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Climate Adaptivity and Field Test of the Space Heating Used Air-Source Transcritical CO_2 Heat Pump

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Abstract: In this study, an innovation of air-sourced transcritical CO_2 heat pump which was employed in the space heating application was presented and discussed in order to solve the problem that the heating performances of the transcritical CO_2 heat pump water heater deteriorated sharply with the augment in water feed temperature. An R134a cycle was adopted as a subcooling device in the proposed system. The prototype of the presented system was installed and supplied hot water for three places in northern China in winter. The field test results showed that the acceptable return water temperature can be increased up to 55℃, while the supply water temperature was raised rapidly by the presented prototype to up to 70℃ directly, which was obviously appropriate to the various conditions of heating radiator in space heating application. Additionally, though the heating capacity and power dissipation decreased with the decline in ambient temperature or the augment in water temperature, the presented heat pump system performed efficiently whatever the climate and water feed temperature were. The real time COP of the presented system was generally more than 1.8 in the whole heating season, while the seasonal performance coefficient (SPC) was also appreciable, which signified that the economic efficiency of the presented system was more excellent than other space heating approaches such as fuel, gas, coal or electric boiler. As a result, the novel system will be a promising project to solve the energy issues in future space heating application.

1. Introduction

Recently, lots of cities in northern China were invaded by haze once the heating season came. A main cause of the strict dilemma in haze is that plentiful of distributed coal-fired boiler in northern China exhausted a mass of dust and smog to atmosphere. Nowadays, policies were presented in China to promote extensively the alternation of the thermal resource from coal to electricity in order to decrease the environment pollution. Comparing with the electric boiler, the very high system performances of air-source heat pump (ASHP) which can absorb the thermal energy from atmosphere and transport to water makes the ASHP become the best choice to provide thermal energy to users in winter.

However, it is the heat radiator instead of the underfloor heating or fan coil which is still the broadest heating facility in China. The heating conditions of circulating water in heat radiator which are 60~80℃ in supply temperature and 40~50℃ in return temperature are far beyond the operating conditions of heat pumps using convenient refrigerants which can only provide the water supply temperature of 40~50℃ [1]. Fortunately, the transcritical air-source CO_2 heat pump which can provide hot water with temperature of higher than 80℃ directly is proper for the working conditions of the heat radiator.

To solve the problem of the space heating used transcritical CO_2 heat pump (SHTCHP) that the system performances deteriorated sharply with the increase in the return water temperature which is 40~50℃ in heat radiator, abundant researches were carried out. Wang [2] illustrated that the transcritical CO_2 system could be applied to wonderful thermal resource due to its huge temperature glide in the gas-cooler. However, the COP of the transcritical CO_2 system is quite lower than systems with convenient refrigerants due to the remarkable irreversibility which caused by the superheated vapor horn and the high throttling losses. The conclusion can be obtained by comparing the results from [2] and [1]. Thus, plenty of improvements were worked out to enhance the efficiency of transcritical CO_2 systems by employing various modifications.

Lots of scholars experimentally discussed the performances of the transcritical CO_2 system with and without the IHX in single-stage transcritical CO_2 systems [3-5] as well as similar investigations were carried out to the two-stage systems [6-7]. The performances of the transcritical CO_2 system with an IHX was assessed by [3] in which they found that the system COP was enhanced by up to 10.5% by adding the IHX since the fluid temperature at gas-cooler inlet decreased from 40 to 25 °C. The performances of an IHX based CO_2 transcritical refrigeration prototype at three various evaporating temperatures (-5 °C, -10 °C and -15 °C) and two gas-cooler outlet temperatures (31 °C and 34 °C) were analyzed by [5] experimentally and they found an augment in system cooling capacity of 12% and the system efficiency up to 12%. As for the two-stage compression cycle from [6-7], the simulation research indicated that IHX rose the specific cooling capacity and declined the optimal discharge pressure, and the adoption of the IHX enhanced the system COP up to almost 10%. The effects of the vapor extraction from the vessel and subsequent injection into the refrigeration cycle in a double-stage system were studied by [8] experimentally and they found that the system COP increased by approximately 16.5%.

