13th IEA Heat Pump Conference
April 26-29, 2021 Jeju, Korea

Energy Efficiency Research of an Air-source Heat Pump for Floor Heating System in Winter Test

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Abstract

An air-source heat pump for floor heating system was developed and tested which produced hot water for heating the floor of residential buildings by heat pump cycle. The field test of the air-source heat pump for the floor heating system has been conducted in winter for 93 days. During the test, the compression ratio (π) was controlled stable based on the exergy performance analysis. The theoretical η_x of the system decreases with the increase of the compression ratio (π). The values of π were maintained as small as possible from 2 to 6 approximately in the ideal control method in the winter test. Based on the control strategy, the system was proved to be robust. The energy efficiency ratio (EER) ranged from 2.32 to 4.71 while the heat capacity ranged from 3.8 to 10 kW. Especially, the performances of the system in the warmest and coldest days were compared. The results indicate good environmental adaptability of the heat pump system.

Keywords: heat pump; floor heating system; field test

1. Introduction

Building energy consumption has become an important aspect that cannot be ignored. Some studies have shown that energy consumption can be saved by 20% to 30% via optimizing system operation and efficient energy management without changing the building structure and internal equipment [1]. Heating ventilation and air conditioning (HVAC) system is the main energy consumption system in residential buildings, which generally accounts for 30%~50% of the total energy consumption [2]. At present, heat pumps are widely used for domestic hot water (DHW) preparation and space heating (SH), which can offer the possibility to substitute conventional heating systems (direct electric, oil boiler, gas boiler, etc.) and reduce the primary energy consumption [3]. Especially, air-source heat pumps (ASHP) are widely used in residential heating and cooling systems due to the low installation costs and high efficiency [4]. In this paper, an air-source heat pump system used for floor heating was proposed and tested. The operation performance was evaluated.

2. System description

A schematic diagram of the proposed air-source heat pump system is shown in Fig. 1. The system consists of an inverter compressor, an electronic expansion valve, a fin-tube heat exchanger in the airside, and a plate heat exchanger in the waterside. When the heating model is operated, hot water is produced through the plate heat exchanger by the heat pump cycle.

The adaptive defrosting strategy is applied in the system for improving the adaptation to environmental variations [5]. The whole defrosting process is conducted in time and barely does not affect the indoor temperature. The operation test was conducted and the main working conditions are shown in Table.1.

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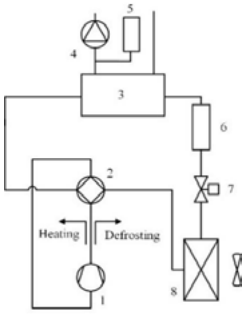


Fig. 1. The air-source heat pump for floor heating system (1.Compressor 2. Four-way reversing valve 3. Plate heat exchanger 4. Circulating water pump 5. Expansion tank 6. Reservoir 7. Expansion valve 8. Fin-tube heat exchanger)

Table 1 Test condition of the air-source heat pump for floor heating system

Nominal Condition	**Outdoor dry/wet bulb temperature (T_{od}/T_{ow}:)**	**Water flow rate(Q)**	**Outlet water temperature (T_o)**	**Refrigerating/ Heating capacity(C_R/C_H)**	**Inlet power(P_{in})**	**EER**
Refrigerating	35/- °C	1.2 m³/h	7 °C	2 kW	2.77 kW	3.08
Heating	7/6 °C	1.2 m³/h	45 °C	9.02 kW	2.87 kW	3.14
			35 °C	9.56 kW	2.38 kW	4.02

3. Application test in winter

The air-source heat pump system is used on the third floor of a 6-storey unit building in Chuzhou, Anhui Province, China， in which the building area is about 120 m². The whole heating load in the winter season is calculated to 2.19KW. In Fig. 2, residential structure and geothermal hot water circuit design can be seen that four separated hot water circuits are respectively installed in the master bedroom A, second bedroom B, master bedroom C and study room D. Besides, the temperatures of the water inlet and outlet were tested outside to control the condensing temperature for the stable air temperature in the room, and the evaporating temperature varied with ambient temperature by the fin-tube evaporator in Fig. 2.

