CO_2 HEAT PUMP HEATING AND WATER HEATER SYSTEM FOR COLD AREA

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ABSTRACT: By incorporating a first-stage expansion intermediate-cooling cycle (gas injection) into a CO_2 natural refrigerant heat pump, we have managed to achieve a significant improvement in heating capacity and energy consumption efficiency under low ambient temperature conditions. By combining this technology with previously fostered technologies developed for CO_2 heat pump application including the heater-less ice formation prevention technology, the “direct heating” technology that provides direct heat exchange between the CO_2 refrigerant and heating antifreeze, and the outlet hot water temperature transition control technology, we have developed a prototype of CO_2 natural refrigerant heat pump heating and water heater system, which enables both heating and hot water to be delivered by a single system. This report describes the results obtained from performance evaluation tests conducted in our environmental test laboratory and from monitoring tests in a cold region (Hokkaido, Japan), and addresses the possibilities of this system for use in a heat pump system application.

Key Words: ***CO_2 Heat Pump, Heating, Water Heater, Cold Area***

1 INTRODUCTION

Stemming from the growing awareness of environmental concerns in recent years, the adoption of heat pump water heaters and other high-efficiency appliances has been increasing in Hokkaido, Japan’s largest cold region. The number of CO_2 natural refrigerant heat pump water heater installations have also risen significantly, from 85 units in 2005 to 237 units in 2006. In addition, the number of new single-family house construction starts in Hokkaido stood at 17,800 units (in 2006), and factors such as the fact that some 50% (8,700 units) of these households have only electrically powered appliances in the kitchen and throughout the home appear more to be an indication of a tailwind that is helping to boost the introduction of high-efficiency heat pump equipment. Nevertheless, when these figures are compared to the scale of the several hundred thousand units installed in regular-climate areas, the cold region market still remains a small one, and thus market penetration cannot be said to be moving very rapidly.
The consumption of energy (used for heating and cooling, hot water supply, lighting, and so on) per household in Hokkaido is approximately 1.4 times greater than that of the nationwide average. This consumption breaks down to about 50% for heating and 20% for hot water supply. In order to achieve energy savings, a two-pronged simultaneous approach to heating and hot water supply with their high energy demands is thought to be necessary, though unfortunately, almost all of the electrically powered high-efficiency appliances that have reached practical application to date are single-function types (HP water heaters and HP heaters), and no systems with integrated high-efficiency heating and water heaters, desired by consumers, have been supplied. This is believed to be the principal reason for the lack of growth in the number of units installed.

Acknowledging this situation, several years ago we began our research and development into systems that can supply hot water and provide heating at the same time, and to date have repeatedly assessed the performance of these systems and conducted validation tests. (Kobayashi et al.2002, 2004) We started with the development of a water heater and moved on to the development of technologies for maintaining compressor reliability and preventing ice formation among others, but assuring heating performance at low ambient temperatures has remained an issue to the very last. (Kuwabara et al.2006) For some time, the injection system has been known as a technology that improves performance at low ambient temperatures. (Goosmann and Zumbro 1928) Recent reports on related research have been increasing. (Lambers et al.2006) (Zha et al.2006) (Cejka 2007) At this time, we have managed to obtain some specific results for the CO_2 natural refrigerant heat pump technology for cold regions in which the split cycle (gas injection) has been incorporated. (Kobayashi et al.2006) (Kuwabara et al.2007) (Itou 2007) As a result, it is now possible to significantly improve the heating capacity and energy consumption efficiency at low ambient temperatures, and the possibility of practically applying this technology in heating and water heater systems has become more viable.

By combining these basic technologies with proven technologies developed for CO_2 heat pump application including the heater-less ice formation prevention technology, the “direct heating” technology that provides direct heat exchange between the CO_2 refrigerant and heating antifreeze, and the outlet hot water temperature transition control technology, we developed a prototype CO_2 heating and water heater system for cold regions and have been carrying out performance evaluation tests in our environmental test laboratory and monitoring tests in ordinary households. Below, using the data obtained so far, we will discuss the current status of CO_2 heat pump heating and water heater systems for cold regions, and examine their future potential.