Another excellent way to enhance the performances of the transcritical CO_2 system is the subcooling technology. The performances of a transcritical CO_2 system with a thermoelectric facility were investigated by [9] theoretically. The results found that the system COP can be increased by up to 25.6%. The energy improvements of the transcritical CO_2 system with a dedicated mechanical subcooling were analyzed theoretically by [10]. The simulation results showed that the enhancements in system COP and system cooling capacity was up to 20% and 28.8%, respectively. Besides, the comparison was carried out by different refrigerants in the subcooling cycle. The combined transcritical CO_2/R134a heat pump system which was dedicated for space heating applications was investigated theoretically and experimentally by [11] and [12], respectively. The results of those

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studies proposed that the combined transcritical CO_2/R134a system can operate efficiently and provide stably hot water with temperature of higher than 70°C. Furthermore, the combined CO_2/R134a system was compared with the cascade CO_2/R134a system.

As mentioned above, although the transcritical CO_2 system showed a good potential on space heating application and hot water production, there is little literatures can be found to test such a machine in specific conditions of space heating field. In this paper, three prototypes of SHTCHPs which is similar with the plant in literature of [12] are installed and tested in three different cities (Tianjin, Shijiazhuang and Zhangjiakou) with different climate conditions. Further, the heating performances of the prototypes like return water temperature, supply water temperature, heating capacity and COP are recorded and analyzed. Additionally, the effects of defrosting procedure is also discussed.

2. System description

2.1 Field test facility

The proposed transcritical CO_2 heat pump system is a combination form of a standard transcritical CO_2 system and a standard R134a refrigeration system. In this form, the standard R134a refrigeration system play a role as a subcooling machine to provide cooling capacity for the refrigerant CO_2 from the gas-cooler which is not cooled enough. The sketch map of the proposed system which includes a transcritical CO_2 cycle, an R134a cycle and a circulating water system is shown in Figure 1a. The ambient temperature is varied from -20 to 10 °C as well as the feed water (from user) varied from 40 to 50 °C. The feed water flows through the three-way proportional valve and then is split into two parts of streams. The first part of feed water enters the condenser of the R134a subsystem in which it is heated up by the high pressure and high temperature R134a refrigerant before it is channeled into the mixing tank. Another part of feed water is channeled into the evaporator of the R134a subsystem in which it is cooled down by the low pressure and low temperature R134a refrigerant. The cold water in the exit of R134a evaporator is then flowed into the gas-cooler of the transcritical CO_2 cycle in which the cold water sub-cools the CO_2 refrigerant and meanwhile is heated up by the high temperature CO_2. This part of hot water is channeled finally into the mixing tank in which it is mingled to the first part of hot water from the R134a condenser. In this system, circulating water system is used to be an intermediate fluid to link the gas-cooler of the transcritical CO_2 subsystem and the evaporator of the R134a subsystem.

The experimental prototype which includes an R134a subsystem and a transcritical CO_2 subsystem is shown in Figure 1b. Three same prototypes were installed in three different places: a hall of a ski resort where the ambient temperature is low to -20°C in Zhangjiakou City, a government building where the average ambient temperature is almost -10°Cin Shijiazhuang City, and a train station where the average ambient temperature is almost 0°C in the whole heating season in Tianjin City.

The test data acquisition system is comprised by various measurement facilities that can measure the system parameters and maintain the system status. For instance, the ambient temperature and relative humidity are measured by a temperature and humidity recorder. The measurement range of the temperature sensor is -40°C~150°C with the accuracy of ±0.5°C and that of the humidity sensor is 0~100%. T-type thermocouples with the accuracy of ±0.5°C are used to measure the circulated water temperatures (both inlet and outlet). The water flow rates are measured by electromagnetic flowmeters with the measurement range of 0~6m3h-1 and the uncertainty is ±1%. The total electric power dissipation from compressors, fans and water pumps is recorded by an electric power meter. More details of the measurement facilities are showed in Table 1.

Table 1. Details of the measurement facilities

Measuring object	Facilities	Measuring range	Uncertainty
Ambient temperature (°C)	Temperature and humidity record	-40~150	±0.5°C
Relative humidity (%)	Temperature and humidity record	0~100	±5%
Water temperature (°C)	T-type thermocouple	-40~150	±0.5°C
Water volume flow rate (m3h-1)	Electromagnetic flowmeter	0~6	±1% of reading
Power dissipation (kW)	Electric power meter	0~100	±0.5% of full scale
R134a pressure (MPa)	Pressure transmitter	0~6	±1% of reading
CO_2 pressure (MPa)	Pressure transmitter	0~16	±1% of reading

