Development of CO_2 Compressor and Its Application Systems

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Abstract

Amidst concerns over global environmental issues, social controls and restrictions on the use of gases that could not only deplete the ozone layer but also possess a greenhouse effect are accelerating. As a result, attention is focusing on natural refrigerants. Among them, CO2 is a non-inflammable, non-toxic, safe refrigerant. Applications that could make the best use of its properties in favor of heat pump heating are under intensive development.

CO2-based refrigeration cycles would involve three to five times higher pressures than do any existing HFC (hydrofluorocarbon)-based cycles, urging the development of a compressor customized for CO2. The CO2 compressor developed by the authors uses two-rolling piston two-stage compression to provide a high discharge pressure while suppressing both vibrations and noises. Moreover, an internal intermediate pressure scheme was employed under which the gas discharged from the first-stage compression unit is introduced into the compressor chamber in order to help reduce the design pressure and reduce compressor dimensions.

As an application system incorporating the CO2 compressor system, a heat pump water heater has been developed. Other developments are drink vending machines and a secondary refrigerant heat transfer systems designed for refrigerated supermarket showcases.

The heat pump water heater can be used in a broad range of geographic regions, including cold districts, because it not only efficiently produces hot water at 90℃ or above, but involves little capacity loss in a low-temperature environment.

Regarding the drink vending machines, as they use hot water, chilled water, and ice, the simultaneous processes of hot water supply and ice making could make significant savings in energy requirements.

The heat transfer systems that use liquefied CO2 as a secondary refrigerant could not only cut their HFC-based primary refrigerant requirements drastically but would allow for the use of copper piping with a smaller outside diameter secondary piping, contributing to a cost reduction on the whole.

This paper describes a CO2 compressor and its system applications with respect to their characteristics and status of development.

1. CO2 Compressors

1.1 Design Considerations

If CO2, with its critical temperature being as low as 31.1℃, is used in a vapor compression refrigeration cycle, it would trigger a supercritical cycle. Moreover, its high working pressure of 12 MPa and low working pressure of 3 MPa make it mandatory to develop a high-pressure-resistant compressor capable of handling up to three to five times higher working pressures than are imposed by any traditional refrigerant.

Specifically, the following solutions need to be found:

1) Solutions to address the high working pressure (12MPa)
2) Solutions to prevent leakage resulting from large pressure differences

A discussion of these solutions follows.

1.1.1 Solutions to address the high working pressure

1) Two-stage compression

The CO2 compressor we developed this time, as a solution to address the high working pressure, employs two-rolling piston to implement two-stage compression. Dividing the compression process into two stages not only narrows the pressure difference between the first- and second-stage compression units but also reduces the pressure ratio, offering possible improvements in both the volumetric and indicated efficiencies. The two-rolling piston two-stage compressor is constructed with the first and the second compression units, each with a different displacement volume. The refrigerant is progressively compressed from the compression unit with the larger displacement volume to the one with the smaller displacement volume. As shown in Figure 1-1, the two-stage compression system smoothens torque variations when compared with single-stage compression, making for low-vibration, low-noise characteristics.

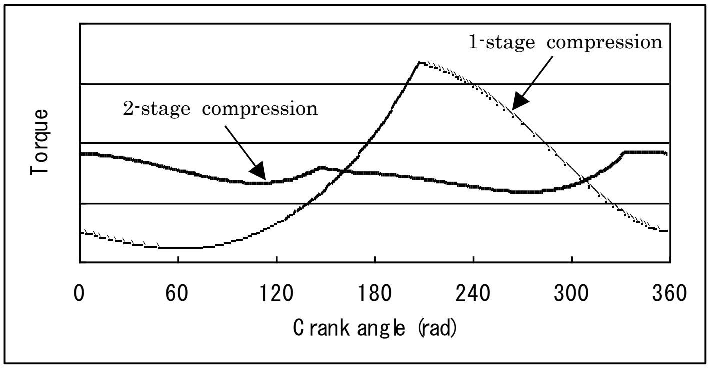

Fig.1-1 Relationship between torque and crank angle

2) Internal intermediate pressure scheme

To reduce case design pressure, an internal intermediate pressure scheme was employed under which the compressed gas in the first-stage compression unit is discharged into the shell case in order to adjust the internal pressure to an intermediate level.

