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Study on the performance of solar-assisted transcritical CO_2 heat pump system with phase change energy storage suitable for rural houses

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ABSTRACT

To alleviate the serious energy waste and air pollution caused by heating of buildings in rural areas, a solar-assisted transcritical CO_2 heat pump system with phase change energy storage (STCHPS-PCES) suitable for rural houses is proposed. In addition to the environmental protection of refrigerants and the matching of heating characteristics with the building heat load, the system also achieves the complementary use and energy storage utilization of natural low-grade heat sources. Taking a 500 m^2 residential building of a rural area in Shenyang, China as the case study building, the mathematical model of the system was established and its performance was simulated. The results showed that the system can meet the demand for building heating and hot water during the heating period. The seasonal coefficient of performance (*COP*) of the system is 2.56. Compared to the traditional decentralized coal-fired heating and hot water supply programmes in rural areas of Shenyang, the system's energy-saving rate is 43.55%.

1. Introduction

In recent years, the problems of high energy consumption, low efficiency, and serious pollution of buildings in severely cold rural areas have drawn increasing attention. The study of new energy-saving air-conditioning and heating technologies applicable to the characteristics of rural buildings (Ignas et al., 2006) has become the key to alleviate this problem. Heat pump technology utilizes nature's low-grade energy, which is an excellent way to replace traditional building energy supply technology and reduce building energy consumption. From the perspective of the current application of heat pump systems, air source heat pump (ASHP) systems (Caihua et al., 2011; Paul et al., 2011) occupy a large market advantage because of their wide application range and reliable performance. At the same time, solar heat pumps (Nagaraju et al., 1999; Tintai et al., 2011) have also developed rapidly in water heaters due to good heating performance. However, ASHP systems have the problems of poor low-temperature heating performance and frosting of the heat exchanger. Due to the low and unstable solar heat flux, this results in a larger collector area and a higher initial investment cost, which limits the further development of solar heat pump technology. On the other hand, with the restrictions and bans on the use of hydrochlorofluorocarbons (HCFCs) and chlorofluorocarbons (CFCs) by various governments (United Nations Environment Programme, 2018); transcritical heat pump systems using natural working fluid CO_2 as a refrigerant have received more and more attention. However, in addition to the common problems of ASHPs, transcritical CO_2 heat pump (TCHP) systems also face the problems of large throttling losses and high operating pressures.

In order to improve the operating performance of TCHP systems, researchers have done a lot of research on its components. Minxia et al. (2016) studied the effect of the number of tubes inside the gas cooler on the heat transfer performance and pressure drop of a TCHP system. Li (2013) optimized the design of gas coolers and intercoolers in a two-stage TCHP system. Liu et al. (2016, 2012) experimentally studied the effects of injectors and compressors on the heating and cooling performance of a TCHP and simulated the influence of the structural parameters of the injectors on its performance. Ahammed et al. (2014) proposed a two-phase injector to replace the throttle valve in a conventional TCHP system. Improvements in component performance have also improved system performance to some extent. But due to limitations in theory and manufacturing technology, it is difficult to further improve the system performance from the component level.

To further improve the performance of TCHP systems, some scholars have begun to study the cyclical mode and control strategy. Goodarzi et al. (2015), Goodarzi and Gheibi (2015) studied and improved the performance of a two-stage TCHP. Qi et al. (2013) conducted an experimental study on the optimal exhaust pressure of a TCHP water heater. In addition, similarly to a traditional ASHP system,

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frosting on the surface of the heat exchanger is also one of the main reasons for the deterioration of the performance of a TCHP system under low-temperature conditions. In order to improve its performance during defrosting, Steiner and Rieberer (2013) simulated and optimized the influence of different system parameters on the reverse cycle defrosting process. After optimization, the defrosting efficiency was improved. But due to the use of reverse cycles, the system could not achieve continuous heating. Therefore, Jang et al. (2013) developed a continuous heating technology that can use a hot gas bypass valve to achieve continuous and efficient heating of the room while defrosting. It has been found that a single heat source heat pump system is difficult to operate continuously and efficiently. Therefore, some scholars have combined multiple heat source coupling technology with a TCHP system and proposed different types of composite systems. Hu et al. (2016) proposed a multi-heat-source TCHP system. Compared with the traditional ground source heat pump, the system's economy and energy savings were improved.

With the rise of solar-assisted heat pump technology, different kind of solar-assisted heat pump systems were proposed and studied. Choi et al. (2014) combined solar energy, geothermal energy, and TCHP and proposed a solar composite ground source heat pump system. Deng et al., (2011) proposed a new type of solar-assisted TCHP system. Compared with the traditional CO_2 system, the annual energy-saving rate of this system reached 19.3%. Moreover, solar collectors have a greater impact on the performance of solar-assisted heat pump systems. To test the performance of solar collectors, Esen and Esen (2005) conducted a comparative study of the performance of a two-phase thermosyphon solar collector charged with different refrigerants. Garnier et al. (2009) studied the water temperature distribution in an integrated collector storage solar water heater. In order to improve the energy efficiency of solar heat pump systems, some scholars added phase change heat storage (PCES) devices in heat pump systems. Esen and Ayhan (1996), Esen et al. (1998), Esen (2000) simulated and optimized the performance of the PCES device in a solar assisted heat pump system, determined its internal energy storage and temperature distribution. Comakli et al. (1993) experimentally studied the performance of a solar-assisted heat pump system with PCES for residential heating.

