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Keywords
study, system, heating, integrated, thermal, energy, storage, space, evaluation, performance, co2, heat, pump

Disciplines
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Study on performance evaluation of CO₂ heat pump system integrated with thermal energy storage for space heating

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Abstract

CO₂ heat pumps have drawn a great deal of attention owing to their advantages of high efficiency and environmental friendly for heating water under low ambient temperature. However, the system performance is not desirable and shows a lower COP for space heating, especially for a radiator as heating terminal, due to the higher inlet water temperature at the gas cooler, which causes a large throttle loss when the refrigerant flow through the throttling device. To tackle this issue, a transcritical CO₂ heat pump system integrated with thermal energy storage (TES) systems was developed in this paper. The heating performance of the proposed system was investigated using TRNSYS 17.0 based on a typical single family rural house located in Beijing, China. The results showed that when only weather change is considered, the margin of error could be acceptable for some applications.

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Keywords: CO₂ heat pump; TRNSYS; Simulation; Space heating; HSPF;

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1. Introduction

In recent years, heat pump has been widely used for space heating because of their advantage of high efficiency, energy-saving and green environmental protection [1][2][3]. Especially in China, in order to track the issues of environmental pollution and energy crisis, the Chinese government has adopted a series of fiscal stimulus packages for promoting the use of the renewable energy technologies. The most significant measure of them is to use ASHP unit replace the coal fired boiler for space heating in northern China. It greatly reduces the consumption of fossil energy resource and greenhouse gas emission [4][5]. However, most heating systems are taken use of radiator as terminal in rural region of northern China, the design of supply/return water temperature are 75/50°C according the Chinese Standard (Design code for heating ventilation and air conditioning of civil buildings, GB 50736-2012). The outlet water temperature of air source heat pumps (ASHPs) unit using conventional refrigerant as working fluid is only 41 °C at the nominal design condition (-12 dry-bulb temperature /-14 wet-bulb temperature) (Chinese Standard < Low ambient temperature air source heat pump (water chilling) packages- Part 2: Heat pump (water chilling) packages for household and similar application > GB/T 25127.2-2010), it does not meet the requirement of supply/return water temperature that leading to the indoor air temperature decrease. In order to improve the indoor thermal comfortable, it is usually to take measures of increasing the numbers of radiator. Yang et al. [6] presented an application of an ASHP replaced the coal fired boiler for heating system in rural region of Beijing, China. The results indicated that the area of the radiators was increased from 19.6 m² to 31.64 m² when the supply/return water temperature was decreased from 70/55°C to 45/45°C. It illustrated that the cost of investment and the difficult of reconstruction was increased when the coal fired boiler was replaced by the ASHP unit for space heating in rural region.

In order to improve this problem, the ASHP unit using CO2 as working fluid has been attracted much attention due to its outstanding features with high values of thermal conductivity and heat capacity and so on. It can reduce the irreversible energy loss that leads to a higher performance because of the refrigerant temperature glide at heat rejection in gas cooler contributes to a very good temperature adaptation for heating a finite stream of water[7], resulting in a fairly large temperature lift in water without significant penalization in coefficient of performance (COP). A host of publications in recent years have demonstrated that the refrigeration/heat pump system using CO2 as working fluid is very competitive with conventional equipment for hot water heating, air-conditioning etc. regard to power consumption, compactness and cost when it operated in the transcritical region[8][9]. However, the system performance of CO2 heat pump is not desirable and shows as lower COP when it is used for space heating, especially in the system with radiator terminal. It is because of the higher inlet water temperature at the gas cooler (i.e. the return water temperature is high) causes a large throttle loss when the refrigerant flowing through the throttle device.

To track this issue, a transcritical CO2 heat pump integrated with thermal energy storage (TES) is developed. Firstly, the schematic diagram of the proposed system is introduced. Secondly, descriptions of a simulation model using TRNSYS 17.0 are reported. This is followed by presenting simulation results and discussions on the system heating performance, and compared with the conventional ASHP system. Lastly, the representative conclusions are summarized.