Because the discharge gas compressed in the first-stage compression unit and the intake gas in the second-stage compression unit equal each other in volume, the intermediate pressure is determined by the ratio of the displacement volume in the first-stage compression unit to that in the second-stage compression unit.

Setting the intermediate pressure below the shutdown equilibrium pressure reduces the design pressure of the shell case, thereby easing the tasks of designing the pressure-resistant shell case and downsizing compressor dimensions.

1.1.2 Solutions to prevent leakage resulting from large pressure differences

1) Optimizing the clearance

A large differential pressure of the CO2 refrigerant could accelerate leakage, which will result in degraded performance. A smaller clearance than that required for the conventional refrigerant compression system is needed to sustain the performance of CO2 compressor. To minimize the effect of deformation strains caused by pressure differences, linear analyses using tetrahedral secondary elements were performed on the shaft and vane on which pressure differences were thought to have more effect than anywhere else.

Since deformation in the AA' cross section in Figure 1-2 was found most pronounced in the shaft as a result of these analyses, the shape of the upper and lower eccentric parts was modified to make the cross-sectional secondary moment larger. Also, it was confirmed that the sheet thickness of the vane has minimum deformation when used in high-pressure conditions.

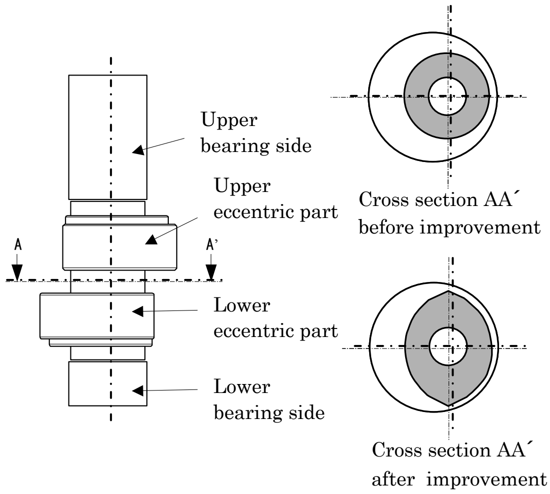

Fig.1-2 Cross section view of shaft

2) Dividing the gas discharged from the first-stage compression unit

As the shell case internal pressure was set to an intermediate level, degradation of lubrication and insufficient sealing for the mechanical components in the second-stage compression unit was anticipated due to shortage of oil circulation when the pressure becomes high. Furthermore, lowering in the oil level in the shell case caused by discharging the high-pressure gas from the second-stage compression unit directly outside the compressor, was concerned.

To solve these problems, this compressor divides the compressed discharged gas from the first-stage compression unit into the two routes (one is discharged into the external piping on the compressor and the other is led into the case chamber) to ensure a supply of oil to the second-stage compression unit and availability of oil in the shell case.

1.2 Basic Specifications

Table 1-1 summarizes the specifications of the CO2 compressor developed to accommodate the design considerations described above. It is essentially equivalent in its physical dimensions to the compressor installed in a home air conditioner rated at a cooling capacity of 1.8 to 5 kW.

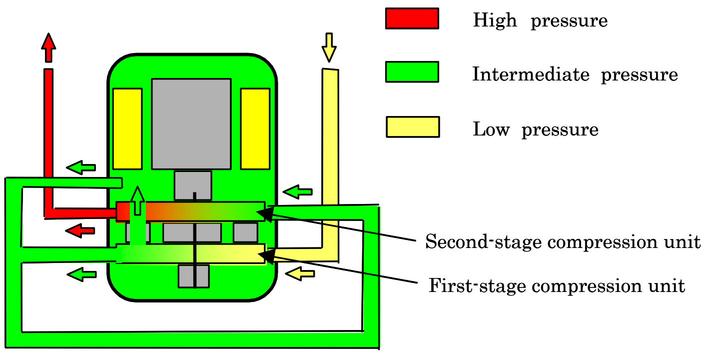

Fig.1-3 Schematic of two-stage compression mechanism

Figure 1-3 shows the arrangement of the compression mechanism superimposed with gas flow. When a low-pressure gas is taken into the first-stage compression unit in the lower part of the compression mechanism, it is compressed to an intermediate pressure and then discharged into the two routes: one is to the external piping on the case and the other is to the case chamber. The two routes of the refrigerant flow merge together again outside the case before the refrigerant is taken into the second-stage compression unit in the upper part of the compression mechanism. After the gas is compressed to a high pressure in the second-stage compression unit, it is discharged to the refrigeration cycle.