Through the above studies, the performance of TCHP systems has been improved, but the overall performance and energy efficiency still need to be improved. Therefore, a solar-assisted transcritical CO_2 heat pump system with phase change energy storage (STCHPS-PCES) is proposed by combining a solar collector with a TCHP and a PCES device. In addition to the environmental protection of refrigerants and the matching of heating characteristics with the building heat load, the system also achieves the complementary use and energy storage utilization of natural low-grade heat sources. Moreover, the system can be connected to hot water radiators for indoor heating. It is especially suitable for heating of decentralized buildings with no central heating, such as rural houses. In this paper, a mathematical model is established to simulate the dynamic performance of the system and the feasibility of the system in severe cold regions is studied.

2. System operation control principle

The STCHPS-PCES is mainly composed of a solar collector, a TCHP unit, a PCES device, a water tank, and circulating water pumps. By controlling the four-way reversing valves and circulation pumps, the system can achieve the independent operation or joint operation of four operating modes: solar heat pump mode (mode 1), ASHP mode (mode 2), PCES mode (mode 3), and phase change heating mode (mode 4). A schematic diagram of the system in different operation modes is shown in Fig. 1.

To maximize the energy efficiency, the system selects the appropriate operating mode based on the outdoor temperature, solar radiation intensity, room heat load, and the system's own state. During the heating period, the system uses different control strategies during the day and at night. Daytime and nighttime are defined as 6:00 to 18:00 h and 18:00 to 6:00 h, respectively.

When the room needs to be heated during the daytime, the system will give priority to operating mode 1 as shown in Fig. 1(a). In mode 1, the water in the solar collector absorbs solar radiation heat to provide low-temperature thermal energy for the heat pump, and the high-temperature hot water produced in the gas cooler is used for heating and domestic water. When mode 1 cannot be activated, the system operates in mode 2, as shown in Fig. 1(b), where the heat pump unit absorbs heat from the air to meet the heating and hot water requirements of the room. When there is no need for heating in the room during the daytime, the system operates in mode 3, as shown in Fig. 1(c), where the excess heat of condensation is absorbed and stored by the PCES device. When the heat pump alone cannot satisfy the heating demand, the system starts mode 4 at the same time as shown in Fig. 1(d). At this time, the phase change material in the PCES device directly exchanges heat with the circulating water to make up for the heat shortage of the heat pump.

When the room needs heating during the night time, the system will give priority to operating mode 4 for heating. When mode 4 alone cannot meet the heating requirement, the system will start mode 2 and mode 4 at the same time. When mode 4 fails to start, the system will run mode 2. At night, the solar collector stops operating. To prevent the pipeline from freezing, the circulating water in the pipeline is discharged by controlling the empting valves.

3. System mathematical model

In order to study the operating characteristics of the STCHPS-PCES, a mathematical model was established for the system charge and major components such as the finned tube evaporators, gas coolers, compressors, and throttle valves.

3.1. Finned tube evaporator model

The refrigerant side of the evaporator usually consists of a two-phase zone and an overheated zone, as shown in Fig. 2(a). To simplify the mathematical model, the following assumptions are made: (1) refrigerant flow in the axial direction in pipes is one-dimensional; (2) the axial heat conduction and friction heat of the refrigerant and the thermal resistance of the tube wall can be ignored; (3) the influence of frost on the pipe wall can be ignored; (4) refrigerant flow distribution is uniform; (5) refrigerant flows as homogeneous flow; (6) the pressure and flow rate of the gas-phase and liquid-phase refrigerants in the two-phase area are the same; (7) the refrigerant pressure drop in the overheating area can be ignored. Based on the above assumptions, a parameter distribution model of the evaporator is established, where the evaporator's micro-segment is as shown in Fig. 2(b).

(1) Flow heat exchange equations of refrigerant in the pipe

$$dQ_{r,e} = m_{re}(h_{ro,e} - h_{ri,e}) = \alpha_{r,e} dA_i (t_{w,e} - t_{rm,e}) \quad (1)$$

where $dQ_{r,e}$ is the amount of heat transfer of the refrigerant, W; $m_{r,e}$ is the mass flow rate of the refrigerant, kg/s; $h_{ri,e}$ and $h_{ro,e}$ are the enthalpy values of the refrigerant in the inlet and outlet of the micro-segment, J/kg; $\alpha_{r,e}$ is the refrigerant-side heat transfer coefficient, W/(m^2·°C); dA_i is the micro-segmental tube inner surface area, m^2; $t_{w,e}$ is the tube wall temperature, °C; $t_{rm,e}$ is the average refrigerant temperature, °C. In the single-phase zone and the two-phase zone, the refrigerant-side heat transfer coefficients are calculated by the heat transfer correlations of Dittus and Boelter (1980) and Gungor and Winterton (1986), respectively.

(2) Flow heat exchange equations of air out of pipe