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ABSTRACT

Great advancements have been made in the ASHPs (air source heat pumps) due to concerns for energy crises and consciousness of sustainable development; however there is still a deadly shortcoming that limits its widespread applications, especially in the frozen areas. However the there are greater need for the ASHPs that can run smoothly in the frigid areas in North China due to the reasons as following:(1) conventional coal-fired heating methods will inevitably lead to severe air pollution beyond toleration in North China;(2) the deadline the government set forth often can’t meet with the real situation, (3) there are great demand for both heating and cooling in many new resident apartment as great achievements have been made in the living standards and there are no coal-fired heating system in these apartments. In this paper the author proposed an improved cycle, an auxiliary cycle is added to the conventional cycle, in this way the heating capacity under low ambient temperature can be greatly improved. The prototype ASHP was validated and ran a whole winter in Beijing, China. The relevant dynamic-performance characteristics were tested and the outcomes show that this new kind of ASHP can work very well under the ambient temperatures as low as -15 ℃, In addition, the efficiency of the improved ASHP under all circumstances was addressed and great energy can be saved through the improved system’s increased efficiency. In this way the applications of ASHP were enlarged and a simple and feasible alternative heating service was developed for heating in North China, which can also partly solve the server and server atmosphere pollution problems in these regions.

NOMENCLATURE

q_{mb}： Mass flow rate of refrigerant in the supplementary circuit (kg/s)

q_{mk}： Mass flow rate of refrigerant in the condenser（kg/s）

q_{m0}： Mass flow rate of the evaporator in the evaporator（kg/s）

ξ： Resistance factor of refrigerant flowing in the supplementary circuit

v_2： Specific volume of refrigerant in state point 2（m3/kg）

p_m：Medium pressure or economizer pressure（kPa）

p_2：Pressure of refrigerant in state point 2（kPa）

R：Gas constant for refrigerant [kJ/（kg·K）]

K：Adiabatic index for refrigerant

T_6:Refrigerant temperature in state point 6（K）

w_{iq}：Isentropic compressing work of process 1-2（kJ/kg）

w_{ih}：Isentropic compressing work of process 2′-3（kJ/kg）

η_m：Mechanical efficiency

η_{mo}：Motor efficiency

INTRODUCTION

With the development of the national economy and improvement in people's living standards, the ASHP has been widely used for heating in Central China and South China in the winter since the 1990s[1-2]. In these regions the outdoor temperature is comparatively high in the winter, thus the conventional ASHP available on the market can satisfy the heating requirement for these regions quite well. However, all theory and practice have showed that the conventional ASHP has several deadly shortcomings that prevent its application in North China: the heating capacity of the ASHP decreases sharply as the outdoor temperature falls, thus in extremely cold climates –where the heat is most needed –heat pumps are least able to supply enough heat, still worse, the conventional ASHP can't work reliably if the ambient temperature is below -5℃ because the COP of the ASHP becomes smaller under this circumstance than in the warmer outside temperature, and the discharge temperature of the refrigerant will continue to increase, leading to the destruction of the compressor unless the ASHP is stopped.

Much research has been conducted to enable the ASHP to run smoothly during winter in cold areas. In Japan, a new kind of ASHP was developed with a kerosene-fired burner either placed in the indoor unit or under the evaporator to improve the performance of the ASHP in low ambient temperature [4]. The packaged ASHP using a scroll compressor which varies the rotary speed according to the heating load and has liquid-injected inlets was proven to work smoothly even under low ambient temperatures of −10 ~ −20℃[5-7]. Since 1999 a SIG was instituted at Tsinghua University to work on technologies enabling the ASHP to work reliably and efficiently in North China. Studies revealed that the heating capacity under low ambient temperatures could be greatly improved through the employment of a scroll compressor with supplementary inlets. A prototype ASHP was developed according to this conception, the dynamic performance characteristics were tested and the outcomes showed that this new kind of ASHP works very well in ambient temperatures as low as -15 ℃. In addition, the efficiency of the improved ASHP under all circumstances was addressed and the improved system's increased efficiency can save energy. In this way the application scope of ASHP was enlarged and a simple and feasible alternative heating service was developed for heat in North China, which can also partly solve the server and server atmosphere pollution problem in these regions where the coal-fired boilers and coal-fired domestic stoves were the primary heating sources

