Study on hybrid ground-coupled heat pump systems

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1. Introduction

Nowadays the ground-coupled heat pump systems (GCHP) are becoming an attractive choice for air conditioning in institutional and commercial buildings due to their advantages of high-energy efficiency and environmental friendliness compared with the conventional alternatives. However, a lot of commercial buildings are cooling-dominated, especially those located in hot-climate regions. Their annual accumulated cooling load exceeds their heating load. If the GCHP systems are installed in cooling-dominated buildings, much more heat is rejected into the ground than that extracted from the ground, which causes heat accumulation in the ground. In summer, this heat accumulation results in increase of ground temperature and then higher temperature of water entering the heat pump and degradation of the GCHP system performance accordingly. This problem may be solved by increasing either the total capacity of the ground heat exchanger (GHE) or the spacing between boreholes, but these methods will increase the initial cost of the whole system significantly. In addition, extra land area requirement is usually not available for many buildings located in urban areas.

A solution to the problem is to use the hybrid GCHP system (HGCHP), which adds supplemental heat rejecters in traditional GCHP systems, such as cooling towers (CT) or surface heat rejecters to interconnect the heat pump and the GHE. A supplemental heat rejecter is sized according to the difference between the cooling and heating loads of the building, and the annual extra heat can then be rejected through the supplemental heat rejecters. The GHE of a HGCHP system can be significantly diminished due to the existence of the supplemental heat rejecter, and the system initial cost can be reduced considerably.

Actually, the amount of heat transferred into and from the ground through the GHE varies with the instant change of building load. This will lead to the fluctuations in the temperature of water entering the GHE and in the coefficient of performance (COP) of the heat pump accordingly so that the hourly energy consumptions of the whole system change according to their corresponding hourly building load. On the other hand, the impact of various control strategies on the performance of the whole HGCHP systems is also complex. Therefore, an hour-by-hour simulation model that can give hourly operation data of the HGCHP system according to the building load. The design methods and running control strategies of the HGCHP system for a sample building are investigated. The simulation results show that proper HGCHP system can effectively reduce both the initial cost and the operating cost of an air-conditioning system compared with the traditional GCHP system used in cooling-dominated buildings.
proposed a design procedure of the HGCHP system for cooling-dominated buildings and offered a series of general guidelines to integrate supplemental heat rejecters into the HGCHP systems. Gilbreath [2] conducted a more detailed study based on a HGSHP system installed in an office building and presented some design suggestions.

The impact of the supplemental heat rejecter on the HGCHP system’s operation was also investigated by Kavanaugh and Rafferty [3], who suggested that a supplemental heat rejecter should be sized according to the peak block loads at design conditions and its nominal capacity was calculated according to the difference between cooling and heating loads. Kavanaugh [4] modified the existing design procedures recommended by the ASHRAE Manual [1] as well as by Kavanaugh and Rafferty [3], and a method to balance the heat transfer in the ground on an annual basis was proposed. The author stated that the economic value of the hybrid systems is apparent in warm and hot climates. Phetteplace and Sullivan [5] described the operating performance of a HGCHP system with 70 vertical closed-loop boreholes and a 275 kW CT using the measured data over a 22-month period. Singh and Foster [6] investigated the initial cost savings of the HGCHP system designed for two sample buildings. Yavuzturk and Spitler [7] presented a system simulation approach to compare the advantages and disadvantages of various control strategies for the operation of the HGCHP systems.

As elucidated above, since many hourly factors affect the design and performance of the HGCHP system, it is necessary to use an hour-by-hour simulation model to design the optimized HGCHP system for a given building under given weather conditions. This paper aims to develop a practical hourly simulation model for the HGCHP system by modeling the heat transfer process of its main components. This computer program is used to calculate the operating data of the HGCHP system according to hour-by-hour cooling or heating load of the building. Different design methods and control strategies of the HGCHP system are also investigated. Both the HGCHP system and the GCHP system are designed for a sample building in eastern China. The initial costs and hourly energy consumptions of the GCHP system and the HGCHP system operated under four different control strategies are calculated.

