

DEVELOPMENT OF MULTI-FUNCTION CO₂ HEAT PUMP WATER HEATER

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Abstract: We have succeeded in the development of a hot water supply system with a high-efficiency heat pump using a natural refrigerant (CO₂), and also succeeded in its mass-production for the first time in the world in 2001. After the first product launch, we have developed a multi-function type, which can be used as a floor-heater and bathroom-heater in addition to the hot water supply. During development, there was a difficulty that the efficiency decreased due to the temperature rise of water returned from the heater. However, we have solved this problem with our ejector cycle technology and the technology of the cascade heat process, which makes it possible to use the hot water in the storage tank for hot water supply or heater selectively. Then, we commercialised this multi-function type in 2003. In this paper, we are reporting the “multi-function CO₂ heat pump water heater” characterized by the ejector cycle technology and the technology of the cascade heat process.

Key Words: CO₂, Natural refrigerant, Heat pump, Water heater, Ejector

1 INTRODUCTION

Today, energy consumption in the residential sector accounts for about 14% of the total energy consumption in Japan, about one-third of which is for air conditioning systems, water heaters, and lighting equipment and household electrical appliances, respectively. Among these appliances, air conditioning systems as well as lighting equipment and household electrical appliances have become significantly more energy efficient in line with tightened rules and regulations concerning energy conservation.

For water heaters, however, improvement in energy efficiency has been slow due to the fact that most of these heaters are of the combustion type in which it is difficult to dramatically improve energy efficiency, though reduction in energy consumption has long been called for. To address this problem, Tokyo Electric Power Company, the Central Research Institute of Electric Power Industry, and DENSO CORPORATION jointly developed, in 2001, a highly energy-efficient, low energy consumption type heat pump water heater (HP water heater). It uses a natural refrigerant (CO₂), and was subsequently commercialized for the first time in the world.

Recently, the authors have developed an HP water heater having a heating capacity of 6.0 kW, in order to meet a variety of market needs. In addition to the functions of conventional 4.5 kW water heaters, the newly developed water heater can meet the needs for such additional functions as floor heating and bathroom drying/heating. In addition to an increase in water heating capacity from 4.5 kW to 6.0 kW, we have successfully improved the energy efficiency of the new type water heater by developing and practically applying the world's first ejector refrigeration cycle for CO₂ refrigerant, and a cascade heat process system which can more effectively manage the storage tank according to applications.

This paper presents the major factors in the development of the HP water heater and the results of field tests performed for performance evaluation purposes.

2 Development of a Multifunctional HP Water Heater and Ejection Cycle

2.1 Ejection Cycle Development

2.1.1 Characteristics of CO₂ Refrigeration Cycle

In a traditionally used refrigeration cycle (expansion valve cycle), the refrigerant undergoes isenthalpic expansion in the expansion valve as shown in Fig. 1, which is different from isentropic expansion in which thermal energy is conserved. In other words, when compared with an ideal refrigeration cycle in which thermal energy is conserved, the traditional refrigeration cycle includes an expansion process in which the refrigerant is subjected to irreversible adiabatic expansion. In this expansion process, kinetic energy is converted to thermal energy and heats the refrigerant. As a result, the amount of heat absorbed from the outside is reduced (expansion loss). It is well known that the expansion loss of the CO₂ refrigerant is intrinsically greater than that of conventionally used chlorofluorocarbon-base refrigerants. We have developed a new ejector refrigeration cycle and put it to practical use, reducing the power requirement of the compressor and thus remarkably enhancing the energy efficiency of the water heaters.

2.1.2 Outline of Ejector's Working Principle

The newly developed ejector refrigeration cycle replaces the conventionally used expansion valve with an ejector and also places an accumulator downstream of the ejector. The accumulator separates the liquid refrigerant from the gas and feeds the refrigerant into an air heat exchanger.

The ejector consists of a nozzle, mixing unit, and diffuser as shown in Fig. 2. The refrigerant entering the nozzle (drive flow) is depressurised and expands in the throttle or "throat" of the nozzle. When the refrigerant pressure drops below the outlet pressure of the air heat exchanger, the suction unit draws the refrigerant that has exchanged heat into the air heat exchanger (suction flow).

