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Performance Characteristics of a Household Refrigerator with Dual Evaporators Using Two-Stage Compression Cycle

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Abstract

The objective of this study is to investigate performance characteristics of a household refrigerator using a two-stage compression cycle. The performance of the two-stage compression cycle was measured by varying the compressor speed, condensing temperature, and evaporating temperature. The COP of the two-stage compression cycle was analyzed and then compared with that of the single-stage compression cycle. The optimum combination of compressor speeds for a low- and a high-stage was determined. The COP of the two-stage compression cycle using a PTC (parallel two-stage compression) method was 5.85% higher than that of a STC (serial two-stage compression) method at optimum operating conditions.

Key words: Two-stage compression, Household refrigerator, R600a, Compressor speed

Nomenclature

C_p : specific heat [J/kg℃]
COP : coefficient of performance [-]
h : enthalpy [kJ/kg]
$\dot{m}$: mass flow rate [kg/h]
W : compressor power consumption [W]

Subscript

2nd : secondary fluid
cond : condenser
evap : evaporator
F : freezer compartment
HP : high pressure
i : inlet
LP : low pressure
o : outlet
r : refrigerant
R : fresh food compartment

1. Introduction

Most household refrigerators use a single-stage compression cycle including a compressor and an evaporator. In the single-stage compression cycle, the volumetric efficiency of the compressor reduces with the decrease of evaporating temperature because of the increase in compression ratio. It is one of serious problems in compressor reliability due to higher discharge temperature.(1) Recently many household refrigerators start to adopt dual evaporators for R(refrigerator)- and F(freezer)-compartment, respectively, using the single-stage compression cycle. In order to control evaporating temperatures for the R- and F-evaporator separately, it is essential to use a compressor with high efficiency and large capacity. In addition, the volumetric efficiency of the compressor becomes low at low evaporating temperature of the F-evaporator. The two-stage compression cycle is expected to have high efficiency at high compression ratio.(2) And a two-stage compression cycle with dual evaporators has been introduced in household refrigerators to enhance compressor efficiency and reduce cycle losses.

There are a few numerical studies on performance

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characteristics for household refrigerators having dual evaporators. Zubair et al. conducted second-law-based thermodynamic analysis on two-stage and mechanical- subcooling refrigeration cycles.[3] Khan et al. studied thermodynamic optimization of finite time vapor-compression refrigeration systems.[4] Shapiro and Rohrer simulated the performance of a two-stage linear compressor with an economizer cycle.[5] They reported the optimum volumetric ratio according to the evaporating and condensing temperature. However, there are few experimental studies on performance characteristics of a household refrigerator with dual evaporators using the two-stage compression cycle.

The objective of this paper is to investigate performance characteristics of the two-stage compression cycle for household refrigerators having dual evaporators. The COP and refrigerating capacity of the two-stage compression cycle were measured by varying the compressor speed, condensing temperature, and evaporating temperature. In addition, the performance of the two-stage compression cycle with two compressor arrangements, which are the STC (serial two-stage compression) and the PTC (parallel two-stage compression) method, was compared with that of the single-stage compression cycle.

2. Experimental setup and test conditions

Fig. 1 shows the schematic diagram of the experimental setup. The test setup consisted of a test section and three constant temperature chillers. The test section had a low-stage compressor, a high-stage compressor, a condenser, a receiver, two needle valves for expansion process, a R-evaporator for a refrigerating compartment and a F-evaporator for a freezing compartment. R600a was used as a refrigerant circulating through the test section. Ethylene glycol-water mixture (50:50) was used as a secondary fluid of constant temperature chillers for the R- and F- evaporators and condenser. The flow rate of the secondary fluid was adjusted by controlling frequency of an inverter-driven pump. The temperature of the secondary fluid was controlled by a constant temperature chiller. In the tests of the single-stage compression cycle, the low-stage compressor and F-evaporator were used. Table 1 shows the specification of the test setup.