A brief introduction to the Improved ASHP System

The principle of the improved ASHP is shown in Fig.1 and the specifications of the prototype ASHP are listed in Table 1. The superheated refrigerant discharged by a scroll compressor as a hot, dense vapor with high pressure (state 3) flows through a heat exchanger, the condenser, which transfers heat from the refrigerant to the water and becomes a saturated or sub-cooled liquid (state 4). Then the refrigerant as a higher pressure, cooled liquid from the condenser, is divided into two parts: One part

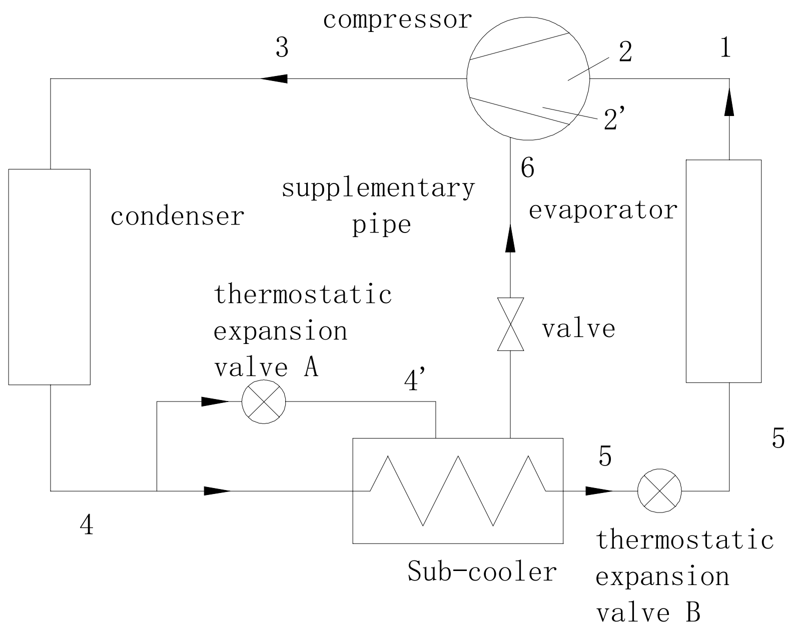

Fig. 1 Principle of the ASHP Cycle

(state 4) flows through thermostatic expansion valve A which causes the pressure to drop, As the pressure falls, the refrigerant expands and partly vaporizes, becoming chilled (point 4'). It then passes through a second heat exchanger, the economizer, which transfers the heat from the refrigerant which directly flows into the economizer (the second part) to the refrigerant of the first part, sub-cooling this second body of the refrigerant (state 5). Then the first part refrigerant flows back to the compressor through the supplementary inlets (called the supplementary circuit). The second part (state 5) flows through thermostatic expansion valve B (state 5') and into a third heat exchanger, the evaporator, which transfers the heat from the air to the refrigerant, reducing the temperature of the air. Finally the refrigerant in the main circuit (state 1) evaporates and flows back to the compressor's suction port. The refrigerant from the suction port is compressed to state point 2 by the scroll compressor where the compression chamber is connected to the supplementary circuit. The two parts of refrigerant are mixed and compressed in the compression chambers as the scroll compressor rotates until the medium pressure is lower than that of the pressure in the compressor, or the compression chamber is cut off with the supplementary circuit (state 2'). Then the mixed refrigerant is compressed to condensing pressure (state 3'). Vapor with a high pressure and temperature from the compressor is discharged to the condenser and then starts another cycle. If there were no supplementary circuit, the improved system would degenerate into a conventional ASHP system and the refrigerant would be compressed to state point 3, in which the refrigerant temperature is much higher than that of the improved cycle under low ambient temperature. The thermodynamic behavior of the improved ASHP system is shown in Fig. 2 (pressure-enthalpy diagram).

