Study of ground source heat pump system with high cost performance

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Abstract
We manufactured a Ground Source Heat Pump (GSHP) system of the test model and investigated the Coefficient Of Performance (COP) of a fabricated GSHP system by experiment in the laboratory. This GSHP system was fabricated using common commercially available equipment. Using such equipment helps to provide at a low price for the market. The COP was estimated by using the refrigerant enthalpy method with pressure and temperature fitting. Furthermore, we evaluated the COP by using the calorimetry method that measures the heat transfer rate of the chilled water in the evaporator and the cooling water in the condenser. As a result, while the estimated COP using the refrigerant enthalpy method decreases with increasing mass flow rate of chilled water in the evaporator, the estimated COP using the calorimetry method showed a peak at 4 kg/min.

Key words: Ground source heat pump system, Coefficient Of Performance, Refrigerant enthalpy method, Calorimetry method

1. Introduction
In GSHP systems, buried pipes are used to extract heat from the ground and vertical boreholes are used as a ground heat exchangers that couple the heat pumps to the ground (Katsura et al., 2008) (Esen et al., 2007). GSHP systems are now widely applied in the commercial sector and buildings in Japan. It is important to design living environments which use renewable energy, especially after the 2011 earthquake off the Pacific coast of Tohoku damaged nuclear power plants and resulted in decreased power-supply. Lots of GSHP systems have already been developed and commercialized in Europe and other countries, so it must be possible to reduce the initial cost of GSHP systems. However, the high initial cost prohibits the progress of GSHP systems in Japan: the cost for excavation is from 10,000 to 15,000 Yen/m. For example, a GSHP system using a borehole of 100 m costs more than 1,000,000 Yen, and takes more than 30 years to recover the initial cost. In general, the market price of the GSHP systems is 100,000 Yen/kW, and therefore it is important to reduce the price in order to spread the use of the system.

In this study, we investigated the COP of a fabricated GSHP system by experiment in the laboratory. This GSHP system was fabricated using common commercially-available equipment. Using such equipment helps to provide at a low market-price. The COP was estimated by using the refrigerant enthalpy method with pressure and temperature fitting. Furthermore, we evaluated the COP by measuring the heat transfer rate of the chilled water in the evaporator and the cooling water in the condenser.

Figure 1 shows the fabricated GSHP system of the test model using common equipment, permitting a low price point for the market. To evaluate this system, we estimated the COP of the fabricated GSHP system by using the refrigerant enthalpy method and the calorimetry method with the heat transfer rate of the chilled water in evaporator and the cooling water in condenser in the environment.
2. Test model of Ground Source Heat Pump system

2.1 Equipment

Figure 2 shows a schematic of the GSHP system. It consists of the compressor, the condenser, the expander and the evaporator. Several operation modes can be realized by switching the flow direction of the refrigerants, however this study only used cooling operation mode. The components of GSHP system are the compressor, the condenser, the expansion and the evaporator. Although the condenser and the evaporator are in reality put underground, the system is kept above ground in this study to control the temperature using a constant temperature reservoir. Measurement positions of temperature and pressure are indicated by 1, 2, and 3. We modified the part of the heat exchanger to use common equipment, and thus the control section is identical with that of typical air conditioners. The GSHP system was targeted to have a cooling capacity of 4 kW.

2.2 Experimental

Table 1 shows the experimental conditions. The mass flow rates of the chilled water in evaporator are 3, 4, and 5 kg/min, and that of the cooling water in the condenser is 8 kg/min. The average temperature of the chilled water in evaporator and the cooling water in the condenser are approximately 27 ℃ and 35 ℃ respectively. As a first step in this study, we evaluated the test condition of cooling capacity based upon Japan Industrial Standards (JIS B 8615).

3. COP

3.1 Enthalpy method

The COP of GSHP is estimated by the enthalpy method the following equation.

\[
\text{COP} = \frac{\dot{m}_m(h_1 - h_3)}{\dot{m}_m(h_2 - h_1)} = \frac{h_1 - h_3}{h_2 - h_1}
\]  

(1)
where, \( \dot{m}_R \) is the mass flow rate [kg/s] of refrigerant, \( h_1 \) is the specific enthalpy [kJ/kg] at the suction port of the compressor (position 1 in Figure 2), \( h_2 \) is the specific enthalpy [kJ/kg] at the discharge port of the compressor (position 2 in Figure 2), \( h_3 \) is the specific enthalpy [kJ/kg] at front of the expansion valve (position 3 in Figure 2). The numerator and denominator of eq. (1) is the cooling capacity of the evaporator and the input power to the compressor.

We used the temperature and the pressure of each point to estimate that specific enthalpy of R410A by REFPROP (NIST, 2013).