2 HEATING AND WATER HEATER SYSTEMS

2.1 Overall configuration

The entire system (refer to **Figure 1**) consists of two 6 kW CO_2 heat pump units (unit A, used exclusively for heating, and unit B for both heating and the water heater), a hot water tank unit (370 liter capacity) and a terminal box. The target specifications call for a 6.0 kW hot water supply capacity and 8.0 kW heating capacity, as well as the ability to maintain these capacities, even at an ambient temperature of –20˚C, and to guarantee operation, even at an ambient temperature of –25˚C.

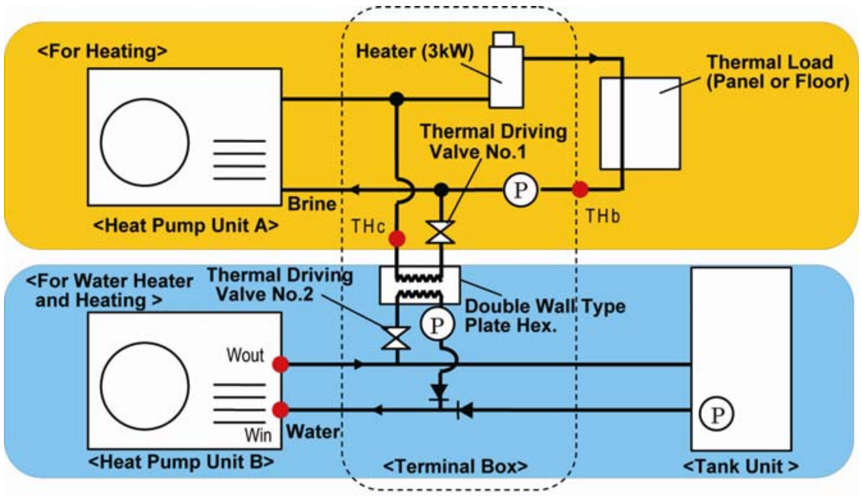

Figure 1: Heating and water heater system

2.2 Direct heating system

A system for ensuring the direct heat exchange between the CO_2 refrigerant and heating antifreeze is used in heat pump unit A, which is used exclusively for heating. Unlike the system used for heating which circulates the hot water in the tank throughout the home through pipes, this system has the advantage that it does not affect the amount of the hot water supplied (there are no concerns that the hot water will run out), and continuous heating operation is possible. For this reason, it can meet the demand of full-fledged heating loads, and enables floor heating and panel radiators to be operated at the same time.

2.3 Terminal box

A double wall type plate heat exchanger is contained in the terminal box, allowing the heat from the water circuit on the hot water supply side to be transferred to the antifreeze circuit on the heating side. When hot water storage (hot water supply) operations are not being undertaken, auxiliary heating operations from heat pump unit B, used for heating and the water heater, are possible. We devised a system that, in the event the heating load is high and heat pump unit A (used exclusively for heating) cannot meet this load demand alone while hot water storage (supply) operations are being undertaken, provides support by way of a 3 kW supplement heater.

2.4 Safety considerations

Taking into account the fact that the system would be installed in households in the Hokkaido area, we considered characteristics that would make it easy to install the system. For instance, we made the tank unit smaller than door dimensions (through which the tank unit can be carried indoors) and designed the terminal box with dimensions (enabling the box to be installed under the floor) to allow the terminal box to be lowered into place from the under floor storage cover. In addition, the aspect of safety was taken into consideration by, for instance, incorporating a water leak detection device. (**Table 1**)

Heat pump unit	Dimensions (H×W×D)	690×840×290 [mm]
	Weight	66 [kg]
Terminal Box	Dimensions (H×W×D)	417×436×438 [mm]
	Weight	28 [kg]
	Heater	3 [kW]
Tank unit	Dimensions (H×W×D)	1800×650×740 [mm]
	Capacity	370 [L]
	Control	Full auto