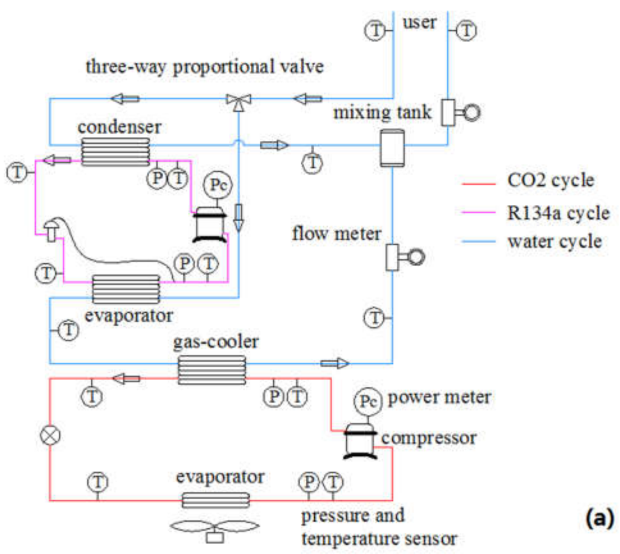

Figure 1a. The sketch map of the space heating used transcritical CO_2 system

Figure 1b. The picture of the experimental prototype

2.2 System analysis and defrost control

In this study, the heating capacity and the power dissipation of the prototype can be measured and then the system COP can be calculated. Since the water volume flow rate and both the inlet and outlet temperatures were measured and recorded, the heating capacity of the prototypes can be calculated from water side accurately, which is written by:

$$Q_i = c_{p,water} \cdot G_i \cdot (t_{out,i} - t_{in,i}) \quad (1)$$

Additionally, in order to assess the system performances of the prototype during all the heating season, the total heating capacity during the total working period τ can be calculated by:

$$Q_{season} = \int_0^{\tau} c_{p,water} \cdot G_i \cdot (t_{out,i} - t_{in,i}) d\tau \quad (2)$$

Since the ambient temperature varies continuously with time under the field test conditions, the instantaneous COP has less significance than the seasonal performance coefficient (SPC) which can be calculated by (all the test data of the prototype will be recorded per 1.5 minutes):

$$SPC = \frac{Q_{season}}{W_{season}} = \frac{\sum_{k=1}^{n}\sum_{h=1}^{j}\sum_{m=1}^{p} Q_{1.5\,min}}{\sum_{k=1}^{n}\sum_{h=1}^{j}\sum_{m=1}^{p} W_{1.5\,min}} \quad (3)$$

where the total power dissipation can be recorded by the electric power meter.

As for the operating logic, the transcritical CO_2 subsystem is the main cycle while the R134a subsystem is the vice-cycle. If the return water temperature is lower than 30°C in which case the CO_2 gas-cooler outlet temperature can be cooled adequately by only the return water, the transcritical CO_2 subsystem is activated to heat the circulating water and the R134a subsystem is unoperated. When the return water temperature reaches 30°C or higher, the R134a subsystem is activated to be the subcooler.

The frosting phenomenon is a serious constraint for all air-source heat pump because the system performances of the heat pump system will be deteriorated sharply with the increase in the thickness of the frost layer. Thus, the defrost method of the proposed system is also significant for the performance assessment in field test conditions. The convenient defrost approach which employ a four way valve to interchange the condenser and evaporator will cause low water supply temperature thereafter decline of indoor temperature during the defrost procedure. Moreover, there is no suitable four way valve which can be used in the transcritical CO_2 system because of the very high operated pressure. Therefore, the hot gas by-pass approach is adopted. The water pump stops immediately and the high pressure and high temperature CO_2 which is discharged from the compressor release little thermal energy to the remained water in the gas-cooler before it flows into the evaporator once the defrost mode is activated. Thus the mass of remained thermal energy in the refrigerant CO_2 which is channeled to the evaporator can be used to melt the frost layer in fins. In this way, the defrost procedure can be controlled stably without a four way valve as well as the supply water temperature and indoor temperature can be kept in constant.

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As for the defrosting control logic, the defrosting mode is activated by reaching three judgments simultaneously as follow:

- The ambient temperature is lower than 0°C;
- The operated period is longer than 45 mins;
- The temperature difference between the ambient and the finned tube is higher than 13°C.

