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ABSTRACT

A dual-loop cycle for a domestic refrigerator-freezer (RF) has a large energy saving potential because of lower compression ratio due to higher evaporating temperature in the fresh food compartment (R)-operation, and the individual optimization flexibility for each loop, compared to a conventional refrigeration cycle. In this study, the optimization of the dual-loop cycle was carried out. The optimization process included both the theoretical analysis and experimental investigation. The theoretical analysis was conducted to optimize the specifications of cycle components for each loop of the dual-loop cycle. Based on the experimental results, the energy consumptions of the optimized dual-loop cycle was decreased by 14.2%, compared with that of a bypass two-circuit cycle in the same RF platform.

1. INTRODUCTION

Recently, various refrigeration cycles designed for domestic refrigerator-freezers (RFs) have been investigated to improve the energy efficiency of the domestic RF. A two-circuit cycle has been considered as a promising solution due to its potential to enhance the energy efficiency and increase the relative humidity of the fresh food compartment (R). Three types of two-circuit cycle have been used in general: a dual-loop cycle, a bypass two-circuit cycle, and a two-circuit cycle with parallel evaporators (known as a parallel cycle). In the bypass two-circuit cycle, the evaporating temperature of the R-evaporator is almost the same as that of the freezer compartment (F)-evaporator. This indicates that the temperature difference between the refrigerant and air in the R-evaporator is relatively large, resulting in large irreversible losses and compression ratios in the R-operation. Therefore, the parallel and dual-loop cycles could be used as alternatives to the conventional refrigeration cycle to improve the system performance by maintaining a higher R-evaporating temperature. Even though the parallel cycle is essentially equivalent to the dual-loop cycle in that it has separate flow paths for the F- and R-operations, it has several weak points. It cannot perform the F-and R-operations simultaneously; therefore, when the F- and R-compartments have to be cooled, each compartment must be cooled alternately. In addition, it requires a refrigerant recovery operation, which causes additional energy consumption, to allow enough refrigerant flow in the R-operation (Mahesh et al., 1998; Yoon et al., 2011). The dual-loop cycle can overcome these problems because two completely separate refrigeration cycles with an additional compressor and condenser can provide cooling for each compartment independently. Generally, the dual-loop cycle shows very high energy saving potential in an RF because each loop can be individually optimized. In addition, the dual-loop cycle can obtain a higher R-evaporating temperature compared to the parallel cycle with the same R-evaporator capacity because it uses smaller compressors with reduced loads. However, the down-sized compressor may cause a loss of energy efficiency because a small compressor is less efficient than a large one. The criteria that determine the optimal design of the parallel and dual-loop cycles may lie on the technical trade-off between the cost and the system performance.

The energy saving potential of the dual-loop cycle increases with the decrease in the compression ratio that occurs due to the higher evaporating temperature in the R-operation. Based on a computer simulation, Bare et al. (1991) reported that a dual-loop cycle with R-12 showed a 19% energy saving over the conventional refrigeration cycle. Pedersen et al. (1986) also predicted a 20% energy saving of a dual-loop cycle with R-12 by using a simulation model. Won et al. (1994) experimentally observed a 4.3% energy saving for a dual-loop cycle over the conventional refrigeration cycle. However, most of the studies on the dual-loop cycle were performed using halocarbon refrigerants, which have ozone depletion potential (ODP) and/or high global warming potential (GWP). This study aims to suggest detailed design guidelines for dual-loop cycles using R-600a by conducting a theoretical design for cycle components. In addition, the performances of dual-loop cycles using R-600a were measured by varying the refrigerant charge and capillary tube length.

2. EXPERIMENTAL SETUP AND TEST CONDITIONS

A household side-by-side (SBS) RF with an internal volume of 0.74 m^3 (740 l) was used in the experiments. The original SBS RF adopted a bypass two-circuit cycle using R-600a. After testing the original RF, the bypass two-circuit cycle was modified to the dual-loop cycle to compare the performance of the dual-loop cycle with that of the bypass two-circuit cycle. Figure 1 shows the schematic diagrams of the bypass two-circuit cycle and the dual-loop cycle. In the bypass two-circuit cycle, a 3-way valve controls the refrigerant flow path for each operation mode. In the R/F simultaneous mode, the refrigerant flows along a serial path through the R-capillary tube and the R- and F-evaporators. When the system operates in the freezer-only mode (F-only mode), the refrigerant passes through the F-capillary tube and then enters the F-evaporator directly. On the other hand, the dual-loop cycle is comprised of two independent conventional single evaporator cycles (F-loop cycle and R-loop cycle). Therefore, the temperature and compressor on-off status for each loop in the dual-loop cycle can be controlled independently.

