PERFORMANCE ANALYSIS OF GROUND SOURCE HEAT PUMPS FOR DOMESTIC HOT WATER SUPPLY IN THE LOW ENERGY HOUSE AND ITS APPLICATION

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Abstract: Two types of the heat pump unit with R410a and CO_2 as the refrigerant for supplying domestic hot water in the low energy house were analyzed. First, performance tests of the GSHP units for DHW were carried out and the COP and heating output according to the inlet and outlet temperature, and flow rate were investigated. Next, it was shown that the GSHP systems for DHW with R410a and CO_2 can reduce 30~ 50 % of CO_2 emission compared to the conventional oil boiler system by simulation with a design tool for the GSHP system. In addition, the authors introduced the ground source CO_2 heat pump system application for the residential cooling and demonstrated that the total COP is 4.21 and this system can provide SC and DHW with high efficiency by simulation.

Key Words: ***Ground Source Heat Pump System, Domestic Hot Water Supply, Low Energy House, CO_2 Heat Pump for Cooling, Performance Test, System Simulation***

1 INTRODUCTION

Recently, supplying domestic hot water (DHW) with high efficiency becomes more important for the low energy house, which saves energy and reduces CO_2 emission. Especially in Japan, energy consumption for DHW account for approximately 30% of total energy consumption in the residential house. Therefore, shipment of CO_2 heat pumps so-called “ECO CUTE”, which can generate DHW with higher efficiency, has been increasing rapidly. Also, heat pumps for DHW utilizing other refrigerants are commercialised. However, there are very few cases where ground source heat pump (GSHP) systems for DHW are applied in Japan although the GSHP system can operate with higher efficiency even for DHW.
In this paper, the authors analyse performance of two types of GSHP for DHW and demonstrate its advantage. The one of heat pumps utilizes R410a and the other uses CO_2 as refrigerants. First, the heating output and COP according to temperature of the primary and secondary side of the heat pumps are evaluated by the performance tests. Next, the authors approximate the heating output and COP according to the temperature. Then, the approximate equation is installed to a computer simulation tool for the GSHP system. In addition, annual performances of the heat pumps are calculated with the simulation tool. Furthermore, feasibility study of the GSHP utilizing CO_2 as the refrigerant for space cooling (SC) is carried out. It is difficult to install refrigerators with CO_2 as the refrigerant for SC in the moderate climate region like Tokyo in Japan. This is due to the extremely lower energy efficiency compared to the conventional refrigerators with Freon. However, it is possible to lower the water temperature in the primary side by injecting the exhaust heat of SC to the ground since the ground temperature is kept almost constant throughout the year e.g. the temperature is around 16~17 °C in Tokyo. This prevents decrease of energy efficiency of the

refrigerator with CO_2 as the refrigerant. In this system, to supply DWH by using the exhaust heat is also possible and it yields higher energy efficiency. The authors make a performance prediction tool for the GSHP system by using the performance test results and simulate the system performance.

2 COMPARISON OF TWO TYPES GROUND SOURCE HEAT PUMP FOR DOMESTIC HOT WATER IN THE LOW ENERGY HOUSE

2.1 Performance test of heat pumps with different refrigerants

2.1.1 Performance test of heat pump with R410a as refrigerant

Figure 1 shows appearance of a heat pump unit with R410 as refrigerant for DHW. As shown in Figure1, the footprint is smaller than the heat pump unit for space heating (SH) and SC and it can be placed next the unit for SH and SC. This unit has an inverter-controlled compressor and can vary the revolution speed and heating output according to heat demand. Shown in Figure 2 is schematic diagram of performance test apparatus. The unit is connected to two water tanks in the apparatus, which imitate primary side as heat source and secondary side for DHW. This apparatus can automatically keep the inlet and outlet water temperature of the unit constant.
In this performance test, temperatures and flow rates at each part indicated in Figure 2 are measured at five-second intervals by keeping revolution speed of the compressor constant. After the temperatures almost reach steady state, additional measurement is conducted for ten minutes. Performance of the unit is analysed by using the average values in ten minutes. Here, heating output Q_2 and COP of the unit are calculated by the following equations.

$$Q_2 = c_{f2}\rho_{f2}G_{f2}(T_{2out} - T_{2in}) \tag{1}$$

$$COP = Q_2 / E \tag{2}$$

First, the performance test was carried out on the conditions indicated in Table 1. Heating output and flow rate in the secondary side are changed. Then inlet and outlet temperatures in the secondary side of the heat pump unit are kept at 17 ^{o}C and 65 ^{o}C, respectively. As the result, Figure 3 shows heat pump COP according to the heating output Q_2. The COP becomes maximum value when the heating output is around 4.7. Therefore, it suggests that the flow rate of around 1.4 L/min is the most adequate. Here, when the flow rate in secondary side is set at 1.4 L/min, the time required to fill the DHW tank of 460 L with hot water of 65 ^{o}C is 5.5 hours.
From this result, additional test was conducted on the condition that flow rate in secondary side is kept at 1.3 L/min constant as shown in Table 2. Also in this experiment, the inlet temperature T_{1in} in the primary side and compressor revolution speed are varied. Figure 4s show heating output, outlet temperature in the secondary side T_{2out}, and COP according to T_{1in}. The heating output increases as the inlet temperature and compressor revolution speed rise. The COP drops when the compressor revolution speed becomes higher. This is because the temperature T_{2out} is raised by the increase of heating output.
As these test results, the authors obtained the following approximate equation of COP according to T_{1in} and T_{2out} by multiple linear regression analysis.