Table1; Components of Heating and water heater System

3 COLD REGION SUPPORT FEATURES

3.1 Refrigeration circuit

One issue associated with CO_2 natural refrigerant heat pumps has been that when the return load temperature (cold water temperature) is high (40˚C or more), the gas cooler outlet temperature (point e) also rises, and energy consumption efficiency (COP) and capacity decline. A high return temperature is a roadblock to applying these heat pumps to heating. For this reason, we implemented a countermeasure to reduce the gas cooler outlet temperature by adding an intermediate heat exchanger and adopting a single-stage expansion intermediate cooling cycle (**Figure 2**).Some of the gas cooler outlet refrigerant is shunted to the bypass circuit, passes through the second expansion device, and is set to the intermediate pressure, after which it is heat-exchanged with the refrigerant flowing through

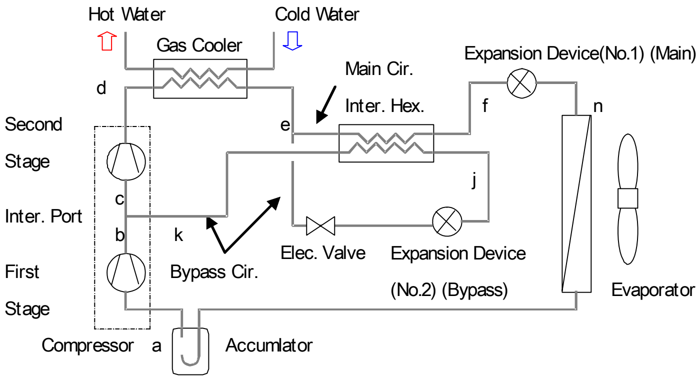

Figure 2: Diagram of heat pump unit

the main circuit, and returned to the intermediate port. The temperature of the refrigerant flowing through the main circuit drops, and since the shunted refrigerant is gasified and returned, a gas injection circuit is formed. Since this circuit splits off at the gas cooler outlet, it is also called a “split cycle.” Hereafter, it will be referred to as the split cycle. Unlike a 2-stage expansion cycle, a separating device, such as a gas-liquid separator, is not used. A number of features are achieved by the design of this circuit: it is possible to reduce the evaporator inlet quality and enhance the refrigeration effect by cooling the gas cooler outlet high-pressure refrigerant in the intermediate heat exchanger; it is also possible to minimize the temperature rise of the second stage discharge gas by mixing low-temperature gas with the first stage discharge gas.

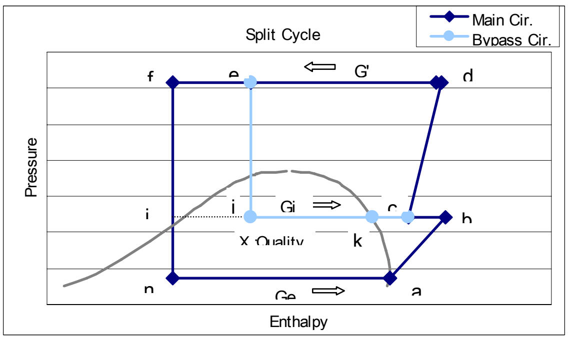

Figure 3: P-h diagram of split cycle

Figure 3 is a pressure vs. enthalpy diagram of the split cycle: here, P is the pressure, h the enthalpy, G’ is the gas cooler refrigerant flow, Ge is the evaporator refrigerant flow, and Gi is the bypass circuit refrigerant flow. The super heat at intermediate heat exchanger outlet point k and evaporator outlet point a for the gas refrigerant was assumed to be zero. In addition, X is the quality at intermediate heat exchanger inlet point j, and point i was made the

