EXPERIMENTAL RESULTS AND EVALUATION OF 2-STAGE COMPRESSION AND EXPANSION CYCLE USING CO2

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ABSTRACT

Natural refrigerants used in the refrigeration and air-conditioning systems are gathering attention as public interest in environmental issues increase. Amongst these, CO_2 is a non-combustible, non-toxic, safe refrigerant. Development of applications for using its beneficial properties for heat generation with a heat pump is currently being pursued. However, CO_2 is not used for low temperature applications as it produces poor cycle efficiency at low temperatures.

We considered a two-stage compression, two-stage expansion cycle as a way of improving cycle efficiency when using CO_2 refrigerant. The two-stage compression expansion cycle has a gas-liquid separator installed at the intermediate pressure point in the middle of the pressure-reducing circuit of a common refrigeration cycle using a two-stage compressor. The refrigerant is separated into gas and liquid, with the gas returning to the compressor's second stage inlet, and the liquid being further reduced in pressure and fed to the evaporator.

Presuming that an identical cycle was used at low temperatures, the two-stage expansion cycle is effective because the CO_2 refrigerant is drier at the entrance of the evaporator than an HFC refrigerant.

The main paper reports the characteristics we investigated using a test version of the two-stage compression two-stage expansion cycle that we built.

Keywords: CO_2, two-stage compressor, two-stage expansion cycle.

1 INTRODUCTION

An effort is underway in Japan to switch from CFC to HFC refrigerants as a way of limiting damage to the ozone layer, and at the same time interest is growing in natural refrigerants with low global warming potential. Among these, CO_2, unlike HC and ammonia, is not a problem in terms of toxicity or flammability, and would be an excellent refrigerant to mitigate the environmental problem described above, but so far little progress has been made applying CO_2 to applications other than heat pump water heaters and mobile air conditioners.

The reason is because of two disadvantages of the refrigerant: the fact that the CO_2 cycle has a low COP (coefficient of performance), and that it requires a high-pressure cycle. Therefore improvements need to be made to make the refrigeration cycle more efficient so that CO_2 may be used, particularly in low temperature applications.

On the other hand, the authors have been developing hermetic type rotary two-stage compressors and are engaged in developing and commercializing a number of applications for CO_2 in Japan, including residential heat pump water heaters.

In this research, we studied a two-stage expansion cycle utilizing a gas/liquid separator using a hermetic type, rotary two-stage compressor to improve COP. The effect of this experiment was estimated, actual compressors were test-produced, and a refrigeration cycle evaluation was performed, all of which is reported in this paper.

2 ISSUUES WITH THE CO_2 REFRIGERATION CYCLE

The following are issues that must be addressed in applications employing the CO_2 cycle at low temperatures.

1) For the CO_2 cycle at low temperatures, a working pressure of 7 - 10MPa at the high-pressure side and 1 - 2MPa at the low-pressure side are necessary, making this a high-pressure cycle also characterized by a high-pressure differential.
2) For the CO_2 cycle at low temperatures, CO_2 has greater quality than other refrigerants and its COP is markedly lower. For example, in an ordinary refrigeration cycle with other refrigerants such as R134a or R600a, when the condensation temperature is 30°C, supercooling is at 0°C and the evaporation temperature is -25°C, the quality of the other two refrigerants is 0.34 and 0.33, respectively. In contrast, in a CO_2 cycle at high-pressure, 7.5MPa, and a temperature of 30°C at the evaporator inlet, the quality is 0.50 and the latent heat region is very small.

Two-stage compressors and a two-stage expansion cycle were examined to resolve the above issues and improve COP. Details are reported below.

