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ABSTRACT

A two-circuit cycle with parallel evaporators (called as "parallel cycle") for a domestic refrigerator-freezer (RF) shows energy saving potential comparing with a conventional cycle with single loop or serial evaporators because of low compression ratio and pressure drop in evaporator during fresh food compartment (R)-operation. Therefore, several home appliance companies adopted this cycle for their household RFs. This study presents the performance of the parallel cycle according to design parameters such as refrigerant charge, R-capillary tube diameter, R-evaporator, and air flow rate. The experiments were conducted in a side-by-side (SBS) RF using R-600a under no load condition at the ambient temperature of 25^{o}C. A quantitative analysis for above design parameters was performed. R-evaporating temperature and refrigerant mass flow rate increased by increasing R-capillary tube diameter from 0.85 mm to 1.4 mm, resulting in the decrease of the compression ratio. Therefore, energy consumption of the optimized parallel cycle was reduced by 7.8% over that of the bypass two-circuit cycle with the same RF platform.

1. INTRODUCTION

Recently, the adoption of a two-circuit cycle for a domestic RF is increasing due to its enhancement potential of the energy efficiency and relative humidity of fresh food compartment (R). Three types of the two-circuit cycle have been used in general: a dual loop cycle with two compressors, a bypass two-circuit cycle, and a two-circuit cycle with parallel evaporators (parallel cycle). Generally, the dual loop cycle needs more operating cost than other two-circuit cycles because it uses two compressors. For the bypass two-circuit cycle, the evaporating temperature of the R-evaporator is almost the same as that of the freezer compartment (F)-evaporator. It means that the temperature difference between the refrigerant and air in the R-evaporator is relatively large, resulting in large irreversible losses and compression ratio in R-operation. The parallel cycle would be a solution to overcome these shortcomings of the above-mentioned cycle (Lu and Ding, 2006). The parallel cycle has been applied by several home appliance companies for their household RFs.

The energy saving potential of the parallel cycle mainly became large at low compression ratio resulted from high evaporating temperature and low pressure drop in the evaporator during R-operation. Kim *et al*. (1995) and Mahesh *et al*. (1998) reported that the parallel cycle showed 2.3~8.5% higher efficiency than the two-evaporator cycle in series (serial cycle). The resistance of the R-capillary tube should be decreased to maximize the energy efficiency of the parallel cycle. However, it causes the increase of refrigerant mass flow rate and the decrease of temperature difference between refrigerant and air in the R-evaporator. Therefore, the heat transfer area and air flow rate in the R-evaporator should be modified.

In this study, the heat transfer area and air flow rate of the R-evaporator were theoretically redesigned by the cycle analysis based on the specification of the bypass two-circuit cycle or serial cycle. The modification of the aforementioned two design parameters is the essential for the performance of the parallel cycle. Therefore, only after their specifications were fixed, the cycle optimization according to refrigerant charge and R-capillary tube diameter was performed experimentally. The R-capillary tube diameter was increased to decrease the flow resistance in the R-capillary tube because this method did not include any change in the length of suction line heat exchanger (SLHX). Based on the experiments, optimum refrigerant charge and R-capillary tube diameter were selected and then design guidelines for the parallel cycle were proposed.

2. EXPERIMENTAL SETUP AND TEST CONDITIONS

Experiments were conducted in a household SBS RF using R-600a. The internal volume of the RF was 0.68 m^3 (680 liters, 24.0 ft^3). The bypass two-circuit cycle was originally adopted for the SBS RF. After performance evaluation of the original bypass two-circuit cycle, the RF was modified into the parallel cycle to compare the performance of the parallel cycle with that of the bypass two-circuit cycle. Figure 1 shows the schematic diagram of the parallel and bypass two-circuit cycles. R-operation in the parallel cycle was performed alone, while that in the bypass two-circuit cycle was conducted along with the early stage of F-operation. After the R-operation of each cycle, the F-operation was carried out alone by the same manner. Table 1 lists specifications of the basic bypass two-circuit cycle.