Fig. 2 shows the schematic diagram of compressor arrangements for the STC and PTC methods. The STC and PTC methods have serial- and parallel-arrangements of two compressors, respectively. Tables 2 and 3 show the test conditions used in this study. The performance of the two-stage compression cycle using the STC and PTC methods (called as "STC system" and "PTC system", respectively) was measured at standard conditions as shown in Table 2 to determine optimum combination of low- and high-stage compressor speeds. As one of two compressors was operated at a constant speed, the other compressor was tested with variable speed. In addition, the performance of the tested system was measured by varying the condensing temperature and evaporating temperatures of two evaporators. The condensing temperature was controlled using the secondary fluid.

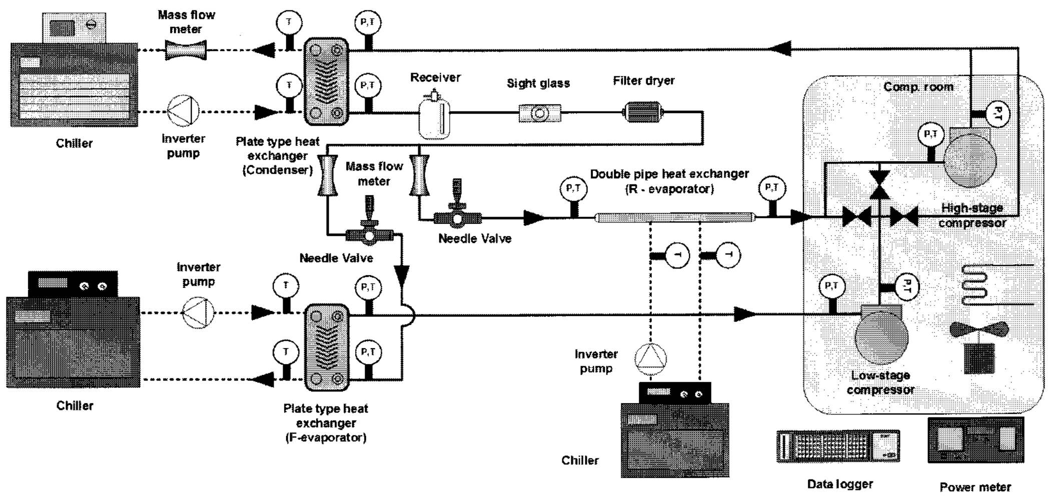


Fig. 1. Schematic diagram of experimental setup.

Table 1. Specifications of the test setup.

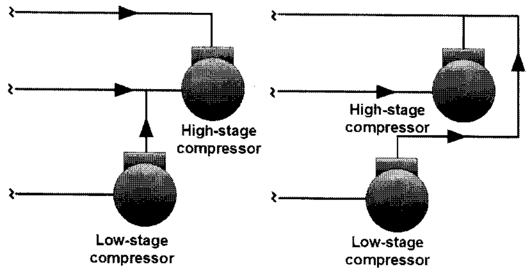
Component	Specification
Refrigerant	R600a
Compressor	Reciprocating inverter compressor, Displacement volume=15 cc
Condenser	Plate type heat exchanger, Capacity=1 kW
Expansion device	Metering valve, Orifice size = 1.19 mm, C_v = 0.024
R-evaporator	Double pipe heat exchanger, Inner: 9.52 mm(OD)×0.5 mm(T)×1500 mm(L) Outer: 12.5 mm(OD)×0.5 mm(T)×1500 mm(L)
F-evaporator	Plate type heat exchanger, Capacity=0.5 kW

Table 2. Test conditions.

Parameter	Standard (Uncertainty)	Test range (Increment)
Comp. ambient temperature(℃)	30 (±2)	-
Condensing temperature(℃)	35 (±0.3)	25~55,(≒3℃ inc.)
R-evaporating temperature(℃)	-20(±0.3)	-25~-10,(≒ 3℃ inc.)
F-evaporating temperature(℃)	-30 (±0.3)	-45~-23,(≒3℃ inc.)
Subcooling(℃)	5 (±2)	-
Compressor speed (rpm)	1600, 1800, 2050, 2450, 2800, 3400, 3600, 3800, combinations	

Table 3. Test conditions in the secondary fluid.