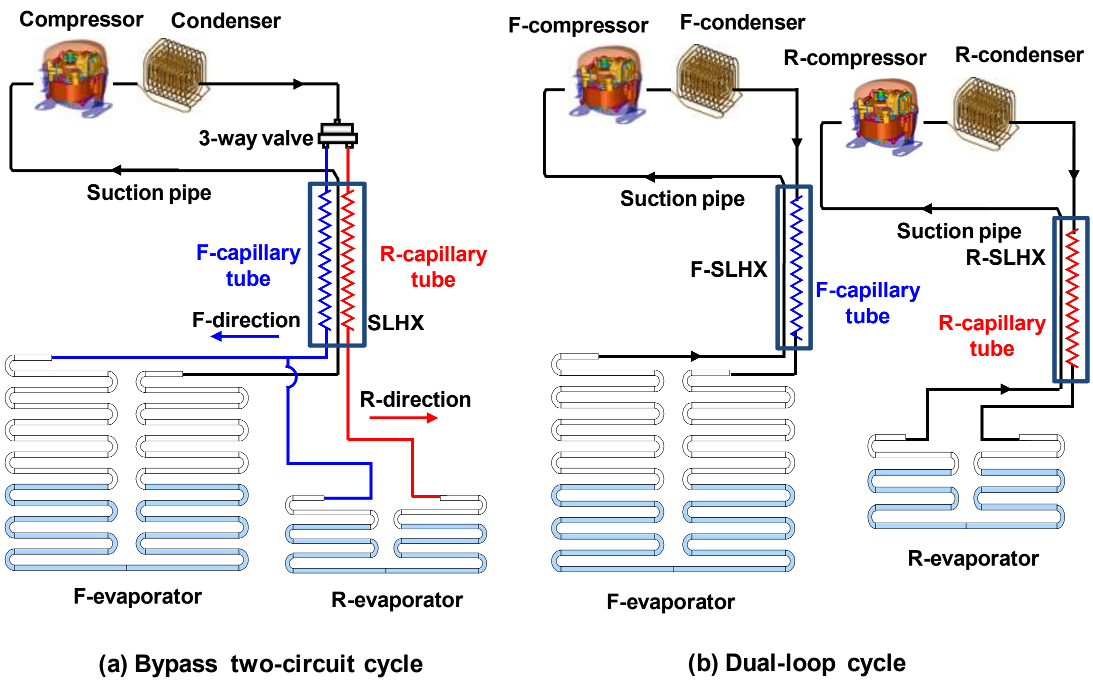

Figure 1: Schematic diagrams of the bypass two-circuit cycle and the dual-loop cycle

Table 1: Specifications of the basic bypass two-circuit cycle

Compressor	Type	Reciprocating, hermetically sealed, low pressure oil sump
	Motor	Inverter-driven BLDC (operation range : 3600~1600 RPM)
	Displacement volume	15.0 cc
	Cooling Capacity	171.5 W at AHAM conditions (40.6, -23.3°C), 1800 RPM 329.1 W at AHAM conditions (40.6, -23.3°C), 3600 RPM
R-Evaporator	Type	Plate fin-tube, aluminum, 1 column 11 rows, forced convection
	Heat transfer area	0.359 m^2
F-Evaporator	Type	Plate fin-tube, aluminum, 2 columns 15 rows, forced convection
	Heat transfer area	1.416 m^2
Condenser	Type	Spiral fin-tube, steal, tube length 16.8 m, forced convection
	Heat transfer area	2.270 m^2
Cycle	Capillary tube	Inner diameter 0.85 mm, length 3300 mm (same for F, R)
	Refrigerant	R-600a, charge amount 96 g
	SLHX length	2000 mm (soldered with F, R capillary tube)
	Control	Programmable micro-computer

In the modification of the original RF, theoretical analysis for the component design was carried out by considering the technical and practical requirements. The dual-loop cycle included an additional compressor, condenser, and suction line heat exchanger (SLHX) as compared to the bypass two-circuit cycle. In addition, the original compressor and condenser were down-sized because of the reduced load in the F-loop cycle, while the R-evaporator was enlarged because of the increase in the R-evaporating temperature. The SLHXs and sub-coolers were installed outside of the RF cabinet with urethane panels so that they could be modified easily. Meanwhile, the RF was controlled by a programmable micro-computer.