On the contrary, the defrosting mode was cancelled by reaching any of three judgments as follow:

- The defrosting period reaches 15 mins;
- The low pressure of the transcritical CO_2 system reaches 3.8MPa;
- The finned tube temperature reaches 7°C.

3. Experimental results and discussion

3.1 Heating performances of the SHTCHP

As mentioned above, three prototypes of SHTCHPs are installed and tested in three different cities (Tianjin, Shijiazhuang and Zhangjiakou) with different climate conditions. The ambient temperature, water temperature, heating capacity and the system COP are recorded and analyzed in this study.

The main parameters and performances of the prototype in Tianjin are shown in Figure 2. The data showed that the ambient temperature kept approximately 0°C during the heating period (almost from 10:00am to 17:00pm) expect for a sudden drop of almost 5°C for almost 1 hour. Meanwhile, the prototype of the SHTCHP was switched to running from almost 10:00am to 17:00pm except for an interruption in 14:30. The return water temperature was remained 40~50°C during most time of the running period, which meant that the water temperature of the whole circulating water system can be heated quickly by the heating activity of the prototype. On the basis of the high return water temperature, the supply water temperature can be heated to 60~70°C once the prototype was running. The return water temperature thereafter the supply water temperature declined slightly due to the increased requirement of the heating capacity which might cause by the sudden drop of the ambient temperature. However, the sudden drop of the ambient temperature effected very little on the system heating capacity as well as the system COP, as shown in Figure 2. On the one hand, the system heating performances should be deteriorated by the drop of the ambient temperature, on the other hand, the heating performances was quite enhanced by the drop of the return/supply water temperature because the decline of the return water temperature decreased the CO_2 gas-cooler outlet temperature, which could improve the system performance significantly. Because of the combination of the two effects, the heating capacity and COP remained almost constant during all the heating period in this day. Additionally, the temperature difference between the supply water temperature and return water temperature was also kept almost constant due to the stable heating capacity and water flow rate.

The main parameters and performances of the prototype in Shijiazhuang are shown in Figure 3. The ambient temperature which was lower than -12°C in morning and evening and higher than -8°C in noon was more regular here than that in Tianjin. During all the running period (9:30am to 17:30pm), the data showed that the return water temperature increased slightly (from almost 35°C to nearly 50°C) from morning to noon due to the continuous heating activity of the prototype. Next, the return water temperature declined sharply first and then kept on constant until the machine was turned off in evening. The sudden drop of return water temperature was caused by the additional heating requirement indoor (a standby fan coil was activated) and the heat releasing capacity of the circulating water was increased. As mentioned above, the decline in return water temperature enhanced the heating performance of the prototype, which can be observed by the trend of COP in Figure 3. Because of the fluctuation of the return/supply water temperature when the water temperature drop occurred, the heating capacity of the prototype which was calculated by the temperature difference of the return/supply water also fluctuated subsequently. If the system performances (COP and heating capacity) can be split into two portions by the fluctuation of the water temperature drop, it can be observed that the system performances increased in the first half with the increase in the ambient temperature and decreased slightly in the second half with the slight decline in the ambient temperature, which is rational with the theory of the air-source heat pump system.