The mixing unit mixes the drive flow containing air and liquid uniformly with the suction flow of the gas refrigerant. The mixture is pressurized as it reduces velocity. It further reduces velocity when it passes through the enlarged cross section of the diffuser, increasing the outlet pressure (pressurization). As described above, the new ejector cycle uses the kinetic energy of the refrigerant effectively to boost the pressure of the mixture and thus to assist in the power required by the compressor.

2.1.3 Principal Points of CO₂ Refrigeration Cycle Development

The principal point of the refrigeration cycle development was how to maintain the refrigerant discharge temperature. The basic function of water heaters is to produce a required heating capacity. To meet this requirement, it is important to maintain the discharge temperature of the refrigerant that has been pressurized by the compressor. In the ejector cycle shown in Fig. 3, the refrigerant reduces discharge temperature as the ejector pressurizes the refrigerant, which reduces heating capacity. A method for securing a required refrigerant discharge temperature had to be devised.

In order to ensure the required refrigerant discharge temperature without sacrificing the effect of the ejector, we studied the usefulness of both an internal and external heat exchanger.

The internal heat exchanger was used to absorb heat then used to heat the refrigerant discharged from the accumulator, while the external heat exchanger was used to absorb heat from the outside air to heat the refrigerant. The evaluation test results for this design are shown in Fig. 4. First, we checked the effect of the internal heat exchanger under winter conditions, during which hot water of the highest temperature should be supplied.

As can be seen from the curve representing the effect of refrigerant discharge temperature on the water supply temperature, the internal heat exchanger worked sufficiently for

supplying high temperature hot water but was very ineffective for supplying low-temperature water. Our analysis showed that the reason for the above result was that the difference between the accumulator outlet temperature and heat exchanger outlet temperature was small when the water supply temperature was low and the amount of heat to be exchanged was reduced accordingly.

One of the methods for alleviating the above unfavourable result was to increase the capacity of the internal heat exchanger. However, this method was expected to deteriorate the ejector's power recovery rate under some conditions. It was necessary to devise a heat absorption system that does not impede the power recovery by the ejector. As a means of absorbing heat without reducing the power recovery rate, we considered absorbing heat from the outside air. To address this challenge, we positioned an additional external heat exchanger in the rear of the accumulator and carried out a test to evaluate the effectiveness of this exchanger. The test result showed that this configuration effectively assured the required heating power even when the water supply temperature was low. Thus we combined an internal heat exchanger and external heat exchanger to effectively absorb heat over the entire hot water supply temperature range.

2.2 Development of the Cascade Heat Process System

2.2.1 Outline of the Cascade Heat Process System

Conventional HP water heaters return heat source water (about 45 °C) for room heating to the lower part of the thermal storage tank used to store hot water for water heating (thermal storage tank). The returned heat source water is mixed with feed water (about 9 °C), forming a low temperature water layer (30 °C) in the lower part of the thermal storage tank. Such low temperature water cannot be used for hot water supply. In order to obtain the low temperature water, it is necessary to use the high temperature water (70 to 90 °C) in the upper part of the thermal storage tank. When the high temperature water is unused, the low temperature water remains in the thermal storage tank, thus reducing the energy efficiency. As shown in Fig.5, the thermal storage tank of the newly developed cascade heat process system is designed to have two inlets used to return heat source water for room heating, in order to prevent the hot water from being mixed with the feed water in the tank. This method produces intermediate temperature water (35 to 75 °C) which can be used for hot water supply, instead of low temperature water which cannot be used for hot water supply. Furthermore, several outlets are located at the upper, middle, and lower parts of the thermal storage tank to allow intake of the intermediate temperature water. Selecting an appropriate outlet port according to where the intermediate temperature is located makes it possible to selectively use the intermediate temperature water.

2.2.2 Design of the Cascade Heat Process System

Determining the positions of the valves of the newly developed cascade heat process system is greatly affected by the positions of the outlets of the intermediate temperature water. We studied the variation in the amount of the intermediate temperature water taken with different positions of the outlets of the intermediate temperature water in IBECL mode, which is the most common model in Japan (average floor area to be heated: 13.12 m², average operating time: 8 hours plus 20 minutes for bathroom heating, average number of family member: 3). The result of the study is shown in Fig. 6. The test indicated that the largest amount of intermediate temperature water could be taken when an outlet was located at the 320 L level.