Table 1-1 Specifications of CO2 compressor

Compressor type	Hermetic-type compressor
Compressing system	two-rolling piston two-stage compression
Motor type	DC inverter motor(20～120Hz)
Rated power	900W
Refrigerant	Carbon dioxide
Size	OD117.2×H244.3mm

2. CO2 Heat Pump Water Heaters

CO2 compressor-based CO2 heat pump water heaters have been commercialized and introduced in Japan. The CO2 heat pump water heater, if used in a warm district like Tokyo, would yield higher energy conservation efficiency, than do combustion water heaters, electric water heaters, and HFC-based heat pump water heaters, and because it has a lower refrigerant GWP, we can say that it is a low-TEWI, environmentally friendly water heater. Moreover, the properties of the CO2 refrigerant are considered to allow little fall in the heating capacity and energy conservation efficiency when the outside air temperature drops. Hence, a CO2 heat pump water heater capable of efficiently serving hot water in cold districts as well has been under development. This chapter reports the results of testing of a CO2 heat pump water heater installed in a cold district in Japan (Hokkaido).

2.1 Testing of a Prototype Model Built to Cold District Specifications

2.1.1 Prototype model specifications

Figure 2-1 shows what the prototype model looks like, with its block diagram appearing in Figure 2-2. Comprising of a CO2 compressor, a gas cooler, an expansion valve, an evaporator, and an accumulator, the refrigerant circuitry is assembled and accommodated in the same casing as the outdoor unit of a room air conditioner of the 4.0 kW class. The CO2 compressor mounted is developed to meet the cold district specifications. The compressor has a larger displacement volume than a CO2 compressor built for use in moderate-temperature districts and has extra motor output to maintain its heating capacity (4.5 kW) at an outside air temperature of -20℃. To prevent the icing of defrosting drain water, the discharged high-pressure refrigerant flow from the gas cooler exit is rerouted to a portion of the piping on the lower part of the evaporator, so that the lower part of the evaporator will always be maintained at around the supply water temperature, thereby preventing icing due to wind and snowfalls and keeping defrosting drainage water from re-icing.

Fig.2-1 Prototype model built to Cold District

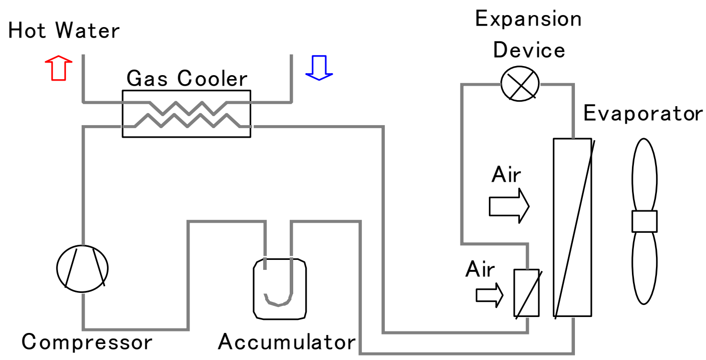

Fig.2-2 Diagram of Prototype Unit

2.1.2 Field installation

The CO2 heat pump outdoor unit installed outdoors is timed to run 16 hours a day (11:00 - 17:00, 19:00 - 21:00, 23:00 - 07:00) to measure its heating capacity and COP (Coefficient Of Performance). The compressor revolutions and the expansion valve opening are automatically set for an instantaneous heating capacity of 4.5 kW and for the maximum COP according to the outside air temperature and the supply water temperature. As indicated in Table 2-4, the supply water is regulated at the average supply water temperature in the Sapporo District. Hot water flow rate was regulated to maintain a hot water temperature at 90 ℃. In addition, defrosting was carried out from time to time as frosting was detected from the difference between the evaporator intake air temperature and the evaporation temperature.