$$COP = 0.0898T_{1in} - 0.0543T_{2out} + 5.66 \tag{3}$$

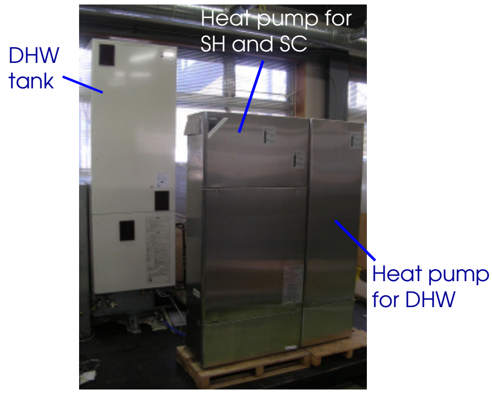


Figure 1: Appearance of heat pump unit for DHW

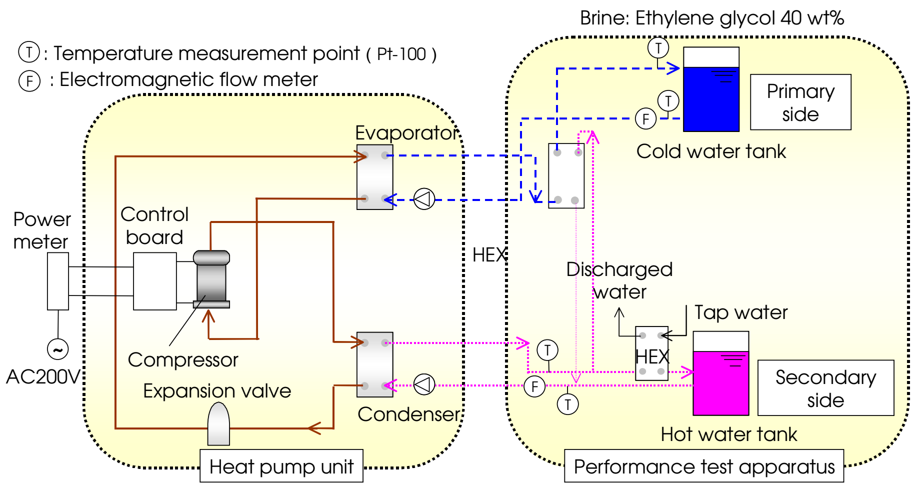


Figure 2: Schematic diagram of performance test apparatus and heat pump unit

Table 1: Experimental conditions (T_{2out} is constant)

	T_{2in} (°C)	T_{2out} (°C)	T_{1in} (°C)	Compressor revolution speed (rpm)	Q_2 (kW)	G_{f2} (L/min)	G_{f1} (L/min)
Exp.1-1	17	65	5	1,800	1.63	0.49	10
Exp.1-2				4,200	4.71	1.41	
Exp.1-3				4,800	5.54	1.66	
Exp.1-4				5,400	6.38	1.91	

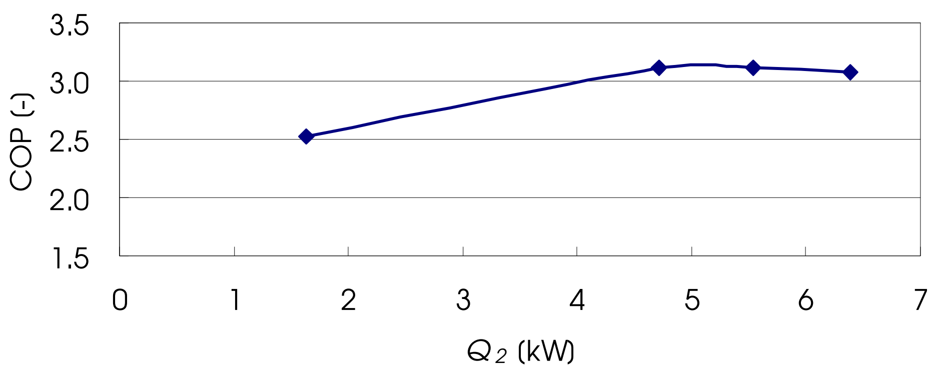
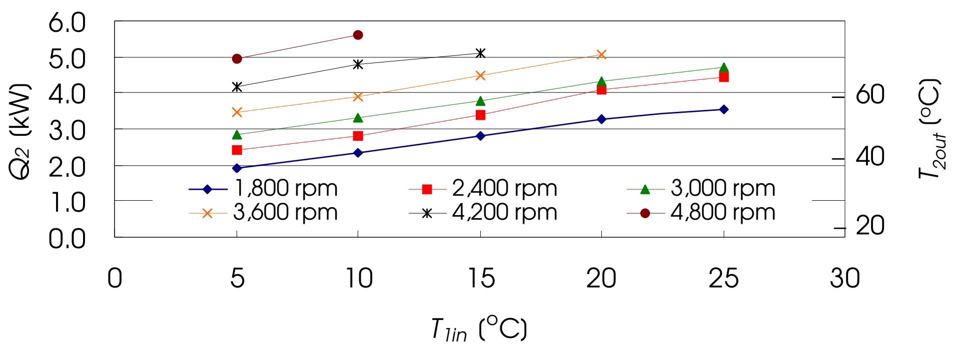


