Part-load Performance of a Screw Chiller with Economizer using R22 and R407C

Young-Soo Chang, Youngil Kim*†, Yong-Chul Lee

Thermal/Flow Control Research Center, KIST, Seoul 136-791, Korea

**Department of Energy System Engineering, Chungju University, Chungbuk 380-702, Korea*

Key words: Screw chiller, Part-load, Economizer, R22, R407C

ABSTRACT: Screw compressor chillers are widely used in refrigeration for capacity over 30 RT. In general, chillers operate under part-load conditions during most of the time. Therefore, information on the characteristics of part-load is very important for better chiller performance and energy economy. In this study, performance tests of screw chiller with economizer using R22 and R407C under part-load conditions have been performed for various secondary fluid temperatures. Adoption of an economizer system increased the cooling capacity and improved COP except for lower part-load condition when economizer volume ratio is 1.01. For the same cooling capacity condition at part-load, COP's of both non-economizer and economizer system showed similar values.

Nomenclature

COP	: coefficient of performance
C_p	: constant pressure specific heat [J/kgK]
h	: enthalpy [J/kg]
$\dot{m}$	: mass flow rate [kg/s]
$\dot{Q}$	: cooling capacity [W]
T	: temperature [℃]
V	: trapped volume of refrigerant [m^3]
VF	: displaced volume flow rate [m^3/s]
v_i	: volume ratio of economizer port
W	: compressor power consumption [W]

Greek symbols

η_s	: isentropic efficiency
η_v	: volumetric efficiency
ρ	: density [kg/m^3]

Subscripts

b	: secondary heat transfer fluid
ci	: starting point of compression process
$comp$	: compressor
$cond$	: condenser
$econ$	: economizer
$econi$	: starting point of injection
$evap$	: evaporator
in	: inlet
$isen$	: isentropic process
o	: reference operation condition
out	: exit
r	: refrigerant

† Corresponding author
Tel.: +82-43-841-5282; fax: +82-43-841-5280
E-mail address: yikim@chungju.ac.kr

1. Introduction

A screw compressor has a good controllability, high efficiency, and high reliability and duration of life. Thus it has been widely used for compressing air and gas replacing the reciprocating type compressor. Especially, it is becoming very popular for large refrigeration capacity over 30 RT.

To improve the capacity as well as COP in refrigeration system with screw compressor or turbo compressor, economizer arrangements are often used. The basic principle of economizer is that an additional refrigerant is allowed to enter the compressor after the original suction process is complete. At the condenser outlet, the refrigerant flow is divided in two: the main flow passes through the economizer heat exchanger, the secondary flow is expanded in the economizer heat exchanger and sub-cools the main flow refrigerant. The evaporated refrigerant enters the intermediate pressure section of the compressor.

A number of research on screw chiller focuses on screw compressor and its performance.[1-3] However, system performance with economizer has not been studied extensively. Jonsson[4] reported that performance of refrigeration system enhances when a flash tank economizer is located in intermediate pressure of two-stage cycle. Högberg and Berntsson[5] investigated theoretically the performance differences between non-azeotropic mixtures (R22/R152a, R22/R142b, R22/R114) and pure fluids in two-stage cycles.

If a refrigeration load changes in a refrigeration system, capacity control is required. There are many ways of achieving capacity control depending on the type of compressor. In small systems, cycling operation of compressor is commonly used. Compressor installed in large plant has ability to alter the compressor throughput while the compressor is in operation.

Conventionally, the design and installation of refrigeration system is based on peak load condition. Furthermore, oversizing heating or cooling systems by engineer is in common practice. Thus, it is known that chillers installed in the field are usually operated under part-load condition. To predict compressor power consumption and system performances, performance parameter such as load pattern and part/full-load efficiencies should be collected and analyzed.[6]

Due to the ozone depletion problem, environmentally friendly refrigerants with zero ozone depletion potential are required to be used in refrigeration system. The HFC (hydrofluorocarbon) refrigerants with zero ozone depletion potential have been recommended as alternatives. Among HFC refrigerants, R407C is considered as a most promising drop-in substitution for R22 in air-conditioning applications because important thermodynamic properties such as vapor pressure and volumetric capacity are quite similar to R22.