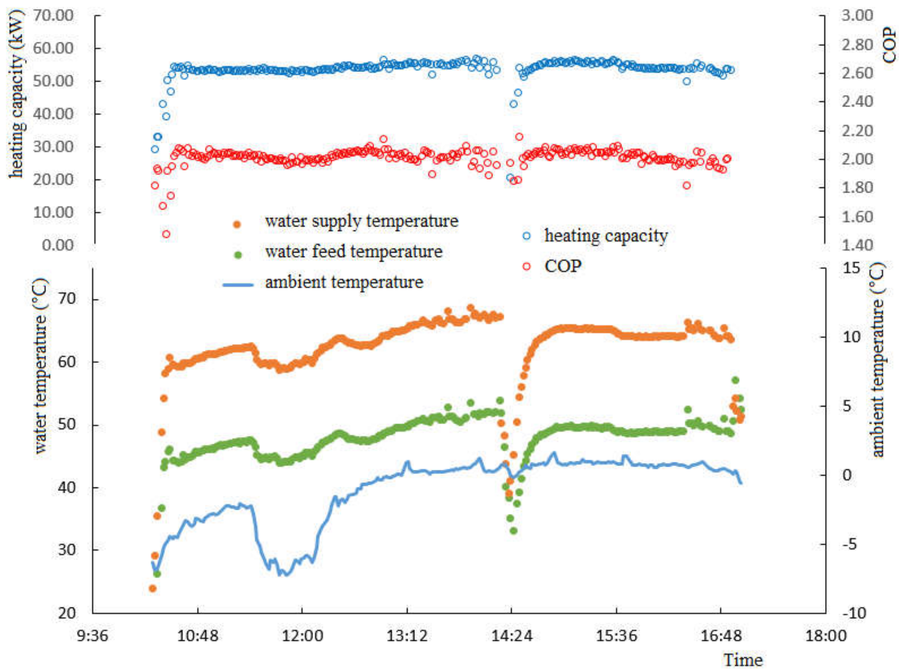

Figure 2. The main parameters and performances of the prototype in Tianjin, China

Similarly, the main parameters and performances of the SHTCHP in Zhangjiakou are shown in Figure 4. It can be seen that the ambient temperature in Zhangjiakou varied in a wide range in this day, which was lower than -15°C in the morning and increased up to -7°C in the afternoon. Very similar to the situation in Shijiazhuang, the frequent switch between start and stop of the standby fan coil caused the fluctuation in the return water thereafter the supply water temperature. However, the system heating capacity approximately increased first and then decreased with the ambient temperature except for several fluctuation, which is rational with the theory of the ASHP. Additionally, it can be noticed that the heating capacity is low while the system COP is quite high during the morning. Because the return water temperature in that time was lower than 30°C and the R134a subsystem was unoperated thereafter the power dissipation was lower. However, with the increase of the return water temperature and the activation of the R134a subsystem, the heating capacity was enhanced and the system COP can be remained at a rational value even the return water temperature is higher than 45°C which can deteriorate the system COP of the convenient transcritical CO_2 system to almost lower than 1.3.

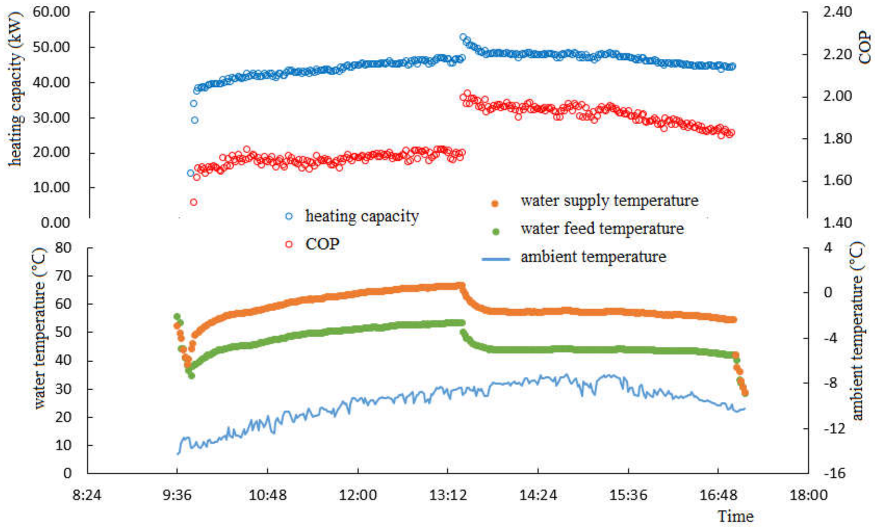

Figure 3. The main parameters and performances of the prototype in Shijiazhuang, China

