2016

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ABSTRACT

CO_2 refrigerant is expected to be used for air conditioners due to its environmental property of GWP = 1, non-flammability and low toxicity and has already been applied to hot water supply system, refrigerators and freezers. However, since theoretical cycle COP of CO_2 is poorer than that of conventional HFC in applying to air conditioners, it cannot be expected to improve the performance significantly by improving the efficiency of components adopted in the conventional single-stage CO_2 cycle. Therefore, it is important to consider efficient refrigerant cycle, especially in the performance of cooling which is a weak point of CO_2. Challenges also lie in reducing weight of components such as compressor and heat exchangers which have relatively heavy weight to resist high pressure and to achieve high efficiency in the system.

In this study, a prototype of CO_2 refrigerant VRF air conditioning system is constructed and evaluated the performance. The system is adopting the four-stage compression cycle applying a new type compressor and heat exchangers and so on aimed for high efficiency cycle and reducing weight.

As a result of the evaluation, it is confirmed that CO_2 cycle performance can be improved in similar extent to that of HFC refrigerant cycle. On the other hand, this high efficiency cycle consists of a lot of components and becomes more complex than HFC refrigerant cycle; therefore the size and weight of outdoor unit increases to settle these components. Further technical development will be necessary to overcome these weaknesses.

1. INTRODUCTION

It is expected that CO_2 refrigerant used for water heater and refrigerator and freezer is also considered to apply to air-conditioner since its prominent property of the Global Warming Potential (GWP) being one, non-flammability and low toxicity. However, when adopting as air-conditioner, The Coefficient Of Performance (COP) is worse comparing with conventional HFC refrigerant and significant improvement of efficiency cannot be expected by improving the efficiency of components in conventional single cycle.

Therefore improvement of cycle efficiency, especially improving the cooling performance where CO_2 refrigerant shows poor performance, is significant issues to overcome. Counter measures can be made by following three approaches. First, compression power would reduce by using multi-stage compression which is operated compressing and cooling repeatedly. Second, the refrigerant expansion loss would be recovered. Third, the cycle efficiency would be enhanced by using injection and installing internal heat exchangers. Several practical measures are found by the literature review to have been taken so far as following way.

Compression power reduces by applying oil injection device and adopting iso-thermal compression process which requires cooling in each multi-stage compression, and ejector which rises suction pressure of compressor. Expansion loss reduces by applying expander as expansion device to recover expansion energy. Economizer cycle consisting injection device and internal heat exchanger increase the capacity of evaporator and reduce compression power [1]-[4].

These measures cause side effect such as increasing the quantity and size of the equipment, complicating the refrigerant cycle by a lot of number of piping. In addition, as CO_2 refrigerant requires high pressure resisting strength of containers and pipes, product weight increases significantly. Therefore, weight reduction of components and simplification of piping are important to suppressing product total weight.

In this study, the CO_2 refrigerant VRF air conditioning system with the four-stage compression cycle applying a new type compressor and heat exchangers and so on is adopted to enable to high efficiency cycle and reduce weight. The prototype is constructed and measured the performance to verify the system improvement.

2. FOUR-STAGE COMPRESSION AND COOLING CYCLE

2.1 System outline

Figure 1 shows the schematic diagram of cooling mode considered in this study. In the cooling mode, dividing compression process into four processes, each process consists of compression and cooling in order, can reduce the compression power. Further reduction of compression power is possible by providing power recovery from the expansion energy. In addition, the compression power in first and second compression can be reduced by using economizer cycle with injection to suction line before third compression. Consequently, compression power can significantly decrease comparing to the single-stage compression cycle.
On the other hand, in the heating mode, cooling in compression process is suspended since the heating energy is extracted into ambient, resulting the heating capacity turns less. Therefore, the inter cooler to cool the exhaust vapor is functioning as evaporator in this case.