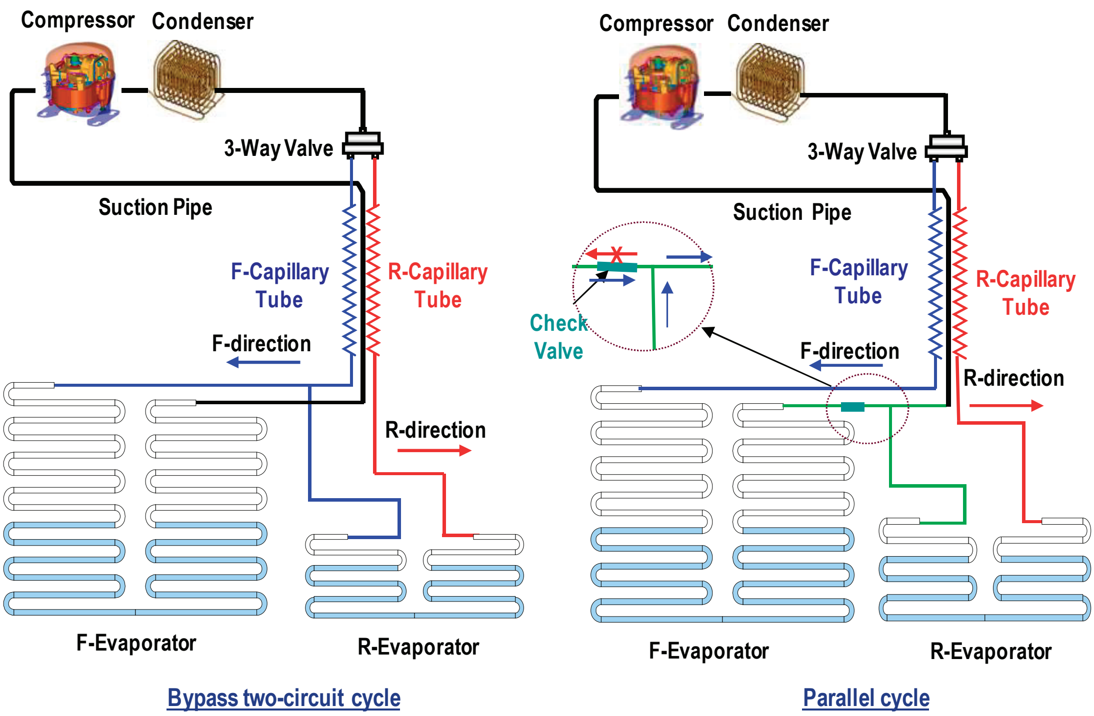

Figure 1: Schematic diagram of the bypass two-circuit cycle and parallel cycle

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	1.626 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 80 g
	SLHX length	2000 mm (soldered with F, R capillary tube)
	Control	Programmable micro-computer

To convert the bypass two-circuit cycle into the parallel cycle, the control logic and SLHX part of the RF should be modified. The SLHX part contains a suction pipe, F/R-capillary tube, and connecting components between the F- and R-evaporator. To analyze and compare the performance of each cycle, the capillary tube also should be changed. Therefore, the SLHX part was installed the outside of the RF cabinet and insulated by urethane panels carefully. Meanwhile, the control of the RF was achieved by a programmable micro-computer.

The experiments were carried out at the ambient temperature of 25°C and relative humidity of 70% (ISO 15502, N-class condition). The ambient condition was maintained by a psychometric chamber. The cabinet internal air temperature of F/R was regarded as an averaged value over 3 positions for each compartment. The above 3 positions were located in the center of the compartment in depth, and equally spaced in height (1/4H, 1/2H, 3/4H of the compartment). All cabinet internal air temperatures were monitored and recorded continuously every 30 sec. Each time-averaged cabinet air temperature of F- and R-compartment was controlled approximately -18°C and 3°C, respectively, for constant experimental conditions. The cabinet air temperature was controlled by intermittent run or cycling control. In this study, the difference in cycle time was less than 5% in all experimental cases and the target value was 60 min to reduce the effect of the cycle time. To achieve this objective, cut-in (set-on) and cut-out (set-off) temperatures were adjusted precisely by the programmable micro-computer. The compressor of the RF was operated at 1800 RPM in all experimental cases. Under the above conditions, the cooling capacity of the compressor was decreased by 48% from the full capacity, which is appropriate condition for the energy saving mode operation in normal thermal-load ambient condition.

