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An Experimental Study on the Performance of CO_2 Air-conditioning Cycle Equipped with an Ejector

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Abstract

As an effort to prevent environmental problems caused by ozone depletion and global warming, alternative refrigerants are being developed, and one of the candidates is carbon dioxide. To overcome slightly low efficiency of CO_2 refrigeration system, air-conditioning cycle using an ejector was suggested. Ejector compensates throttling loss in an expansion device by reducing compression work. In this study, the ejector refrigeration cycle using CO_2 as a refrigerant is investigated to understand the effect of the mixing section diameter and refrigerant charge amount on the performance. If mixing section diameter is too large or too small, either cases show low performance. The optimum refrigerant charge amount which gives the best performance is found for standard operating conditions. The air-conditioning cycle was analyzed for several operating conditions.

Key words: Carbon dioxide, Refrigeration system, Ejector, Expansion device, Performance

Nomenclature

h : specific enthalpy [J/kg]
$\dot{m}$: mass flow rate [kg/s]

Subscript

m : motive nozzle inlet
s : suction nozzle inlet
$diff$: diffuser

1. Introduction

Recently, industrial development increased human comfort in many aspects, but it also caused many environmental problems. Among them, ozone depletion problem and global warming problem are appearing as issues of environmental threat. Some researchers warn that if the problems progress, there will be considerable changes which cannot be recovered. The refrigeration industry is also concerning about reducing environmentally harmful effect which refrigeration systems have resulted in.

One of them is to replace conventional refrigerants. Until about 1980s, CFCs were the most commonly used refrigerants, but they have large ozone depletion effect and global warming effect. Because of its high ozone depletion potential (ODP), it should be phased out by international agreement. HCFCs were developed with much lower ozone depletion effect, however it still contains ozone depletion effect and global warming effect.

In these days, HFCs are being considered as zero ODP refrigerant. It has no Chlorine atoms which deplete ozone layer, but it cannot be environmentally appropriate because of its high global warming potential (GWP).

CO_2 was proposed first as a refrigerant in late 19th century and now it is considered again as an environmentally benign refrigerant. It has been verified to be safe for both environment and human, being a substance that has existed in nature.

One of the characteristics of refrigeration system using CO_2 is that it forms trans-critical cycle owing to its critical temperature of 31℃ and critical pressure of 7.2 MPa. Because of this, the system performance is very sensitive to outdoor temperature and slightly worse than that of conventional cycle.

The system using ejector was proposed to enhance

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the performance of refrigeration system whose refrigerant is CO_2. In an ejector system, ejector is installed to use kinetic energy of high speed flow during expansion process. Before entering the compressor suction, the refrigerant is compressed by the kinetic energy and the compressor work can be reduced.

In 1931, Gay[1] devised a refrigeration system which uses a two-phase ejector for a replacement of an expansion valve. Domanski[2] evaluated COP of an ejector system theoretically in 1995 and Elbel et al.[3] validated an ejector expansion cycle experimentally in 2008.

In this study, the performance of CO_2 refrigeration system using an ejector is experimentally investigated. The system uses ejector as an expansion valve and has no additional expansion device. Hence, ejector suction mass flow was enhanced. The steady state system performance characteristics were investigated while ejector mixing section diameter and refrigerant charge amount were varying.

The compressor is selected as an inverter-driven type to control the cooling capacity and system performance. The optimal operating strategy is possible with the variable speed compressor system. Hua et al.[4] devised PI control logic to apply to variable speed refrigeration system in 2007 and Lee et al.[5] investigated fuzzy control logic for the CO_2 variable speed refrigeration system.

2. CO_2 air-conditioning cycle with an ejector

2.1 Ejector system cycle and the test rig

The pressure-enthalpy diagram of ejector system is represented in Fig. 1. Each numbered point represents the state at the corresponding position (1: compressor suction, 2: gas cooler inlet, 3: internal heat exchanger inlet of high pressure side, 4: ejector motive nozzle inlet, 5: phase separator, 6: internal heat exchanger inlet of low pressure side, 6': evaporator inlet, 7': evaporator outlet). The basic concept of the system is to use kinetic energy of the refrigerant which is made during expansion process. After the high pressure side internal heat exchanger, the refrigerant is accelerated and expands in the motive nozzle of the ejector. The expanded refrigerant flow stream is mixed with the stream from the ejector suction port, forming corresponding equilibrium pressure. Pressure of the suction stream rises to the equilibrium pressure and is more pressurized in diffuser. After the diffuser, the liquid phase refrigerant is separated from the mixed