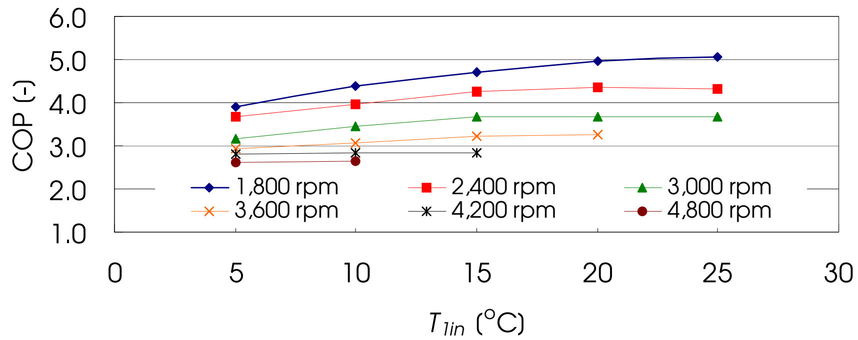
Figure 3: COP according to Q_2

Table 2: Experimental conditions (G_{f2} is constant)

	T_{2in} (°C)	T_{1in} (°C)	Compressor revolution speed (rpm)	G_{f2} (L/min)	G_{f1} (L/min)
Exp.2	17	5~25	1,800~4,800	1.3	10



(a) Q_2 according to T_{1in}



(b) COP according to T_{1in}

Figure 4: Q_2 and COP according to T_{1in}

2.1.2 Performance test of heat pump with CO_2 as refrigerant

Performance test of a heat pump unit with CO_2 as refrigerant was conducted at the test field in Sapporo city of Japan. Figure 5 shows schematic diagram of the field test. The unit is connected to a borehole ground heat exchanger (GHEX) in the primary side and a DHW tank in the secondary side. The GHEX has double U-tube and the length is 100 m. The DHW tank is generally used one. The inlet and outlet water (or antifreeze liquid) temperatures are measured by Pt-100 temperature sensors as shown in Figure 5. The flow rates are also observed by the flow sensors. Power meters set at the compressor of the heat pump unit and circulation pump in primary side measure the electric power.
In this experiment, the heat pump unit automatically operates in nighttime (11:00 pm ~7:00 am). If the inlet temperature in the primary side becomes over 50 °C, the unit stops. Table 3 indicates the experimental conditions. In the first test, the outlet temperature in the secondary side T_{2out} is kept at 65 °C constant. The temperature T_{2out} is up to 90 °C in the second test. Additionally, the temperatures in the secondary side T_{2out} and T_{2in} are varied in the next test. Temperatures, flow rates, and electric powers, whose points indicated in Figure 5, are measured at one-minute interval. The average values in ten minutes are analyzed. The

heating output Q_2 and COP of the unit are calculated by Equation (1) and Equation (2) as well as in **Chapter 2.1.1**.
Figure 6s and Figure 7s are Q_2, E, and COP according to T_{1in} in Exp.1 and Exp.2, respectively. These results indicate that the unit provides stable heating output both in Exp.1 and Exp.2. The heating outputs Q_2 are more than 5 kW in both cases and almost equal to each other on a condition of the same the temperature T_{1in}. However, more electric power of the compressor is required in Exp.2 so that the COP is lower than that in Exp.1. Thus, the operating condition of T_{2out} = 65 °C is more efficient from a viewpoint of energy saving. Additionally, the following approximate equations of COP according to T_{1in} in both cases of T_{2out} = 65 °C and T_{2out} = 90 °C are obtained from the test results.

$$T_{2out} = 65\ ^{\circ}\mathrm{C}:\ COP = 0.0519\,T_{1in} + 3.268 \tag{4}$$
$$T_{2out} = 90\ ^{\circ}\mathrm{C}:\ COP = 0.0344\,T_{1in} + 2.803 \tag{5}$$

Shown in Figure 8 are heating output Q_2 and extracted heat in the primary side Q_1 according to T_{1in}. Here, Q_1 is expressed by the following equation.