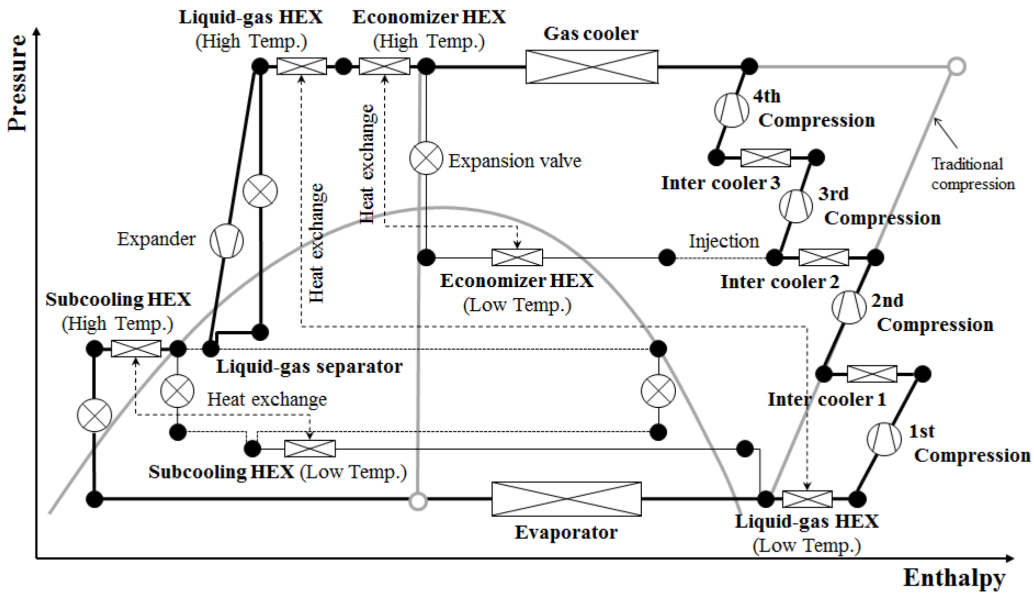


Figure 1: P-h diagram of high efficiency four-stage compression and cooling system

2.2 Theoretical performance

Figure 2 shows the results of theoretical cooling COP. Although the two-stage compression and internal cooling system practically used in refrigeration and freezer certainly enhances the efficiency against single-stage compression, theoretical COP reaches to only over 80% of R410A even adopting the expander in the system. It is hard to accomplish the same efficiency with R410A in two-stage compression and internal cooling system.

However, the four-stage compression and cooling with adopting expander can improve COP to 37pt from single-stage cycle without expander. Therefore, aiming to the same capacity with HFC refrigerant cycle, reduction of compression power by four-stage compression and cooling and recovery of expansion energy by using expander is indispensable.

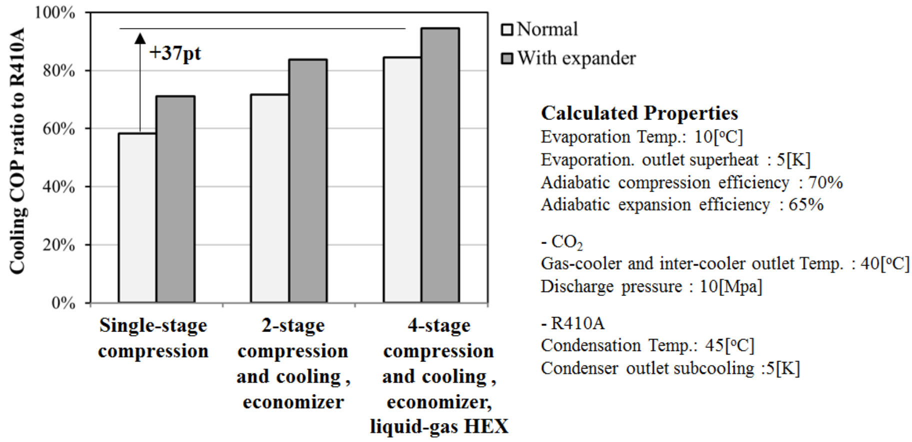


Figure 2: Theoretical cooling COP

3. TEST SYSTEM

3.1 Refrigerant circuit

Figure 3 shows the refrigerant circuit of this test system. The circuit is considered to obtain high efficiency for both cooling and heating mode for an air-conditioner. Passage switching mechanism and check valve are used in switching cooling and heating operation. Thereby three intercoolers in cooling mode can work as evaporator in heating mode. Refrigerant flows in parallel with gas cooler and in series with intercooler.

Bridge circuit which composes of check and expansion valves is adopted to utilize expansion recovery by expander and economizer effectively in the heating mode. Thereby, in heating mode, the high pressure refrigerant flow after heat exchanger of indoor unit can pass through expander and economizer in an adequate order.