Table 2-4 Inlet / Outlet Water Temperature[℃]

	Target	Measured
Inlet(Jan.)	3.1	About 4
Inlet(Feb.)	2.4	About 3
Outlet	90.0	–

2.1.3 Results of field-testing

Figure 2-3 plots the relationships between the outside air temperature and the heating capacity, indicating the average of the heating capacities for a period of 150 minutes when defrosting did not occur for a period of 150 minutes after the start of operation or end of defrosting cycle and the average of the heating capacities several cycles after the start of operation or end of defrosting when defrosting occurred during a period of 120 minutes after the start of operation or end of defrosting. Where defrosting did not occur, heating capacities stood between from 4.0 and 4.5 kW, virtually achieving the design value. However, a fall in the average heating capacity to as low as 3.0 kW or so was observed where defrosting took place. Figure 2-4 summarizes the COPs in a like manner. The COP showed a linear change in relation to the outside air dry-bulb temperature where defrosting did not take place, demonstrating basic agreement with the results of environment lab testing. When defrosting was carried out frequently, the COP fell by as much as about 20%. Averaging by time zone (11 to 17, 19 to 21, 23 to 7 hours), the COP recorded for the duration of outdoor testing was found about 5% down from the results of environment lab testing.

No icing of the evaporator drainage water was observed for the duration of outdoor testing.

The effect of the rerouted piping was proved.

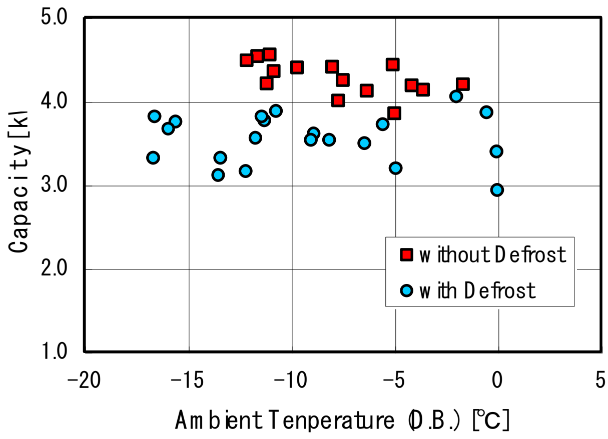

Fig.2-3 Capacity of Unit in the Open Air

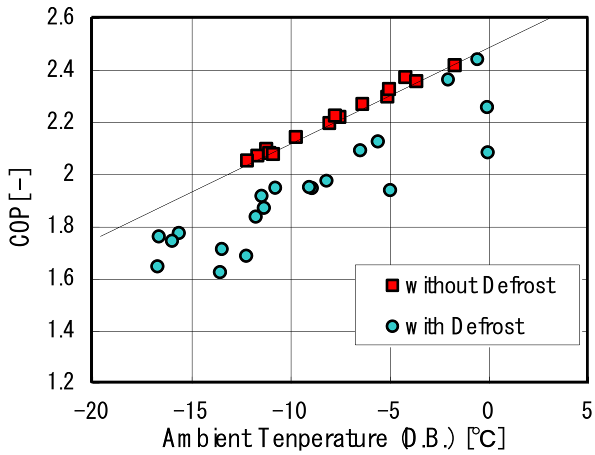

Fig.2-4 COP of Unit in the Open Air

2.2 Evaluation by Way of TEWI

2.2.1 Yearly hot water supply load

Assuming that a water heater has an IBEC-L mode equivalent of hot water supply load throughout the year as it is used by an average Japanese household (four member-family), its hot water supply load was calculated by allowing for the temperature of the city water available in the Sapporo District. Further, the heating load of the water heater was computed by assuming that the rate of discharge from the hot water storage tank remains constant at 10% of the hot water supply load. Figure 2-5 plots the daily heating loads by month. The total heating load a year (365 days) amounts to 6978 kWh.

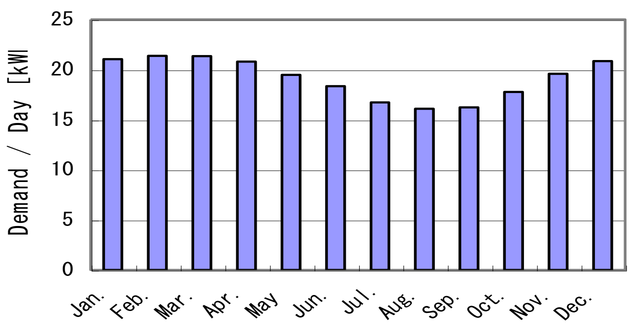

Fig.2-5 Heat Demand for Hot Water Supply
(IBEC L-mode Model Base)