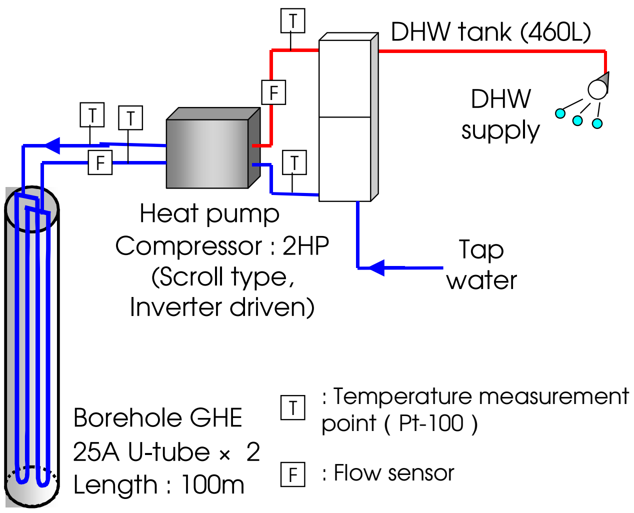
$$Q_1 = c_{f1}\rho_{f1}G_{f1}(T_{1out} - T_{1in}) \tag{6}$$

Although inlet temperature in the secondary side T_{2in} varies, this result suggests that variation of T_{2in} does not affect the heating output and heat extraction rate. Figure 9 shows flow rate in secondary side G_{f2} according to T_{2in}. Since this unit can automatically vary the flow rate, it can keep the heating output almost constant even in the case where T_{2in} varies. Figure 10 and Figure 11 point COP according to T_{1in} and T_{2in}, respectively. The value COP1 is the ratio of extracted heat to the electric power and COP2 is the one expressed by Equation (2). Both of the COP1 and COP2 become larger as T_{1in} rises and descend with increase of T_{2in}. From these results, the following approximate equation of COP while DHW supply on a condition of T_{2out} = 65 °C according to T_{1in}, G_{f2}, and Q_2 is derived by multiple linear regression analysis.

$$COP = 0.117\,T_{1in} - 0.531\,G_{f2} + 0.295\,Q_2 + 1.1425 \tag{7}$$

Also, the authors investigate COP while cooling by using the value of COP1. As the result, the following approximate equation of COP according to T_{1in}, T_{2out}, and G_{f1} is obtained. Here, the primary side and secondary side change place each other during cooling operation.

$$COP = -0.0275\,T_{1in} + 0.0331\,T_{2out} + 3.41\,G_{f1} + 1.984 \tag{8}$$



Measurement point

Temperature (Pt-100)	•Inlet and outlet of heat pump unit in primary side •Inlet and outlet of heat pump unit in secondary side
Temperature (Thermo couples)	Ambient air
Flow rate (Flow sensor)	•Heat pump primary side •Heat pump secondary side
Electric power (Power meter)	•Compressor of heat pump unit •Circulation pump

Figure 5: Schematic diagram of performance test apparatus

Table 3: Experimental conditions

	Days	T_{2in} (°C)	T_{2out} (°C)	G_{f2} (L/min)	G_{f1} (L/min)
Exp.1	21	10~15	65	1.3~2.0	
Exp.2	35	10~15	90	1.0~1.5	8~10
Exp.3	2	10~50	65	1.0~3.0	

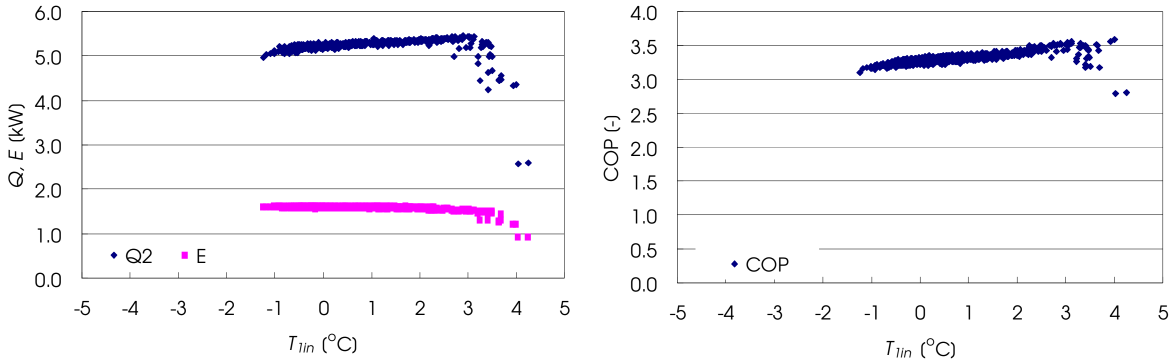


Figure 6: Q_2, E, and COP according to T_{1in} (In Exp.1)