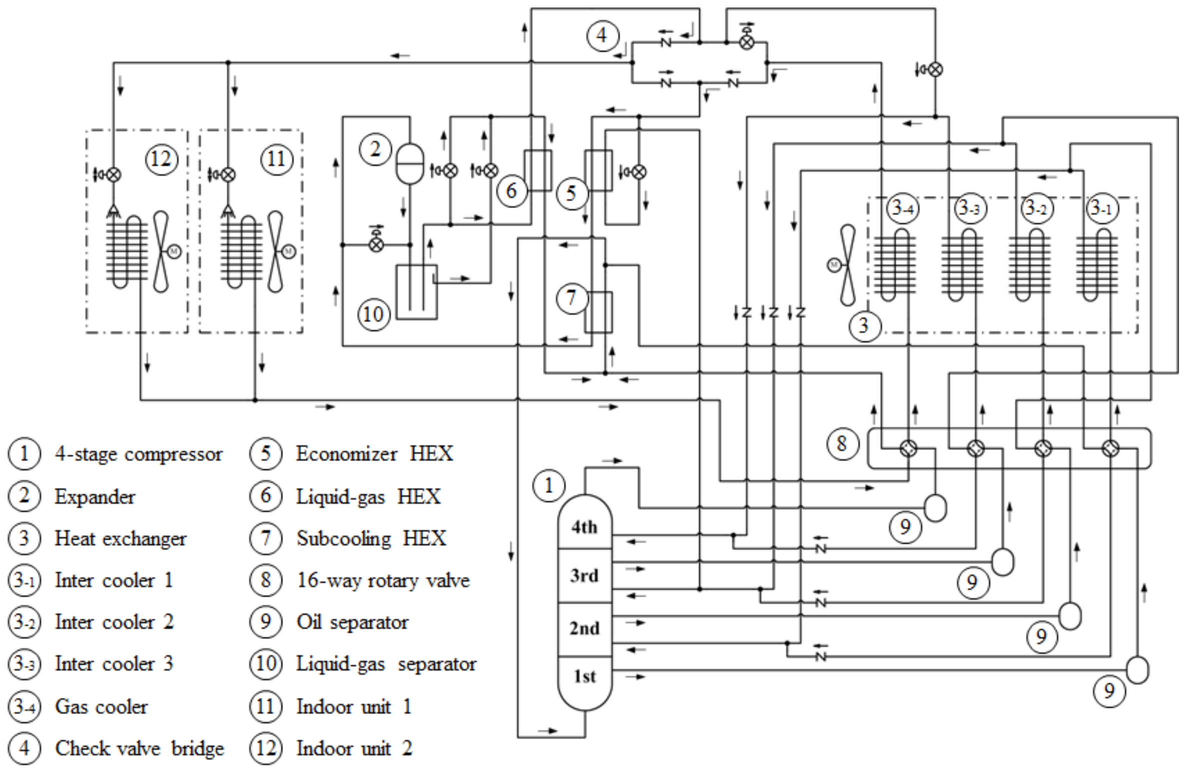


Figure 3: Refrigerant circuit of test system

3.2 Outdoor unit

The specifications of outdoor unit are listed in Table 1. Both cooling capacity is 14 kW and face area and length of heat exchanger are the same as the one of conventional R410A air conditioner. Specifications of each component are far different from conventional one to enable high efficiency, weight reduction and simplification of piping.

The casing size is 300mm wider than conventional one to install additional equipment, such as three internal heat exchangers and oil separators and an expander. As a result, the volume of the outdoor unit is 146% larger than that of R410A as shown in Figure5. It can be seen that internal spaces are well tidy packed with many components and pipes.

Table 1: Specifications of typical Outdoor unit of R410A regular product and CO_2 test equipment

Refrigerant			**R410A**	**CO_2**
Nominal capacity	cooling	[kW]	14.0	
	heating	[kW]	16.0	
Casing dimensions	height	[mm]	1,680	
	depth	[mm]	765	
	width	[mm]	635	930
Compressor			Single-stage scroll type	4-stage new type
Expander			Non	Single-stage swing type
Heat exchanger			Plate finned coil	Micro channel
Internal heat exchanger	economizer		Copper double tube	Aluminum double tube
	subcooling		Non	
	liquid- gas		Non	

Figure 4: Outdoor unit in test operation

Figure 5: Relative internal structure

3.3 Compressor

Simple four single-stage compressors or two two-stage compressors are necessary in compressing with four stage compression. However, adopting these, size and weight of outdoor unit increase. Moreover, management of lubricant oil is very difficult in series displace multiple compressors. To tackle this problem, we introduced new compression mechanism that four-stage compression housing in one compressor. While conventionally one compression room forms in one piston, the new-type compressor can form four compression rooms in one piston.

The operation mechanism is shown in Figure 6. In detail, swing piston with sliding vane and bush is assembled into cylinder groove, slides according to the rotating piston angle to accomplish proper compression work. Total eight compression rooms are formed within two pistons and two compression rooms with 180° phase difference are allowed to each stage compression to decrease pressure pulsation occurring at the inlet and outlet section.