point at which middle pressure line j-c is extended and intersects with line f-n. Symbol a, b, c, d, e, f, n indicate the main circuit, and symbol e, j, k indicate the bypass circuit. The gas cooler outlet refrigerant is cooled, and moves from point e to f. The enthalpy at evaporator inlet point n is reduced, and Δh (the enthalpy differential between point n and point a) increases. At the same capacity, therefore, flow Ge of the refrigerant flowing through the evaporator can be lower than G' and the input can be reduced. Conversely, if flow Ge of the refrigerant flowing through the evaporator is fixed, the capacity increases. At the both cases COP increases.
The effect of the split cycle is dependent upon quality X: the higher quality X is, the greater the effect of the split cycle. Also, the lower the low pressure or the higher the gas cooler outlet temperature, the higher quality X will be, so in terms of the operating conditions, the lower the ambient temperature or the higher the return temperature from the heating load, the greater the effect will be.

3.2 Element technology

We developed an intermediate heat exchanger and a 2-stage compression rotary compressor that supports gas injection. (Refer to **Table 2**.) We estimated the capacity of the intermediate heat exchanger while anticipating the highest bypass volume conditions, pursuing a compact construction, and determined its specifications. The compressor is a hermetic internal intermediate-pressure 2-stage compression type rotary compressor, the same system as before, but since the gas refrigerant is injected into the intermediate pressure port via the intermediate heat exchanger, the intermediate pressure rises, and the balance between the high pressure and low pressure changes. As a result of simulations, in a form in line with the new refrigeration circuit, we reset the displacement volume ratio to a point where it is very efficient (around 70%). Some of the motor winding and valve specifications have been changed to allow the rotational speed to be increased for use in cold regions. Furthermore, in cold regions, there is always the concern of ice forming in or on the evaporator. We therefore incorporated an ice-forming prevention coil at the very bottom on the outside of the evaporator, and adopted a construction to run the high-pressure refrigerant in order to prevent ice from forming. (Heater-less ice formation prevention technology)

3.3 Heating outlet water temperature control that supports CO_2 heat pumps

In exercising remote control over the heating, the heating water supply temperature can be set manually to 50˚, 60˚ or 70˚C, after which the control transfers to automatic temperature control. When the heating return temperature rises to a certain extent during a heating operation, this is interpreted to be a reduction in the heating load, and control is exercised to reduce the supply temperature setting in steps. Conversely, when a difference between the heating supply and return temperatures becomes greater, this is interpreted to be an increase in the heating load, and control is exercised to increase the temperature setting in steps. In this way, by incorporating a function that ascertains the increase or reduction in the heating load and adjusts the heating water supply temperature, system efficiency is improved by the automatic transition to a state in which the efficiency is always high.

Table 2; Specifications of intermediate heat exchanger and CO_2 compressor

4 SYSTEM EVALUATION

4.1 Independent heat pump performance tests

After connecting the heat pump units to the water supply system, we set the expansion devices to manual and took measurements in a calorie test chamber. **Figures 4 and 5** show the results of the maximum heating capacity tests in the split cycle and conventional cycle. The test conditions were a heating outlet water temperature of 55˚C and heating return temperature of 40˚C.