The performance of the dual-loop cycle was measured by varying the refrigerant charge amount and the capillary tube length at an ambient temperature of 25°C and a relative humidity of 70% (ISO 15502, 2005, N-class condition). The ambient conditions were controlled by an environmental chamber. The refrigerant charge was varied until the optimum point appeared for each capillary tube length. The capillary tube length was also increased until final cycle matching was achieved. During the tests, the time-averaged air temperatures for the F- and R-compartments were maintained at approximately -18°C and 3°C, respectively. The internal air temperature of the F/R cabinet was determined as an average value of those yielded by three temperature sensors located deep in the center of the compartment and equally spaced in the vertical direction. The cabinet air temperature was controlled by a cyclic operation. In this study, the cut-in (set-on) and cut-out (set-off) temperatures were maintained at a constant value. For all tests, the compressor speed of the RF was maintained at 1800 RPM, which yielded 52% of the full cooling capacity. This condition was appropriate for the energy saving mode operation of the RF. The performance of the RF was measured under the cyclic steady condition, in which the difference between the time-averaged cabinet air temperatures of adjacent cycles was less than 0.2°C. T-type copper-constantan thermocouples were used for the temperature measurements. All thermocouples were calibrated to an accuracy of ±0.2°C in a constant temperature bath.

3. RESULTS AND DISCUSSION

3.1 Optimal Design Guide for the Dual-Loop Cycle

The two compressors of the dual-loop cycle were redesigned to optimize the system performance. As shown in Table 1, the displacement volume of the baseline compressor in the bypass two-circuit cycle was 15 cc. If this volume is simply divided by the heat load ratio of each compartment, the displacement volumes of the F-and R-compressors in the dual-loop cycle should be 9 cc and 6 cc, respectively, because the ratio of the F- and R-heat loads was typically 60:40. However, since the volumetric capacity of the refrigerant varies according to the evaporating temperature, the displacement volume of each compressor has to be adjusted for each operating condition, yielding a constant refrigeration capacity for each cycle (Stoecker and Walukas, 1981).

The volumetric capacity is given by Equation (1),

$$VC = \frac{h_{sat,vapor} - h_{sat,liquid}}{v_{suc}} \tag{1}$$

The evaporating temperature of the bypass two-circuit cycle was about -20°C in the R/F simultaneous mode, which can be regarded as the standard condition for the bypass two-circuit cycle. In the dual-loop cycle, the F-evaporating temperature can be estimated as -27°C by considering the evaporating temperature in the F-only mode of the bypass two-circuit cycle (Table 2). On the other hand, the R-evaporating temperature has to be determined as a target design condition. Although a higher evaporating temperature allows a larger energy saving potential, it requires a smaller compressor and/or a larger evaporator capacity. Therefore, the upper limit of the R-evaporating temperature has to be determined by the possible ranges of the compressor and evaporator capacities. In this study, the target R-evaporating temperature was determined as -10°C by considering the above two factors. As shown in Figure 2, at evaporating temperatures of -27°C and -10°C, the volumetric capacity of R-600a was decreased by 25% and increased by 48%, respectively, compared with that at the evaporating temperature of -20°C. The optimum displacement volumes of the F- and R-compressors were estimated at 12 cc and 4 cc, respectively, by considering these effects. However, the smallest available displacement volume was 4.8 cc among R-600a compressors. Therefore, the displacement volumes for the F-and R-compressors were chosen as 12 cc and 4.8 cc, respectively. It should be noted that the energy efficiency of the F-compressor was the same as the baseline, while the R-compressor had a 9.1% lower efficiency at 1800 RPM.

The machine room for the dual-loop cycle should be redesigned to contain two compressors and two condensers. Figure 3 shows the R-condenser attached onto the RF cabinet and the redesigned arrangement of the machine room. The R-condenser with a length of 11.3 m was installed in the RF cabinet wall so that it can utilize the steel cabinet wall as a heat transfer area. In addition, the capacity of the F-condenser was reduced by about 40% corresponding to the decrease in the heat load, resulting in a smaller occupied volume.