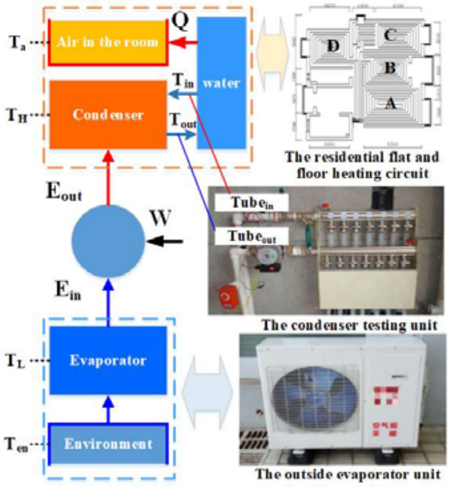


Fig. 2. The diagrammatic sketch of the principle of the air-source heat pump for floor heating system

The Energy efficiency ratio (*EER*) of the heat pump unit is the long-established evaluation parameter that is the ratio of heating energy and energy consumption of the system. Energy consumption (*W*) is the inlet electric energy, and heating energy (Q) is the energy from the environment (E_{in}) plus electric energy (W) shown in Fig. 2. W is measured by the power meter every one minute.

3.1. Exergy efficiency

For the vapor compression system, the external power is supplied by the compressor for the major components: evaporator, condenser, and expansion valve. Heat transfer between the chiller and the environment shown as Qcond in Fig. 3 takes place at a finite temperature difference, which is a major source of energy losses for the cycle. Irreversibility causes system performance to degrade. The irreversibility in the cycle is supposed to be evaluated considering individual thermodynamic processes that composed the cycle. To show the individual process energy balances, exergy calculations provide increased and deeper insight into the process for improvements. Exergy analysis of the heat pump system for heating floor is performed by analyzing the components of the system separately. Identifying the main sites of exergy destruction shows the direction for potential improvements. The goal of exergy analysis for the system is to find the minimum power required for a certain desired result. The expressions for the exergy efficiency and exergy loss for the individual processes that make up the cycle as well as the energy efficiency ratio (EER) and second law efficiency for the entire cycle are analyzed.

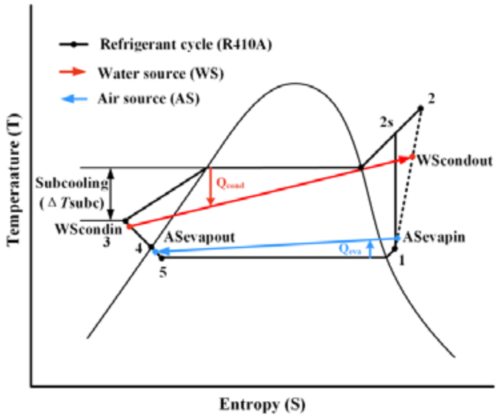


Fig. 3. T–s diagram of the refrigerant cycle and heat sources (water and air sources).

T–S diagram of the system is shown in Fig. 2. In the diagram, process 1-2s is isentropic compression in the compressor. The real compression process is the state 1-2. The other states viz. 2-3, 3-5, and 5-1 show condensation, throttling in the expansion valve, and evaporation in the evaporator respectively. The special lines in red and blue of the refrigerant (R410A) states in Fig. 2 indicate the heat exchange processes of air source and water source.

The mathematical formulation for exergy analysis in different components can be arranged in the following way:

Specific exergy in any state is:

$$\psi = (h - h_d) - T_d(s - s_d) \tag{1}$$

where h is the specific enthalpy, s is the specific entropy, and T_d, h_d and s_d are the temperature, the enthalpy, and the entropy at the dead state.

For evaporator, the heat addition in the evaporator is:

$$Q_{evap} = \dot{m}(h_1 - h_5) \tag{2}$$

The exergy destruction of the evaporator is:

$$I_{\text{evap}} = \dot{m}(\psi_5 - \psi_1) + Q(1 - \frac{T_{\text{d}}}{T_{\text{ev}}}) \\ = \dot{m}(h_5 - h_1) - T(s_5 - s_1) + Q(1 - \frac{T_{\text{d}}}{T_{\text{ev}}}) \tag{3}$$