It is already known that returned heat source water for room heating forms a boundary layer having a different temperature range and remains stable in the thermal storage tank. The temperature difference and thickness of the layer are shown in Fig. 7. Generally, the amount of water in the boundary layer accounts for 6 to 8% of the total capacity of the thermal storage tank. The figure shows that the boundary layer contains 30 to 40 L, which is 6 to 8%

of the total thermal storage tank capacity of 460 L, as mentioned above, and that the boundary layer forms when the temperature difference exceeds 15 °C. The returned heat source water (about 45 °C) and the high temperature water (70 to 90 °C) exist separately and stably in the tank. It is, therefore, recommended that the most appropriate position of the inlet of the returned heat source water be located at a level lower than the outlet port of the intermediate temperature water by 40 L. Accordingly, the position of the inlet port was determined to be 280 L.

The position of the bottom of the thermal storage tank was determined to be 100 L, which is the water level when water is heated during the night, and the position of the inlet of the returned heat source water was determined to be 60 L, which is lower than the bottom of the thermal storage tank by 40 L.

2.3 Evaluation of Hot Water Supply Performance

2.3.1 Performance Evaluation at Environment Testing Laboratory

We constructed a field-test HP water heater to evaluate its performance. Assuming an average family of four as the user of this test machine, we set the hot water supply load at a value equivalent to IBEC L mode or 421 L/day (in terms of 43°C hot water) and set the operating conditions in accordance with standards established by the Japan Society of Refrigerating and Air Conditioning Engineers (JRA4050-2001). We also carried out a performance evaluation test under a heating load for a multifunctional water heater we have recently commercialised. We assumed that this product would be used for room heating during winter, and set the test ambient condition according to the standard operating conditions specified by the Japan Society of Refrigerating and Air Conditioning Engineers (JRA4050-2001). The heating load for the test is shown in Table 1.

The performance of the mass-produced '01 model 4.5 kW HP water heaters (using an expansion valve cycle) is also shown in Fig.8 for comparison purposes.

When compared with the expansion valve type HP water heaters, the newly developed ejector type heater increased the heating capacity by 33% and improved COP by 20% under the rated operating conditions. The test results also showed that expansion loss increased during the last part of the heating process when hot water supply temperature is attained. This further enhanced the effect of the ejector and improved COP by about 50%.

Since the mass-produced '01 model HP water heaters did not have a room heating function, the comparison data on heating load is not contained in the figure.

2.3.2 Performance Evaluation by Trial Users

Fig. 9 shows the correlation between the water heating and room heating performance evaluation test results we described in the preceding subsection as well as the evaluation results obtained by trial use of the newly commercialised products during January through July 2003.

This figure, which marks atmospheric temperature on the horizontal axis and COP on the vertical axis, shows that both evaluation results coincide with each other. The heating load condition for trial use is almost the same as that shown in Table 1. (Number of trial users: 7; average number of family members: 3.4; location of trial users: Kanagawa pref., Nagoya city, etc.; hot water supply load: 470 L/day at 43°C; floor area to be heated: equivalent to 14 mats; average operating time: 8 h per day.) COP under the ambient condition (winter, 7°C) specified in JRA4050, which can be determined from the graph, is nearly equal to that which we measured at the environmental testing laboratory. For COP in the intermediate season and summer, there is no significant difference between the evaluation test result and trial use result.

2.3.3 Estimation of Yearly Average COP of Multifunctional System

Based on the fact that the result of the performance evaluation of the multifunctional field test machine at the environmental testing laboratory and the results of test use coincided closely we estimated the yearly average COP of a multifunctional HP water heater by using a simulation. The results are shown in Table 2.

The table shows that the yearly average COP was 3.0 or higher after the following factors were taken into account: heat dissipation from hot water storage tank (approx. 10%), power consumed to defrost the storage tank, heat loss during the end of water heating (mixing loss from feed water that is highly heated during the end of heating process, which usually causes COP to drop), and reduction in efficiency due to feed water temperature rise in room/floor heating mode.