Parameter	Condition
2nd fluid inlet temperature of condenser (℃)	Variable
2nd fluid inlet temperature of R-evaporator ℃)	3 (±0.3)
2nd fluid inlet temperature of F-evaporator ℃)	-8 (±0.3)
2nd fluid flow rate of condenser (℃)	Variable
2nd fluid flow rate of R-evaporator *l*/min)	0.75 (4 or over)
2nd fluid flow rate of F-evaporator *l*/min)	0.75



(a) STC (b) PTC

Fig. 2. Compressor arrangements of STC and PTC method.

The evaporating temperatures for the R- and F-evaporators were simultaneously controlled using the needle valve opening and compressor frequency. All tested data were measured for 20 minutes with a 0.5 second interval when the test conditions reached to steady state.

Eq. (1) was used to calculate the secondary fluid-side heat transfer capacity in the condenser and evaporator. The refrigerant-side heat transfer capacity in the condenser was determined by Eq. (2). The refrigerant-side heat transfer capacity was validated against the secondary-side heat transfer capacity. As shown in Fig. 3, both test results were matched within ±5%. The refrigerating capacity in the evaporator was calculated by Eq. (3). The COP of the system was calculated by Eq. (4).

$$Q_{cond,2nd} = \dot{m}_{2nd} Cp_{2nd} (T_{cond,2nd,o} - T_{cond,2nd,i}) \tag{1}$$

$$Q_{cond} = \dot{m}_r (h_{cond,i} - h_{cond,o}) \tag{2}$$

$$Q_{evap} = \dot{m}_r (h_{evap,o} - h_{evap,i}) \tag{3}$$

$$COP = \frac{Q_{evap}}{W} \tag{4}$$

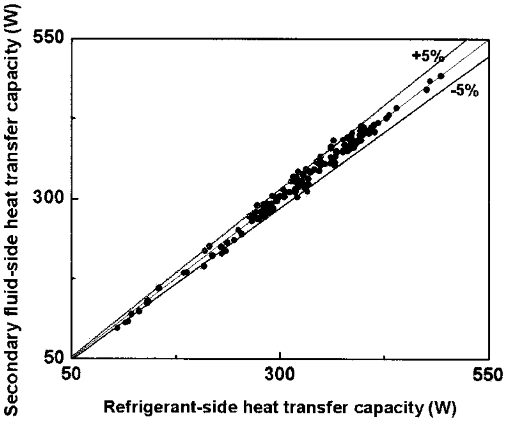


Fig. 3. Energy balance between refrigerant-side and secondary-side in the condenser.

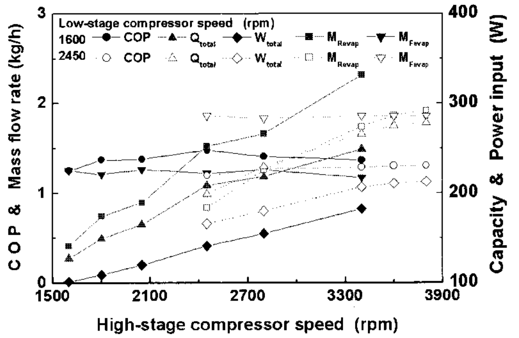


Fig. 4. Performance of the STC system with high-stage compressor speed.