The schematic diagram of mechanism with cross sectional view is shown in Figure 7. Since simple machinery with less moving parts is possible for many compression rooms owing to this mechanism, the number of parts and compressor size can decreases.

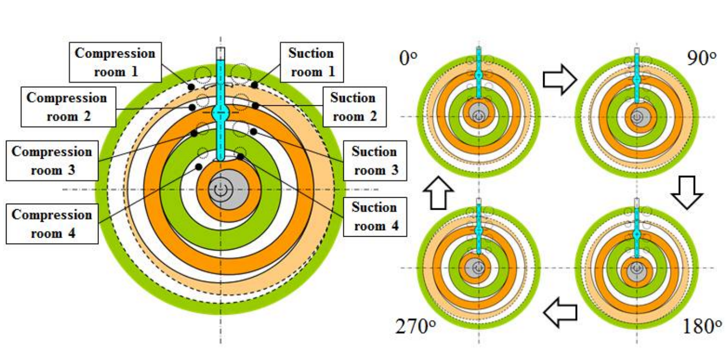


Figure 6: Operation mechanism

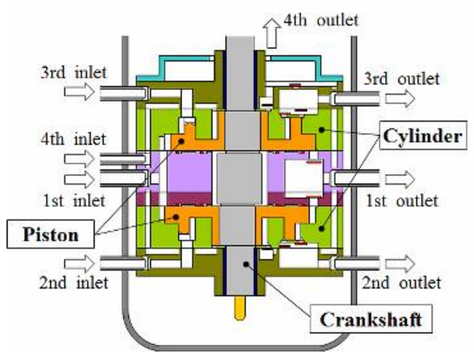


Figure 7: Structure outline

Figure 8 shows the piston and vane with bushes assembled in the compressor. Figure 9 shows the appearance of the prototype compressor. Although the outer diameter of the body is fixed with the single-stage compressor in the same class, the height is about 25% higher than the single-stage compressor. Eight ports are protruded in the body while suction and discharge lines of two pistons are in common.

The four-stage compressor has many restrictions in design as many parts should be compactly stored in one casing. Within this restriction, volumetric ratio of cylinder of each stage, the back vapor pressure of the piston and the shape of the refrigerant pass are optimized. Figure 10 shows the compressor test result. From the figure, it is seen that the results of four-stage compressor are satisfactory.

The loss of sliding contact becomes larger since many sliding sections exist in the new mechanism. Refrigerant pressure drop also becomes larger since internal refrigerant pass is complicated and suction and discharge parts increases. Therefore, the indicated efficiency becomes lower than that of single-stage compressor. On the other hand, dividing the compression process into two, leakage from compression room to suction line and between cylinders are less. Therefore the volumetric efficiency increases more than single-stage compressor. As a result, compression efficiency reached satisfactory level and especially superior in lower rotating speed.