2.4 Conclusion and Future Task

The authors successfully enhanced the energy efficiency of hot HP water heaters by 1: recovering the energy that has traditionally been wasted during expansion of refrigerant in the refrigeration cycle and 2: developing a cascade heat process system which uses different temperature zones of heat source water stored in the storage tank according to application (water heating or room heating).

From the trial use results of the HP water heaters with room/floor heating functions, we confirmed that new products with an ejector cycle can attain a yearly average COP of 3.0 or higher in and around Tokyo, though COP fluctuates somewhat depending on ambient temperature, feed water temperature, hot water supply load, and heating load (which varies depending on the heat insulation performance of each residence).

We will continue to add further value to the natural refrigerant (CO_2) HP water heater we have developed by enhancing its energy efficiency, thereby expanding the market size of water heaters of this kind.

3 TABLES AND GRAPHICS

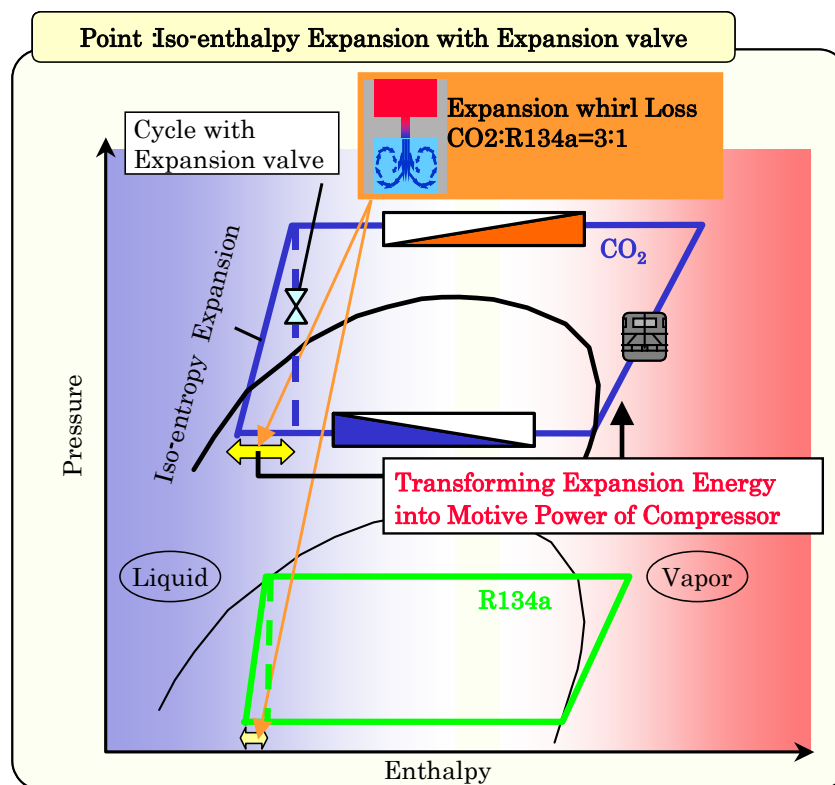


Figure 1: Improvement in Efficiency of Refrigeration Cycle

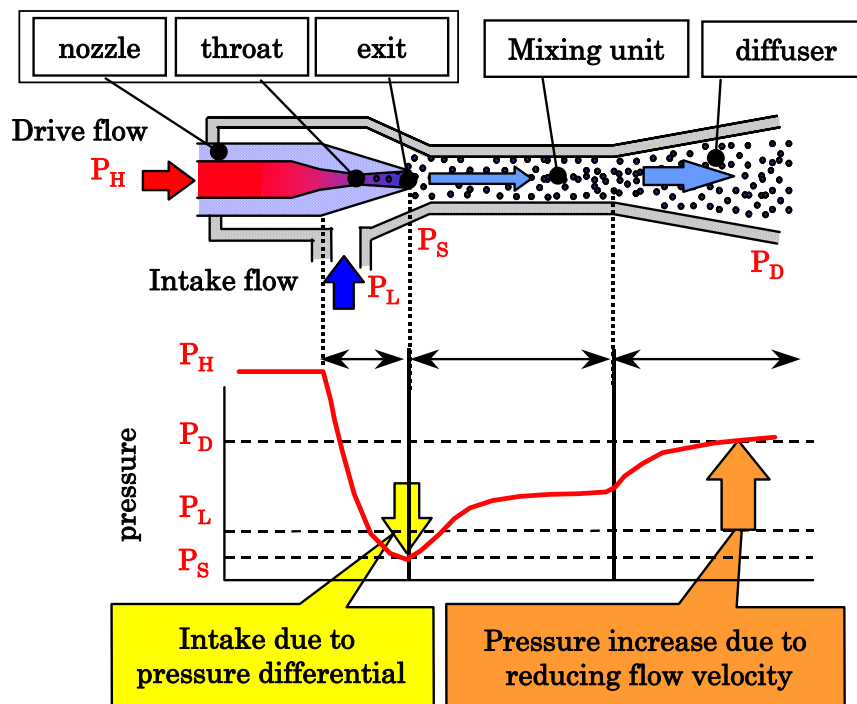


Figure 2: Schematic Illustration of Ejector and Pressure Change inside the Ejector