2.1 Two-Stage Compressor

For the compression system, an internal high- type rotary two-stage compression system with two rotors was used to deal with the high-pressure and high pressure differential of the CO_2 cycle described above. The advantages of this system are as follows:

1) Two separate compression processes were created, which helped reduce the pressure differential in each compression unit, and this system is also expected to improve efficiency since the compression ratios can also be reduced.
2) Heat exchangers are placed in the first-stage discharge and second-stage suction piping areas. These are expected to make intermediate refrigeration possible, reduce the temperature of second-stage discharge gas, and increase reliability, while also raising the efficiency by lowering electric power for compression.
3) This system can also be applied easily to the two-stage expansion cycle, for example, by connecting return piping for the gas phase refrigerant from the gas/liquid separator in the first-stage discharge and second-stage suction piping.
4) An internal high-pressure system was used, such that second-stage discharge gas was discharged to a shell case and the pressure inside the case was high. Because this enables the use of the space in the shell case as a separator space, the amount of discharging oil may be reduced.
5) Different displacement volumes are set for each of the first and second-stage compression units in a two-stage compression system with two rotors; this construction compresses refrigerant in order, from the compression unit with high displacement volume to the compression unit with low displacement volume. The result is low noise and low vibration, because of the smoothing of torque fluctuations, an advantage of a two-rotor system.

2.2 Two-Stage Expansion Cycle

Figure 1 shows a P-h diagram of the two-stage expansion cycle. The two-stage expansion cycle is a cycle in which the expansion portion of the ordinary refrigeration cycle is performed by two expansion mechanisms and a gas/liquid separator is placed in the intermediate pressure portion between them. The two-phase refrigerant is separated into the gas phase and liquid phase, the gas phase is sent to the compressor, and the refrigerant liquid is further depressurized and sent to an evaporator. The effect of this system is that the refrigerating effect is enhanced because quality is reduced on the evaporator side, and the electric power for compression is reduced, resulting from the inflow of gas refrigerant. As the above indicates, in the CO_2 cycle, under low temperature cycle conditions, the quality of the low-pressure portion is greater as compared to other refrigerants, and therefore the two-stage expansion cycle can be expected to be very effective.

Figures 2 and 3 show the major circuit diagrams of the two-stage expansion cycle. Figure 2 is a circuit diagram of a system with a combined two-stage compressor and two-stage expansion cycle; the gas phase refrigerant from the gas/liquid separator is made to converge in the two-stage compressor's first-stage discharge and second-stage suction piping. In the circuit diagram, in order to make the CO_2 cycle more efficient, an internal heat exchanger that could conceivably be located on a regular basis is placed so that heat exchange occurs between the piping in front of the first expansion valve (from the GC outlet) and the compressor suction piping from the evaporator, as well an intercooler is placed in the second-stage suction piping from the first-stage discharge area. Similarly, Figure 3 is also a circuit that combines a two-stage compressor with a two-stage expansion cycle, but the gas phase refrigerant from the gas/liquid separator is injected into the cylinder of the first-stage compression unit of the two-stage compressor. These two-stage expansion cycles are cycles in which the gas refrigerant, separated by the gas/liquid separator, is returned to the compressor side by the pressure differential. In the cylinder injection type cycle shown in Figure 3 it is possible to have a cycle in which the pressure within the gas/liquid separator is lower than in the case of the convergent piping type cycle of Figure 2. So these cycles have great merits, for example, for applications where the required low temperature regions fluctuate because of seasonal or other factors.

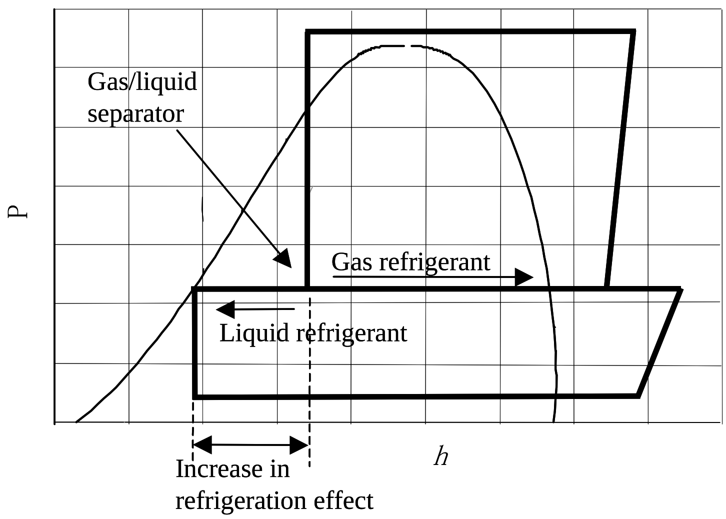

Fig. 1. P-h diagram of the two-stage expansion cycle.