![Fig. 2 Schematic of the GSHP system. Dashed line box is the pseudo environmental side. T, G, and FM show the measurement positions of the temperature, the pressure, and the flow rate of refrigerant, respectively. Labels 1, 2, 3, i, o, COND, and EVA represent positions 1, 2, and 3, the inlet water, the outlet water, the COND, and the EVA, respectively. COMP, COND, EXP, and EVA show the compressor, the condenser, the expansion valve, and the evaporator, respectively.](image)

<table>
<thead>
<tr>
<th>No.</th>
<th>Mass flow rate of water [kg/min]</th>
<th>Average temperature of water [^\circ\text{C}]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Evaporator</td>
<td>Condenser</td>
</tr>
<tr>
<td>1</td>
<td>5.0</td>
<td>8.0</td>
</tr>
<tr>
<td>2</td>
<td>4.0</td>
<td>8.0</td>
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<tr>
<td>3</td>
<td>3.0</td>
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</tbody>
</table>

### 3.2 Calorimetry method

The COP of GSHP is estimated by the heat transfer rate of the pseudo environmental evaporator and condenser with the following equation.
\[ \text{COP} = \frac{Q_{EVA}}{Q_{COND} - Q_{EVA}} \]  

(2)

where,

\[ Q_{EVA} = \dot{m}_{EVA} c_{pEVA} \Delta T_{EVA} \]  

(3)

\[ Q_{COND} = \dot{m}_{COND} c_{pCOND} \Delta T_{COND} \]  

(4)

where, \( Q_{EVA} \) and \( Q_{COND} \) are the heat transfer rate [kW] of the evaporator and condenser respectively under pseudo environmental conditions. \( \dot{m}_{EVA} \) and \( \dot{m}_{COND} \) is the mass flow rate [kg/s] of the chilled water in the evaporator and the cooling water in the condenser, \( c_{pEVA} \) and \( c_{pCOND} \) is the specific heat \([\text{kJ/(kg·K)}]\) of water (JSME, 1999) calculated using the average temperature of the inlet and outlet ([\( T_i + T_o \)/2], where \( T_i \) and \( T_o \) are inlet and outlet temperature of the evaporator and condenser respectively) and \( \Delta T_{EVA} \) and \( \Delta T_{COND} \) [K] is the temperature difference of an inlet and an outlet of the chilled water in the evaporator and the cooling water in the condenser.

4. Results and discussion

Figure 3 shows the time-series data of the temperature and the pressure with experiment No. 1, 2, and 3, respectively. No. 1 (Figure 3 (a)), 2 (Figure 3 (b)), and 3 (Figure 3 (c)) data each display stable temperature and pressure after 1000 s, 3000 s, and 3000 s, respectively. Therefore, we calculated COPs and the heat transfer rate of the chilled water in the evaporator and the cooling water in the condenser under pseudo-environmental conditions using this stable average data.

Figure 4 shows the time-series data of the temperature and mass flow rate of the chilled water in the evaporator and the cooling water in the condenser with experiment No. 1, 2, and 3, respectively, along with the mass flow rate of the refrigerant. Figure 5 shows the time-series data of the heat transfer rate of the chilled water in the evaporator and the cooling water in the condenser with experiment No. 1, 2, and 3, respectively. In figure 3 (b) and (c), figure 4 (b) and (c), it can be confirmed that the refrigerant mass flow rate changed at approximately 1500 s. The causes were thought to be unstable temperature and pressure at this time.

Figure 6 details the COP of GSHP obtained by the refrigerant enthalpy method and calorimetry method, calculated with the heat transfer rate of the mass flow rate of the chilled water in the evaporator and the mass flow rate of the cooling water in the condenser, versus the mass flow rate of the chilled water in the evaporator. First of all, we focus on the COP by the refrigerant enthalpy method. These values decrease with increasing the mass flow rate of the chilled water in the evaporator, and the maximum COP is approximately 10; a huge value. We considered that this was due to measurement error of the refrigerant temperature in the two-phase gas-liquid area (Position 1).

Next, we focus on the COP by heat transfer rate of the chilled water in the evaporator and the cooling water in the condenser under environmental conditions. The maximum average COP is 7.1 with the mass flow rate of the chilled water in the evaporator of 4 kg/min. The increasing COP is commonly caused by increasing heat transfer rate of the chilled water in the evaporator. The optimum mass flow rate of the heat exchanger with the chilled water in the evaporator is approximately 4 kg/min.
Fig. 3 Time-series data of the temperature and the pressure. (a): Experiment No. 1, (b): Experiment No. 2, (c): Experiment No. 3. The Position 1, 2, and 3 are shown a compressor inlet, a compressor outlet, and an expander inlet.
Fig. 4  Time-series data of the temperature and mass flow rate of the chilled water in the evaporator and the cooling water in the condenser with experiment No. 1, 2, and 3, respectively.  (a): Experiment No. 1, (b): Experiment No. 2, (c): Experiment No. 3.
Fig. 5  Time-series data of the heat transfer rate of the chilled water in the evaporator and cooling water in the condenser with experiment No. 1, 2, and 3, respectively. (a): Experiment No. 1, (b): Experiment No. 2, (c): Experiment No. 3.
5. Conclusion

We fabricated the ground source heat pump system by using common equipment, and investigated the COP of that system. The COP was estimated by using the refrigerant enthalpy method with pressure and temperature fitting, and by the calorimetry method with measurement of the heat transfer rate of the chilled water in the evaporator and cooling water in the condenser. As a result, the average COP of experiment No. 1, 2, and 3 by the refrigerant enthalpy method decreased with an increasing mass flow rate of the chilled water in the evaporator. However, these COPs were estimated to have a huge value, because it was difficult to measure the temperature at each point of the refrigerant. On the other hand, the maximum COP estimated by the calorimetry method showed a mass flow rate of the chilled water in the evaporator of 4 kg/min.

References

Journal of the Japan Society of Mechanical Engineers, Steam Tables, JSME, (1999), p. 120.

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