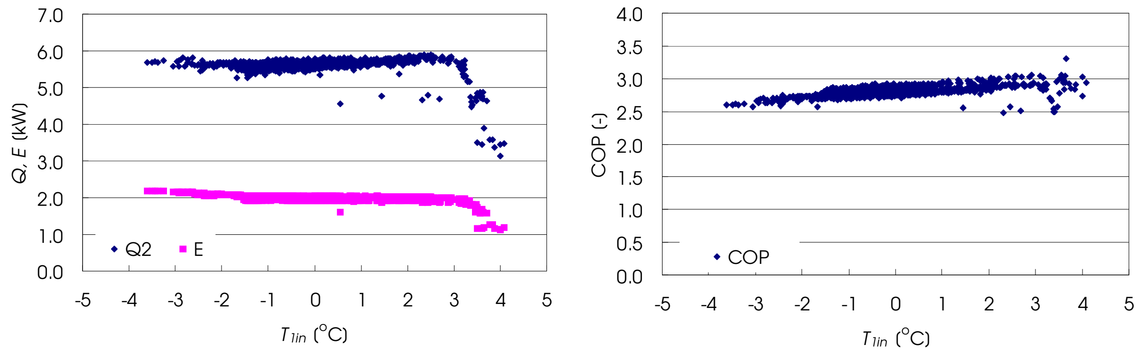


Figure 7: Q_2, E, and COP according to T_{1in} (In Exp.2)

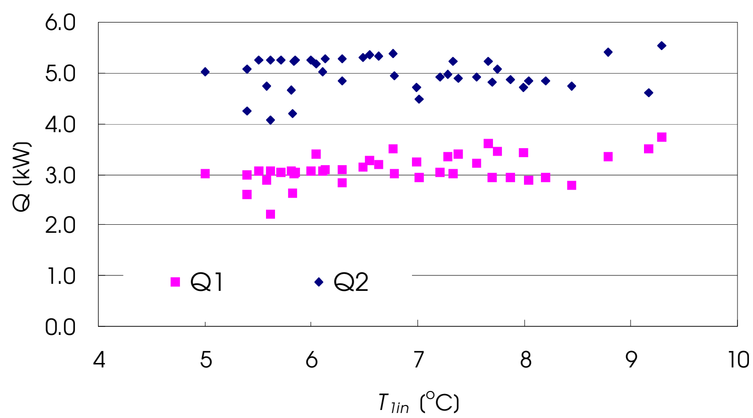


Figure 8: Q_2 and Q_1 according to T_{1in} (In Exp.3)

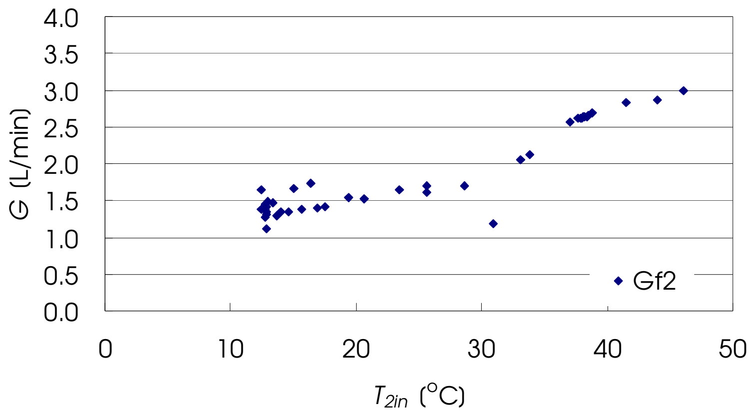


Figure 9: G_{f2} according to T_{2in} (In Exp.3)

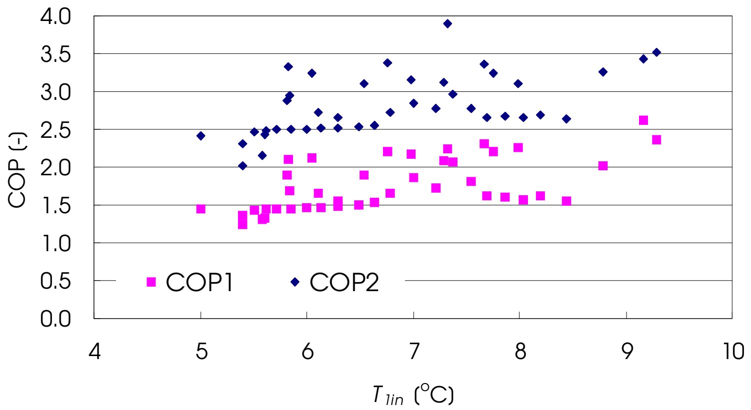


Figure 10: COP according to T_{1in} (In Exp.3)

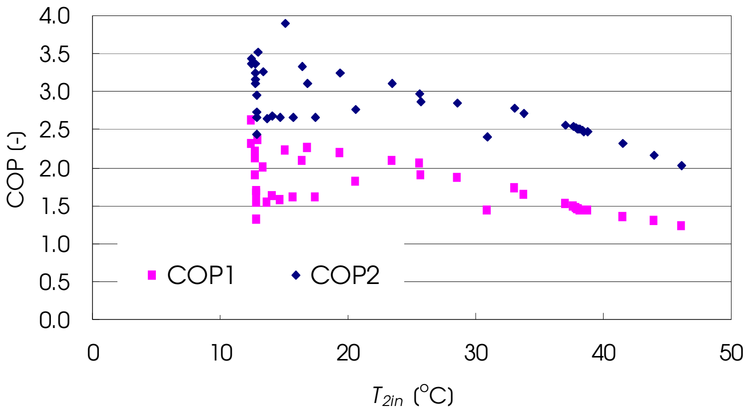


Figure 11: COP according to T_{2in} (In Exp.3)