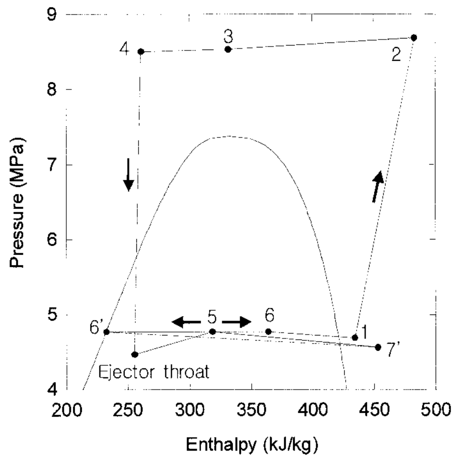

Fig. 1. Ejector refrigeration cycle.

two-phase state refrigerant and enters to evaporator. Finally, it returns to the ejector suction port. The gas phase refrigerant, from the separator, is sent to compressor with higher pressure than that of evaporator. Consequently the compression work can be reduced. In this study, pressure drop at expansion device was not considered but only pressure drop in evaporator was recovered because the system has no additional expansion devices.

The experimental apparatus is shown in Fig. 2. The system consist of a refrigerant loop and two water loops. Refrigerant loop has internal heat exchanger to enhance the performance of CO_2 refrigeration system.

A model TCS113 semi-hermetic reciprocating CO_2 compressor made by Dorin Company was used with an inverter to control the operation frequency. Each heat exchanger made by copper is coaxial counterflow type and water was chosen as a secondary fluid. Gas cooler and evaporator consist of 20 and 13 straight sections with a length of 0.90 m, respectively. Two Oval Coriolis type mass flow meters are placed to measure the refrigerant mass flow rates of induced flow and motive flow. T-type thermocouples and pressure transducers are installed for the measurement of each property.

2.2 Ejector configuration

In Fig. 3, ejector configuration is shown. The motive nozzle throat diameter was set to 0.8 mm and mixing section diameter was set to 2.5, 3.0, 4.0 and 5.0 mm for each case.

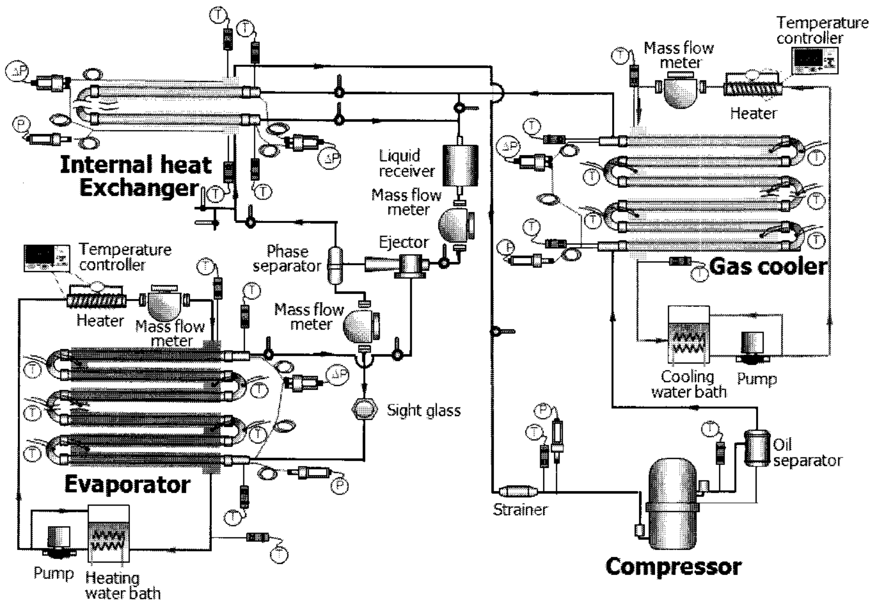

Fig. 2. Experimental apparatus.