For the compressor, the compressor work is:

$$W_{\text{com}} = \dot{m}(h_2 - h_1) \tag{4}$$

For non-isentropic compression, the electrical power is:

$$W_{\text{el}} = \frac{W_{\text{com}}}{\eta_{\text{mech}}\eta_{\text{el}}} \tag{5}$$

where the mechanical efficiency (η_{mech}) and the electrical efficiency (η_{el}) of the motor are 90% respectively.
The exergy efficiency (η_{x}) is:

$$\eta_{\text{x}} = \frac{\psi_1 - \psi_5}{W_{\text{el}}} \tag{6}$$

The energy efficiency ratio (*EER*) is:

$$EER = \frac{h_1 - h_5}{W_{\text{el}}} \tag{7}$$

To improve the *EER* of the heat pump, the refrigerant cycle should be controlled for enlarging (h_1-h_4) based on equation (7) and reducing (h_2-h_1) based on equations (4) and (5). It could be concluded that h_1 is positively correlated with *EER*. Based on equations (1), (5), and (6), the η_x could be improved by controlled for the proper h_5 which is mainly controlled by point 3 in the cycle under varied ambient conditions. To improve the *EER* of the heat pump, the refrigerant cycle should be controlled for enlarging (h_1-h_4) based on equation (7) and reducing (h_2-h_1) based on equations (4) and (5). It could be concluded that h_1 is positively correlated with *EER*. Based on equations (1), (5), and (6), the η_x could be improved by controlled for the proper h_5 which is mainly controlled by point 3 in the cycle under varied ambient conditions. In conclusion, points 1 and 3 should be controlled for closing the states of inlet air (point ASevapin in Fig. 3) and inlet water (point Wcond in Fig. 3) to decline the temperature difference between the heat sinks and the heat sources. Based on the researches for the best performance according to the energy and exergy efficiency of the system, exergy efficiency can be improved by controlling sub-cooling temperature to 5 °C. It can be done for reducing the pressure rises in the compressor and hence reduce the irreversibility in the compressor because it is found that most of the irreversibility occurs in the compressor parts. The temperature difference of the evaporating and condensing temperature should be reduced because exergy loss increases as the temperature of the evaporator decrease. For a fixed indoor temperature heated by the condenser, the condensing temperature varies with the inlet water source temperature. The components that contribute most to the heat pump exergy efficiency improvement with the sub-cooling state are the expansion valve and the condenser. To decline irreversibility which can lead to the reducing exergy destruction ratio (*EDR*) and the increasing exergy efficiency of the heat pump system, the sub-cooling temperature should be maintained close to 5 °C, also the compression ratio (π) is kept stable. The inlet water temperature for the test in winter varied from 2 to 12 °C, and the inlet air temperature varied from -10 to 12 °C. Therefore, π of the compressor unit should be maintained from 2 to 6 and the actual π in the practical test is controlled from the range of 2.67 to 5.73 by the expansion valve and the variable frequency compressor in the test.

3.2. Operation test

The approach of the air-source heat pump for the floor heating system has been implemented in TRNSYS software for accurately modeling the dynamic simulation of the system. The simulating data for 93 days in winter were performed in Fig. 5 (a) to verify the theory of the ideal control method. The field test of the air-source heat pump for the floor heating system has been conducted comparing with the simulating data in winter for the 93 days. Based on the fixed compression ratio (π) controlled by the system, the tested heating capacity (*HC*(Test)) and energy efficiency ratio (*EER*(Test)) of the heat pump system, which nearly are the same as those of simulation (*HC*(Sim) and *EER*(Sim) in Fig. 5 (a)), showed opposite variation trends. The tested heating capacity of the system ranged from 3.8 kW to 10 kW, while *EER* ranged from 2.32 to 4.71. It means that the required heating capacity varied with the change of environmental temperature, and the relatively stable temperature field can be obtained due to the fixed π during the heating period.

As shown in Fig. 5 (b), the average measured temperatures at different height of 0.1m, 1m, 1.5m, and 2m were 20.5°C, 19.5°C, 19°C, and 18.4°C in the heating periods, during which the outdoor temperatures were ranging from -10°C to 14°C with the average temperature of 3.7°C. Therefore, people could feel comfortable owing to the relatively stable temperature field in the house.