However, it is reported that the decrease of heat transfer coefficient of R407C compared with that of R22 results in the cooling capacity and COP decreases of R407C by 10~20% and 20~30%, respectively.[7] Therefore, performance enhancement by adopting an economizer is considered a good method for compensating the performance degradation of R407C.[8]

In this study, performance of R407C is investigated in a screw chiller with economizer at various part-load conditions and compared with that of R22.

2. Description of the screw chiller

In this study, performance test is performed for a commercial screw chiller with a nominal cooling capacity of 30 RT, which is originally designed for R22. A drop-in performance test of R407C is carried out without any modification except refrigerant.

The chiller is equipped with a screw compressor, two expansion valves, a condenser, an evaporator, and an economizer as shown in Fig. 1. The screw compressor has a secondary port (economizer port) between the suction and discharge ports. Flash refrigerant from the economizer can be introduced to this port, which sub-cools the main refrigerant flow. The

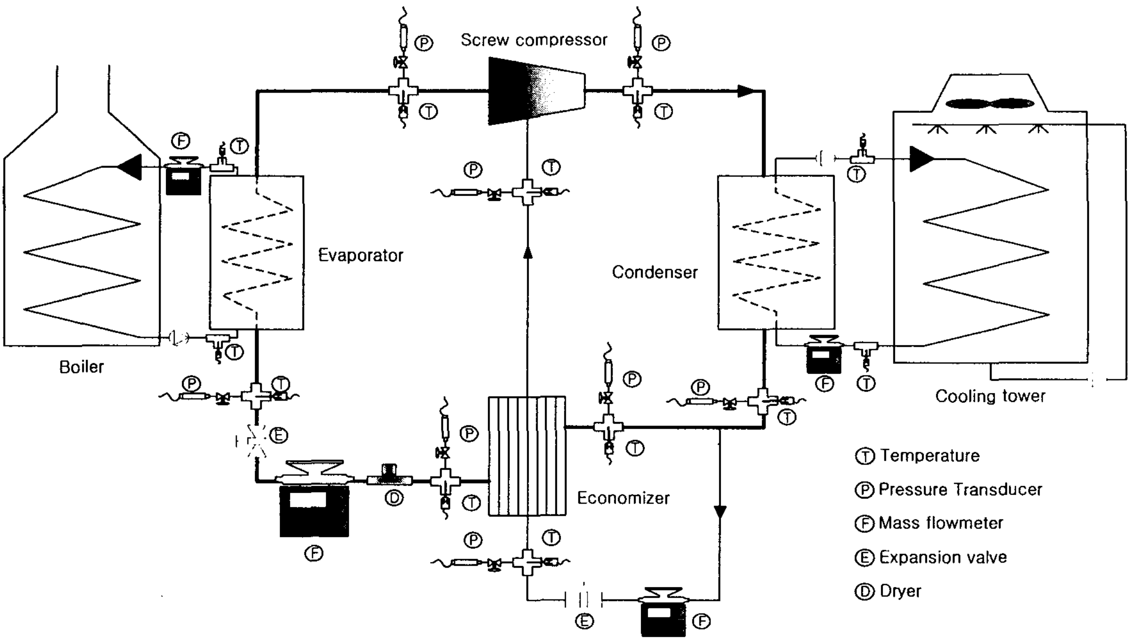


Fig. 1 Schematic diagram of the system.

rate of mass flow injected to the compressor depends on suction and discharge pressures, economizer port geometry and position. The volume ratio of the economizer port is defined as the ratio of volume of the thread at the start of the compression process to the volume of the same thread when it begins to close the center of economizer port as follows.

$$v_i = \frac{V_{ci}}{V_{econi}} \quad (1)$$

The screw compressor has two economizer ports, which are located at volume ratios of 1.01 and 1.17. The details of the compressor are described in Table 1.