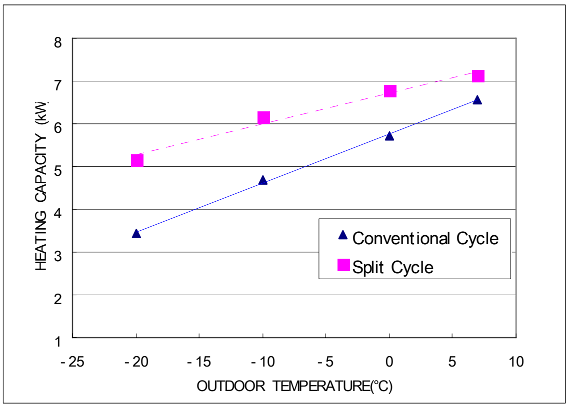

Figure 4: Heating Capacity of split and conventional cycle

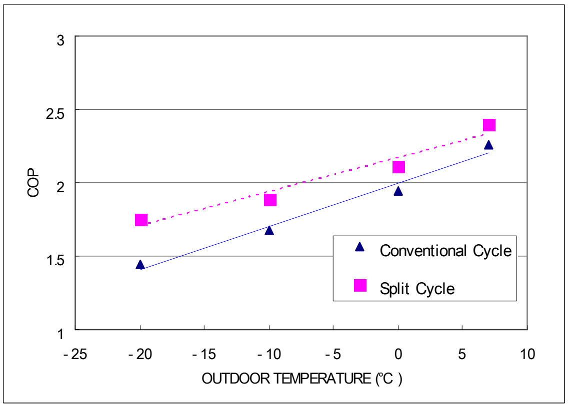

Figure 5: COP of split and conventional cycle

Intermediate Hex.	Construction	Direct Contact Type (Welded)
	CO_2 middle Pressure side	OD=6.35 [mm] t=1.2 [mm] (Bare tube)
	CO_2 high Pressure side	OD=6.35 [mm] t=1.2 [mm] (Bare tube)
	Length	2.8[m]
Compressor	Mechanism	Rolling piston type two-stage Compression
	Motor Type	DC Inverter driven
	Nominal	1300 [W]
	Displacement Volume	3.90 / 2.80 [cm^3] (Ratio: 70%)
	Dimensions	Outside Diameter:118 [mm]、Height:217[mm]
	Weight	9.3 [kg]

Compared with the conventional cycle, it is clear that the heating capacity in the split cycle increases significantly and that the COP also increases at the same time. Furthermore, the lower the ambient temperature, the higher the rate of increase in the heating capacity and COP. The tests we performed proved the effect to be great at a low ambient temperature.

4.2 Auxiliary heating capacity tests using heat pump unit B

It was possible to output a heating capacity of 5 to 7 kW using one heat pump unit A, but the use of one heat pump unit A is not enough to meet the demands of the highest heating loads. Heat pump unit B is what performs the auxiliary heating operation for the highest heating loads. (However, when heat pump unit B is engaged in a hot water supply operation, the demands of the load are met by the supplement heater inside the terminal box instead.) We confirmed that when the flow rate of the primary heat medium (water) and the secondary heat medium (brine) of the double wall type plate heat exchanger in the heating and water heater system shown in **Figure 1** are 2 liters/minute and 3 liters/minute respectively, it is possible to maintain a capacity of 3 kW with an outlet water temperature (Wout) of 75˚C and return temperature (Win) of 50˚C, and a capacity of 4.3 kW with an outlet water temperature of 75˚C and return temperature of 40˚C.

4.3 Simulation results

We calculated the annual average COP while factoring in the switches between the operation modes such as the auxiliary heating operation by heat pump unit B. The conditions for the heating calculations included a hypothetical operation site in Sapporo City, a floor area of 140 square meters and an insulation efficiency of 1.24 W/m^2K, and we used the SMASH residential building thermal load calculation program. (Refer to **Table 3**.) By calculating the heating load from the weather data and comparing the performance characteristics for the heating load, we extracted the operation modes corresponding to the loads, calculated the total input and divided the total heating load to obtain the annual average COP. Furthermore, we fixed the flow at 3 liters/minutes and performed this calculation both when the outlet water temperature is set to a constant 60˚C, regardless of the fluctuations in the heating load, and when it is switched in five steps from 40˚C to 60˚C in accordance with the load. For the hot water supply load, we simulated the IBEC-L mode, and calculated the annual average COP in the same way as for heating. Based on field test data obtained in the past, the figures representing the drop in performance due to the defrosting cycle were also incorporated into the simulation, and consideration was given to bringing these figures into alignment with the actual equipment.

Table 4 shows the results of the simulation. When the heating water supply temperature is controlled in five steps from 40˚ to 60˚C, improvements of about 42% with heat pump unit A used exclusively for heating, about 38% with the heating system, and about 26% for the overall system including the water heater were obtained, compared to when the outlet water temperature is fixed at 60˚C. It can be understood that the effects of outlet water temperature control on the annual average COP are immense.