2. System description

Fig. 1 shows the schematic diagram of a transcritical CO2 heat pump system integrated with TES. It mainly consists of a CO2 heat pump unit, TES (L represents the low melt point PCM; H represents the high melt point PCM) (10), a radiator (11), a water pump (12) and solenoid valves (8, 9, 13). The system works in two modes: (1) the heat charging mode (Fig. 1a) and (2) heat discharge mode (Fig. 1b), the detailed work process is presented as follows:

1. Heat charging mode. The solenoid valves (8, 9) are open while the solenoid valve (13) is close. Water is first heated by the CO2 ASHP unit, after the supply water flow is divided into two streams by a 3-way damper: one stream passes through the main pipe line and the rest is bypassed to the TES (10-H), where the two supply water streams are mixed before enters into the radiator (11) and releases the heat for space heating. The return water (the supply water after pass through radiator is regard as return water) passing through the water pump (12) and then flows through TES (10-L), the waste heat is absorbed by TES (10-L) and resulting in a further reduction in the return water temperature. Finally, the return water enters into the CO2 heat pump unit. The refrigerant temperature
before the throttle valve (5) is decreased due to the decrease in the temperature of the return water, leading to reduce the loss of throttling, resulting in an increase in the performance of the CO₂ ASHP unit.

Fig. 1 Schematic diagram of the transcritical CO₂ heat pump heating system integrated with TES.

(2) Heat discharging mode. During the night time, the performance of the CO₂ ASHP unit decreases because of the low ambient temperature. In this situation, to improve the system efficiency, the ASHP unit is turned off and the heating system should be switched to the TES heat discharge mode.

During the heat discharging mode, the ASHP unit is turned off, and the solenoid valve (13) is open while the solenoid valves (8, 9) are closed. Firstly, the return water from the radiator (11) enters into the TES (10-L) to absorb the stored heat, and it then flows through the TES (10-H) and the water temperature rises further that can meet the temperature requirement of supply water. Finally, the supply water flows through the radiator (11) and releases heat for space heating.

3. Model and boundary conditions

3.1. Building model

The building concerned in this study was a typical single family rural house located in Beijing, China. The total heated space area was 75 m². In order to meet the requirement of indoor temperature and thermal comfort, the indoor temperature was set 18±2°C, the house was used the model of Type 56. The thermal transmittance values of the main building elements are present in Table 1.

Table 1 Thermal transmittance values of main building elements

<table>
<thead>
<tr>
<th>NO.</th>
<th>Building envelope element</th>
<th>U-value (W/m²·K)</th>
<th>Structure</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Exterior wall</td>
<td>0.4</td>
<td>Concrete brick 370mm; polystyrene thermal insulating board 50 mm</td>
</tr>
<tr>
<td>2</td>
<td>Roof</td>
<td>0.8</td>
<td>Cement + adhesive power polystyrene 100 mm;</td>
</tr>
<tr>
<td>3</td>
<td>Ground floor</td>
<td>0.039</td>
<td>Concrete slab 100 mm</td>
</tr>
<tr>
<td>4</td>
<td>Windows (g=0.5)</td>
<td>2.8</td>
<td>Double lay plastic steel window;</td>
</tr>
</tbody>
</table>

3.2. CO₂ heat pump model

For the CO₂ heat pump performance modeling Type 941 was a performance map with the data of heating capacity and power consumption, which were the functions of the ambient temperature and inlet water temperature in the gas cooler, for a range of testing points, i.e. the heat output and COP of the heat pump were calculated in the model through interpolation between these points.

The technical characteristics of the CO₂ heat pump unit were provided by the manufacturer. The heating capacity (Eq. 1) and power consumption (Eq. 2) can be fitted as follows:

\[ Q_{\text{max}} = 19.02 + 0.402 T_a + 0.05 T_{\text{in}} + 0.00112 T_a^2 - \frac{0.00887 T_a}{T_{\text{in}}} + 0.0012 T_{\text{in}}^2 - 2.3 \times 10^{-5} T_a^3 - 3.06 \times 10^{-5} T_a^2 T_{\text{in}} + 8.59 \times 10^{-5} T_a T_{\text{in}}^2 - 9.56 \times 10^{-5} T_{\text{in}}^3 \]  

(1)
The main building elements are present in Table 1.

Indoor temperature was set 18±2°C, the house was used the model of Type 56. The thermal transmittance values of model through interpolation between these points.