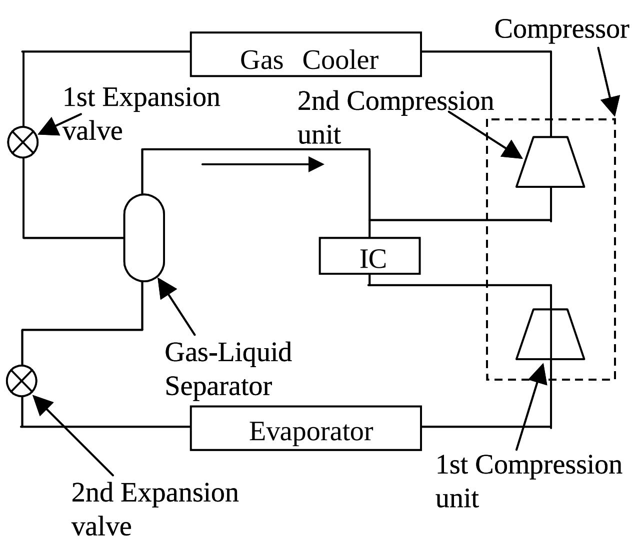

Fig. 2. Convergent piping type two-stage expansion cycle.

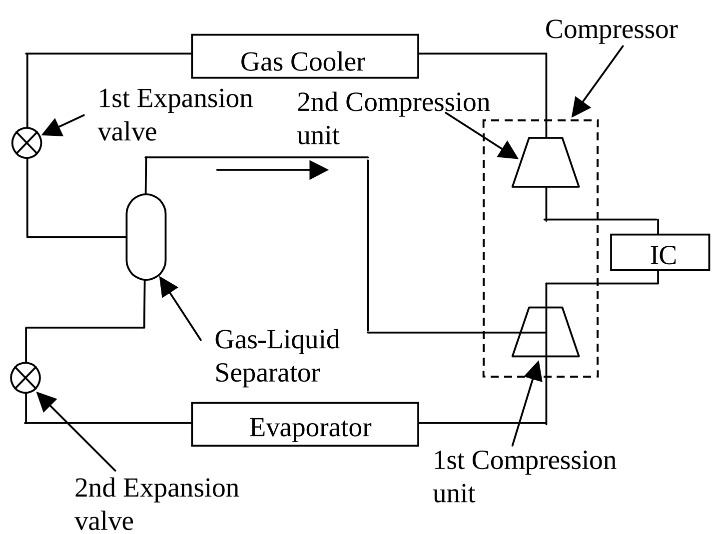

Fig. 3. Cylinder injection type two-stage expansion cycle.

3 CALCULATION OF COP OF TWO-STAGE EXPANSION CYCLE USING TWO-STAGE COMPRESSOR

The COP of the two-stage compression + two-stage expansion cycle shown in Figures 2 and 3 was calculated. For comparison, the COPs for two-stage compression cycles (Figure 4) and single-stage compression cycles (Figure 5) were also calculated. Conditions of the calculations are as follows:

- High pressure is 7.5MPa, gas cooler outlet temperature is 30°C, intercooler outlet temperature is 30°C, and suction gas temperature is 30°C
- Heat recovery by the internal heat exchanger is equivalent to 80% of superheat enthalpy
- There is no loss of pressure
- The gas compression is adiabatic (isentropy)
- The gas/liquid separator causes complete separation of gas and liquid, and all of the resulting gas refrigerant flows to the compression side