The energy consumption of the RF was measured by a power meter with an uncertainty of ±0.2% and integrated for four cycles. The performance of the RF was investigated at steady cycling condition, which was defined that all time-averaged cabinet air temperatures of adjacent cycles were closer than 0.2°C. T-type copper-constantan thermocouples were used for temperature measurement. All thermocouples were calibrated in a constant temperature bath with an accuracy of ±0.2°C.

3. RESULTS AND DISCUSSION

3.1 Theoretical Design of R-evaporator

Redesign of the heat transfer area and air flow rate in the R-evaporator is essential to enhance energy saving potential in the parallel cycle. The parallel cycle shows relatively higher evaporating temperature and refrigerant mass flow rate in R-operation over the bypass two-circuit cycle or serial cycle. Therefore, theoretical design for the heat transfer area and air flow rate was performed for the parallel cycle based on the bypass two-circuit cycle specifications. Table 2 shows input conditions. In addition, the following assumptions were applied in the cycle analysis: 1) The pressure drops in the evaporator and condenser are neglected, 2) When the F- and R-compartments of the bypass two-circuit cycle operate simultaneously, the ratio of refrigerating effects of F- and R-evaporators is 30:70, 3) Condenser outlet is in saturated liquid state, and 4) The ambient and cabinet internal air temperatures are the same with those of the experiments.

The energy balance for the R-evaporator is given by equation (1).

$$Q = UA_{R,evap}\Delta T_{LMTD} = \dot{m}_{ref}\,\Delta h_R = \dot{m}_{air}\,C_{P,air}(T_{air,in} - T_{air,out}) \tag{1}$$

The heat transfer area and air flow rate of the R-evaporator can be expressed by equations (2) and (3).

$$A_{R,evap} = \frac{\dot{m}_{ref}\,\Delta h_{R,evap}}{U_{R,evap}\Delta T_{LMTD}} \tag{2}$$

$$\dot{m}_{air} = \frac{\dot{m}_{ref}\,\Delta h_{R,evap}}{C_{p,air}(T_{air,in} - T_{air,out})} \tag{3}$$

Where $U_{R,evap}$ and $C_{p,air}$ are regarded as constant, and $\dot{m}_{ref}$ and ΔT_{LMTD} are calculated by equations (4) and (5), respectively.

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

$$\Delta T_{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \qquad (\Delta T_1 = T_{air,in} - T_{ref,out}\text{ , } \Delta T_2 = T_{air,out} - T_{ref,in}) \tag{5}$$

Where C is the clearance volume ratio and PD is the displacement rate of the compressor. The modified heat transfer area and air flow rate for the parallel cycle should compensate the increased refrigerant cooling potential and the decreased temperature difference between refrigerant and air. Based on the cycle analysis, the design parameters were determined in the form of magnification ratio over that of the bypass two-circuit cycle. Table 3 shows the results of the cycle analysis. In the respect of system performance, the bypass two-circuit cycle and the serial cycle showed the same specifications of the R-evaporator regarding the design parameters. Therefore, the results of this analysis can be applicable to the serial cycle.

Table 2: Input conditions for the cycle analysis

Parameters	Value specified	Technical ground for the value
Inlet air temperature of R-evaporator	3°C	Internal cabinet air temperature
Outlet air temperature of R-evaporator	5°C above R evaporating temperature	Measured data for the bypass two-circuit cycle
Condensing temperature	30°C	Measured data for the bypass two-circuit cycle at 1800 RPM
Evaporating temperature in R-operation	-20°C (bypass two-circuit cycle)	Measured data for the bypass two-circuit cycle in R-operation
	-15°C, -10°C (parallel cycle)	Target design conditions
Suction temperature	25°C	Anti-sweat condition for suction pipe
Clearance volume ratio (*C*)	0.02	Estimated value for R-600a

Table 3: Results of the cycle analysis

R-evaporating temperature	P_{cond} / P_{evap}	$A_{R,evap,parallel} / A_{R,evap,bypass}$	$\dot{m}_{air,parallel} / \dot{m}_{air,bypass}$
-20°C (Bypass two circuit cycle)	5.57	Base	Base
-15°C (Parallel cycle)	4.53	2.07	2.47
-10°C (Parallel cycle)	3.71	3.12	4.98