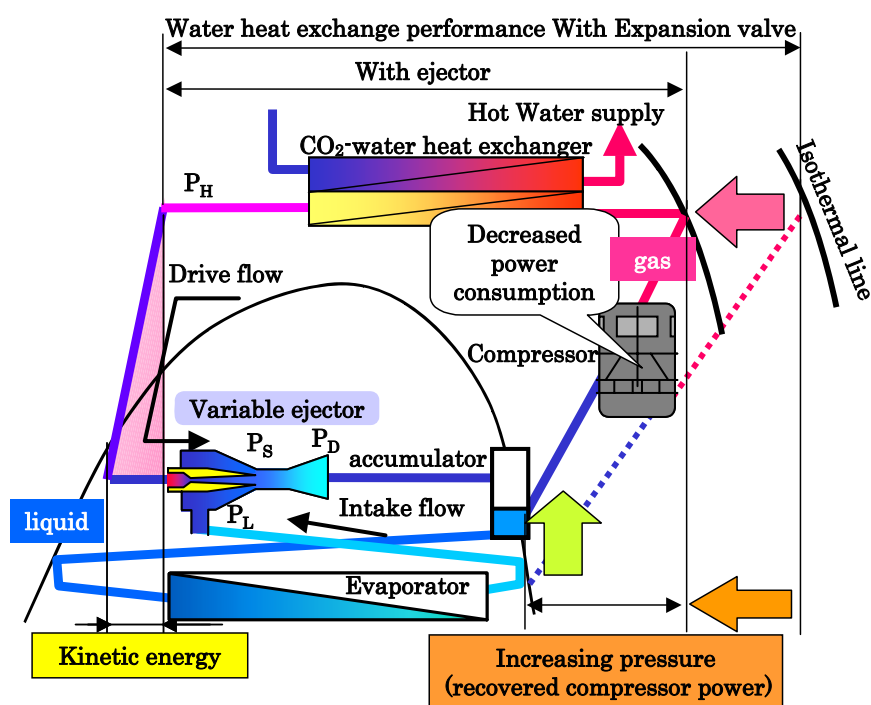


Figure 3: Pressure-temperature Balance in Heat Pump Ejector Cycle

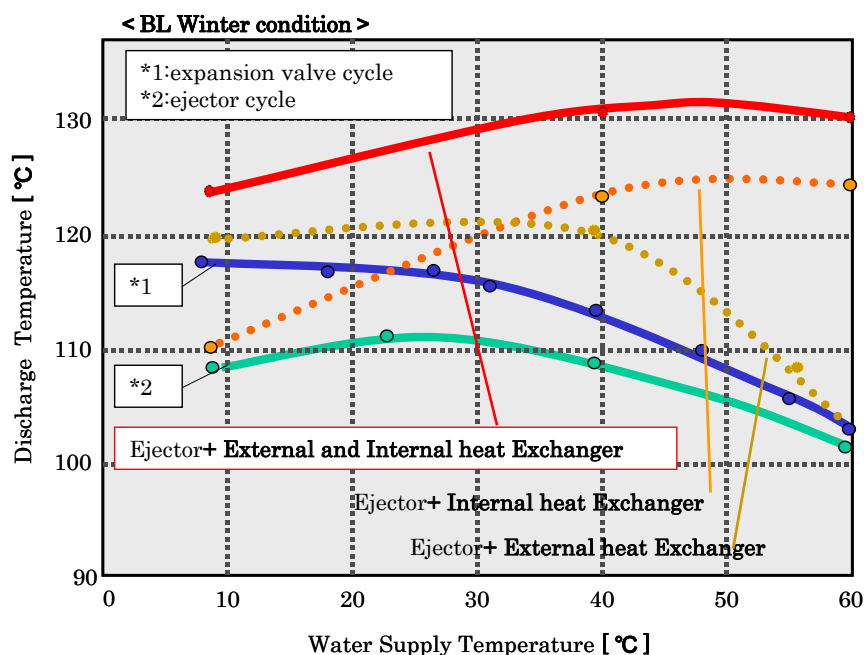


Figure 4: Effects of External and Internal Heat Exchangers

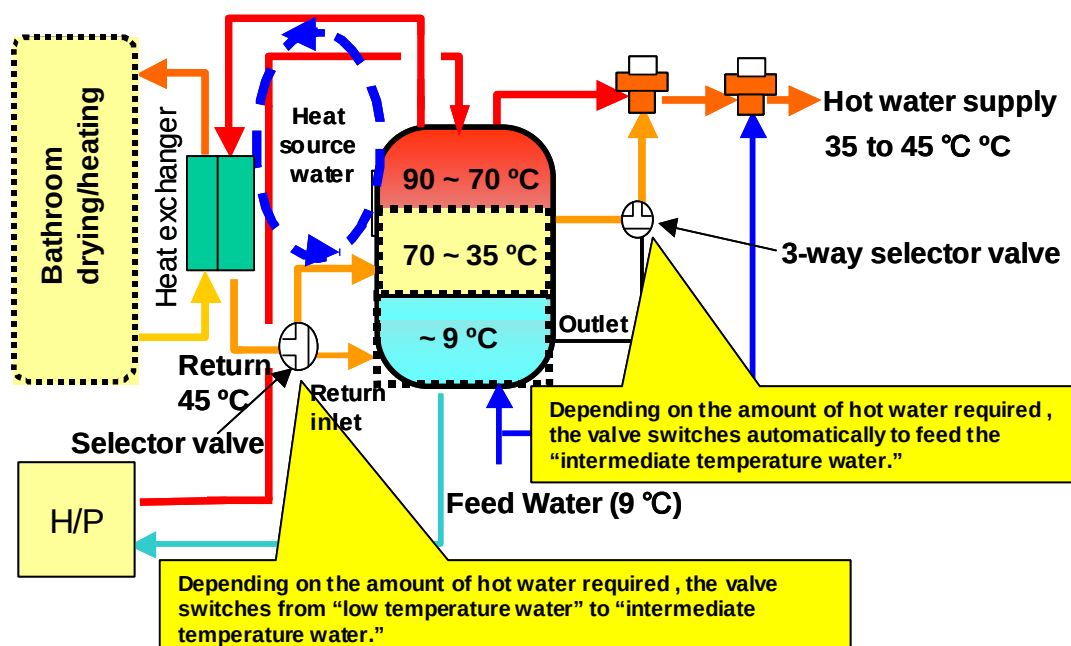