Table 3; Calculation conditions of annual average COP

Weather Data	Sapporo City
Calculation Program	SMASH
Floor Area	140 m^2
Insulation efficiency	1.24 [W/m^2K] (Next Gen. Standard Region Ⅰ)
Heating Load (Details)	11304 [kWh] (Unit A: 10271, Unit B: 615, Heater: 419)
Water Heater Load	6774[kWh]

COP	Heating Unit (A)	Heating	Water Heater	Total
Without outlet water Temperature Control (60˚C: Constant)	1.36 (100)	1.33 (100)	2.44 (100)	1.60 (100)

With outlet water Temperature Control (40 - 60˚C: 5 stage)	1.93 (142)	1.83 (138)	2.44 (100)	2.02 (126)

Table 4; Calculation results of annual average COP at Sapporo city

5 MONITORING TESTS

5.1 Installation conditions

We installed the systems in ordinary households in Sapporo (Refer to **Table 3**.)and Asahikawa, which are major districts in Hokkaido, and over a 1-year period, including the winter season, we conducted monitoring tests with a view to assessing system performance and assuring its reliability. An example of the installation conditions is shown in **Figure 6** and an example of a system on which the monitoring tests were conducted is shown in **Figure 7**.

Figure 6: Heat pump units installed in Asahikawa

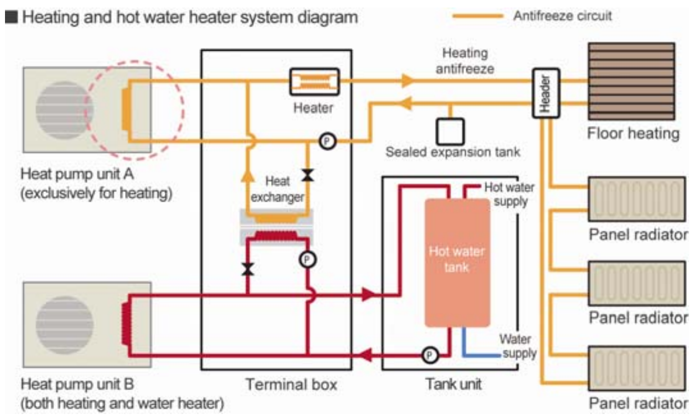

Figure 7: Total system of heating and water heater

5.2 Operation control

Based on **Figure 8**, we verified that the water heater capacity of heat pump unit B, used both for heating and as a water heater, was 5 to 6 kW at temperatures of –8˚C and below, and based on **Figure 9**, verified the heating capacity of heat pump unit A, used exclusively for heating, to be 4 to 5 kW at ambient temperatures of –15˚C and below. This system is designed in such a way that when the ambient temperature drops and the heating load increases to a point beyond the heating capacity of heat pump unit A, the auxiliary heating operation turns on. If, at this time, heat pump unit B is operating to supply hot water, the supplement heater turns on first, and provides support by intermittent operation using an ON/OFF switching system. When time passes and the hot water operation of heat pump unit B ends, the supplement heater turns off, and operation switches over to the auxiliary heating provided by heat pump unit B. We were able to verify this sequence from **Figure 9**.

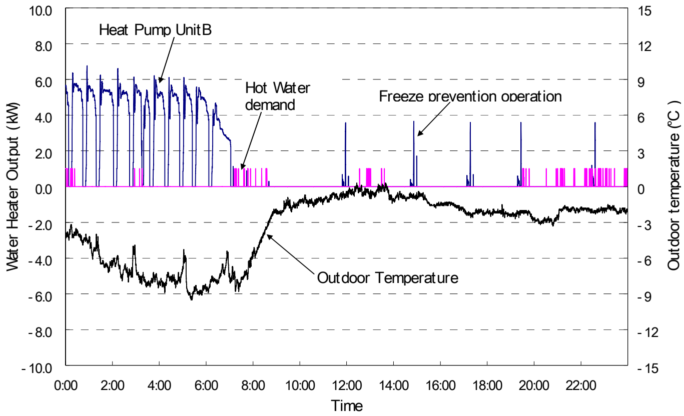

Figure 8: Water heater operation (Sapporo: 2007.Feb.8)