2.2.2 Yearly COP calculations

The yearly average COP of the heat pump water heater was calculated by allowing for changes in the environmental conditions in effect during hot water storage operation, such as outside air temperature and humidity, supply water temperature, and hot water temperature, under the following assumptions:

<1>The hot water temperature remains constant at 90℃ throughout the year.
<2>Assuming that the COP is determined by the outside air dry-bulb temperature and the supply water temperature, it is approximated by solving a linear expression on the basis of the results of testing to be separately carried out in the environment lab.
<3>The heating capacity of the water heater is maintained at 4.5 kW, regardless of operating conditions.
<4>When the outside air dry-bulb temperature is 5.5℃ or below, the COP will drop by a flat rate of 6% due to a need for defrosting.
<5>The inlet water temperature of the water heater is equal to the supply water temperature, and no mixing in the hot water tank is assumed.
<6>The wattage requirements for components other than the outdoor unit, such as the water circulation pump, are assumed constant at 90W and added to the compressor input to arrive at the total power consumption.

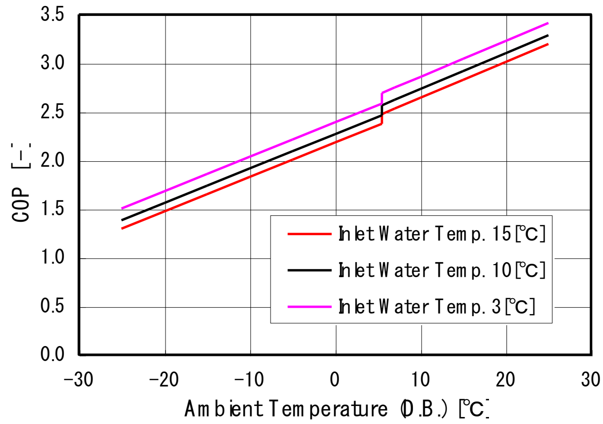

Fig.2-6 Estimated COP of Unit with Defrost

Table 2-5 Annual Average COP [-]

No.	Operation Time	COP
1	[11p.m.–7a.m.]	2.53
2	[3a.m.–7 a.m.] and [0p.m.–4p.m.]	2.60

Figure 2-6 shows how the COP of the prototype model can be calculated on the basis of the results of measurement in the environment lab, by approximating it with a linear expression with the outside air dry-bulb temperature and the supply water temperature as parameters and allowing for extra input required for defrosting. The yearly average COP was calculated with regard to two cases. Table 2-5 summarizes the operating time zones and calculation results.

2.2.3 TEWI calculations

Table 2-6 summarizes the CO2 emission coefficient of electric power in the Hokkaido District and other fuels. The unit requirements of electric power are the all-day averages of CO2 discharges per at-market electric energy.

TEWI is expressed as a sum of the direct warming effect of the refrigerant used in the equipment and the warming effect of CO2 that is discharged as a result of the production of energies that are indirectly consumed by the equipment.

Table 2-6 CO_2 Emission Coefficients for Various Fuels [kgCO_2/kWh][6)7)]

Kerosene	0.242
LPG	0.217
City Gas (13A)	0.184
Electricity	0.46

As a direct effect of the CO2 heat pump water heater, it was assumed that the refrigerant would be used up without leakage till expiration of the equipment service life and would be discharged into air in its total quantity when the equipment was scrapped. The equipment service life was assumed 13 years. As an indirect effect of the CO2 heat pump water heater, the rate of discharge was calculated from the yearly power consumption and the unit requirement of CO2 discharge.

For water heaters operating from other combustion principles, the heating load was calculated from the hot water supply load and the equipment efficiency. Further, the rate of discharge was computed from the unit requirement of CO2 discharge. Table 2-7 summarizes the calculation conditions, and Figure 2-7 presents the calculation results.

According to the calculation results, the CO2 heat pump water heater, when used in a cold district, marked a lower TEWI than water heaters operating from other combustion principles using a city gas (13A) and other fuels. When the heating load is low as in a summer season, the hot water temperature may be set low to achieve a further reduction in the TEWI.