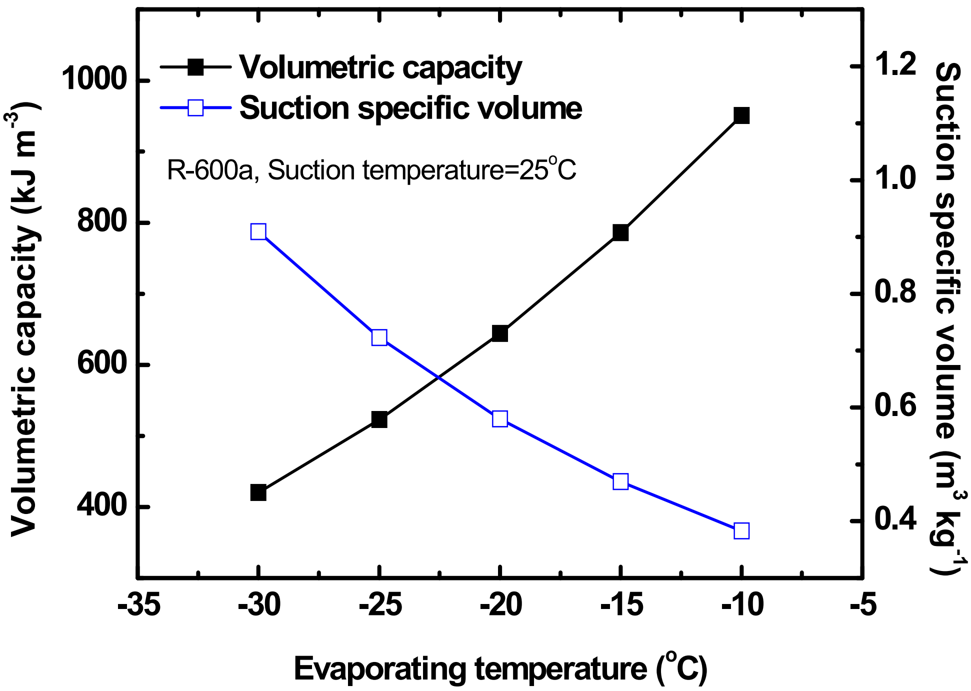

Figure 2: Variations of the volumetric capacity and suction specific volume of R-600a

As mentioned before, the dual-loop cycle requires a larger heat transfer area for the R-evaporator due to the smaller temperature difference between the refrigerant and air. The energy balance without latent cooling for a heat exchanger is given by Equation (2),

$$\dot{Q} = UA\Delta T_{LMTD} = \dot{m}_{ref}\Delta h_{ref} = \dot{m}_{air}C_{P,air}(T_{air,in} - T_{air,out}) \tag{2}$$

Therefore, the heat transfer area of a heat exchanger can be expressed by Equation (3).

$$A = \frac{\dot{m}_{ref}\Delta h_{ref}}{U\Delta T_{LMTD}} \tag{3}$$

For the R-evaporator of the dual-loop cycle, the effect of the volumetric capacity, expressed as the numerator in Equation (3), can be mostly compensated by the down-sizing of the R-compressor. Therefore, the rate of increase of the heat transfer area can be determined by the reduced temperature difference between the refrigerant and air (ΔT_{LMTD}) only. On the other hand, for the R-evaporator of the parallel cycle, the heat transfer area should be enlarged to take into account both effects, which requires that a much larger heat transfer area is used. On the other hand, for the R-evaporator of the parallel cycle, the heat transfer area should be enlarged to take into account both effects, which requires that a much larger heat transfer area is used.