2.2 Comparison of heat pumps with different refrigerant by simulation

2.2.1 Calculated conditions

Table 3 indicates calculated conditions for comparison of heat pumps for DHW with different refrigerant. It is assumed that the GSHP system for DHW is installed in a low energy house in Sapporo, which is a representative city in cold climate region of Japan. Outlines of heat pump system are shown in Figure 12. The heat pump unit is connected to the borehole GHEX in the primary side and the DHW tank of 460 L in the secondary side. Hot water with the temperature of 65 °C is supplied to the DHW by operation of the heat pump unit at nighttime (11:00 pm ~ 7:00 am). The hot water in the DHW is used according to the demand indicated in Figure 12. By installing Equation (3) and Equation (4) to a design tool for the GSHP system that the authors developed and using the design tool, the GSHP system operation for two years is simulated hourly.

Table 4: Calculated conditions

City			Sapporo
Soil		Ground temperature (°C)	11.0
		Effective thermal conductivity (W/m/K)	2.5
		Density (kg/m^3)	1500
		Specific heat (kJ/kg/K)	2.0
Ground heat exchanger	Borehole with double U-tube	External diameter of U-tubes (m)	0.032
		Internal diameter of U-tubes (m)	0.026
		Borehole diameter (m)	0.15
		Grout thermal conductivity (W/m/K)	1.8
Heat pump unit		Compressor	2-horse power
Electric power consumption of circulation pumps (W)			Primary side : 150 W
Heat carrier fluid (Brine)			Organic acid group 40%

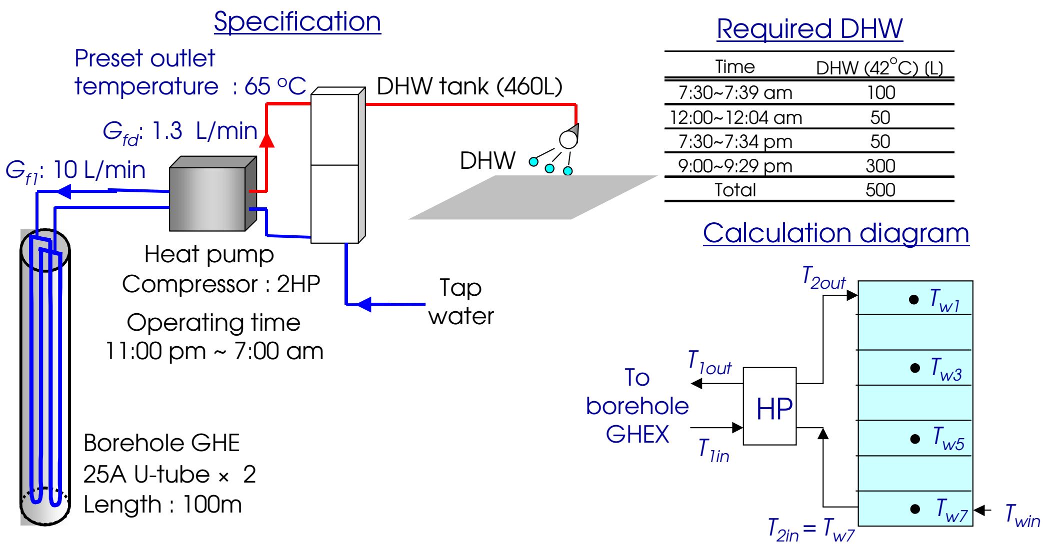


Figure 12: Outlines of calculation

2.2.2 Results and discussions

As the result of simulation in the second year, Table 4 shows heating output, electric power, COP and annual performance of the GSHP systems for DHW, which have heat pumps with different refrigerant. The results demonstrate that the GSHP with CO_2 as refrigerant provides higher energy efficiency for the DHW. In addition, CO_2 emissions to operate the GSHP system in annual are shown in Figure 12. In Figure 12, CO_2 emission by operation of a conventional oil boiler system is also indicated. The GSHP with CO_2 as refrigerant has great possibility of reduction of CO_2 emission as shown in Figure 12. Installation of the GSHP with R410a also yields to reduce more than 30% of CO_2 emission compared to the conventional oil boiler system.