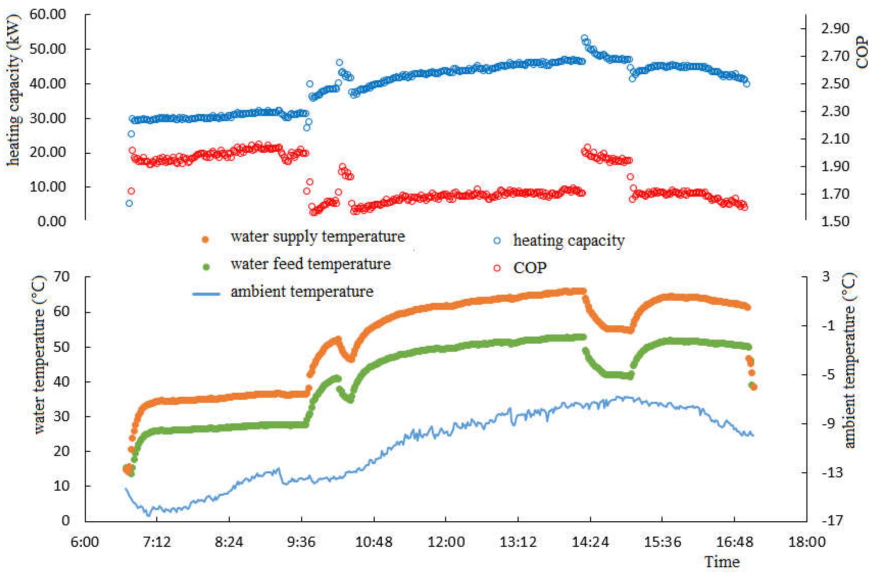

Figure 4. The main parameters and performances of the prototype in Zhangjiakou, China

According to the Figures 3~5, the system COP of the prototypes increased with the increase in ambient temperature and the decline in return water temperature. Besides, the system COP can reach to 2.0 if the operated conditions (ambient and water temperatures) were proper, and the system COP can also remain to more than 1.6 when the ambient temperature is lower than -15°C and the return water temperature is higher than 50°C. The results showed that this system can be used to supply efficiently hot water with more than 70°C in the extreme working conditions. Upon equation (3), the

SPC of the prototypes in three different cities can be obtained as follow: SPC=2.04 for prototype in Tianjin where the average ambient temperature is 1.8℃; SPC=1.81 for prototype in Shijiazhuang where the average ambient temperature is -8.6℃; SPC=1.68 for prototype in Zhangjiakou where the average ambient temperature is -15.2℃ during the heating season.

3.2 Effects of the defrosting procedure on system performance

It is well-known that the frost phenomenon and defrosting procedure that is mainly determined by the operated conditions (ambient temperature and relative humidity) could deteriorate the performance of the heat pump system in winter. In this section, the parameters of the prototype during the defrosting procedure were analyzed under different operated conditions.

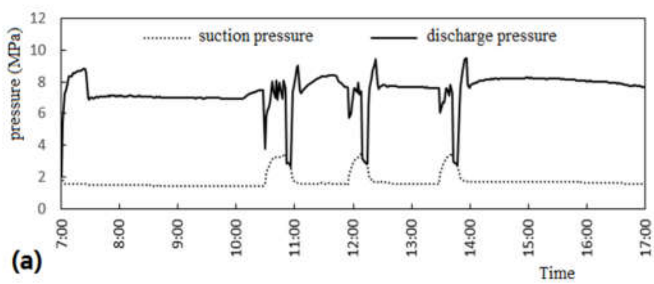

Figure 5a. The CO_2 pressures during several defrosting periods

Figure 5b. The picture of the frosting of system

The discharge and suction pressure of the transcritical CO_2 subsystem during a heating period is showed in Figure 5a. The operated conditions were given as follow: the ambient temperature was low to -20.4℃ in the morning and -19.9 in the evening while it was up to -13.9℃ in the noon. Besides, the relative humidity was up to 82% in the noon and decreased to lower than 70% in the morning and evening. There were three defrosting procedure occurred during all the heating period in this day, and the times of the three defrosting procedure were gathered from 10:30am to 14:00pm when the ambient temperature and the relative humidity were high. However, although appropriate ambient temperature and high relative humidity are able to cause frequent frosting phenomenon, the proposed system in which the hot gas by-pass defrosting approach was employed could defrost the frosted layer quickly and keep the heat pump system operated stably on the basis of the data in Figure 5a.

4. Conclusion

According to the field test results mentioned above, the space heating used transcritical CO_2 heat pump performed excellent at low ambient temperature condition and provide great potential to replace the coal-fired boiler for space heating application in northern China in winter.

On the basis of the field test data, the system performance increased with the increase in ambient temperature and the decline in return water temperature. The system COP can reach to more than 2.0 if the operated conditions were proper, and it can also remain to more than 1.6 when the ambient temperature is lower than -15℃ and the return water temperature is higher than 50℃ (water supply temperature can be obtained by higher than 70℃). The seasonal performance coefficient was higher than 2.0 at the average ambient temperature of 0℃.

Besides, the proposed system in which the by-pass defrosting approach was employed could defrost the frosted layer quickly and keep the heat pump operated stably during all the heating period.

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