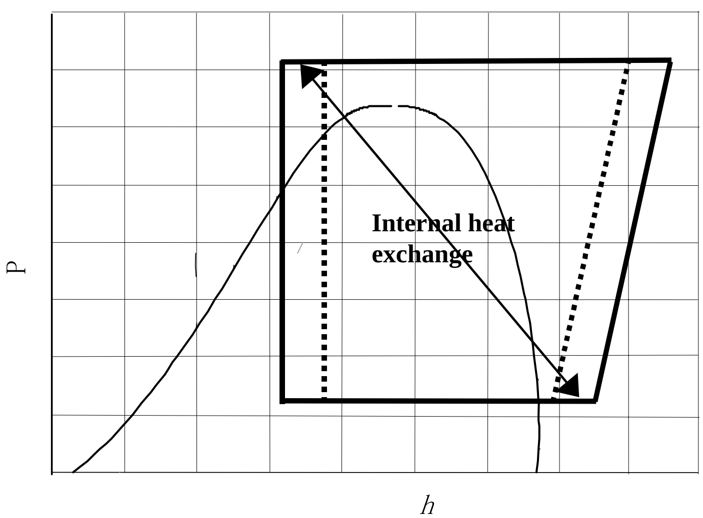

Fig. 4. P-h diagram of ordinary cycle.

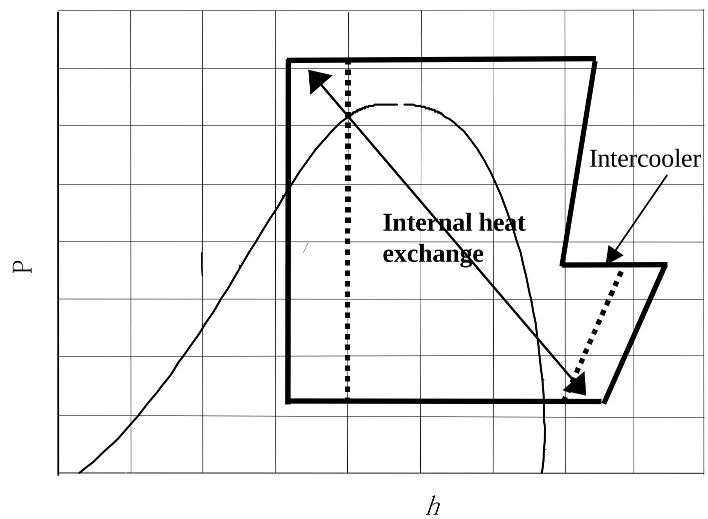

Fig. 5. P-h diagram of two-stage compression cycle.

The efficiency of these two-stage expansion cycles depends on the intermediate pressure of the gas/liquid separator portion. When the intermediate pressure is low, the amount of gas refrigerant that returns to the compressor from the gas/liquid separator is small, the increase in refrigerating effect is also small and the overall effectiveness is minimal. If the intermediate pressure is high, the liquid refrigerant from the gas/liquid separator also returns to the compressor and efficiency declines, so an optimal value must exist. Additionally, when using two-stage compressors the intermediate pressure is determined by the displacement volume ratio of the first and second stages.

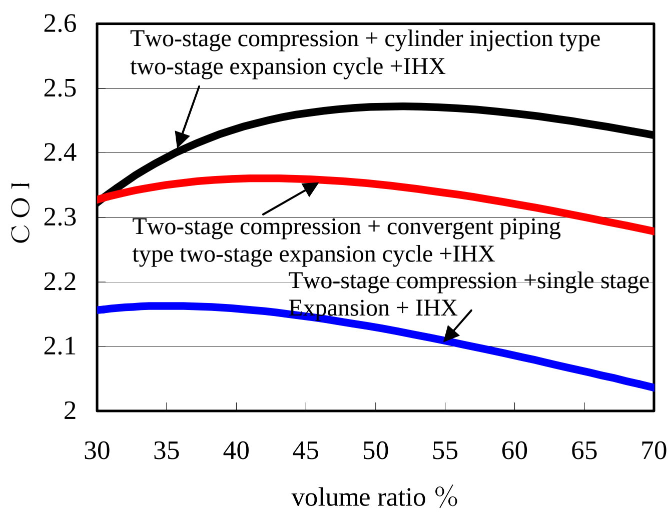

Fig. 6. Correlation of compressor volume ratio with each cycle's COP

Table 1. Max COP and optimal volume ratio calculation results for each cycle

	MAX COP	Optimal volume ratio
Ordinary cycle (with IHX)	1.88	—
Two-stage compression cycle (with IHX)	2.16	35.0%
Two-stage compression + convergent piping type two-stage expansion cycle (with IHX)	2.47	50.8%
Two-stage compression + cylinder injection type two-stage expansion cycle (with IHX)	2.36	42.5%

The relationship between COP and the volume ratio of each cycle is shown in Figure 6. Table 1 gives the maximum COP for each cycle and the volume ratio that occurs at that maximum COP. A maximum COP value can be ascertained for each cycle of two-stage expansion, and an optimal volume ratio can exist.