Table 1 Specification of screw compressor

Number of male/female rotor lobes	5/7
Outer diameter of male rotor	107.50 mm
Outer diameter of female rotor	104.92 mm
Wrap angle of male rotor	244°
Rotor length	122 mm
Built-in volume ratio	2.4
Economizer port built-in volume ratio	1.02, 1.17
Economizer port diameter	12 mm

The condenser is a shell-and-tube heat exchanger in which the refrigerant flows shell side, whereas the evaporator is dry expansion type shell-and-tube heat exchanger with refrigerant flowing inside the tubes. The tubes are arranged as two passes in both the con-

Table 2 Specification of condenser

Shell inside diameter	299.7 mm
Number of tubes	76
Tube length	1,209 mm
Tube outside diameter	19.4 mm
Tube inside diameter	15.9 mm

Table 3 Specification of evaporator

Shell inside diameter	277 mm
Tube outside diameter	15.9 mm
Tube layout pitch	20 mm
Overall nominal tube length	1,664 mm
Number of tubes	116
Number of tube passes	2
Number of baffles	14

Table 4 Specification of economizer heat exchanger

Size	520 mm × 110 mm × 76 mm
Plate number	28
Each plate area	0.0572 m^2
Plate pitch	2.7 mm

denser and the evaporator.

The specifications of the condenser and evaporator are listed in Tables 2 and 3, respectively. A plate-type heat exchanger is installed as an economizer, and Table 4 summarizes the details of the economizer.

Traditionally, the capacity of a screw compressor is controlled by a sliding valve which is a solid block as shown in Fig. 2. Activated by a control mechanism, the piston valve can move in the direction parallel to the axis of the rotors. At full-load (100% load factor), as all solenoid valves close, the front end of the piston valve is in contact with its facing plane on the stationary part of the rotor housing by external oil pressure, and the refrigerant in the compression chamber is totally enclosed after the suction port closes.

At partial loads, the piston valve is pulled a distance toward the exhaust end of the compressor and a bypass port is created in front of the valve. This makes the refrigerant in the compression chamber to flow back to the suction side of the compressor. Both the flow capacity and the compression ratio change according to the piston valve movement, which

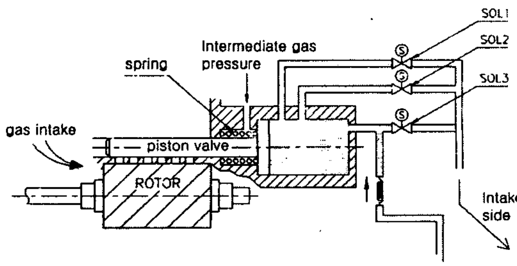


Fig. 2 Schematic diagram of capacity control system.

Table 5 Properties of lubricant

Property	Value
Density at 15℃	974 kg/m^3
Flash point	250℃
Viscosity at 40℃	170 mm^2/s
Viscosity at 100℃	17.2 mm^2/s
Pour point	−24℃

is determined by solenoid valve operations. Screw compressor in this study can modulate its capacity in 4 steps: Full-load (100% load) and 3 part-load (75%, 50%, and 25% load factor) conditions are possible by setting the piston valve position. The closer the piston valve is to the exhaust port of the compressor, the more refrigerant bypasses through the bypass port. 75%, 50% and 25% load factor conditions do not mean the cooling capacity of the compressor, but they indicate the part-load operation by piston valve positions.

The same POE (polyolester oil) lubricant is used for both R22 and R407C, and its properties are described in Table 5. At the discharge side of the screw compressor, an additional oil separator is installed separate the oil from the refrigerant. This is intended to minimize oil circulation through the heat exchangers.