2. Heating and cooling loads of the sample building

The sample building is an office estate located in eastern China where the weather is hot in summer and cold in winter with extreme temperatures of 37 and \(-10\) °C, respectively. The total floor area of this building is 4850 m\(^2\). The typical meteorological year weather data is used and the building loads are calculated on an hourly basis for 1 year by the DEST, which is a convenient and reliable tool to calculate the air-conditioning load of buildings located in various climate regions. According to the calculation results shown in Fig. 1, this sample building is a cooling-dominated one and the ratio between its annual accumulated cooling load and heating load is 1.59. This means the total amount of heat to be rejected into the ambient environment from the building is larger than that to be extracted from the ambient environment into the building in order to ensure required indoor temperature.

![Fig. 1. Hourly heating and cooling loads of the sample building.](image1)

![Fig. 2. Schematic diagram of the HGCHP system.](image2)
3. The HGCHP systems

As shown in Fig. 2, the HGCHP system is mainly consist of a traditional heat pump, a ground heat exchanger (GHE) and a supplemental heat rejecter (a plate heat exchanger and a cooling tower). The difference between the HGCHP system and a GCHP system is that, the HGCHP system integrates the GCHP system with a supplemental heat rejecter.

4. Model of the supplemental heat rejecter

The supplemental heat rejecter is an open-circuit CT cooperated with a plate heat exchanger. A schematic diagram of the supplemental heat rejecter is shown in Fig. 3.

Many studies have presented the heat and mass transfer analysis of the CT. Early researchers include Lewis [8] in the 20th century, and then Robinson [9] who reported a series of fundamental concepts and established the general principles for designing the CT. Merkel [10] developed a practical model for simulating the CT's performance based on an enthalpy difference theory. Baker and Mart [11] utilized a complicated unit-volume iteration procedure to account for the temperature and enthalpy change during the running of the CT. Baker and Shryock [12] analyzed the heat transfer of the CT based on the energy equilibrium.

These models concentrate on analyzing the complicated heat and mass transfer processes in the CT, and have provided theoretical basis for simulating the HGCHP systems which utilizing the CT as their supplemental heat rejecter. Some of them, however, are too intricate to be employed in simulating the performance of an even more comprehensive HGCHP system for long-term operation on hourly intervals. Our aim is to simulate the effective cooling capacity and the effusing water temperature of the CT on an hour-by-hour basis. Thus, a compromise must be taken between the model precision and simulation efficiency. A concise model based on Merkel's enthalpy theory and the Effectiveness-NTU method is used in this study. This model predigests the running of CT into a steady-state one and it can be used to simulate the hourly operating data of CT by iterative method according to the mass flow rate of air and water, the wet-bulb temperature of ambient air and the temperature of inlet water. The precision of this model has been validated by comparing with manufacturers' data. The cooling capacity and effectiveness curves of the CT are shown in Fig. 4.

5. Model of the heat pump

The performance of the heat pump affects the efficiency and energy consumption of the whole HGCHP system. Therefore, a heat pump simulation model is needed to calculate the COP and effusing fluid temperature of the heat pump according to its entering fluid temperature on an hour-by-hour basis, and then the hourly energy consumption of the heat pump can be analyzed.

In the literature, there are a number of theoretical and experimental works on the performance of heat pumps. Blanchard [13] first analyzed the heat pump cycles and utilized the COP to study a Newton’s law Carnot heat pump. Goth and Feidt [14], Philippi and Feidt [15] deduced the optimal COP for the fixed load of a Newton's law Carnot heat-pump. Wu et al. [16] developed an optimized relationship between the specific load and COP of the heat pump. These researches focus on inner cycles of the heat pump, and they are too complex to be utilized in present study to simulate the hourly operation of the whole HGCHP system. In order to simulate the operation performance of a heat pump hour-by-hour, it is more feasible to fit the functions of its COP and effusing fluid temperature versus entering fluid temperature according to the heat pump manufacturer’s data.

The capacity of the heat pump is determined according to the peak cooling load of this cooling-dominated sample building, and
then the entering fluid temperature of the heat pump, its corresponding COP and the effusing fluid temperature can be determined from manufacturer specifications. The least square method is used and the 2nd power polynomial curves are fit to the data. The COP and effusing fluid temperature curves versus entering fluid temperature of the heat pump are given in Fig. 5. The COP of the heat pump varies obviously with the temperature of fluid entering the heat pump, especially when the heat pump operates in cooling circulation. Thus, one valid method for increasing the COP and decreasing the energy consumption of the heat pump is to provide the entering fluid in moderate temperature, and this is one of the targets in this study.