Table 2-7 Conditions of TEWI Calculation

	CO_2 Heat Pump	Others
Radiation Loss [%]	10	–
Stack Loss [%]	–	15
Direct Effect [kg_CO_2]	0.82	0

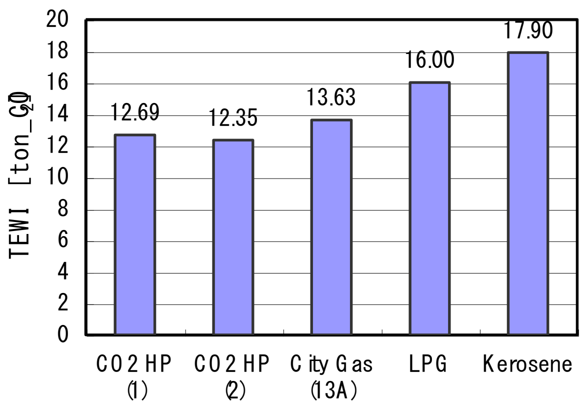

Fig.2-7 TEWIs for Verious Fuels

3. Drink vending machines

The drink vending machines are vending machines that simultaneously serve hot drinks, such as coffee, in cups and cold drinks, such as coke and therefore require both hot water for hot drinks and chilled water and ice for cold drinks. The existing drink vending machines use an electric heater to serve hot water and a refrigeration cycle using an HFC-based refrigerant to serve chilled water and ice. CO2 cycles are ideal for heating water, as for hot water supply, allowing clean, efficient heat pump cycling. An enhanced application of the CO2 heat pump cycle of a water heater can be found in the drink vending machine, in which water is heated and chilled and ice is made all in a single CO2 heat pump cycle; that is, heat radiating from the high-pressure end of the heat pump is used as a heat source for hot water supply, while heat absorbed from the low-pressure end of the heat pump is used as a heat source for chilling water. The authors fabricated a tester for drink vending machines using the CO2 heat pump cycle to probe into its possibilities.

3.1 Experimental Apparatus

Figure 3-1 schematically shows the experimental apparatus. The basic components of the apparatus consist of a CO2 compressor, four heat exchangers for hot water supply (gas cooler), chilled water, ice making, and air, an internal heat exchanger that exchanges heat between the refrigerant at the gas cooler exit and that at the low-pressure heat exchanger exit to augment the

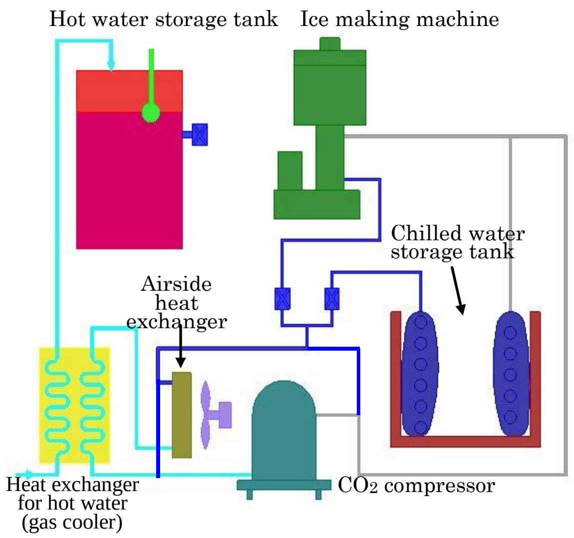

Fig.3-1 experimental apparatus

exchanger capacity, an expansion valve, a hot water tank, and an ice maker. Drink vending machines should be capable of supplying hot water, ice, and chilled water separately because their requirements vary according to the status of drink sales. The air heat exchangers are used as a gas cooler or as an evaporator depending on the pattern of machine operation.

3.2 Test Results

Figure 3-2 shows a typical cycle diagram of the heat pump cycle run at a supply water temperature of 25 ℃ and a hot water temperature of 90 ℃ (temperature-enthalpy chart). Testing was carried out by supplying hot water and ice making simultaneously at an outside air temperature of 27 ℃ and an evaporation temperature of -10℃.