Table 5: Predicated performance

	R410a	CO_2
D1. Heating output for DHW (kWh)	7290	7290
D2. Electric power consumption of compressor for DHW (kWh)	2581	1998
D3. Electric power consumption of circulation pumps (kWh)	321	321
D4. Average COP of heat pump unit during operation (=D1/D2)	2.82	3.65
D5. Annual performance factor (=D1/(D2+D3))	2.51	3.14

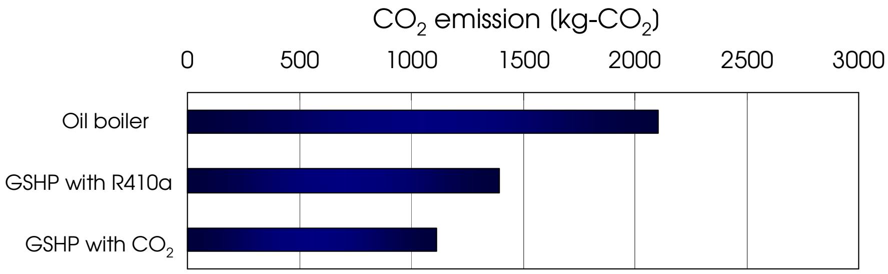


Figure 13: Comparison of CO_2 emissions

3 GROUND SOURCE CO_2 HEAT PUMP SYSTEM FOR RESIDENTIAL COOLING

3.1 System description

Figure 13 is a system diagram of the ground source CO_2 heat pump system for residential cooling. It is difficult to operate heat pumps with CO_2 as the refrigerant for cooling with high efficiency because of the refrigerant's characteristics. Especially, the performance drops when the temperature of heat sink becomes higher. However, it is possible to lower the water temperature in the primary side by injecting the exhaust heat of SC as shown in Figure 13. This improves energy efficiency of the refrigerator with CO_2 as the refrigerant. In addition, total performance of the system is advanced by using the exhaust heat to supply DHW. When the heat demand for SC is not generated, the GSHP system operates as well as the GSHP system only for DHW.

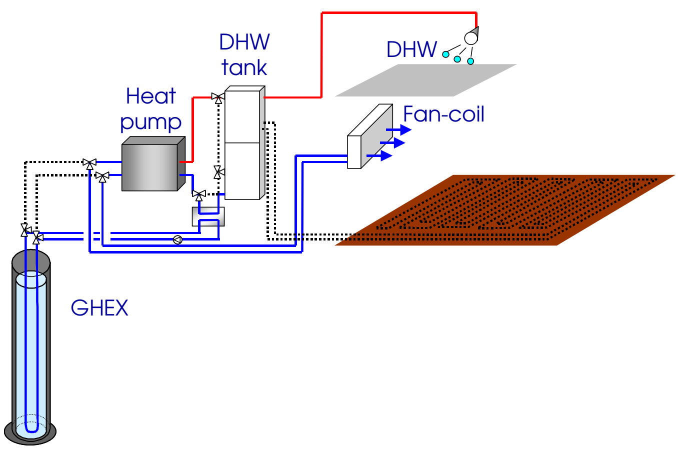


Figure 13: System diagram of the ground source CO_2 heat pump system for residential cooling

3.2 Feasibility study of ground source CO_2 heat pump system for residential cooling

3.2.1 Calculated conditions

Calculated conditions for feasibility study are indicated in Table 6. It is assumed that the GSHP system is installed in a residential house in Tokyo, which has larger heat demand for SC than the one in Sapporo. The hourly heat demand for SC was calculated by a building heating and cooling demand calculation software “SMASH for Windows”. Total cooling demand is 10.0 GJ and the GSHP system is operated to satisfy the cooling demand. A heat pump unit with CO_2 as refrigerant has a three-horse power compressor and the COP can be estimated by Equation (7) and Equation (8). Number of 20 steel foundation piles with the length of 8 m is used as the GHEX.

Table 6: Calculated conditions

City			Tokyo
Building		Heat loss coefficient (W/m^2/K)	2.57
		Total floor area (m^2)	130
Cooling period			Jun. 9th ~ Sep. 25th
Total cooling demand (GJ) (Maximum demand (kW))			10.0 (6.3)
Secondary system for SC			Fan-coil × 4
Soil		Ground temperature (°C)	16.5
		Effective thermal conductivity (W/m/K)	1.5
		Density (kg/m^3)	1500
		Specific heat (kJ/kg/K)	2.0
Ground heat exchanger	Steel pile with single U-tube	External diameter of U-tubes (m)	0.032
		Internal diameter of U-tubes (m)	0.026
		External diameter of steel pile (m)	0.14
		Internal diameter of steel pile (m)	0.13
		Length and number	8m × 20 piles
		Grout thermal conductivity (W/m/K)	1.8
Heat pump unit		Compressor	3-horse power
Electric power consumption of circulation pumps (W)			Primary side : 150 W Primary side for cooling : 150 W
Heat carrier fluid (Brine)			Organic acid group 40%
Volume of DHW tank			460 L

3.2.2 Results and discussions

Two years’ operation of the GSHP system is calculated by using the design tool. As the result, Figure 14 shows variations of heating output for DHW Q_{2DHW}, cooling output Q_{2SC}, and extracted heat Q_1 in the second year. In the cooling period, the heating output Q_{2DHW} is generated by exhaust heat of cooling operation. Thus, if the cooling output Q_{2SC} becomes larger, the heating output Q_{2DHW} also increase. The system operation for SC and DHW at the same time reduces the extracted heat Q_1. Next, variations of temperature at each part in the primary side are shown in Figure 15. The maximum temperature of T_{1out} is around 27 °C and it indicates that the GSHP system can operate for long term. Finally, the performance of GSHP system is summarized in Table 7. Although the COP of heat pump unit for SC of 2.20 is not so high, the COP for DHW is fairly good. The total COP of 4.21 is obtained so that this system can provide SC and DHW with high efficiency. Additionally, the system performance will be increased by improving the heat pump unit and the operation method.