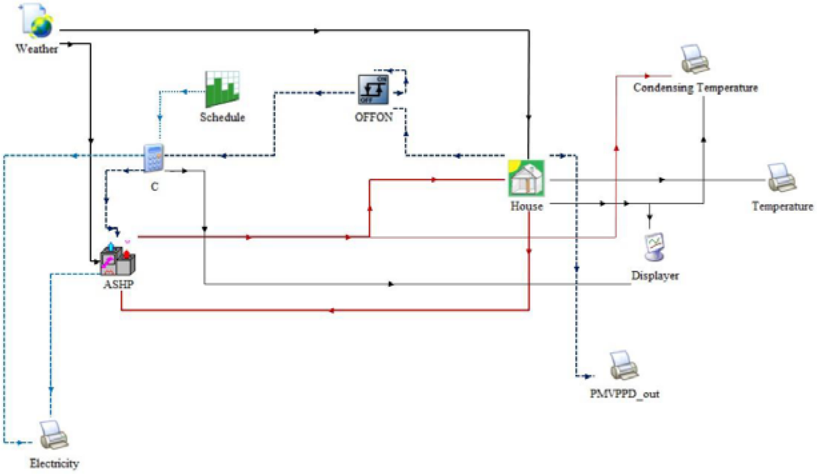
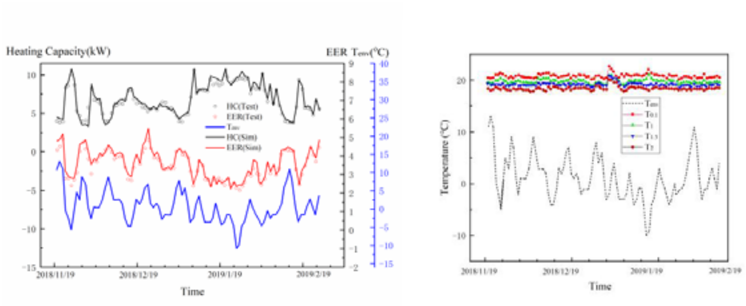


Fig. 4. The diagrammatic sketch of air source heat pump for floor heating system in TRNSYS software



(a) (b)

Fig. 5. The performance of the air-source heat pump in winter (a. Tested temperatures b. EER and *Q* in winter)

The operation performances of the heat pump system in the coldest and warmest days were listed in Table.2 based on the tested data. The results show that: 1) In the whole period, there was a great difference between the outdoor temperatures on the coldest and warmest days. The indoor temperature keeps relatively stable at about 19 ºC which indicates good environmental adaptability of the heat pump system. 2) The heating capacity and *EER* on the coldest day are relatively stable, while the fluctuation is relatively larger on the warmest day. The average *EER* in the warmest day is much higher than that of the coldest day by 159%. 3) The difference between suction pressure and exhaust pressure in the coldest day and the warmest day is small and within a reasonable range based on the required stable indoor temperature.

Table 2 The operation data of the heating system on the typical days during the heating period

Typical weather day	Environmental temperature(T_{env})	Heating capacity(Q_h)	EER	Suction/Exhaust pressure(P_s/P_e)	Condensing/Evaporating temperature(T_c/T_e)
The coldest day(2019/1/23)	-4.3~-10.0ºC	8.6~9.61 kW	2.01~2.54	0.44/2.52MPa	48.3/-16.3 ºC
The hottest day(2018/11/21)	8~16 ºC	2.1~6.03 kW	3.62~5.33	0.79/2.11 MPa	37.9/-0.3 ºC

4. Conclusion

The air-source heat pump system for floor heating was presented. Based on the theoretical analysis of the exergy efficiency (η_{ex}), the related simulation and operation tests were conducted and its performance was analyzed. Some conclusions are shown as follows.

(1) The theoretical η_x of the system decreases with the increase of the compression ratio (π). According to the actual demand, the sub-cooling temperature should be maintained close to 5 ºC, also the compression ratio (π) should be kept stable theoretically. The values of π were maintained as small as possible from 2 to 6 approximately in the ideal control method in the winter test.

(2) The average measured temperatures in different heights show a relatively stable and uniform indoor temperature field during the heating period in the winter test. Under the different outdoor temperatures, the heating capacity of the presented heat pump varied from 3.8 to 10 kW with the *EER* changed from 2.32 to 4.71. The required heating capacities in the coldest and warmest days can be both satisfied owing to the controlled compression ratio (π), which shows the good environmental adaptability of the system.

(3) The actual *EER* is not only correlated with η_{ex} but also the coefficient of performance and rate of the heating of evaporator, condenser, and water source. There is a clear positive correlation among them, and thus η_{ex} could be the only objective function theoretically. The real gap between *EER* and η_{ex} is the ignored error in measurement and the product of the coefficient of performance and rate of the heating source which could be researched further.

Acknowledgments

The work of this paper is financially supported by China YangziGroup Chuzhou Yangzi Air Conditioner Co., LTD and Independent Innovation Specialized Fund Project of Anhui Province (2012XKJH0003).

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