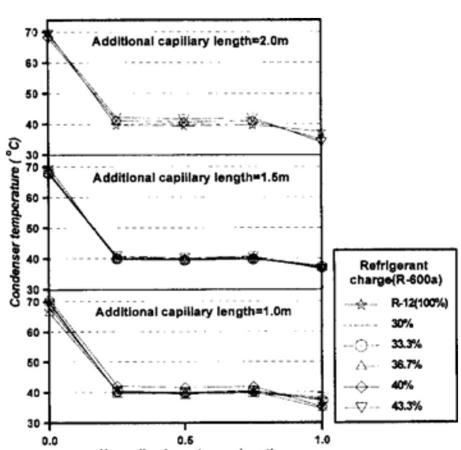

Fig. 10 Condenser wall temperatures for different capillary tubes.

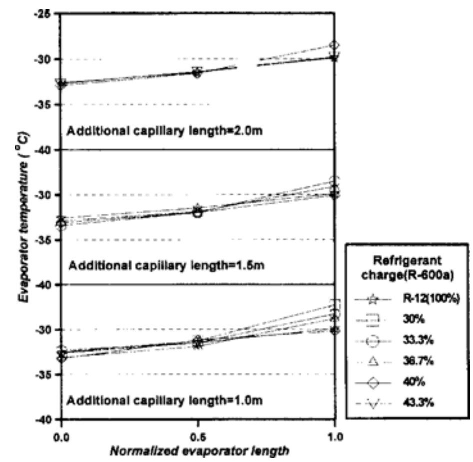

Fig. 11 Evaporator wall temperatures for different capillary tubes.

thereby improving compressor reliability as compared to a R-12 system.

Figures 10 and 11 show the condenser and evaporator wall temperatures at different locations with variation of the refrigerant charge amount. The condenser temperature and the subcooling increase, and the superheat at the evaporator exit decreases, with an increase of the refrigerant charge amount for the same cepillary tube.

5. Conclusions

The following conclusions were made from the theoretical cycle analysis of R-12 and its alterna tive refrigerants and the performance tests with R-12 and isobutane refrigerant R-600a in a household auto-defrost refrigerator/freezer.

(1) The theoretical COP of R-600a is 6% greater than that of R-12, but the compressor displacement must be increased 91% because the VCR for R-600a is 52% of R-12.

(2) The energy savings with R-600a is achieved by 1～11% compared to the baseline R-12 unit, and the compressor on-time ratio for the R-600a unit is equal to or a little bit lower than that for the R-12 unit.

(3) The cooling speed of R-600a is improved by 3～10% compared to R-12, and the compressor reliability could be improved because the discharge temperature with R-600a is 5℃ lower than

that of R-12.

(4) The cyclic losses and safety issues associated with refrigerant flammability can be reduced, because the charge amount of R-600a is only 30 to 50% of R-12.

(5) R-600a is suggested as a good alternative refrigerant to R-12. To apply it to the commercialized auto-defrost refrigerator/freezer, the additional research is needed in the areas of flammability and risk assessments.

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