To increase the evaporating temperature from -20°C to -15°C, the heat transfer area and air flow rate of the parallel cycle should be increased by 107 and 147% over those of the bypass two-circuit cycle, respectively. From the practical point of view, it is very difficult to obtain the evaporating temperature of -10°C in the parallel cycle because the air flow rate should be increased by 398% over the bypass two-circuit cycle or serial cycle. Therefore, the design target for the R-evaporating temperature was chosen as -15°C. Fortunately, a compact R-evaporator with smaller fin pitches can be applied into the parallel cycle because a forced convection defrosting process has been used in the R-evaporator. The forced convection defrosting can be achieved by internal air circulation through the R-evaporator when the refrigerant does not pass through the R-evaporator tube. The R-evaporator, fan, and air duct of the bypass two-circuit cycle were modified based on the results of the cycle analysis. Figure 2 shows the R-evaporator for each cycle. The heat transfer area of the redesigned R-evaporator became 2.5 times larger than that of the bypass two-circuit cycle, and the air flow rate was increased from 0.38 to 0.87 m^3/min (CMM) by modifying the fan and air duct.

3.2 Effects of Refrigerant Charge and Capillary Tube Diameter

The experiments were conducted to measure the performance of the modified SBS RF by varying refrigerant charge amount and R-capillary tube diameter. The parallel cycle needs refrigerant recovery operation before R-operation, which is called pump-down (P/D) operation. The refrigerant in the F-evaporator should be recovered to allow enough refrigerant flow through the R-evaporator by operating the compressor without having any flow through the 3-way valve. The operation of the RF was controlled by the following sequence: R-operation, F-operation, compressor off, and P/D operation. The P/D operation time was approximately 120 sec. Generally, when the refrigerant in the F-evaporator is fully recovered during the P/D operation, the suction temperature approaches the ambient temperature. Therefore, the P/D operation time was determined by monitoring the suction temperature.

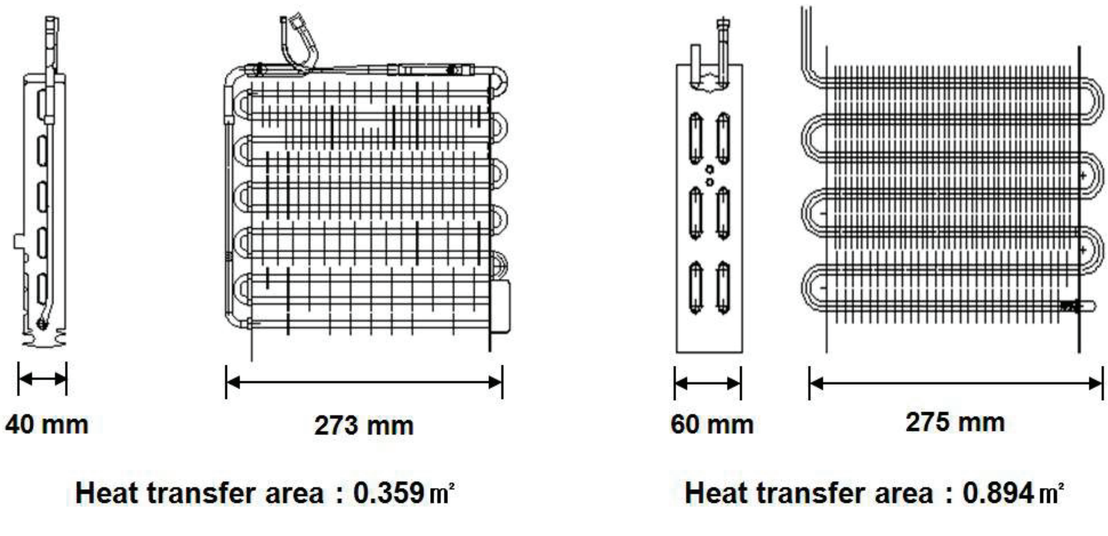

Figure 2: R-evaporator for (a) the bypass two-circuit cycle, and (b) the parallel cycle