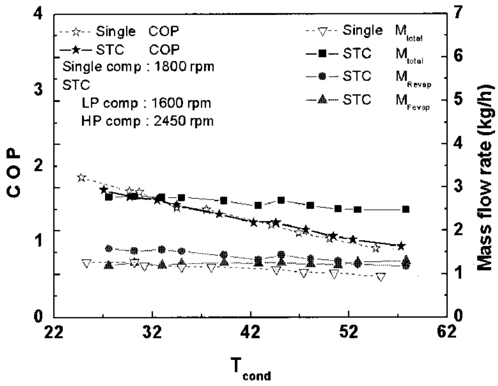


Fig. 5. Performance of the STC system with condensing temperature.

3. Results and discussion

3.1 Performance of the STC system

3.1.1 Effects of compressor speed

Fig. 4 shows the performance of the STC system with respect to high-stage compressor speed. As the high-stage compressor speed increased, the mass flow rate of the R-evaporator significantly increased but the mass flow rate of the F-evaporator slightly decreased. This is because the increase of the high-stage compressor speed causes the decrease of the mass flow rate of the F-evaporator.[6] The COP decreased because of higher compressor power consumption as compared with the increase of the total refrigerating capacity.

As a result, the optimum low- and high-stage compressor speeds were 1600 and 2450 rpm, respectively, which yielded the highest COP of 1.48. In addition, the COP at the low-stage compressor speed of 1600 rpm was 18.9% higher than that at the compressor speed of 2450 rpm because the increase of the total refrigerating capacity was higher than that of the compressor power consumption.

3.1.2 Effects of condensing temperature

Fig. 5 compares the performance of the STC system with that of the single-stage cycle according to condensing temperature. As the condensing temperature increased from 25 to 50℃, the COPs for the single-stage cycle and the STC system decreased by 43.9% and 36.7%, respectively, because of the decrease of the total mass flow rate. This results in the decrease of the refrigerating capacity for the R-evaporator and the increase of the power consumption of the high-stage compressor. The COP of the single-stage cycle was 4.4% higher than that of the STC system at condensing temperatures below 35℃. However the COP of the single-stage cycle was 2.3% lower than that of the STC system at condensing temperatures above 35℃. Therefore, the COP of the STC system was higher than that of the single-stage cycle at standard conditions.

3.1.3 Effects of F- and R-evaporating temperature

Fig. 6 compares the performance of the STC system with that of the single-stage cycle according to F-evaporating temperature. In the single-stage cycle, as the F-evaporating temperature increased, the total refrigerating capacity increased due to the increase in the total mass flow rate with the decrease of specific volume at the compressor suction. The COP of the STC system was higher than that of the single-stage cycle at lower F-evaporating temperature. As the F-evaporating temperature increased from -47 to -23℃, the COP of the STC system decreased by 7.0%.

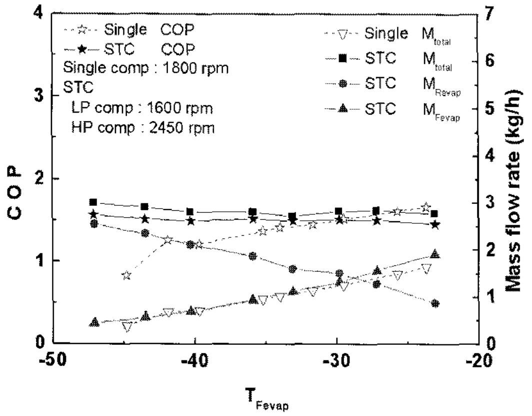


Fig. 6. Performance of the STC system with F-evaporating temperature.

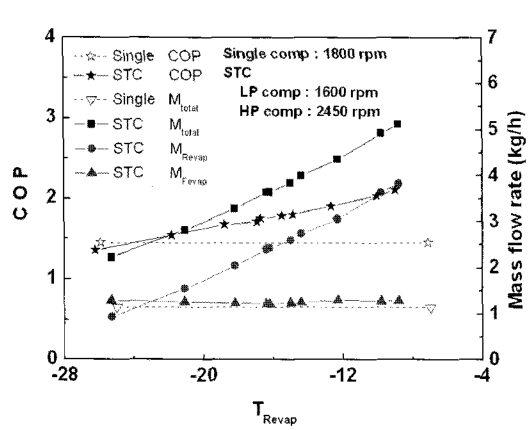


Fig. 7. Performance of the STC system with R-evaporating temperature.