6.1. Heat transfer outside boreholes

The transient heat transfer around boreholes of the GHE is analyzed here by a 2D model, which assumes the ground as a homogeneous semi-infinite medium with a uniform initial temperature, and assumes that the borehole as a line-source with finite length releases heat at a constant rate per length. The analytical solution of the non-dimensional excess temperature of the borehole wall has been obtained by Zeng et al. [17].

The GHE is used for extracting or injecting heat from or into the ground. It consists of sealed loop U-tube pipes buried in boreholes, which are connected to the heat pump through circulating water pipelines. Heat is transferred between the heat pump and the ground by circulation water through the heat exchange in the GHE. It is crucial to calculate the heat transfer of the GHE in order to simulate the operation of the whole HGCHP system. Due to its complications and long-term effect, the heat transfer process is usually separated into two parts: the process in solid soil or rock outside the borehole, where the heat transfer is treated as a transient process; and the process inside the borehole, where the heat transfer is considered as steady process.

6. Model of heat transfer in the GHE

The GHE is used for extracting or injecting heat from or into the ground. It consists of sealed loop U-tube pipes buried in boreholes, which are connected to the heat pump through circulating water pipelines. Heat is transferred between the heat pump and the ground by circulation water through the heat exchange in the GHE. It is crucial to calculate the heat transfer of the GHE in order to simulate the operation of the whole HGCHP system. Due to its complications and long-term effect, the heat transfer process is usually separated into two parts: the process in solid soil or rock outside the borehole, where the heat transfer is treated as a transient process; and the process inside the borehole, where the heat transfer is considered as steady process.

6.1. Heat transfer outside boreholes

The transient heat transfer around boreholes of the GHE is analyzed here by a 2D model, which assumes the ground as a homogeneous semi-infinite medium with a uniform initial temperature, and assumes that the borehole as a line-source with finite length releases heat at a constant rate per length. The analytical solution of the non-dimensional excess temperature of the borehole wall has been obtained by Zeng et al. [17].

Usually, the GHE contains more than one borehole, and each borehole has its own borehole wall excess temperature according to its position in the GHE (the borehole located in the centre of GHE has the higher excess temperature compare with the borehole located in the verge of GHE). In order to ensure the effective heat transfer capacity of the GHE, the largest borehole wall excess temperature is employed to represent the excess temperature of the whole GHE in this study:

\[
\theta_e = \frac{q_i}{4k\pi \int_0^1 \left( \text{erfc} \left( \frac{\sqrt{(r_b/H)^2 + (0.5 - H)^2}}{2\sqrt{\pi t/H^2}} \right) - \text{erfc} \left( \frac{\sqrt{(r_b/H)^2 + (0.5 + H)^2}}{2\sqrt{\pi t/H^2}} \right) \right) dH + \sum_{j=1, j \neq i}^N \text{erfc} \left( \frac{\sqrt{(r_j/H)^2 + (0.5 - H)^2}}{2\sqrt{\pi t/H^2}} \right) - \sum_{j=1, j \neq i}^N \text{erfc} \left( \frac{\sqrt{(r_j/H)^2 + (0.5 + H)^2}}{2\sqrt{\pi t/H^2}} \right) \right)
\]

where \(r_b\) denotes the radius of the borehole, \(H\) denotes the depth of the borehole, \(r_j\) denotes the distance from other boreholes to the concerned borehole.

In order to integrate the analytical solution into our computer program, we use the \(f\)-function in this study to represent the non-dimensional temperature response of GHE. The \(f\)-function is defined as

\[
f(\tau - \tau_{i-1}) = g(\tau - \tau_{i-1}) - g(\tau - \tau_i) \quad \text{or in another form,}
\]

\[
f(\tau - (i - 1) \Delta \tau) = g(\tau - (i - 1) \Delta \tau) - g(\tau - i \Delta \tau)
\]

where \(g(\tau) = 2\pi k\theta_e q_i\), and \(q_i\) denotes the heat releases rate of the line-source per length, \(\theta_e\) can be calculated by Eq. (1). According to its definition, the \(f\)-function of a certain GHE only varies with two parameters: the time \(\tau\) and the time interval \(\Delta \tau\). When the time interval \(\Delta \tau\) is set as 1 month, the hourly \(f\)-function curves of the GHE with boreholes of 80 m depth, 5 m apart and arranged in \(3 \times 10\) as well as \(5 \times 10\) configurations are shown in Fig. 6. Both of these two types of the GHEs are employed in present study, and the \(m \times n\) denotes the boreholes configuration, i.e. \(m\) lines and \(n\) rows.