4 EXPERIMENTAL RESULTS

4.1 Test specifications

Based on the above study, compressors, related components, etc. were test-produced to evaluate refrigeration cycles.

4.1.1 Compressors

Specifications of the compressors that were test-produced at this time are shown in Table 2. Two compressors were made: one for cycles with convergent piping and one for the cylinder injection type.

Table 2. Specifications of compressors used

	Convergent piping type	Cylinder injection type
Compressor type	Hermetic type compressor	
Compression form	Two-stage compression rotary type	
Motor	DC inverter motor	
Displacement volume	First-stage compression unit 1.0cc	First-stage compression unit 1.0cc
	Second-stage compression unit 0.65cc	Second-stage compression unit 0.45cc
Volume ratio (2nd/1st)	57%	45%
Oil	PAG oil	
Size	OD ϕ117.2 x H195.4mm	

Figure 7 shows the flow of refrigerant in these internal high-pressure two-stage compressors. Low-pressure refrigerant is taken into the first-stage compression unit located in the lower part of the compressor and compressed. The gas, now under intermediate pressure, is discharged one time into the piping outside the case. Passing through the piping, the gas is then taken into the second-stage compression unit located in the upper portion of the compressor. The gas put under high-pressure by the second-stage compression unit is discharged to the case chamber and subsequently discharged, from the case, by the discharge pipe at the upper part of the motor.

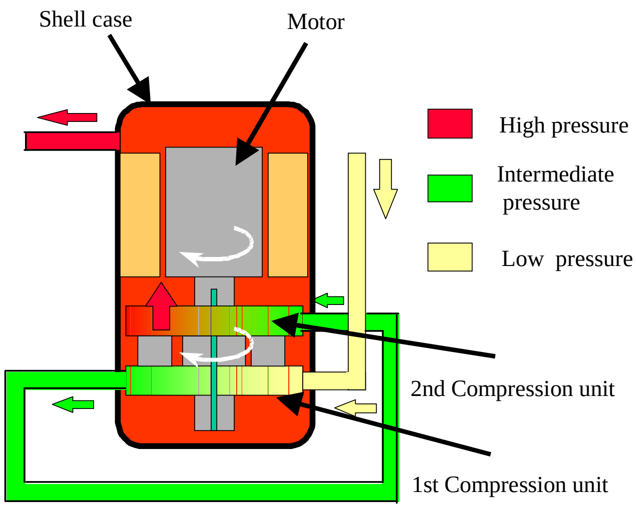

Fig. 7. Internal high-pressure type two-stage compressor

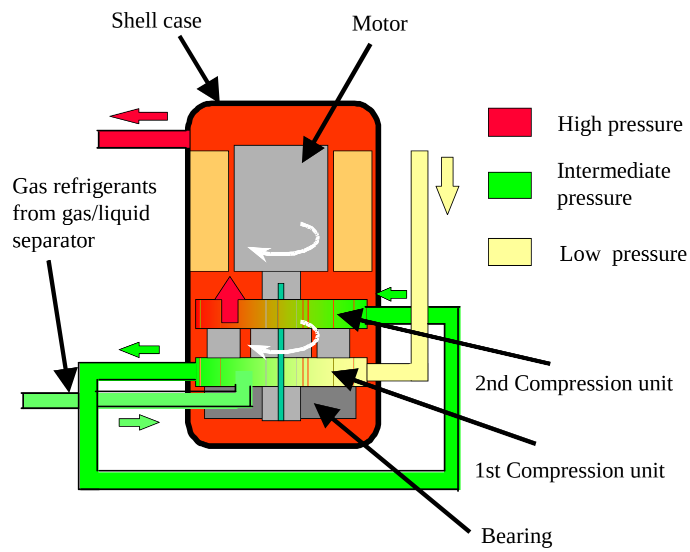

Fig. 8. Two-stage compressor for cylinder injection type two-stage expansion cycle

In the case of the convergent piping type unit, the gases returning from the gas/liquid separator converge in the first-stage discharge and second-stage suction line and are taken into the second-stage compression unit. If instead a cylinder injection type is used, a suction port is placed in the lower bearing so that gases will be injected into the first-stage compression unit, as illustrated in Figure 8. The movement of the roller causes the injection port to open and close, but because of such factors as the need to prevent backflow to the gas/liquid separator, there is believed to be an optimal port position and size. In this prototype, positions and diameters were chosen randomly for the injection port.