3. Experimental method and conditions

Cooling capacity and COP of screw chiller are main results of our experiments. Cooling capacity can be obtained by measuring inlet and outlet temperatures and mass flow rate of the secondary heat transfer fluid through the evaporator, and is defined as follows.

$$Q_{evap} = m_{evap,b} C_{p,b} (T_{evap,in} - T_{evap,out}) \quad (2)$$

Compressor power consumption is measured using power watt meter. Based on the values of cooling capacity in Eq. (2) and compressor power consumption, COP is calculated as

$$\mathrm{COP} = \frac{Q_{evap}}{W} \tag{3}$$

The accuracy of the measurement was verified in previous study[7] and measurement error of cooling capacity and COP is within 5%.

A closed-type cooling tower is installed to control the inlet temperature of the secondary heat transfer fluid for the condenser, and a boiler is applied to control the secondary heat transfer fluid inlet temperature to the evaporator. The secondary fluid temperature in condenser is controlled by speed modulation of the cooling tower fan and on-off control of the spray water. The secondary fluid temperature in evaporator is controlled by modulating the fuel flow rate supplied to the oil burner of the boiler.

The reference operation condition in this study is set according to standard condition for performance measurement of a liquid chiller: the inlet temperature of secondary heat transfer fluid at condenser is 32℃ and the secondary heat transfer fluid temperature at evaporator outlet is 7℃. Since the performance of the chiller is largely dependent on operating temperatures, four cases of temperature conditions are tested. Condenser inlet temperature of secondary heat transfer fluid is set at 32 and 27 ℃, and evaporator outlet temperature of secondary heat transfer fluid is maintained at 7 and 2℃. The inlet and outlet temperature differences of secondary heat transfer fluid at condenser and evaporator are adjusted to be 5℃ by modulating mass flow rate of secondary heat transfer fluids. For each temperature condition, full-load (100% load) and 3 part-load (75%, 50%, and 25% load) tests are performed.

A metering expansion valve is used to regulate mass flow rate of refrigerant injected in the compressor through the economizer port. Cooling capacity and COP are observed with respect to variation of mass flow rate of injected refrigerant through two economizer ports individually.

For all temperature conditions, compressor load condition, and economizer port position, performance enhancement using economizer is tested for R22 and R407C, while maintaining the superheat at economizer at 5℃.

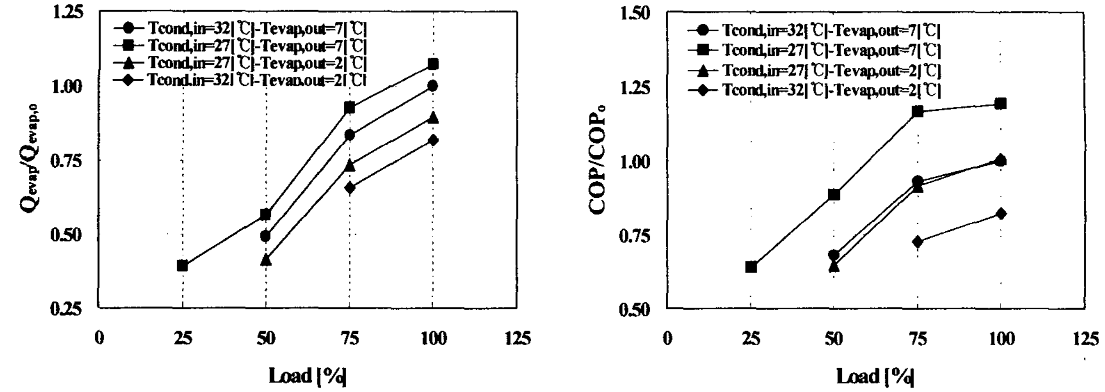
The superheat of the refrigerant is calculated as the difference between the refrigerant temperature and the saturation temperature evaluated from the pressure measured at that point. The superheat of evaporator is maintained at 5℃ throughout the tests.

All the signals from the sensors and transducers are gathered by a data acquisition system and transferred to a personal computer. In the experiment, the measurement for the performance evaluation begins when the temperature variations are maintained within ±0.2 ℃ at every point in the system. Then, the data are collected for 10 min at an interval of 10 s to be averaged for reliable values.