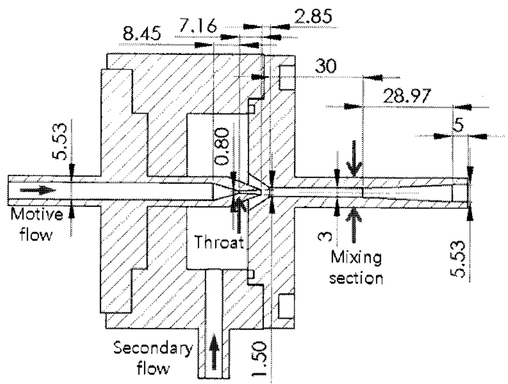

Fig. 3. Ejector configuration.

2.3 Experimental procedure and test conditions

Before operating the system, it was made vacuous by vacuum pump and proper amount of refrigerant was charged. The system was started and when it reaches steady state, the data were collected. After receiving sufficient amount of data, the compressor frequency was changed for the next test condition. After a set of tests, more refrigerant was charged and the experiment for the varied refrigerant charge amount was performed.

Test conditions are represented in Table 1. The temperatures of secondary fluid were selected follow ing the KS standard[6].

Table. 1. Experimental conditions.

Parameters	Values
Gas cooler water inlet temperature [℃]	30
Gas cooler water flow rate [g/s]	80
Evaporator water inlet temperature [℃]	27
Evaporator water flow rate [g/s]	100
Refrigerant charge amount [kg]	3.0-3.5
Compressor inverter speed [Hz]	25-60

The uncertainties of pressure measurements are estimated as ±0.25% and ±0.5℃ for the temperature measurements. Mass flow rate measured in the study has ±0.1% of uncertainty.

3. Results and discussion

3.1 Steady state system performance

The experimental results for the ejector system without expansion valve are shown in Fig. 4.

Cooling capacity was calculated by water temperature differences and compressor work was obtained by digital power meter.

Steady state system performances are shown with varying compressor frequency and different refrigerant charge amount. The mixing section throat diameters are set to 2.5, 3.0, 4.0 and 5.0 mm for each case.