4.1.2 Gas/liquid Separator

Specifications of the gas/liquid separator used in the two-stage expansion cycle are given in Table 3. In this experiment, a mesh was placed near the connection to the gas refrigerant piping (upper part) with the intent of assuring separation at the gas/liquid separator.

Table 3. Specifications of gas/liquid separator

Size	OD ϕ70 × H220
Capacity	600cc
Other	Mesh placed at connection to refrigerant gas outlet piping (upper part)

4.1.3 Refrigeration Circuit

The experiment was done with the circuit configuration shown in Figure 9; the amount of refrigerant in circulation was found by calculating from the heat exchange volume at the evaporator. Intermediate pressure is set by adjusting the opening of the two expansion valves located in front of and behind the gas/liquid separator.

Additionally, in order to simplify the evaluation of the two-stage expansion cycle, the experiment was performed without an internal heat exchange or intercooler (IC).

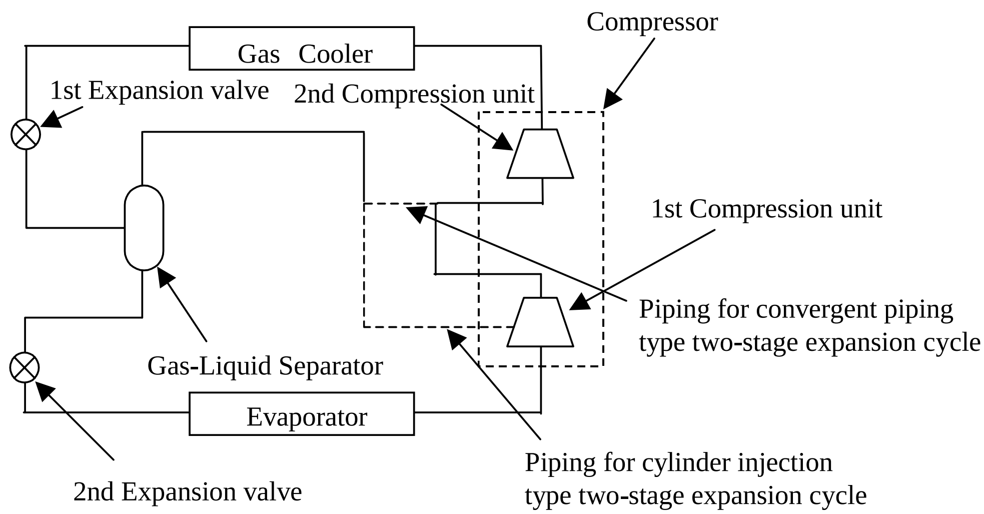

Fig. 9. Convergent piping type two-stage expansion cycle.

4.2 Results

4.2.1 Convergent Piping Cycle

Figure 10 shows the relationship between refrigerating capacity and COP when the intermediate pressure was varied in the convergent piping cycle. As the figure indicates, the increases over ordinary two-stage compression cycles at intermediate pressure with the greatest COP were 37.9% and 21.8% respectively. As the intermediate pressure declines, the refrigerating capacity also declines. This is believed to be because the amount of returning gas refrigerant declines as intermediate pressure