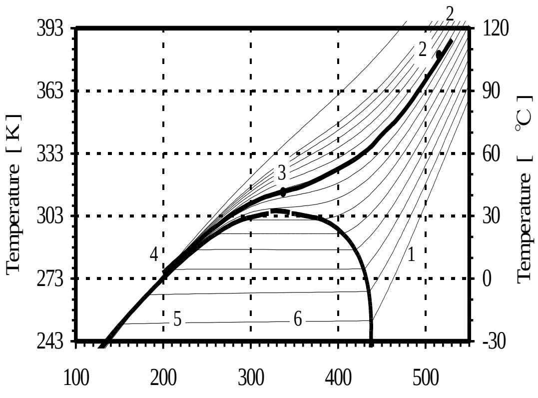

Fig.3-2 *T-h* chart at water heating + ice making mode

Table 3-1 summarizes the results of performance testing carried out in various operation modes at a supply water temperature of 25℃, a hot water temperature of 90℃, a pre-ice making water temperature of 25℃, and an outside air temperature of 27℃. Due to thermodynamic properties of the refrigerant, the CO2 heat pump lost to the existing vending machines using HFC's or HCFC's as a refrigerants in making ice alone, but by far outperformed the electric heater with a COP of 1.0 when it was dedicated to supplying hot water, manifesting the benefit of a CO2 heat pump system. Extra efficiency is expected in operations in which the hot water supply and ice making processes are carried out simultaneously.

Table 3-1 Performance of CO_2 heat pump

Operation mode	Capacity [W]	COP [-]	Evap.temp [℃]
Water heating	1014	2.19	8.7
Ice making	325	1.18	-11.1
Water heating + Ice making	639(heating) 306(Ice making)	2.35	-10.2

3.3 Conclusions

A CO2 heat pump that targets drink vending machines was prototyped for performance evaluation purposes. The prototype model demonstrated a COP of 2.19 in supplying hot water under normal conditions of use for drink vending machines. In an operation in which the hot water supply and ice making processes were carried out simultaneously, the prototype model registered a COP of 2.35, thus promising an energy conservation effect in the heat pump cycle.

4. CO2 Heat Transfer Systems

Large refrigeration systems, such as refrigerated showcases for supermarkets and commercial refrigerated warehouses, have high power and refrigerant requirements, calling for higher efficiencies and the use of natural refrigerants. Among available refrigerants, however, NH3 and HCs are used in these systems in such large quantities that legal restrictions are imposed on their use. Their use is not advisable from a safety viewpoint. Hence, we need to approach the use of a CO2-based natural refrigerant.

Because a CO2-based direct expansion system would be prone to a significant drop in the COP, a two-stage refrigeration expansion system is required in which CO2 is used in the second-stage loop and another refrigerant is used in the first-stage compression unit.

Generally, a two-stage refrigeration system comes less efficient and more costly than a direct expansion system but can rival one if use is made of certain properties of CO2, such as high latent heat and low pressure drop in low temperature regions. Moreover, the use of refrigeration piping with a smaller diameter can easy and economize installation work.

This time, a refrigerated warehouse of the 2.5 kW class was built, and an R22-based direct expansion system and a CO2-based heat transfer system (CO2 pump system) were prototyped to compare them in terms of power consumption.

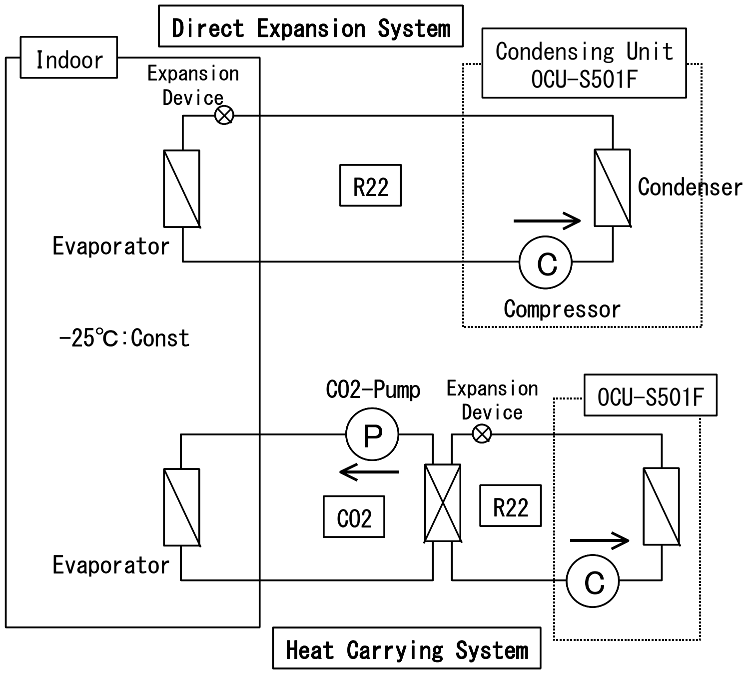

Fig.4-1　Experimental System of CO2 Pump System

Table.4-1　Specification of Cupper Cube at the Experimental System

	Length [m]	Diameter of Liquid Tube [mm]	Diameter of Gas Tube [mm]
R22 Direct Expansion	10	9.53	19.05
CO2 Pump System	10	6.35	9.53