Figure 5: Cascade Heat Process System

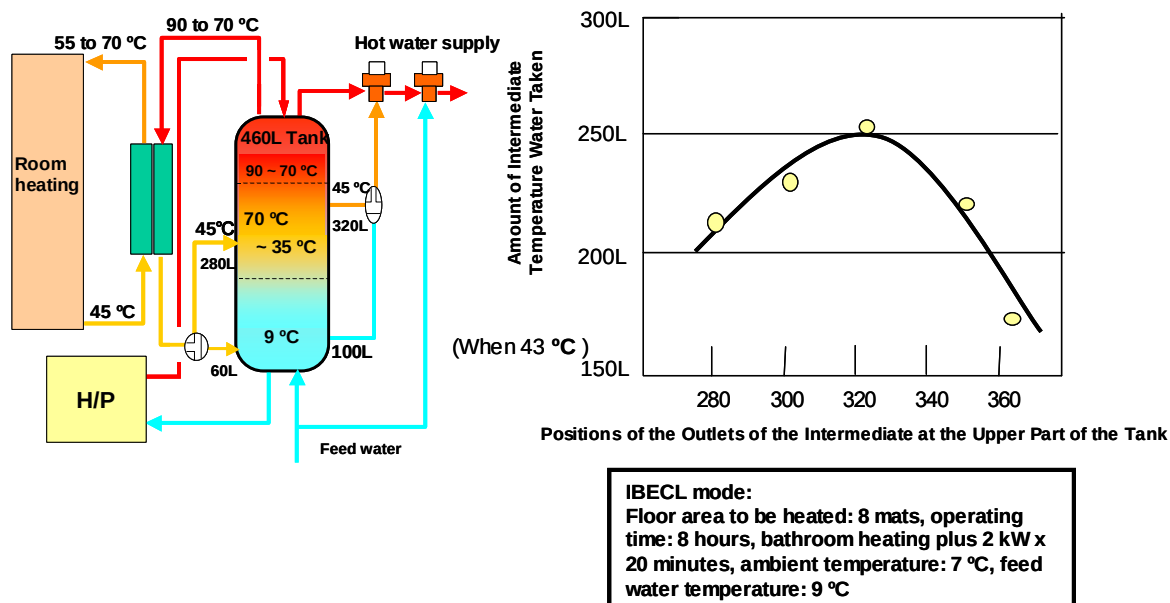


Figure 6: Positions of the Outlets of the Intermediate Temperature Water

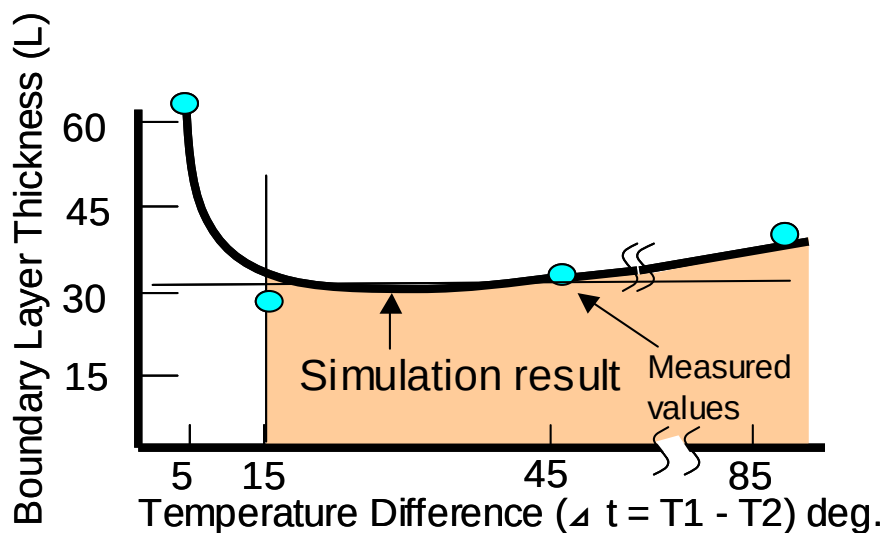


Figure 7: Boundary Layer Formation

Table 1 : Heating Load Condition

Hot Water Supply Load		IBEC L Mode
Floor Heater	Heat Loss Efficiency	3.4W/m ² · k
	Rate of Heating Upper Surface	70%
	Operating Time (6:00 - 10:00, 19:00 - 23:00)	8 Hr
Bathroom Heating Driver	Operating Time	20 Min.
	Power	2 kW
Outside Air Temp.		7°C (JRA4050:2001)
Water Temp.		9°C