4. Results and discussion

4.1 Performance without economizer

Performances of the screw chiller without economizer are described first in this section. Variations of cooling capacities and COP's of R22 for various temperature conditions are shown in Fig. 3 as a function of compressor load factor. The performance is normalized by the performance value using R22 at the reference operation condition, i.e. 32℃ of the secondary heat transfer fluid temperature at the condenser inlet and 7℃ of the secondary heat transfer fluid temperature at the evaporator outlet. As load factor decreases, cooling capacity and COP decrease, because the performance of the compressor is degraded when operated at low load factors. As load factor changes from 100% to 75%, decrease of cooling capacity becomes 16%. As load factor shifts from 75% to 50%, the reduction of the cooling capacity is 34%, which is relatively large. The



(a) Cooling capacity (b) COP

Fig. 3 Cooling capacity and COP with respect to load condition of R22 (reference case with subscript o: R22, $T_{cond,in}$=32℃, $T_{evap,out}$=7℃).

change of COP at low load factor is greater than that at high load factor.

When external load condition changes, the load factor of screw chiller is adjusted automatically to meet the required cooling capacity. Thus from a practical point view, the relation between cooling capacity and COP which is shown in Fig. 4 is very important. The decrease in cooling capacity causes a decrease in COP, which can be expected from Fig. 3. COP variations due to condenser and evaporator temperatures is greater for large cooling capacity than those for low cooling capacity.

The trend of COP change can be explained by thermodynamic analysis. Thermodynamic irreversibility generated in a screw chiller mainly stems from friction loss and heat transfer by finite temperature difference. The main contributor to the loss is the compressor followed by the condenser and evaporator. The irreversibilities due to heat transfer at condenser and evaporator are proportional to total heat trans-

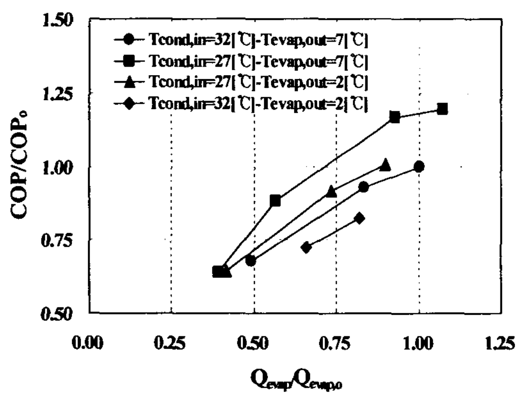


Fig. 4 COP change with respect to cooling capacity of R22 (reference case with subscript o: R22, $T_{cond,in}$=32℃, $T_{evap,out}$=7℃).

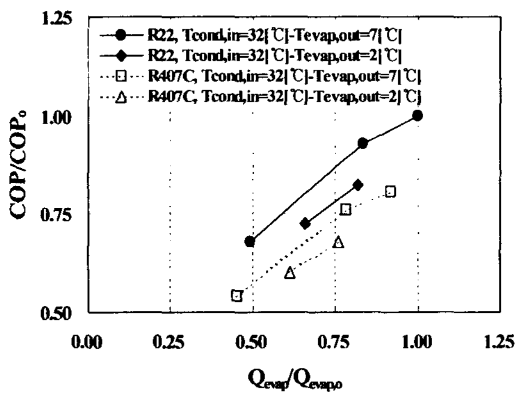
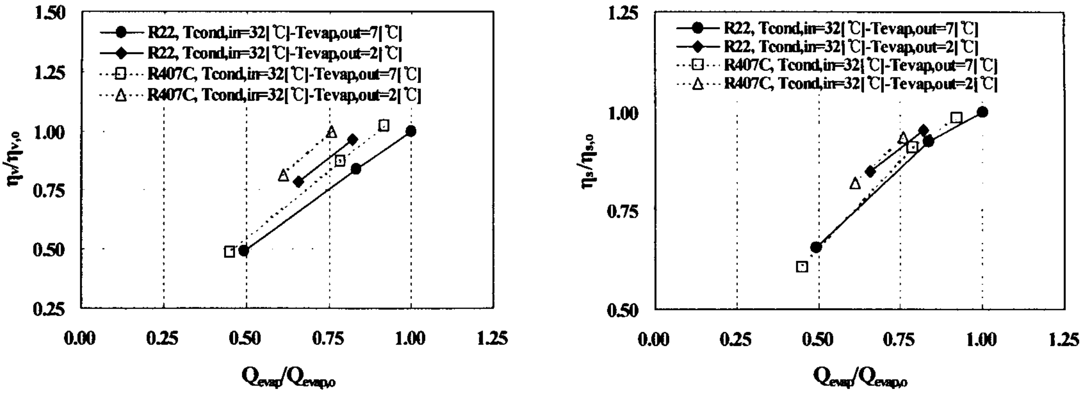