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3.2. CO2 heat pump model

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</tr>
<tr>
<td>4 Windows (g=0.5)</td>
<td>2.8</td>
</tr>
<tr>
<td>5 Window putty</td>
<td>5.0</td>
</tr>
</tbody>
</table>

The technical characteristics of the CO₂ heat pump unit were provided by the manufacturer. The heating performance of the CO₂ heat pump was investigated using the FORTRAN program based on ASHP performance map. The model was validated using the experimental data which were presented literature [19], and the results showed that the simulated data were identical with the experimental data in trend, and the maximum relative error between the experimental data and simulated data was 7.2%.

3.3. TES model

TES is a promising technology for sub-cooling that reduce the throttle loss and improve the refrigeration system performance[15][16][17]. In the present study, a cylindrical tank (tube-in-tank arrangement) filled with PCMs as the TES was considered and the mathematical model was formulated using ε-NTU method based on Tay’s [18] research. The model was validated using the experimental date which were presented literature [19], and the results showed that the simulated data were identical with the experimental data in trend, and the maximum relative error between the experimental data and simulated data was 7.2%.

3.4. Operation modes and control strategies

Table 2 Main TRNSYS components and parameters setting.

<table>
<thead>
<tr>
<th>Name</th>
<th>Component type</th>
<th>Main parameters</th>
<th>Name</th>
<th>Component type</th>
<th>Main parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Building</td>
<td>Type 56</td>
<td>Room air exchange rate: 0.5 h⁻¹; Controlled temperature: 18±2°C; Design capacity 5 kW; Design surface temperature: 55°C.</td>
<td>T_air: 21°C; T_l: indoor real-time temperature</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Radiator</td>
<td>Type 1231</td>
<td>Room air exchange rate: 0.5 h⁻¹; Controlled temperature: 18±2°C; Design capacity 5 kW; Design surface temperature: 55°C.</td>
<td>Controller Type 2b</td>
<td>The upper dead bands: 5°C; The lower dead bands: 1°C;</td>
<td></td>
</tr>
<tr>
<td>ASHP</td>
<td>Type 941</td>
<td>Room air exchange rate: 0.5 h⁻¹; Controlled temperature: 18±2°C; Design capacity 5 kW; Design surface temperature: 55°C.</td>
<td>Weather data Type 15</td>
<td>Input Beijing typical meteorological years data;</td>
<td></td>
</tr>
<tr>
<td>Pump</td>
<td>Type 3d</td>
<td>Room air exchange rate: 0.5 h⁻¹; Controlled temperature: 18±2°C; Design capacity 5 kW; Design surface temperature: 55°C.</td>
<td>Weather data Type 15</td>
<td>Input Beijing typical meteorological years data;</td>
<td></td>
</tr>
<tr>
<td>Flow diverter</td>
<td>Type 11f</td>
<td>Room air exchange rate: 0.5 h⁻¹; Controlled temperature: 18±2°C; Design capacity 5 kW; Design surface temperature: 55°C.</td>
<td>Weather data Type 15</td>
<td>Input Beijing typical meteorological years data;</td>
<td></td>
</tr>
</tbody>
</table>

The model of the proposed system was established in TRNSYS 17.0, it was showed in Fig. 2. The majority component models used were the standard models which were provide in the TRNSYS library. The TES model, which was a nonstandard component in TRNSYS that was written in FORTRAN, according to the equations proposed by Tay et al. [18], and linked to TRNSYS to enable an integrated simulation of the house and its ASHP heating system as a whole.

The simulation period started from November 15th to March 15th of the next year. An on/off controller (Type 2b) was employed to model the operation. Due to the fact that the ambient temperature was relative low at night which led to a reduction in the system heating capacity and COP, in order to improve the heating system performance in over heating period, the CO₂ heat pump unit was operated for heating from 6:00 to 22:00, and the unit was turned off from 22:00 to the next day of 6:00 and heated by the TES. Table 2 shows the main TRNSYS components and parameters settings.

\[
P_{\text{max}} = 3.35 + 0.076 T_a + 0.12 T_m + 6.14 \times 10^{-4} T_a^2 - 0.00195 T_a T_m - 0.003 T_m^2 + 2.29 \times 10^{-6} T_a^3 - 1.57 \times 10^{-5} T_a T_m^2 + 1.9 \times 10^{-5} T_a T_m + 2.62 \times 10^{-5} T_m^3
\]
4. Results and discussion