Figure 8: Piston and vane with bush

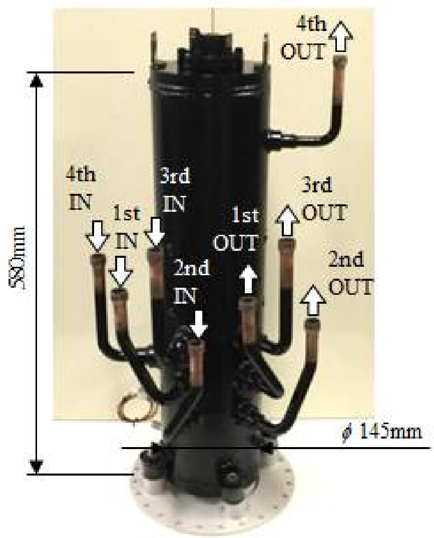


Figure 9: Four-stage compressor with one casing

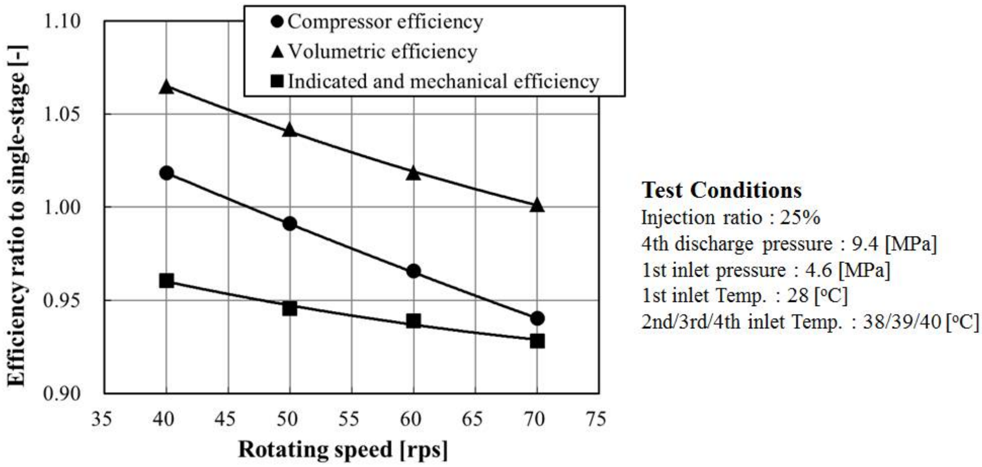


Figure 10: Compressor test result

3.3 Expander

Separate type of expander with electric power generator type is adopted in this system as shown in Figure 11. The mechanism is that the expansion energy of refrigerant convert to electricity and the electricity is applied to compressor power.

Figure 12 shows the appearance of prototype generator type expander. A muffler is equipped to mitigate the pulsation at the refrigerant inlet with high density. The swing type mechanism of expander is adopted since this type of compressor is available in our company and applied to manufacturing for the expander relatively easily.

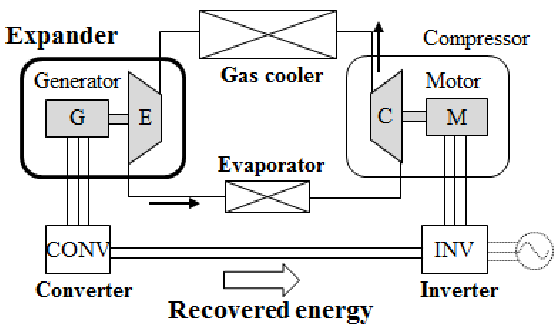


Figure 11: Energy recovery of expander

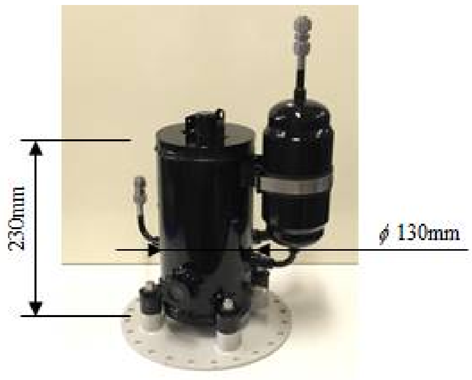


Figure 12: Generator type expander

3.4 Heat exchanger

Figure 13 shows the all-aluminum Micro Channel Heat Exchanger (MCHX) with high pressure resisting strength. The all-aluminum MCHX can reduce the product weight, while conventional fin-and-tube heat exchanger likely increases weight significantly since the tubes are thickened for resisting high pressure. The all-aluminum MCHX integrate three inter cooler and one gas cooler in cooling mode which are all functioning as evaporator by switching valves .

As for internal heat exchanger, since the system requires three internal heat exchangers, size and weight would increase if conventional double-tube heat exchangers made of copper are adopted. To solve this problem, all-aluminum new-type double tube heat exchangers are adopted and coil it up in installation as shown in Figure 14.

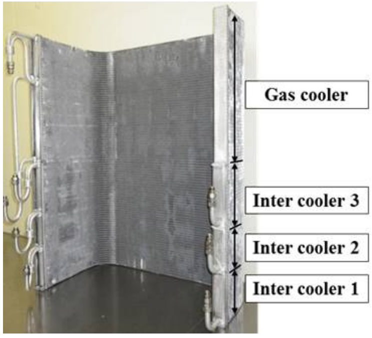


Figure 13: Micro channel HEX

Figure 14: Double tube internal HEX

4. EVALUATION RESULTS

The high efficiency system described above was tested for evaluating its performance. Several sensors, such as 30 thermocouples, 22 pressure gages and 5 mass flowmeters, were attached at the refrigerant circuit shown by Figure 3 for measuring clearly complicated operating conditions of the high efficiency cycle based on four-stage compression and cooling.

During the test, refrigerant enthalpy difference between inlet and outlet of each indoor unit heat exchanger was derived from temperature and pressure, and cooling (heating) capacity was calculated by the enthalpy difference multiplied by mass flow rate. Then, the COP of the outdoor unit was calculated by the cooling (heating) capacity divided by total input power of the outdoor unit without including that of the indoor units.