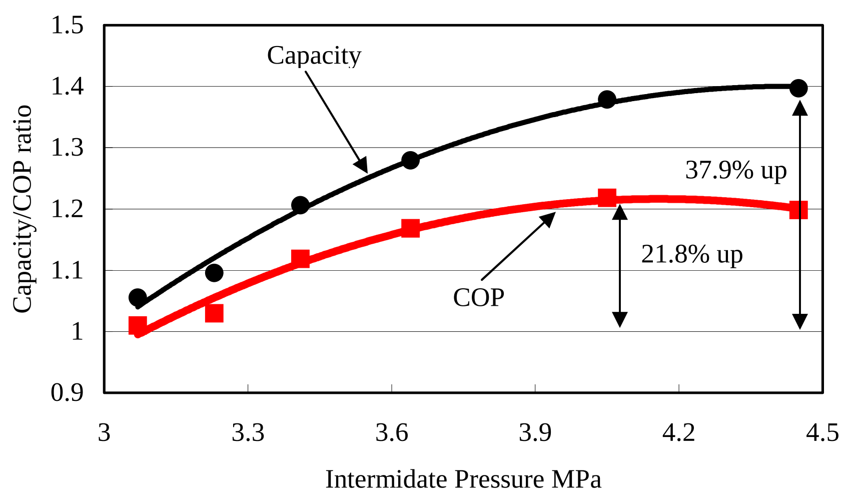

Fig. 10. Capacity and COP ratio in convergent piping type two-stage expansion cycle

declines, thereby minimizing the increase in refrigeration effectiveness. The experiment confirmed that an optimal value exists for refrigeration COP according to changes in intermediate pressure.

4.2.2 Cylinder Injection cycle

Figure 11 shows the relationship between refrigerating capacity and COP when the intermediate pressure was varied in the cylinder injection cycle. As the figure clearly indicates, the increases over the ordinary two-stage compression cycles at intermediate pressure with the greatest COP were 20.0% and 12.6% respectively, and the COP improvements anticipated from calculations were found. This is believed to be because the diameter and the location of the compressor's injection port was suitable, making it possible to get enough gas refrigerant returning to the compressor across the range of test conditions.

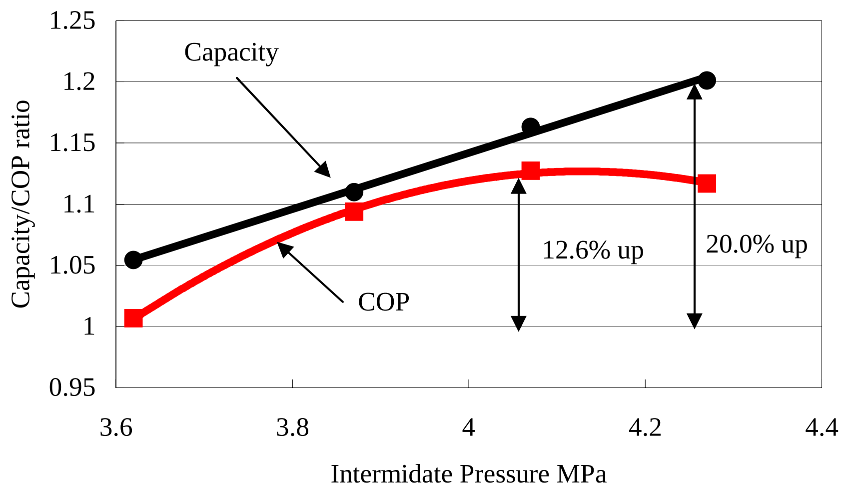

Fig. 11. Capacity and COP ratio in Cylinder injection type two-stage expansion cycle

5 CONCLUSION

A two-stage expansion cycle using a two-stage compressor was examined as a way of improving the COP of the CO_2 cycle and the following conclusions were drawn.

1. In the convergent piping type two-stage expansion cycle, in which the gas refrigerant is made to converge in the second-stage suction, the increase in refrigerating capacity was found to be 37.9% and the increase in COP, 21.8%, over ordinary cycles.
2. In the cylinder injection type two-stage expansion cycle, in which the gas refrigerant is injected into the cylinder of the first-stage compression unit, the increase in refrigerating capacity was found to be 20.0% and the increase in COP, 12.6%, over ordinary cycles. Further COP improvements can be expected by optimizing the location and diameter of the gas refrigerant injection port of the compressor.

This experiment has demonstrated the COP improving effect of two-stage expansion cycles using two-stage compressors. The authors will continue to develop this technology, evaluating it in more realistic cycles that include an internal heat exchanger, optimizing the compressor and refrigeration circuit, seeking to further improve COP and help expand low-temperature applications of CO_2.

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