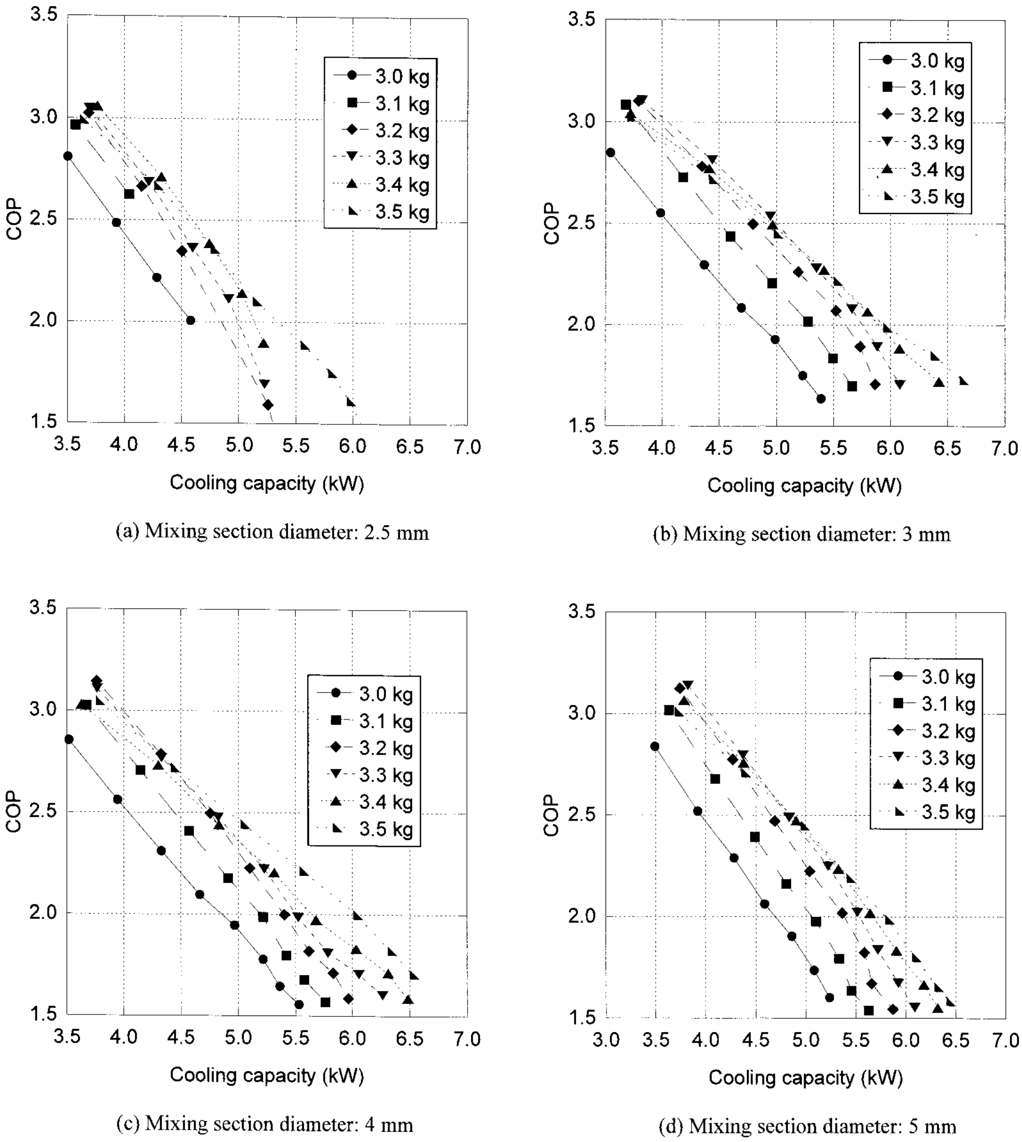

Fig. 4. Relation between cooling capacity and COP for several mixing section diameter.

For all cases, when the compressor frequency is increased, the cooling capacity is increased and COP is decreased like most of conventional refrigeration system.

According to mixing section diameters, the system, which has the mixing section diameter of 3.0 mm, shows the best performance. The main parameter decided by mixing section diameters is entrainment ratio which is defined by the ratio of the suction mass flow rate to the motive mass flow rate as shown in Eq. 1.

$$\varepsilon = \frac{\dot{m}_s}{\dot{m}_m} \tag{1}$$

When we select 3.0 mm as a mixing section diameter, the entrainment ratio was maximum. With the over-sized mixing section diameter the recirculation effect occurs in mixing section and the induced mass flow is reduced. Otherwise, if the mixing section diameter is too small, induced gas refrigerant is chocked at the mixing section inlet. Consequently, the optimum mixing section diameter exists for corresponding throat diameter and operating conditions.

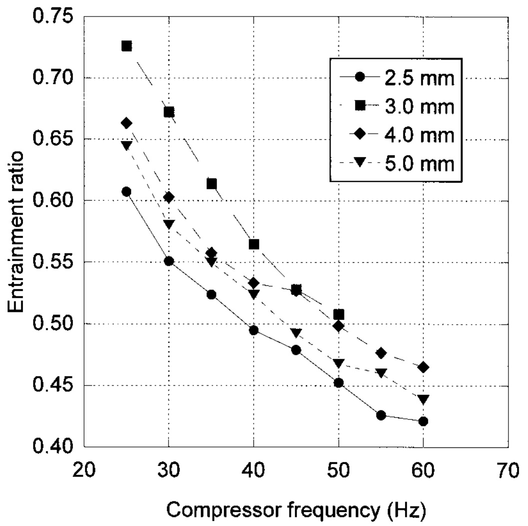

Fig. 5. Enrainment ratio when the mixing section diameter is 3 mm and refrigerant charge amount is 3.4 kg.

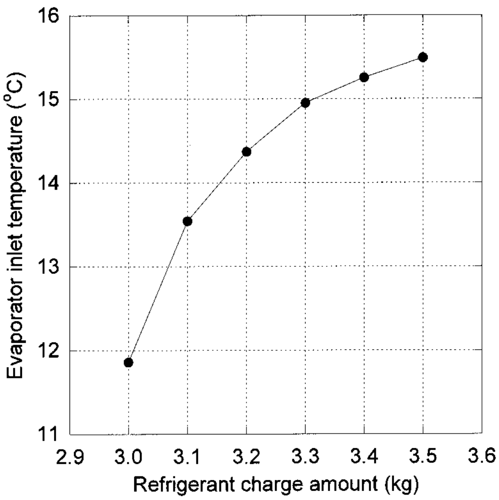

Fig. 6. Evaporator inlet temperature of refrigerant (mixing section diameter: 3 mm, compressor frequency: 30 Hz).