4.1 Optimum pressure on the high pressure side

Since high pressure side is beyond critical pressure in the CO_2 refrigerant cycle, two-phase condensation process in the HFC refrigerant cycle does not exist. The optimum high pressure exists in cooling operation. Figure 15 shows the experimental result of cooling COP ratio with varying high pressure in each gas cooler outlet temperature. In this figure, COP ratio indicates the ratio to the maximum COP at 32 °C and the pressure ratio indicates the ratio to the pressure in which the maximum COP at 32 °C is obtained.

The maxima are taken in each gas cooler outlet temperature and high pressure affects COP ratio badly at high pressure area especially when gas cooler outlet temperature is low. Therefore, it is significantly important to control high pressure as well as the equipment design and cycle configuration to obtain high efficiency. During the system evaluation test, the high pressure varies thoroughly in order to obtain maximum COP in each condition.

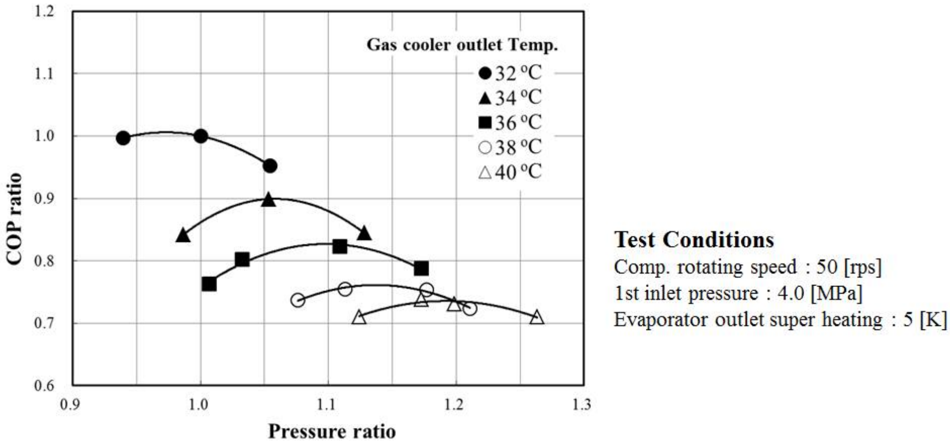


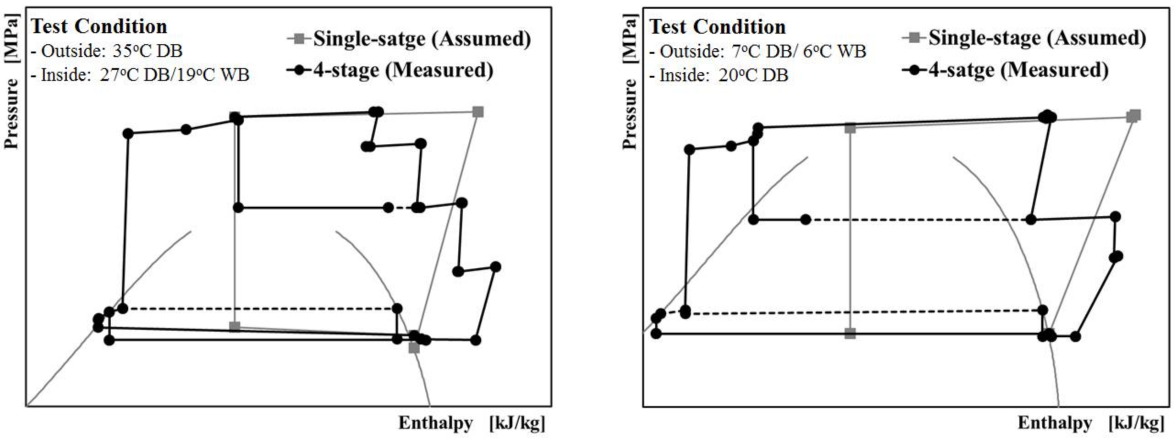
Figure 15: Cooling COP ratio in each outside temperature

4.2 Effect of the input reduction and outdoor unit COP

Figure 16 shows pressure-enthalpy diagram of cooling operation. From the figure, in cooling mode, it is seen that four-stage compression and cooling works effectively since balanced compression process distributed into four stages properly. In cooling of second-stage discharged refrigerant, both intercooler and refrigerant injection by economizer are contributed. The former is much dominant than the latter since the heat of refrigerant from economizer exchange sufficiently with the refrigerant after gas cooler pass through the other pass of economizer. The capacity of evaporator increases due to low inlet enthalpy of evaporator, since pre-expansion temperature is significantly lower than the outside temperature by using economizer and liquid-gas heat exchangers.