Fig. 7 compares the performance of the STC system with that of the single-stage cycle according to R-evaporating temperature. As the R-evaporating temperature increased from -25 to -8℃, the COP of the STC system increased by 35.6% because the refrigerating capacity of the R-evaporator linearly increased, although the power consumption for the high-stage compressor slightly increased. As shown in Figs. 6 and 7, the effects of R-evaporating temperature on the performance of the STC system were more dominant than those of F-evaporating temperature.

3.2 Performance of the PTC system

3.2.1 Effects of compressor speed

Fig. 8 shows the performance of the PTC system with respect to low-stage compressor speed. As the low-stage compressor speed increased at high-stage compressor speeds of 1600 and 1800 rpm, the total

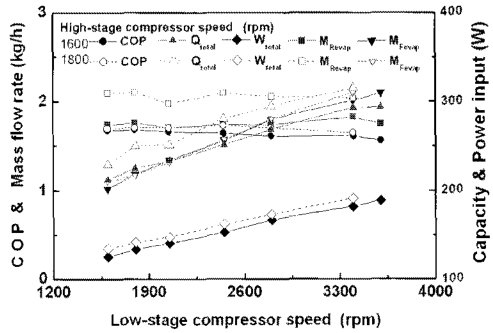


Fig. 8. Performance of the PTC system with low-stage compressor speed.

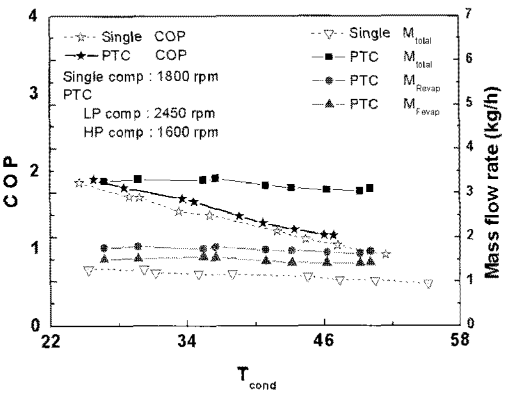


Fig. 9. Performance of the PTC system with condensing temperature.

refrigerating capacity increased by 27.7% and 20.2%, respectively, and the compressor power consumption increased by 33.2% and 27.9%, respectively. Therefore, the COP decreased with the increase of low-stage compressor speed. In addition, the COP of the PTC system at the high-stage compressor speed of 1800 rpm was 5.2% higher than that at 1600 rpm. As a result, the optimum low- and high-stage compressor speeds were 2450 and 1800 rpm, respectively.

3.2.2 Effects of condensing temperature

Fig. 9 compares the performance of the PTC system with that of the single-stage cycle according to condensing temperature. As the condensing temperature increased from 25 to 50℃, the COPs of the single-stage cycle and the PTC system decreased by 43.9% and 38.5%, respectively, because of the decrease in the total mass flow rate. This results in the decrease of the refrigerating capacity for the R-evaporator and the increase of the high-stage compressor power consumption. The COP of the PTC system was 9.8% higher than that of the single-stage cycle.

3.2.3 Effects of F- and R-evaporating temperature

Fig. 10 compares the performance of the PTC system with that of the single-stage cycle according to F-evaporating temperature. As the F-evaporating temperature increased from -45 to -23℃, the COP of the PTC system increased by 25.4% because the refrigerating capacity of the F-evaporator significantly increased with the increase of the total mass flow rate although the low-stage compressor power consumption increased slightly.