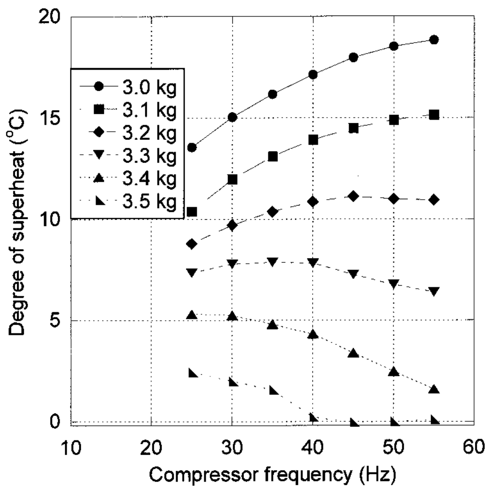

Fig. 7. DSH at compressor suction when mixing section diameter is 3 mm.

The entrainment ratio with different mixing section diameter when the refrigerant charge amount is 3.3 kg is shown in Fig. 5.

There were some cases when the COP reached its maximum with the variation of the refrigerant charge amount. In low cooling capacity region, 3.2 or 3.3 kg of refrigerant charge shows the best performance. But above 5.5 kW of cooling capacity, 3.5 kg of refrigerant charge shows the best performance.

With large refrigerant charge amount, more refrigerant flows and cooling capacity is augmented. On the contrary, at low compressor frequency, the evaporation temperature is not low enough. Moreover, evaporation temperature gets higher with more refrigerant charge, and as a result, large refrigerant charge amount and low compressor operating frequency result in inadequately low heat transfer efficiency in the evaporator. The evaporator inlet temperatures of refrigerant when the mixing section diameter is 3 mm are shown in Fig. 6.

At large cooling capacity, the performance gets better with more refrigerant charge. However, at 3.5 kg of refrigerant charge amount, degree of superheat (DSH) reaches zero at compressor suction. The DSHs for each condition are shown in Fig. 7.

3.2 The cycle analysis in the steady state

The pressure-enthalpy diagram of the cycle with respect to compressor frequency is presented in Fig. 8 when the mixing section diameter is 3 mm and refrigerant charge amount is 3.4 kg. The dashed line represents 27℃ line which is the temperature of the secondary fluid at the inlet point. Each point is shown based on the calculated enthalpy and pressure using REFPROP 7.0[7]. The enthalpy at the ejector's diffuser outlet was calculated by the following equation.

$$h_{diff} = \frac{h_m \dot{m}_m + h_s \dot{m}_s}{\dot{m}_m + \dot{m}_s} \qquad (2)$$

With higher compressor frequency, refrigerant mass flow rate and cooling capacity are augmented.

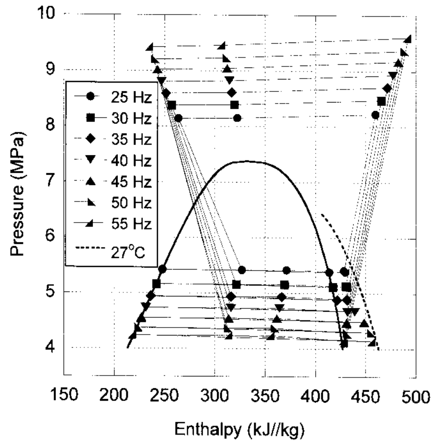

Fig. 8. P-h diagram with respect to compressor frequency (mixing section diameter: 3 mm, refrigerant charge: 3.4 kg)

However, the compressor work is also increased to reduce COP.

Evaporation pressure and temperature fall with compressor frequency increased and heat transfer in evaporator is enhanced. On the other hand, the lowered evaporation pressure and large DSH at evaporator outlet make refrigerant specific volume larger. The refrigerant is not induced easily at suction port because of the large specific volume, and refrigerant mass flow rate is reduced.

The pressure-enthalpy diagram is shown in Fig. 9 when the refrigerant charge amount is varied. As more refrigerant is charged, the total system pressure rises and refrigerant mass flow rate is increased. In the figure, the evaporative latent heat is shown to be decreased as evaporation pressure becomes higher.