In heating mode, apparent two-stage cooling cycle is composed by injecting the two-phase refrigerant into suction line before third-stage compression without intercooler. As the specification of compressor such as the cylinder volume and its ratio etc. are designed for mainly cooling mode, cylinder volume of fourth-stage cylinder is half size of first-stage cylinder. Therefor it is impossible without proper counter measure to intake all discharge refrigerant gas from third-stage cylinder and to control excessive pressure increase.

The new-type compressor equipped volume control mechanism by set the relief valve into the discharge section of third compression. Thereby this system allows controlling high pressure in heating mode properly.



(a): Cooling operation **(b):** Heating operation

Figure 16: P-h diagram in each operating mode

The effect of compressor input power reduction in each operating mode is tabulated in Table 2. Each value indicates the ratio when the compressor input for single-stage compression cycle is 100%. The COP is estimated by assuming the same compressor efficiency and pressures of gas cooler outlet and evaporator inlet with the four-stage compression cycle. It is assumed also that the enthalpy difference of single-stage compression of cooling side is less than that for four-stage compression. Therefore outlet enthalpy of gas cooler of single-stage compression becomes same as that in cooling mode, and larger than that in heating mode for four-stage compression.

As for cooling mode, compression power can reduce to 20.1% by intermediate cooling effect by intercooler which significantly decreases the suction temperature of each stage. In addition, using recovery power generated from expander, 5.3% to compressor input is supplied from expander. 25.4% reduction of input power can be obtained in total within outdoor unit comparing to the single-stage compression cycle.

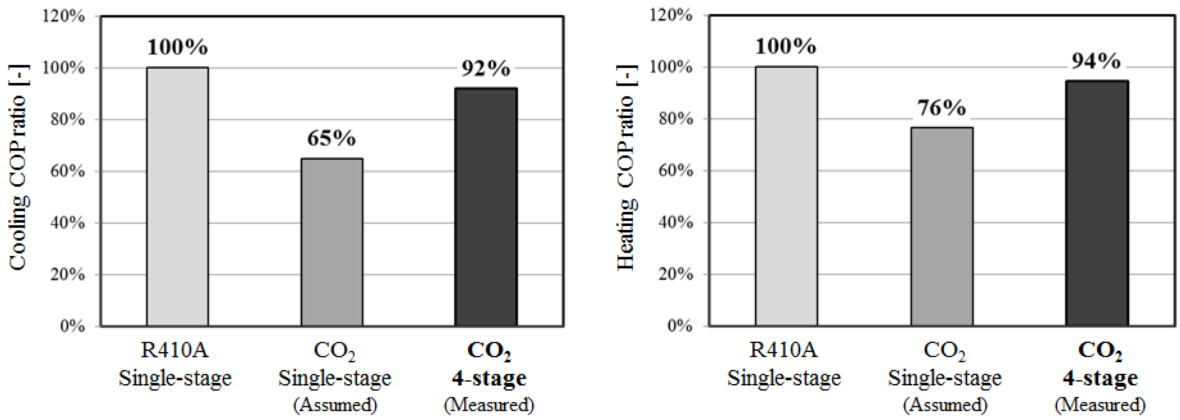
As for heating mode, since theoretical power recovery decreases in expansion process, power recovery becomes 3.5%. In addition, owing to the effect of intermediate cooling by injection, 19.4% reduction of the total power in outdoor unit can be obtained comparing to the single-stage compression cycle.

Table 2: Input reduction in each operating mode

Operating mode	Cooling		Heating	
Cycle type	Single-stage (Assumed)	**4-stage (Measured)**	Single-stage (Assumed)	**4-stage (Measured)**
Compressor input ratio	100	**79.9**	100	**88.3**
Expander output ratio	-	**5.3**	-	**3.5**
Total ratio	100	**74.6**	100	**80.6**

From the effect of the input reduction above mentioned, COP of outdoor unit is shown in Figure 17. Comparisons are made assuming the COP of R410A is 100%. From the figure, it is found that high efficiency cycle gives 92% improvement comparing to 65% for single-stage compression cycle in cooling mode. Similarly, 94% improvement is obtained in heating mode.

Above all, it is found that CO_2 cycle performance can be improved in similar extent to HFC refrigerant cycle due to reducing compression power with divided into four-stage compression process and recovering expansion energy with expander.