Since the \(f\)-function represents the non-dimensional temperature response of the GHE to the pulse heat input, the temperature of borehole wall can be calculated according to the hourly cooling load. The temperature response of the borehole wall can be
deduced using the superimposing theory:

\[ \theta = \frac{1}{2\pi k} \sum_{i=1}^{\infty} q_i f(t - \tau_{i-1}) \]  

(3)

6.2. Heat transfer inside boreholes

Taking the fluid axial convective heat transfer and thermal “short-circuiting” among U-tube legs into account, a quasi-3D model for simulating the heat transfer inside boreholes of GHE has been established, and its analytical solutions of the fluid temperature profiles along the borehole depth have been obtained (Zeng et al. [18]).

The temperature of water entering and effusing the GHE can be derived by the heat transfer resistance:

Temperature of fluid entering the GHE:

\[ T_i = T_b + \frac{q_i H}{MC_p} \left( \frac{1}{1 - \Theta^*} \right) \]  

(4)

Temperature of fluid effusing the GHE:

\[ T_i'' = T_b + \frac{q_i H}{MC_p} \left( \frac{\Theta^*}{1 - \Theta^*} \right) \]  

(5)

where \( T_b \) denotes the temperature of borehole wall, which can be calculated by model of heat transfer outside boreholes; \( \Theta^* \) denotes the non-dimensional effusing fluid temperature.

7. Different control strategies of the HGCHP system

The control strategies are selected according to the local climate characteristics and can be categorized into three groups: (1) the set point control strategy, i.e. to activate the CT when the temperatures of water entering the heat pump exceeds a set value; (2) the temperature difference control strategy, i.e. to activate the CT when the temperature difference between the water entering the heat pump and the ambient air exceeds a set value; (3) the running time control strategy, i.e. to activate the CT during the set time. Four concrete control strategies employed in the simulations include:

1. The set point control strategy: to activate the CT when the temperature of the water entering the heat pump exceeds 32°C.
2. The temperature difference control strategy: to activate the CT when the temperature of the water entering the heat pump is higher than the dry-bulb temperature of ambient air for 3°C in cooling circulation. In addition, the CT is activated when the temperature of the water entering the heat pump exceeds 32°C.
3. The temperature difference control strategy: to activate the CT when the temperature of the water entering the heat pump is higher than the wet-bulb temperature of ambient air for 3°C in cooling circulation. In addition, the CT is activated when the temperature of the water entering the heat pump exceeds 32°C.
4. The running time control strategy: to activate the CT from 00:00 a.m. to 6:00 a.m. in summer. In addition, the CT is activated when the temperature of the water entering the heat pump exceeds 32°C.

8. Results and discussions

Two systems’ performance, i.e. the common GCHP system and the HGCHP system for the sample cooling-dominated building are calculated using the developed program based on the hourly simulation model. The initial ground temperature of this study is 14.7°C. The common GCHP system designed for the sample
As shown in Fig. 7, the borehole wall temperatures of the common GCHP system increases after 1 year’s operation, and the increase becomes more obvious after long time operation. This is because the traditional GCHP system cannot balance the heat rejected into and extracted from the ground, and the excess heat accumulates in the ground year by year. Since the temperature of the fluid entering the heat pump increases with the temperature rise of the borehole wall, the COP of the heat pump will decrease and its energy consumption is getting higher, as shown in Fig. 8.

According to the simulation results, the traditional GCHP system experiences obvious performance degradation over time. Increasing the total length of the GHE or the distance between adjacent boreholes is one of the solutions, but it will also result in higher system initial cost and the GCHP system may become economically unattractive. In order to overcome the drawbacks mentioned above, the HGCHP system is proposed for this sample cooling-dominated building.