Fig. 11 compares the performance of the PTC system with that of the single-stage cycle according to R-evaporating temperature. As the R-evaporating temperature increased from -25 to -14℃, the COP of the PTC system increased by 17.3% because the refrigerating capacity for the R-evaporator increased, although the high-stage compressor power consumption slightly increased. This is because the refrigerating capacity for the F-evaporator and the low-stage compressor power consumption were constant regardless of R-evaporating temperature. In addition, the effects of R-evaporating temperature on the COP of the STC system was higher than those of the PTC system because of higher compression ratio.

Fig. 12 compares the COP of the single-stage cycle with those of the STC and PTC systems at standard conditions. The COPs of the STC and PTC systems were 2.1% and 11.6%, respectively, higher than that of the single-stage cycle because of less cycle losses and better compression efficiency. In addition, the COP of the PTC system was 9.7% higher than that of the STC system.

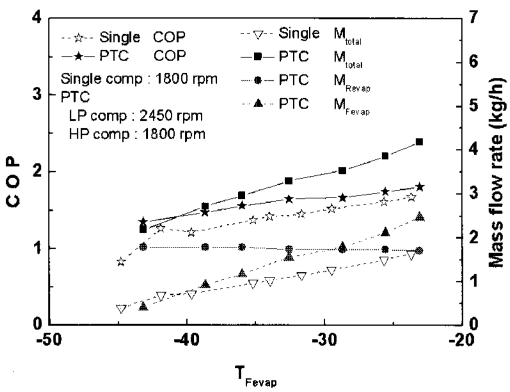


Fig. 10. Performance of the PTC system with F-evaporating temperature.

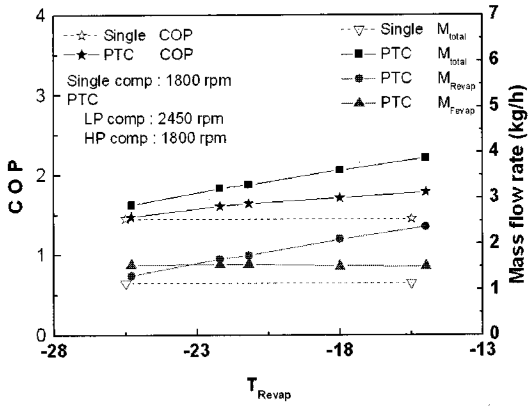


Fig. 11. Performance of the PTC system with R-evaporating temperature.

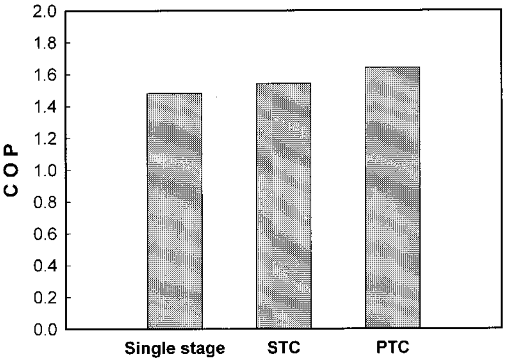


Fig. 12. Comparison of COP with the single -stage cycle, STC and PTC at standard conditions.

4. Conclusions

Performance characteristics of the household refrigerator with dual evaporators using the two-stage compression cycle were investigated by varying the compressor speed, condensing temperature, and evaporating temperature. The COPs of the STC and PTC systems were compared with that of the single-stage cycle.

(1) The optimum compressor speed in the STC system was 1600 rpm for the low-stage and 2450 rpm for the high-stage. For the PTC system, it was 2450 rpm for the low-stage and 1800 rpm for the high-stage. However, these optimum values may be applicable to the present setup only.

(2) As the condensing temperature increased from 25 to 50℃, the COP of the single-stage, STC system, and PTC system decreased by 43.9%, 36.7%, and 38.5%, respectively.

(3) The effects of R-evaporating temperature on the performance of the STC system was more dominant than those of F-evaporating temperature.

(4) The COPs of the STC and PTC systems were 2.1% and 11.6% higher than that of the single-stage cycle, respectively. In addition, the COP of the PTC system was 9.7% higher than that of the STC system.

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