Figure 4-1 shows a block diagram of the test circuit.　Comparative testing with our condensing unit (refrigerant R22) was carried out (refrigeration circuit shown in the upper part of the figure). The testing is made easier because the same condensing unit is used on the primary side of the heat transfer system as well.　The compressor was switched on and off to keep the refrigerator temperature constant at -25℃.　Then, the average power consumption was measured at 1-hour intervals with a sampling cycle of 60 seconds and the ambient temperature as parameters.　Table 4-1 gives the specifications of the piping from the condensing unit to inside the refrigerator.

4.1 Comparative Testing with an Existing Refrigerator

Figure 4-2 plots the average power consumption of the heat transfer system under test.　It was found to maintain an equivalent of the power consumption of the existing refrigerator in an outside temperature range of 21 to 33℃.　When the outside air temperature lowers, the power consumption of the primary refrigerator falls but that of the liquefied CO2 pump does not change. Falls in outside air temperature would be disadvantageous to the heat transfer system, but at an ambient temperature of 32℃, a typical measurement condition common in Japan, the difference is as narrow as 2%.

4.2 Cost Calculations

Two-stage refrigeration cycling would entail an increase of about 10% in the equipment cost.

Using CO2 as a secondary refrigerant, however, would make it possible to use copper piping with a smaller outside diameter, halving the work time, with the benefit of a saving of 50% or so on the installation cost.

The total cost could be cut by 15% or more in Japan where labor accounts for a greater share of

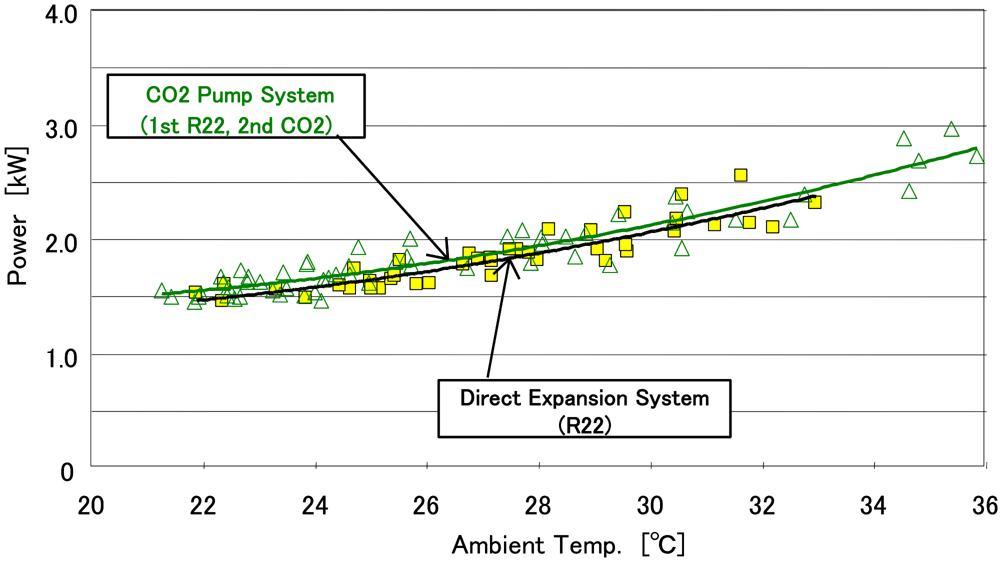

Fig.4-2 Comparison with CO2 Pump System and R22 Direct Expansion System

Table.4-2 Comparison with CO2 Pump System and R22 Direct Expansion System

Ambient Temp.	Indoor Temp.	System	Power	Ratio of CO2/R22
32[℃]	-25[℃]	CO2 Pump	2332[W]	1.02
		R22 D.E	2283[W]	
20[℃]	-25[℃]	CO2 Pump	1476[W]	1.07
		R22 D.E	1383[W]	

the cost than anywhere else.

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