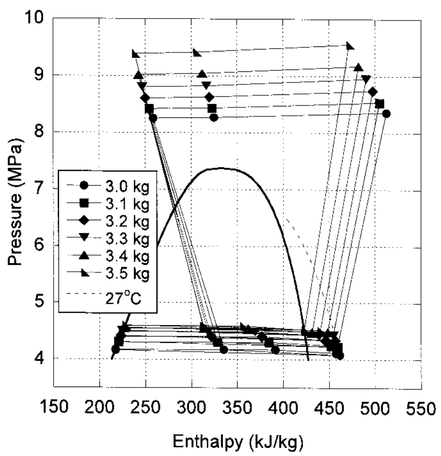

Fig. 9. P-h diagram with respect to refrigerant charge amount (mixing section diameter: 3 mm, compressor frequency: 45 Hz).

Because of that, the specific cooling capacity is reduced.

At compressor suction, the DSH drops with more refrigerant charge amount as shown in Fig. 7. For isentropic compression process, the enthalpy difference between before and after the compression becomes larger when the gas temperature is elevated. Hence, the more compression work is needed for higher DSH, and lower DSH means more efficient compression. Because of the lower DSH, specific compression work is reduced with increase of refrigerant charge amount, in spite of increased pressure ratio. Fig. 10 is representing the specific compression work with respect to different refrigerant charge amount when the mixing section diameter is 3.0 mm and the compressor frequency is 45 Hz.

However, the lower DSH at compressor suction means lower gas cooler inlet temperature leading to negative effect to heat exchange at gas cooler. With the fixed secondary fluid inlet temperature, if the gas cooler inlet temperature of refrigerant becomes lower, the amount of heat rejection is decreased. Fig. 11 shows the gas cooler inlet temperature and specific heating capacity. Consequently, COP increases to a certain value, but after that value, it does not become higher as shown in Fig. 12.

4. Conclusions

The steady state performance characteristics of CO_2 refrigeration system using ejectors were investigated. Ejector with mixing section diameter of 3 mm shows the best system performance and confirmed the

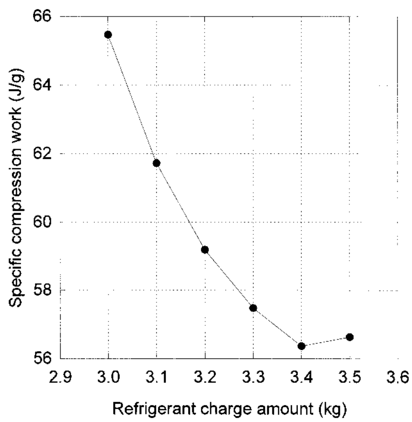

Fig. 10. Specific compression work with respect to refrigerant charge amount (mixing section diameter: 3 mm, compressor frequency: 45 Hz)

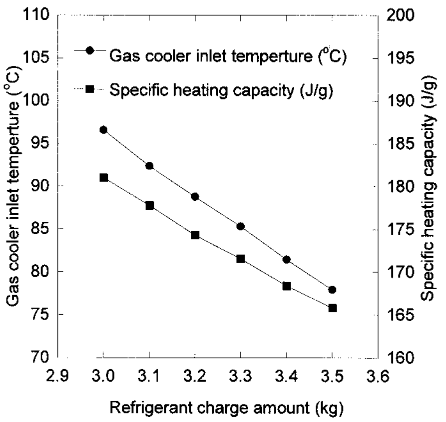

Fig. 11. Gas cooler inlet temperature and specific heating capacity (mixing section diameter: 3 mm, compressor frequency: 45 Hz).

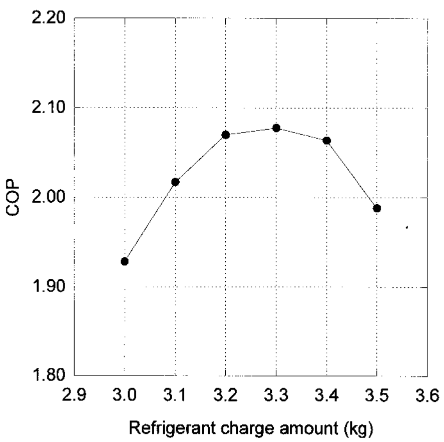

Fig. 12. COP with respect to refrigerant charge amount (mixing section diameter: 3 mm, compressor frequency: 45 Hz).

existence of optimal point. Refrigerant charge amount was limited to 3.5 kg because beyond that point the DSH at compressor suction was not guaranteed. The influence of compressor frequency and refrigerant charge amount was analyzed qualitatively. At high compressor frequency, refrigerant specific volume makes the induction at the ejector suction port poor and heat exchange in gas cooler is reduced with increased refrigerant charge.

Acknowledgment

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