Fig. 5 COP change with respect to cooling capacity of R22 and R407C (reference case with subscript o: R22, $T_{cond,in}$=32℃, $T_{evap,out}$=7℃).



(a) Volumetric efficiency (b) Isentropic efficiency

Fig. 6 Compressor efficiency with respect to cooling capacity (reference case with subscript o: R22, $T_{cond,in}=32℃$, $T_{evap,out}=7℃$).

fer rate that occur in the condenser and evaporator, respectively. Thus, as cooling capacity decreases, the relative loss in the condenser and evaporator become smaller than those of other components. Performance variations due to condenser and evaporator temperature changes at lower cooling capacity become smaller.

The chiller performance using R407C is compared with that using R22 in Fig. 5. This figure shows COP versus cooling capacity at various temperature conditions. As in the previous figures concerning the performance, the performance of R407C is shown normalized by the performance value using R22 at the reference operation condition. R407C shows similar trend with R22, but the values are lower by 12~15%.

To compare the compressor efficiencies of R407C with those of R22, volumetric and isentropic efficiencies of a compressor as a function of cooling capacity are presented in Fig. 6. The volumetric and isentropic efficiencies are defined as follows,

$$\eta_v = \frac{\dot{m}_r}{\rho_{comp,in} VF} \tag{4}$$

$$\eta_s = \frac{m_r(h_{comp,out,isen} - h_{comp,in})}{W} \tag{5}$$

where $h_{comp,out,isen}$ is specific enthalpy after isentropic compression.

It is found that the volumetric and isentropic efficiencies of R407C is greater than those of R22 despite higher compression ratio owing to the lower heat transfer coefficient of R407C. It is considered that this is due to the difference in the refrigerant-oil solubility between R22 and R407C. Refrigerant is dissolved in the oil during the compression, and the refrigerant laden oil that is discharged from the compressor is separated from the refrigerant vapor stream by the oil separator. The separated liquid is then returned to the compressor suction space, where the reduction in pressure cause the refrigerant to flash from the oil. The vapor released from the oil fills part of the suction space and so reduces the effective volume of vapor drawn from the suction line. Consequently, this process results in decreases of both volumetric and isentropic efficiencies of compressor. Typically, the solubility of R22 in POE oil is reported quite higher than that of R407C,[9,10] which results in lower compressor efficiency.