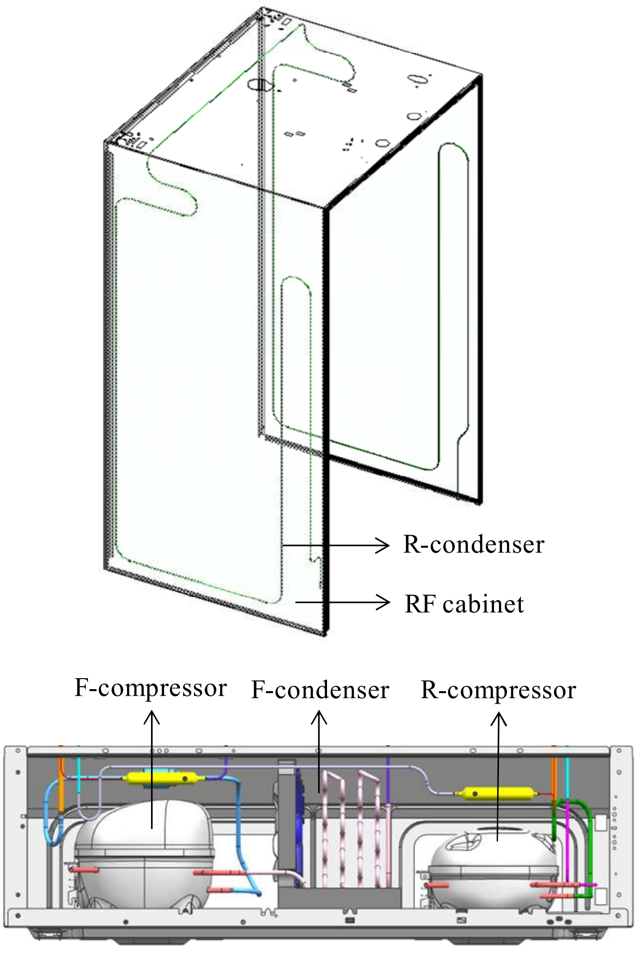

Figure 3: R-condenser attached onto the RF cabinet and the redesigned machine room

Yoon et al. (2011) reported that the heat transfer area of the R-evaporator in the parallel cycle should be enlarged by 107% and 212% to increase the evaporating temperature from -20°C to -15°C and -10°C, respectively. However, the dual-loop cycle requires only a 77% increase of the heat transfer area in order to obtain the evaporating temperature of -10°C. This required increasing rate is even lower than that of the parallel cycle for the R-evaporating temperature of -15°C. Therefore, the R-evaporating temperature of the dual-loop cycle can be at least 5°C higher than that of the parallel cycle with the same heat transfer area. In this study, the heat transfer area of the R-evaporator was increased by 100%, which was 23% larger than the estimated value. The additional increase may compensate for the fact that the capacity of the R-compressor is larger than the optimum value. The F-evaporator of the bypass two-circuit cycle was kept and an additional SLHX having the same specification as the baseline was added for the R-loop cycle.

3.2 Experimental Results

Figure 4 represents the effects of the refrigerant charge and the capillary tube length on the energy consumption of the F- and R-loop cycles using R-600a. The optimum refrigerant charge for each capillary tube length was determined by the minimum energy consumption in each case. For each cycle, the optimum refrigerant charge increased with the increase in the capillary tube length. As the capillary tube length was increased with a fixed refrigerant charge, the evaporating temperature decreased due to more refrigerant accumulation in the condenser, which caused a decrease in the refrigerant flow rate and the cooling capacity. The decrease in the cooling capacity due to the longer capillary tube has to be compensated by an increase in the refrigerant charge. The optimum capillary tube lengths of the F-and R-loop cycles were 4.8 m and 5.8 m, respectively, with the same internal diameter as the baseline. These optimum capillary tube lengths were much longer than those of the bypass two-circuit cycle because the refrigerant flow rate through each cycle was decreased by the down-sizing of the compressor.

In the R-loop cycle, despite the increase in the evaporating temperature, the refrigerant flow rate through the down-sized compressor was decreased by 50.7%, based on the estimation made using Equation (4).

$$\dot{m}_{ref} = [1 + C - C(\frac{P_{cond}}{P_{evap}})^{\frac{1}{n}}]\frac{PD}{v_{suc}} \tag{4}$$

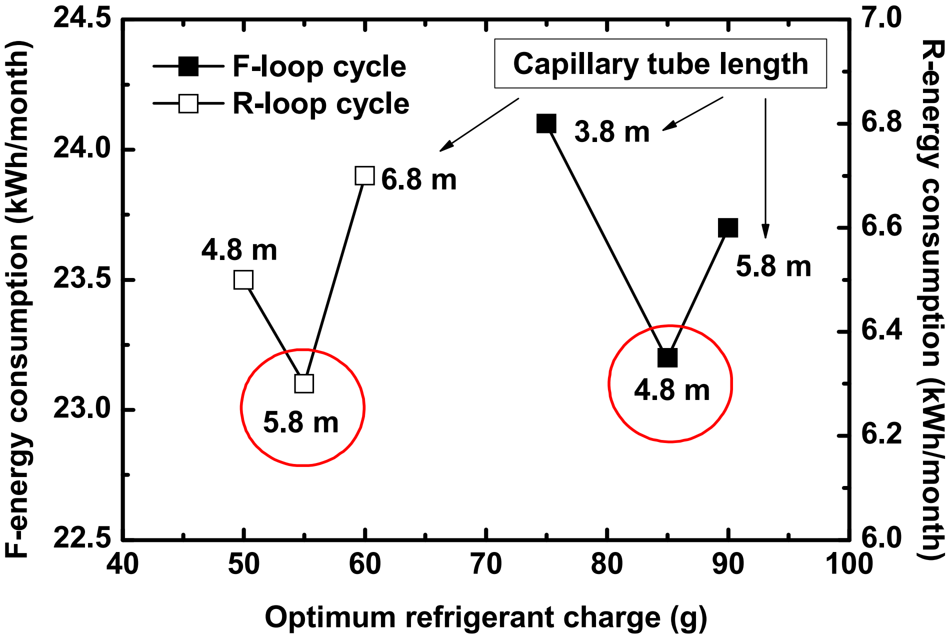

Figure 4: R-condenser attached onto the RF cabinet and the redesigned machine room