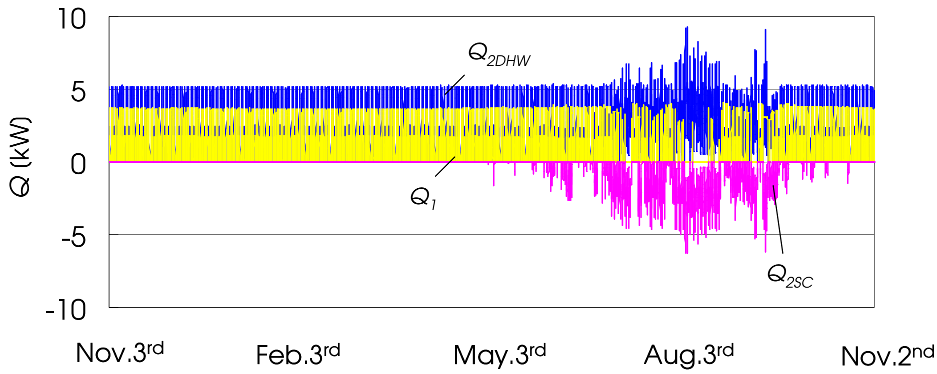


Figure 14: Variations of *Q_{2DHW}*, *Q_{2SC}*, and *Q_1* in the second year

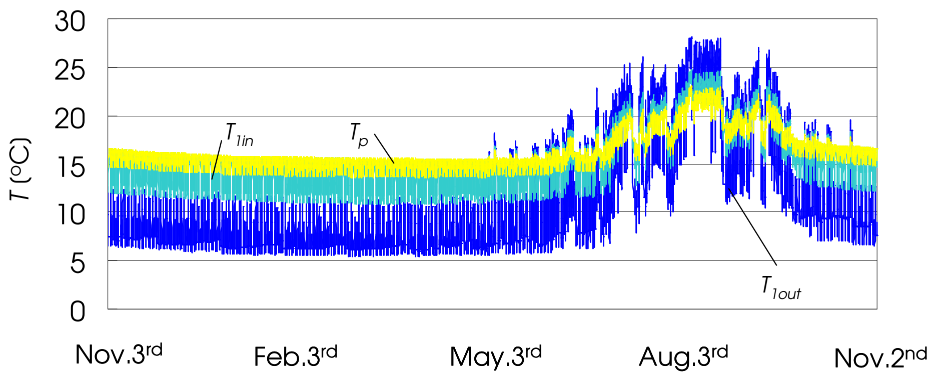


Figure 15: Variations of temperatures in the primary side in the second year

Table 7: Predicted performance

SC	C1. Cooling output for SC (kWh)	2774
	C2. Electric power consumption of compressor (kWh)	1261
	C3. Average COP of heat pump unit (=C1/C2)	2.20
DHW	D1. Heating output for DHW (kWh)	10401
	D2. Electric power consumption of compressor for DHW (kWh)	3127
	D3. Electric power consumption of circulation pumps (kWh)	690
	D4. Average COP of heat pump unit (=D1/D2)	3.33
Total	Average COP of heat pump unit (=(C1+D1)/D2)	4.21
	Annual performance factor	3.45

4 CONCLUSIONS

1) Performance tests of the GSHP units for DHW were carried out and the COP and heating output according to the inlet and outlet temperature, and flow rate were investigated. As the result, approximate equations to evaluate the COP of each unit were obtained.
2) By simulation with a design tool for the GSHP system, performance of the GSHP systems for DHW with R410a and CO_2 as the refrigerant was compared. The result demonstrated that the GSHP can reduce 30~ 50 % of CO_2 emission compared to the conventional oil boiler system. Especially, the GHSP with CO_2 provides higher energy efficiency for the DHW in the low energy house.

3) The authors introduced the ground source CO_2 heat pump system application for the residential cooling and conducted feasibility study of the system by simulation. Although the predicted COP of heat pump unit for SC of 2.20 is not so high, the total COP of 4.21 is obtained. Thus, this system can provide SC and DHW with high efficiency.

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NOMENCRATURE

c: Specific thermal capacity [kJ/kg/K], *E*: Electric power [W], *G*: Flow late [m^3/s], *L*: Length [m], *Q*: Heating output [W], *T*: Temperature [^oC], *t*: Time [h], *ρ*: Density [kg/m^3]

Subscript

DHW: DHW, *dt*: DHW tank, *f*: Thermal medium, *h*: Heat exchanger, *in*: Inlet, *out*: Outlet, *p*: Pipe (Ground heat exchanger), *SC*: Space cooling, *1*: Primary side, *2*: Secondary side, *-in*: Inside, *-out*: Outside