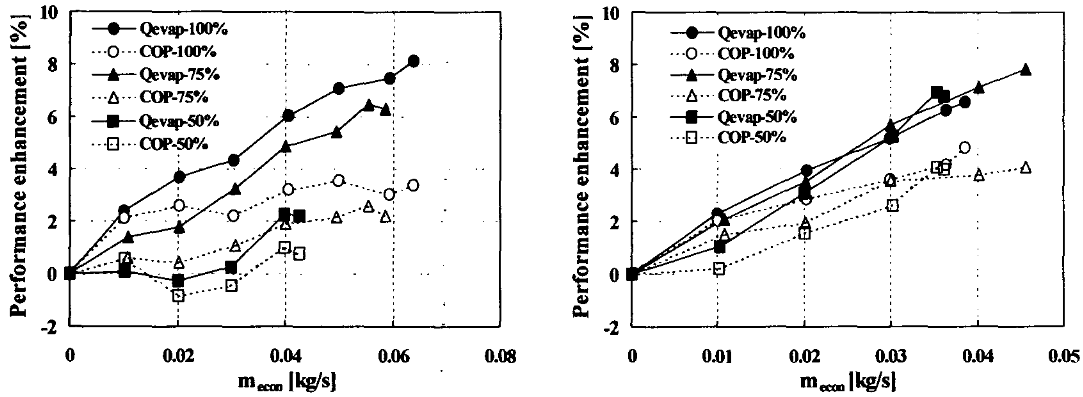
Therefore, as is presented in the previous study,[7] the performance degradation of R407C is mainly due to decrease in the heat transfer coefficient of R407C compared with that of

R22, and this is considered to be an appropriate explaination for the performance degradation of R407C at part-load condition.

4.2 Performances with economizer

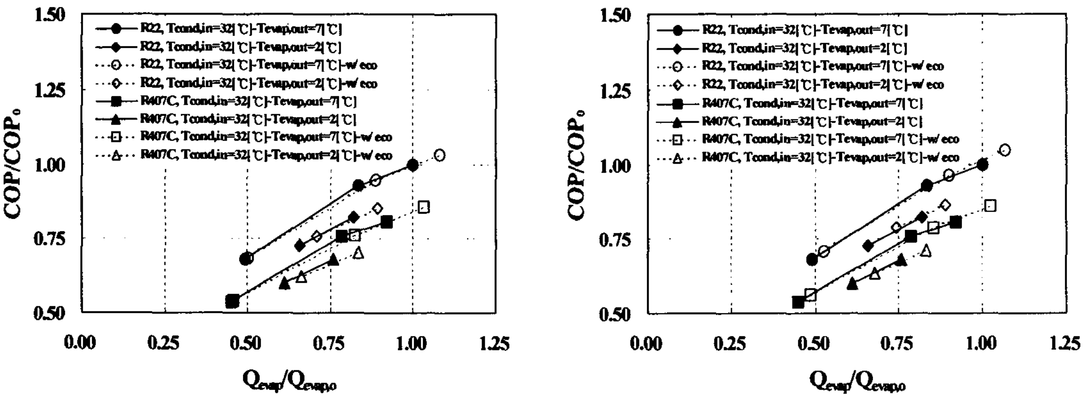
The results of performance tests with economizer are presented and analysed next. Figure 7 shows the influence of mass flow rate of injected refrigerant in economizer port on cooling capacity and COP. The reference is chosen with R22 at reference operation condition. Economizer volume ratio of 1.01 means that the economizer port is positioned just after the suction process for 100% load fraction. When the economizer volume ratio is 1.17, the economizer port position is closer to the discharge port than that of economizer volume ratio of 1.01.

As a load factor of screw compressor decreases, the point of compression moves further along the rotor. This has the effect of connecting the economizer port directly to the suction port instead of the trapped volume. This occurs when an economizer port is po-



(a) v_i=1.01 (b) v_i=1.17

Fig. 7 Performance test with variable economizer mass flow rate under part-load condition of R22 (reference case with subscript o: R22, $T_{cond,in}$=32℃, $T_{evap,out}$=7℃).



(a) v_i=1.01 (b) v_i=1.17

Fig. 8 COP with respect to cooling capacity (reference case with subscript o: R22, $T_{cond,in}$=32℃, $T_{evap,out}$=7℃).

sitioned just after the start of the compression process.

As the cooling capacity decreases, performance increase due to economizer becomes small and under 50% of cooling capacity, effect of economizer becomes insignificant. Since the economizer port is closer to the discharge port for an economizer volume ratio of 1.17, the performance is less affected by the part load than that of 1.01. In general, rate of cooling capacity and COP increase becomes maximum when the injection mass flow rate of economizer is greatest. This trend applies both to R22 and 407C.