Fig. 3 shows the comparison of the heating capacity between the proposed system and baseline system at different month during the heating period. It can be seen that the heating capacity of the both system was the highest in January. This can be explained that the heat load of the house was the highest since the ambient temperature was lower than the other month. Also, the heating capacity of the proposed system was lower than that of the baseline system. In January, the heating capacity of the proposed system was 2013 kW that was 22% lower than that of the baseline system which was 2581 kW. This was because the CO₂ heat pump unit was operated in day time and it was turned off at night time. The CO₂ heat pump had a high efficiency during day time as the ambient temperature was higher than night time. Furthermore, the energy for space heating at night was from TES. During the entire heating
period, the total heating capacity of the proposed system and baseline system were 7021 and 8868 kW respectively. It was 20.8% lower than the baseline system.

![Fig. 3 Comparison of the heating capacity between the proposed system and baseline system during the heating period](image)

Fig. 3 Comparison of the heating capacity between the proposed system and baseline system during the heating period

Fig. 4 shows the comparison of the energy consumption between the proposed system and baseline system during the heating period. From the figure, it can be found that the variation trend of energy consumption of the both system showed a similarity with the trend of the heating capacity during the heating capacity. The energy consumption was the highest in January and relative low in November and March in the next year. It can be explained by two reasons, the one was that the ambient temperature in November and March was higher than in January, that leading to a low heat load. The other reason was that the heating time was the half of in January. It can also be seen that the energy consumption of the proposed system was lower than the baseline system. For instance, the energy consumptions of the both system were 887 and 1195 kW in January, and were 2968 and 3892 kW in total during the heating period. They were 26 and 23.7% lower than the baseline system. The reason for this phenomenon was that the ambient temperature in daytime was much higher than night time, leading to a high heating efficiency. Furthermore, the energy for space hearing at night time was from TES, which reduced the time of frosting accumulated on the surface of the heat exchanger.

![Fig. 4 Comparison of the energy consumption between the proposed system and baseline system during the heating period](image)

Fig. 4 Comparison of the energy consumption between the proposed system and baseline system during the heating period.

![Fig. 5 Comparison of system COP and HSPF between the proposed system and baseline system during the heating period](image)

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The system COP and HSPF (Heating Seasonal Performance Factor) were two important evaluation coefficients.
for the heating performance of ASHP unit. The system COP is defined as a ratio of the heating capacity of the system to energy consumption including power consumption of compressor, water pump and blower in every month. For the entire heating period, the higher the HSPF rating of a unit, the more energy efficiency it is. The HSPF, defined as a ratio of heating over the heating season to power consumption used, is expressed as in Eq. (3).

\[
\text{HSPF} = \frac{\int_0^T Q_{\text{total}} \, dt}{\int_0^T P_{\text{total}} \, dt}
\]  

(3)

where, \(Q_{\text{total}}\) is the total heating capacity, [kWh]; \(P_{\text{total}}\) is the total energy consumption including heat pump (compressor + control + blower) and water pump, [kWh].

Fig. 5 gives the system COP of the proposed system and baseline system during the heating period. It can be seen in this figure, the system COP and HSPF of the proposed system were higher than that of the baseline system. For instance, the system COPs of the both system were 2.27 and 2.16 respectively, and the HSPFs were 2.37 and 2.28. They were 5.1 and 3.9% higher than the baseline system. It can be explained by the fact that the proposed system provided a higher hearing capacity than the baseline system. On the other hand, the proposed system was operated in day time that reduced the frost accumulated on the outdoor heat exchanger, resulted in a reduction in power energy for defrosting. The results showed that the proposed system provided an important in performance as compared to the baseline system in cold region.

5. Conclusions

In this paper, a CO\(_2\) heat pump system integrated with TES was developed, the system performance for space heating in a rural single family house, located in Beijing, China, was investigated using TRNSYS 17.0 software. The results showed that the heating capacity and energy consumption of the proposed system were 8868 and 2968 kWh, respectively, during the entire heating period, which were decreased by 21 and 24%, respectively, in comparison with the baseline system. The HSPF of the proposed system was 2.37 during the entire heating period, which was increased by 4% compared with the baseline system.

Acknowledgements

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