Table 2: Performance comparison between the dual-loop cycle and the bypass two-circuit cycle

Parameters	Dual-loop cycle		Bypass two-circuit cycle	
	R-loop cycle	F-loop cycle	R/F-operation	F-operation
Operating ratio	30.2%	54.9%	23.5%	43.1%
Averaged input power	25.8W	54.8W	74.4W	65.3W
Averaged evaporating temperature	-9.2°C	-28.0°C	-19.1°C	-27.7°C
Averaged condensing temperature	30.5°C	29.1°C	30.8°C	30.2°C
Total energy consumption	29.5 kWh/month		34.4 kWh/month	

In Equation (4), *C* is the clearance volume ratio and *PD* is the displacement rate of the compressor. According to the generalized capillary tube correlation proposed by Wolf and Pate (2002), the increase of the evaporating temperature from -20°C to -10°C decreased the refrigerant flow rate through the capillary tube by only 14% in the two-phase inlet condition. Therefore, the capillary tube length has to be increased to yield the estimated reduction of the refrigerant flow rate. On the other hand, in the parallel cycle, the R-capillary tube length has to be decreased with the increase in the R-evaporating temperature because of the increased refrigerant flow rate through the baseline compressor (Yoon et al., 2011).

Table 2 summarizes the performance of the dual-loop cycle and the bypass two-circuit cycle using R-600a at the optimum specifications. The total energy consumption of the optimized dual-loop cycle using R-600a was 14.2% lower than that of the bypass two-circuit cycle. In the dual-loop cycle, the R-evaporating temperature of -9.2°C was similar to the target design condition, which was about 10°C higher than that of the bypass two-circuit cycle. This yielded an approximately 33% lower compression ratio, resulting in much higher energy efficiency in the R-operation. In addition, in the bypass two-circuit cycle, the F-only mode operated with an over-sized compressor and condenser capacity because these components were designed based on the operating conditions of the R/F simultaneous mode. Therefore, in the dual-loop cycle, the down-sized F-compressor yielded a decrease in the energy consumption during the F-operation. As shown in Table 2, the condensing temperature in the F-loop of the dual-loop cycle was 1.1°C lower than that in the F-only mode of the bypass two-circuit cycle, in spite of the smaller condenser capacity.

4. CONCLUSIONS

In this study, the optimal design of the dual-loop cycle using R-600a was conducted by a theoretical analysis of the cycle components. As a result, the capacities of the F- and R-compressors in the dual-loop cycle were down-sized by 20% and 72%, respectively. In addition, the heat transfer area of the R-evaporator was enlarged by 77% with the optimized R-compressor to compensate the decrease in the temperature difference between refrigerant and air. The performances of dual-loop cycles using R-600a were also measured by varying the refrigerant charge, capillary tube length, and mixture composition. In the dual-loop cycle, the F-and R-capillary tube lengths have to be increased to yield the estimated reduction of the refrigerant flow rate by compressor down-sizing. The total energy efficiency of the optimized dual-loop cycle using R-600a was improved by 14.2% over that of the bypass two-circuit cycle.

NOMENCLATURE

A	heat transfer area	(m^2)		**Subscripts**
C	clearance volume ratio	(–)	*air*	air
C_p	constant pressure specific heat	(J/kg K)	*cond*	condenser
F	freezer compartment	(–)	*evap*	evaporator
h	enthalpy	(J/kg)	*in*	inlet
$\dot{m}$	mass flow rate	(kg/s)	*LMTD*	log mean temperature difference
P	pressure	(Pa)	*out*	outlet
PD	displacement rate	(m^3/s)	*ref*	refrigerant

Q	heat transfer rate	(W)	*sat,liquid*	saturated liquid state
R	fresh food compartment	(–)	*sat,vapor*	saturated vapor state
RF	refrigerator-freezer	(–)	suc	suction
T	temperature	(^oC)		
U	overall heat transfer coefficient	($W/m^2 K$)		
v	specific volume (m^3/kg)			
VC	volumetric capacity (J/m^3)			

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