Figure 3 represents the effect of refrigerant charge on the energy consumption of the parallel cycle. The optimum refrigerant charge decreased with the increase of R-capillary tube diameter because the increase of the R-capillary tube diameter leaded to the rise of refrigerant flow rate. Therefore, it was possible to obtain appropriate cooling capacity in R-operation even though refrigerant charge was lowered. In addition, the increase of the R-capillary tube diameter can lead to have lower compression ratio because of a rise in the evaporating temperature and drop in the condensing temperature. As shown in Figure 4, the evaporating temperature increased with the increase of the R-capillary tube diameter. The optimum R-capillary tube diameter of the parallel cycle was 1.4 mm, which was 65% larger than that of the bypass two-circuit cycle in the same RF platform. The optimum refrigerant charge in the parallel cycle was 75 g, which was similar to the optimum value of 80 g in the bypass two-circuit cycle. When the parallel cycle was optimized in terms of refrigerant charge and R-capillary tube diameter, the energy consumption was reduced by 7.8%, and the R-evaporating temperature was increased by 4.6°C over the bypass two-circuit cycle. Table 4 summarizes the performances of the parallel cycle and the bypass two-circuit cycle.

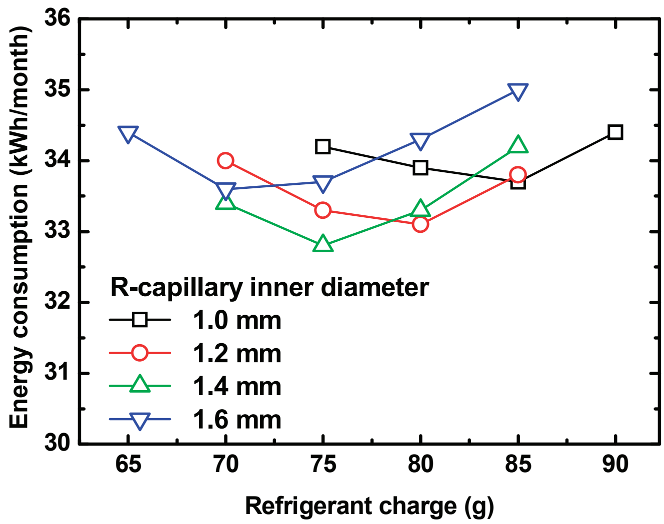

Figure 3: Variation of energy consumption with refrigerant charge and R-capillary tube diameter

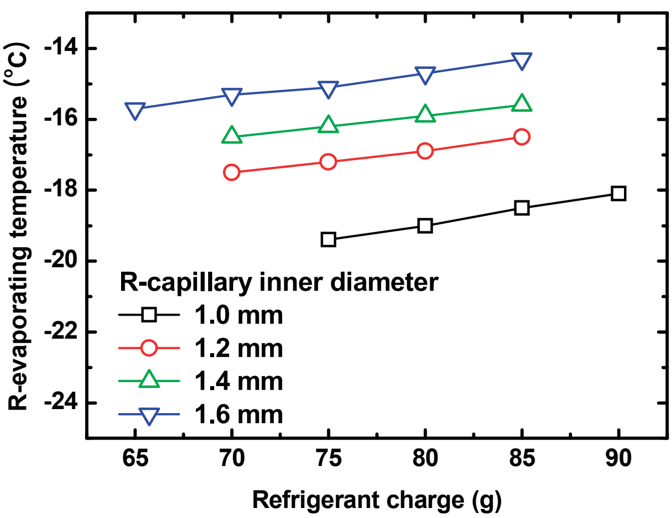

Figure 4: Variation of R-evaporating temperature with refrigerant charge and R-capillary tube diameter

Table 4: Performance comparison between the parallel cycle and the bypass two-circuit cycle

Parameters	Parallel cycle		Bypass two-circuit cycle	
	R-operation	F-operation	R/F-operation	F-operation
Operating ratio	12.2%	53.0%	20.7%	48.8%
Averaged input power	76.0W	61.4W	75.8W	62.8W
Averaged evaporating temperature	-16.2°C	-26.1°C	-20.8°C	-27.0°C
Averaged condensing temperature	31.3°C	29.1°C	31.2°C	29.4°C
Total energy consumption	32.8 kWh/month		35.5 kWh/month	