A cooling tower is utilized as supplemental heat rejecter of the HGCHP system. At the beginning of the design, proper length of its GHE and the capacity of its cooling tower must be determined. The criterion for designing the GHE and cooling tower is that the peak entering fluid temperature (EFT) to heat pump unit from the GHE must be lower than the required peak value, which is chosen according to the heat pump unit utilized in the system and this value is 32°C in this study. Another criterion for the HGCHP system
Two ways to design a HGCHP system are investigated. The first one is to satisfy the annual heating load of the cooling-dominated building just by the GHE and to satisfy its annual cooling load by the GHE cooperated with the CT. The total length of the GHE can be chosen based on the annual heating load and the capacity of the CT can be sized according to the difference between cooling and heating load. Another way is to lengthen the GHE utilized in the first way and employs it to handle some unbalanced cooling load besides the heating load, so that the capacity of the CT can be reduced compared with the first way. In this study, two HGCHP systems with boreholes of 3 × 10 configurations and 4 × 10 configurations are designed based on these two design ways, respectively. The hourly operating data of these two different systems operated under four control strategies mentioned above are simulated and compared as shown in Fig. 9. The comparison of the total cost between these two HGCHP systems is shown in Table 1.

The HGCHP system designed by the first way has better performance in terms of ensuring the EFT under its peak value and decreasing the system total cost for the four investigated control strategies.

After the HGCHP system is properly designed, the complex effect of various control strategies on the system performance also needs to be analyzed in detail. In order to determine which control strategy is better, the hourly operating data of the HGCHP systems designed by the first way and operated under four investigated control strategies are simulated and compared. The hourly temperature responses of the borehole wall and the fluid effusing the GHE are shown in Fig. 10.

According to the average price of local energy and equipments, the initial and operating costs of the traditional GCHP system and the HGCHP system operated under four control strategies are simulated by our computer program. The calculation results are compared in Table 2.

The comparison shows that, the HGCHP system running under different control strategies has obviously different operating costs, and the lowest operating cost can be obtained when the HGCHP system running under control strategy 3, e.g. the CT is controlled based on the difference between the temperature of water entering the heat pump and the wet-bulb temperature of ambient air. This

### Table 1
Comparison of total cost between two different HGCHP systems

<table>
<thead>
<tr>
<th>HGCHP system</th>
<th>Configuration of GHE</th>
<th>Initial cost ($)</th>
<th>Operating cost ($)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Control strategy 1</td>
</tr>
<tr>
<td>Type 1</td>
<td>3 × 10</td>
<td>63928.6</td>
<td>3367.6</td>
</tr>
<tr>
<td>Type 2</td>
<td>4 × 10</td>
<td>75183.7</td>
<td>4345.5</td>
</tr>
</tbody>
</table>

### Table 2
Cost comparisons between GCHP system and HGCHP system operated under four control strategies

<table>
<thead>
<tr>
<th>Item</th>
<th>HGCHP system</th>
<th>GCHP system</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Control strategy 1</td>
<td>Control strategy 2</td>
</tr>
<tr>
<td>Annual operating time of CT (h)</td>
<td>290</td>
<td>555</td>
</tr>
<tr>
<td>Highest temperature of water entering the heat pump (°C)</td>
<td>31.77</td>
<td>31.6</td>
</tr>
<tr>
<td>Annual energy consumption (kWh)</td>
<td>106,794</td>
<td>101,385</td>
</tr>
<tr>
<td>Annual operating cost ($)</td>
<td>6,675</td>
<td>6,337</td>
</tr>
<tr>
<td>Borehole number of GHE</td>
<td>30</td>
<td>50</td>
</tr>
<tr>
<td>Initial cost of GHE ($)</td>
<td>30,000</td>
<td>50,000</td>
</tr>
<tr>
<td>Initial cost of CT ($)</td>
<td>312.5</td>
<td>312.5</td>
</tr>
<tr>
<td>Initial cost of heat pump and circulation pump ($)</td>
<td>25,625</td>
<td>25,625</td>
</tr>
<tr>
<td>System initial cost ($)</td>
<td>55,938</td>
<td>75,625</td>
</tr>
<tr>
<td>Sum of Initial cost and 10 years operating cost ($)</td>
<td>122,684</td>
<td>119,303</td>
</tr>
</tbody>
</table>
strategy activating the CT under the most advantageous weather condition can make full use of ambient air to cool the cooling water and reject the imbalanced heat effectively so that higher COP of the heat pump can be obtained and the degradation of system performance can be avoided in the long term running.