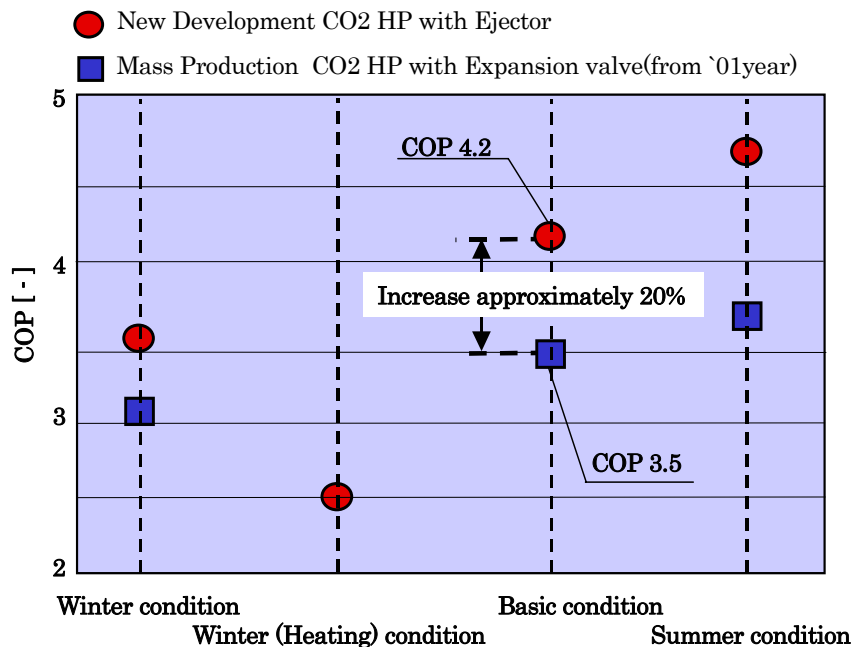


Figure 8: Performance Comparison between Production Model and Newly-developed HP Water Heater

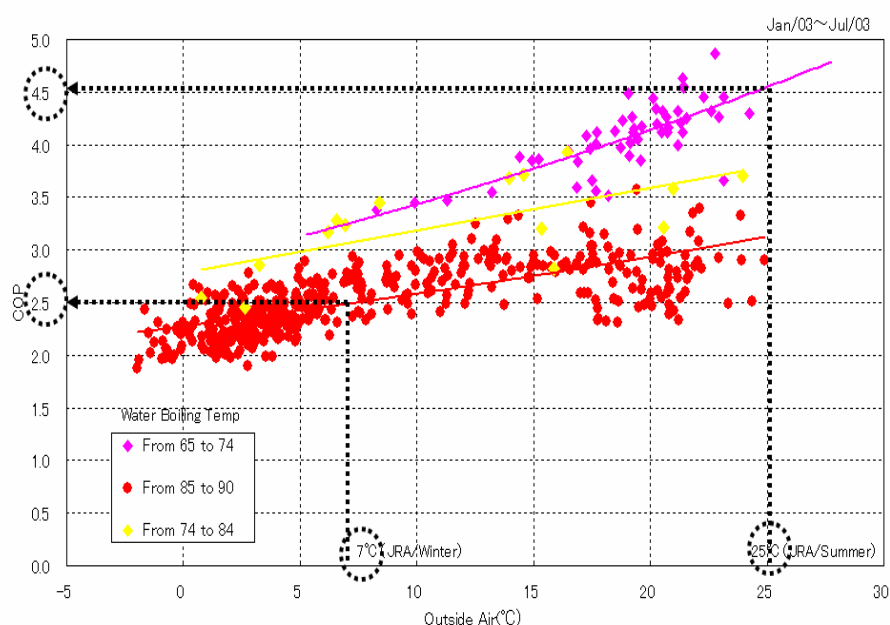


Figure 9: Performance Obtained by Field Test

Table2 : Annual Average COP Calculation

	Winter(Heating)	Intermediary		Summer	Total
	Heating	Heating	-	-	-
Month	Dec.Jan.Feb.	Nov.Mar	Oct.Apr.May.Jun.	Jul.Aug..Sep.	-
Days	90	60	120	95	365
Outside AirTemp (°C) DB/WB	7/6	16/12	16/12	25/21	-
Water Temp (°C)	9	17	17	24	-
Hot Water Load (MJ)	6,143	3,152	6,109	3,534	18,938
Heating Load (MJ)	4,267	1,389	0	0	5,656
Total Load (MJ)	10,410	4,541	6,109	3,534	24,594
Electric Power (MJ)	4,314	1,598	1,508	795	8,215
COP	2.4	2.8	4.1	4.4	3.0

NOMENCLATURE

- P_h : inlet pressure of Nozzle
 P_d : outlet pressure of diffuser
 P_l : pressure of suction flow refrigerant
 P_s : inlet pressure of Mixing section
 Q : refrigerant performance
 L : Motive Power of compressor

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