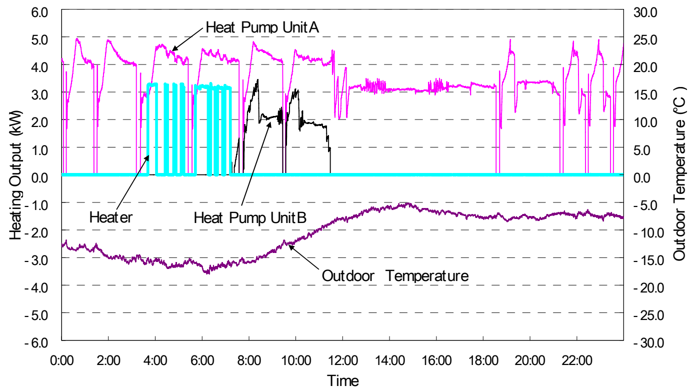

Figure 9: Heating operation (Asahikawa: 2007.Feb.10)

5.3 Room temperature control

The heating outlet water temperature control was exercised as follows: when the heating return temperature rose to a certain extent, this was interpreted to be a reduction in the heating load that reduced the supply temperature in steps and, conversely, when the difference between the heating supply and return temperature increased, this was interpreted to be an increase in the heating load that increased the temperature setting in steps. Based on **Figure 10**, it is possible to confirm that the outlet water temperature also shifts slightly upward as the ambient temperature drops and the heating load increases. In this case as well, it is clear that the room temperature is controlled to the extent where it slightly exceeds 20˚C.

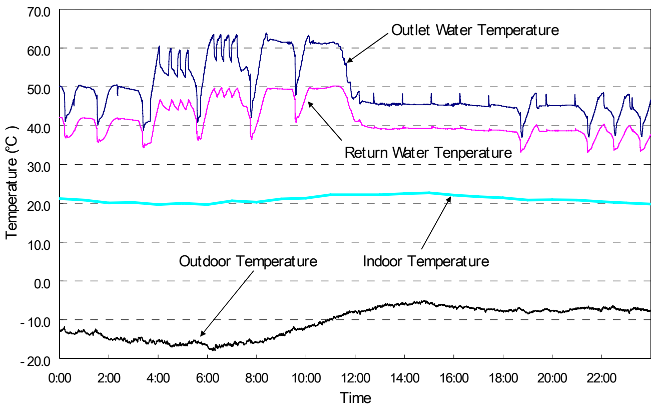

Figure 10: Room temperature control (Asahikawa: 2007.Feb.10)

5.4 Reliability

We performed the monitoring tests on a total of six units and so far no issues have arisen in the main parts used in the system. We were also able to verify during these monitoring tests that the installation of wind-tunnel prevention covers on the evaporators in locations where wind and snow are severe is effective in preventing ice formation, and that the software that prevents ice from forming on water pipes during shutdown of the water heaters can be customized using the software for the water heaters currently destined for cold regions, among other things. In this way, based on the verification results obtained from both a hardware and software perspective of the system, we believe that we have generally achieved a level of reliability that is thought to be required for the practical application of the system.

6 CONCLUSION

Incorporating a first-stage expansion intermediate cooling cycle (a split cycle) in heat pump unit A, used exclusively for heating, and heat pump unit B, used both for heating and as a water heater, has made it possible to drastically improve performance at low ambient temperatures. While laying the foundations for its basic performance, we devised a heating and water heater system consisting of two heat pump units and a 3 kW supplement heater, and carried out verification tests on this system in ordinary households in major districts in Hokkaido. As a result, we ascertained that this system can more than adequately support the hot water supply and heating loads anticipated in cold regions. Although issues such as a review of a control system for the partial heating load change need to be resolved item by item in the future, we believe we have generally attained our objectives if we basically assess matters from a practical application standpoint of the system.

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