(a): Cooling operation **(b):** Heating operation

Figure 17: COP of outdoor unit

4.3 Prospect of outdoor unit size

However the prototype cannot obtain the same or higher performance comparing to that of R410A. To obtain the same or higher performance with R410A, further improvement of cycle efficiency is possible by increasing heat exchanger of outdoor unit.

Figure 18 shows the result of outdoor unit weight at 10HP, based on the 5HP results obtained in this study. Not only adopting the technical improvement such as four-stage compressor and expander but increasing size of heat exchanger up to about 150% of R410A is necessary to accomplish the same performance with R410A. Moreover, it is estimated that the weight of outdoor unit becomes about 250% comparing to R410A by increasing components such as expander and internal heat exchangers, the amount of pipes and casing size of outdoor unit.

Becoming heavy product weight causes a big burden in transportation and installation as well as increasing manufacturing cost. Therefor even more breakthrough technical seeds are necessary to enable to further reduce product weight and size for CO_2 VRF system to spread widely into commercial.

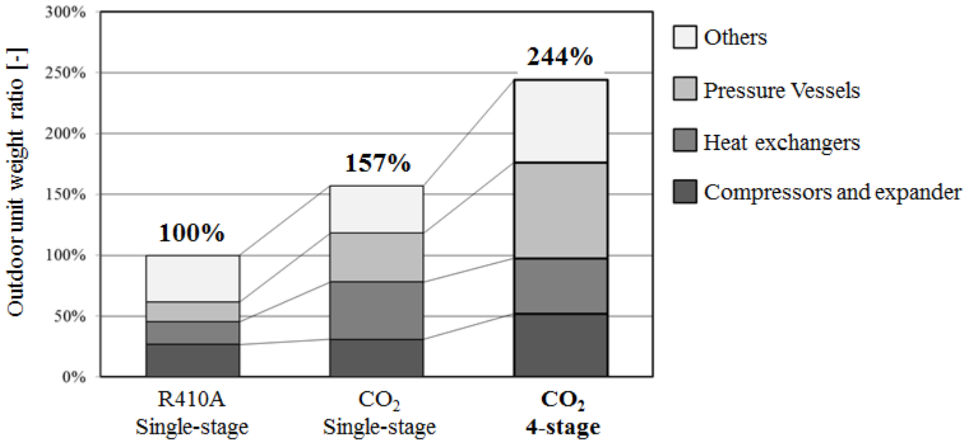


Figure 18: Outdoor unit weight at 10HP

5. CONCLUSIONS

The following results were revealed during our examination:

- To realize the high efficiency CO_2 refrigerant system with multi-stage compression and cooling, we introduced new type compressor which had new mechanism for four-stage compression housing in one casing. Test system was built and new mechanisms are experimentally validated.
- The test system based on high efficiency four-stage compression and cooling cycle leads to reduce the compression work and with recovering the expansion energy during the expansion process can improve the performance and reached the similar level of COP of R410A.
- Not only adopting the technical improvement such as four-stage compressor and expander but also increasing size of heat exchanger up to about 150% of R410A is necessary to accomplish the same performance with R410A.
- It is estimated that the weight of outdoor unit becomes about 250% compare to that of R410A by additional components such as an expander, internal heat exchangers, and the amount of pipes despite weight saving effect in heat exchanger.
- Breakthrough technical seeds are necessary to enable to further reduce product weight and size for CO_2 VRF system to spread widely into commercial.

REFERENCES

[1] C. Toublanc, M. Clausse, M.F. TERRIER, F. MEUNIER. CO_2 CYCLE BASED ON COMPRESSION COOLING. *International Congress of Refrigeration 2007*.

[2] Stefan ELBEL, Predrag HRNJAK. Ejector Refrigeration: An Overview of Historical and Present Developments with an Emphasis on Air-Conditioning Applications. *International Refrigeration and Air Conditioning Conference at Purdue 2008.*

[3] Masakazu OKAMOTO, Eiji KUMAKURA, Tetsuya OKAMOTO, Michio MORIWAKI, Katsumi SAKITANI. Development of Two-Phase Flow Expander for CO_2 Heat Pumps. *International Refrigeration and Air Conditioning Conference at Purdue 2008, Short Course*.

[4] Takashi INOUE, Hiroshi NISHIKAWA, Hideaki SASAKI, Ken KAWAKUBO, Shinichi NAKASHIMA. Development of non-flon refrigeration system using Carbon Dioxide Refrigerant. *JRAIA International symposium 2014*.

ACKNOWLEDGEMENT

This study was funded by New Energy and Industrial Technology Development Organization in Japan (NEDO) from 2011 to 2016.