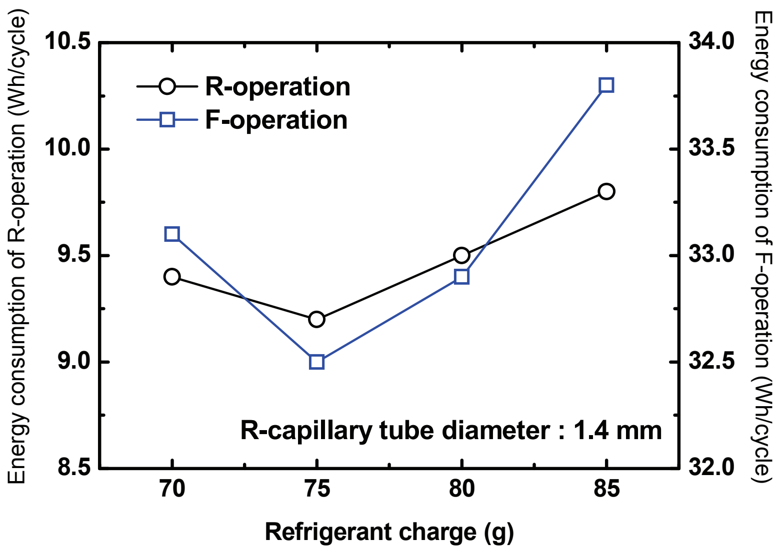

Figure 5: Variation of energy consumption per cycle with refrigerant charge

The optimum refrigerant charge decreased according to the increase in the R-capillary tube diameter. Therefore, the larger R-capillary tube diameter over a certain level causes the difference in the optimum refrigerant charge between F- and R-operation. Figure 5 represents the effect of refrigerant charge on the energy consumption of F- and R-operation per cycle in the optimum R-capillary tube specification. These results show that the modified parallel cycle has the same optimum refrigerant charge for F- and R-operation. Generally, the optimization of refrigerant charge in F-operation is more important in the respect of total energy efficiency because the energy consumption of F-operation is much greater than that of R-operation. Therefore, when the optimum charge for F-operation is relatively larger, it is recommended to increase internal volume and/or heat transfer area of the R-evaporator.

4. CONCLUSIONS

In this study, the heat transfer area and air flow rate of the R-evaporator in the parallel cycle were designed by modifying the specification of the bypass two-circuit cycle or the serial cycle based on the simple cycle analysis. The performance of the parallel cycle was measured and analyzed by varying refrigerant charge and R-capillary tube diameter. In addition, the performance of the optimized parallel cycle was compared with that of the bypass two-circuit cycle. To increase the evaporating temperature from -20°C to -15°C, the heat transfer area and air flow rate of the parallel cycle should be increased by 107 and 147% over those of the bypass two-circuit cycle, respectively. The optimum refrigerant charge in the parallel cycle decreased with the increase of R-capillary tube diameter because the increase of the R-capillary tube diameter leaded to the rise of refrigerant flow rate. In the parallel cycle, the optimum R-capillary tube diameter was 1.4 mm, which was 65% larger than that of the bypass two-circuit cycle in the same RF platform. The optimum refrigerant charge in the parallel cycle was 75 g, which was similar to the optimum value of 80 g in the bypass two-circuit cycle. When the parallel cycle was optimized in terms of refrigerant charge and R-capillary tube diameter, the energy consumption was reduced by 7.8%, and the R-evaporating temperature was increased by 4.6°C over the bypass two-circuit cycle.

NOMENCLATURE

A	heat transfer area	(m^2)
C	clearance volume ratio	(–)
C_p	constant pressure specific heat	(J/kg K)
F	freezer compartment	(–)
h	enthalpy	(J/kg)
$\dot{m}$	mass flow rate	(kg/s)
P	pressure	(Pa)
PD	displacement rate	(m^3/s)
Q	heat transfer rate	(W)
R	fresh food compartment	(–)
RF	refrigerator-freezer	(–)
T	temperature	(°C)
U	overall heat transfer coefficient	(W/m^2 K)
v	specific volume	(m^3/kg)

Subscripts

air	air
bypass	bypass two-circuit cycle
evap	evaporator
in	inlet
LMTD	log mean temperature difference
out	outlet
parallel	parallel cycle
R	fresh food compartment
ref	refrigerant

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