For the HGCHP system operated under control strategy 1 and 2, the imbalanced heat cannot be rejected into the ambient air effectively due to the poor weather condition. For the HGCHP system operated under control strategy 4, the CT activated only at summer night cannot reject the imbalance heat in time so that the temperature of the borehole wall in cooling mode will increase significantly and the annual COP of the heat pump is lower when utilizing the control strategies 1, 2 and 4. If the HGCHP system is operated using the control strategy 3, the temperature of the borehole wall is moderate in cooling circulation, and the COP of the heat pump remains high in long term running. In conclusion, the HGCHP system can obtain the optimal performance when operated under control strategy 3.

As shown in Table 2, the HGCHP system is designed in terms of sizing the length of the GHE according to the heating load, and determine the capacity of CT based on the difference between cooling and heating load, operated under the control strategy 3, which activates the CT when the temperatures of water entering heat pump is higher than the wet-bulb temperature of ambient air for 3°C in cooling circulation, is the most economical choice. Comparisons between this optimal HGCHP system and the traditional GCHP system denote that the optimal HGCHP system can reduce 35% initial cost and 20.79% operating cost in the first year. Due to the performance degradation of the GCHP system year by year, the economical benefit of the HGCHP system is more obvious in long term running. The optimal HGCHP system can save 44.69% operating cost and 40.05% total cost compared with the common GCHP system in 10 years operating. The comparison between HGCHP system and GCHP system on the temperature of borehole wall, on the power of heat pump and on the COP of system in 10 years operating are shown in Figs. 11–13, respectively.

As shown in Figs. 11–13, the lower ground temperature and therefore the higher system performance compared with the common GCHP system can be obtained by the HGCHP system. As expected, for this cooling-dominated building, the borehole wall temperature of the GCHP system will increase and the performance of the whole system will experience obvious degradation over time. By contrary, the temperature of the ground around GHE in the HGCHP system keeps the same as the initial value, and accordingly, the COP of the heat pump remains high after 10 years.
operation. There are some rapid fluctuations of COP at the beginning and at end of each half year due to the heat pump changes its operation mode from heating circulation to cooling circulation. The simulation results indicate that, the HGCHP system can avoid the increase of borehole wall’s temperature and the degradation of the heat pump’s performance by offsetting the annual imbalance load of the cooling-dominated building. Therefore, a properly designed HGCHP system can achieve good performance of air-conditioning system in cooling-dominated buildings.

9. Conclusions

This paper reports a practical hourly simulation model of the HGCHP system by analyzing and modeling the heat transfer process of its main components on an hour-by-hour basis. The hourly operation data of the HGCHP system are calculated by our computer program based on the hourly simulation model. The impacts of four different control strategies on performances of two different HGCHP systems designed for a sample cooling-dominanted building are compared, and an analysis of system investment considering the initial and operating cost is conducted based on the hourly calculation results.

According to the design models and control strategies of the HGCHP system investigated in present study, the optimal HGCHP system design model which sizing the length of the GHE based on annual heating load and determining the capacity of the cooling tower based on difference between cooling and heating loads can obtain the significantly lower initial cost and operating cost compare with other design model. And the best control strategy for this optimal HGCHP system is the temperature difference control strategy which activating the cooling tower whenever the temperatures of water entering the heat pump is higher than the wet-bulb temperature of ambient air for 3 °C in cooling circulation.

By applying the supplemental heat rejecters to offset the annual imbalanced load of cooling-dominated buildings, the properly designed HGCHP system can efficiently balance the heat reject into and extract from the ground, accordingly, it can keep the stable excellent system performance and has benefits in reducing the initial and operating costs compare with the common GCHP system.

Based on the hourly simulation results, the optimal HGCHP system for the sample cooling-dominated building (the ratio between its annual cooling load and heating load is 1.59) can save 35% initial cost and 20.79% operating cost in first year running compare with the common GCHP system. Due to the performance degradation of GCHP system year by year, the economical benefit of HGCHP system is more obviously in long-term running, and the HGCHP system can save 44.69% operating cost as well as 40.05% total cost compare with the common GCHP system in 10 years operating. The analyses based on hourly simulation results indicate that, the HGCHP system utilized in buildings whose cooling and heating load ratio is larger, such as buildings located in hot climate regions, has more attractive benefits in saving the system initial and operating costs.

Acknowledgement

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References

[2] C.S. Gilbreath, Hybrid ground-source heat pump systems for commercial applica-