From a practical point view, the important thing is the relation between cooling capacity and COP as shown in Fig. 4. In Fig. 8, the chiller performances are compared with and without economizer for both R22 and R407C.

Adoption of an economizer system increases the cooling capacity and COP. When chiller runs with economizer at full-load condition, the cooling capacity and COP exceed the rated nominal performance values. However, at partial load, the use of economizer results in the increase of both COP cooling capacity condition. Thus for the same cooling capaicty, COP increase due to the economizer is insignificant. When the compressor is operated at economizer volume ratio of 1.01 and lower load factor condition, COP may rather decrease compared with that of non-economizer system under the same cooling capacity condition. This is because the enhancement of cooling capacity exceeds that of COP.

By installing an economizer, chiller using R407C has similar value of cooling capacity with that of non-economizer chiller using R22, even though the COP of R407C is lower by 14% than that of R22.

5. Conclusions

In this study, performance test is performed for a screw chiller using R22 and R407C, and the effects of part-load operation and economizer system on chiller performances are analyzed.

The decrease in cooling capacity of chiller causes an decrease in COP at part-load operation. For large cooling capacity, the effect of condenser and evaporator temperatures on COP is large. However, as cooling capacity decreases, this effect becomes small.

Adoption of an economizer system increases the cooling capacity and COP except for lower compressor load factor when economizer volume ratio is 1.01. This results because at this operation condition, the economizer port is connected directly to the suction port.

By installing an economizer, the cooling capacity and COP can be increased over the rated nominal performance, and the cooling capacity of chiller with economizer using R407C is found to be similar with that of non-economizer chiller using R22.

References

1. Singh, P. and Patel, G., 1984, A generalized performance computer program for oil flooded twin-screw compressors, International Compressor Engineering Conference, Purdue, pp. 544-551.
2. Fujiwara, M., Kasuya, K., Matsunaga, T. and Watanabe, M., 1984, Computer modeling for performance of rotary screw compressor, International Compressor Engineering Conference, Purdue, pp. 536-543.
3. Hanjalić, K. and Stošić, N., 1997, Development and optimization of screw machines with a simulation model—Part II: Thermodynamic performance simulation and design optimization, ASME J. Fluids Engineering, Vol. 119, pp. 664-670.
4. Jonsson, S., 1991, Performance simulations of twin-screw compressors with economizer, Int. J. Refrig., Vol. 14, pp. 345-350.

5. Högberg, M. and Berntsson, T., 1994, Non-azeoptropic mixtures as working Fluids in two-stage economizer-type heat pumps, Int. J. Refrig., Vol. 17, pp. 417-429.
6. Chang, Y.-S., Shin, Y., Kim, Y. and Baik, Y.-J., 2002, In-situ performance analysis of centrifugal chiller according to varying conditions of chilled and cooling water, KSME J (B), Vol. 26, No. 3, pp. 482-490.
7. Lee, D., Ahn, Y., Kim, Y., Kim, Y., Chang, Y.-S. and Nam, L.-W., 2002, Experimental investigation on the drop-in performance of R407C as a substitute for R22 in a screw chiller with shell-and-tube heat exchangers, Int. J. Refrig., Vol. 25, No. 5, pp. 575-585.
8. Park, S.-H., Chang, Y.-S., Kim, Y., Kim, Y. and Nam, L.-W., 2002, Simulation of the economizer performance of a screw compressor using R22 and R407C, KSME J (B), Vol. 26, No. 3, pp. 465-473.
9. Hewitt, N. J., McMullan, J. T., Mongey, B. and Evans R. H., 1996, From pure fluids to zeotropic and azeotropic mixtures: the effects of refrigerant-oil solubility on system performance, Int. J. Energy Research, Vol. 20, pp. 57-67.
10. Macaudiere, S. and Rolotti, G. D., 1997, Refrigeration lubricants for R-407C, Proceedings of the ASME, Advanced Energy Systems Division, Vol. 37, pp. 405-410.