Compared with a conventional ASHP system, the improved ASHP system has some remarkable characteristics :

1) There is a supplementary circuit in the improved system starting from the outlet of the condenser and ending in the supplementary inlet of the scroll compressor. The refrigerant pressure in the economizer is between the evaporating pressure and the condensing pressure, so it is called the medium pressure or economizer pressure.

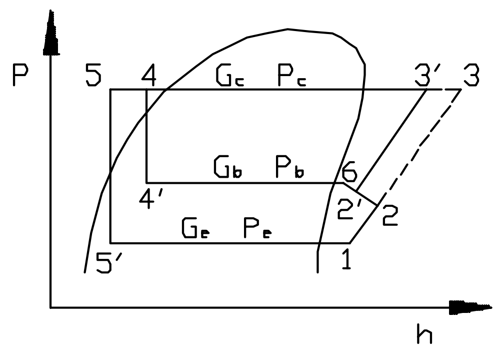

Fig. 2 The cycle in *lg*p~h diagram

Table 1 Specifications of the Prototype

Parameters		Values
Power supply	Voltage	380V
	Frequency	50Hz
Compressor	Type	Scroll
	Displacement	64cm^3
Water heat exchanger	Type	Plate
	Area	1.98m^2
Glycol heat exchanger	Type	Plate
	Area	1.65m^2
Thermostatic expansion valve A		3TR
Thermostatic expansion valve B		1TR

2) The improved ASHP system has an economizer, and there is heat exchange between the refrigerant through the main circuit and through the supplementary circuit, which can sub-cool the refrigerant in the main circuit if the supplementary circuit is on.

3) When the ambient temperature is relatively high, the prototype ASHP operates the same way a conventional ASHP does through cutting off solenoid valve B in the supplementary circuit. When the ambient temperature is below a set point, solenoid valve B is opened and the supplementary circuit is in use. In this way the operating range of the improved ASHP system is greatly enlarged and the efficiency of the improved ASHP system was addressed under all circumstances, especially when the ambient temperature is below a set point.

Theoretical Analysis of the Improved ASHP System

As is shown in the pressure-enthalpy diagram for the improved ASHP system, Process 1-2-2′-3’ is the compressing process while the supplementary circuit is on. Process 1-2 is the compressing process of the refrigerant compressed from the compressor suction part (state 1) until the compression chambers is connected with the supplementary inlets, then the refrigerant in state 2 is mixed with the gaseous refrigerant in state 6 from the supplementary pipe. The two parts of refrigerant are then mixed and compressed in the compression chambers as the scroll compressor rotates until the medium pressure is higher than that of the pressure in the compressor or the compression chamber is cut off with the supplementary circuit. This process ends in state 2’. Then the mixed refrigerant is compressed to the condensing pressure (point 3’). Process 3-4 is the condensing process, and Process 4-4′ is the expansion process through the thermostatic expansion valve A in the supplementary circuit. There is heat exchange in the economizer between the refrigerants in the supplementary and main circuits. Process 4-5 is the sub-cooling process in the main circuit and Process 4′-6 is the evaporating process in the supplementary circuit. Process 5-5′ and Process 5′-1 are the expansion and evaporating processes in main circuit.

Because the suction process through the supplementary inlets conducts quickly, it can be regarded as an adiabatic and isentropic process. The dimensionless flow rate through supplementary circuit α can be expressed as

$$\alpha = \frac{q_{mb}}{q_{mk}} = \frac{q_{mk} - q_{m0}}{q_{mk}} = \frac{\xi v_2 \left(p_m - p_2\right)}{RkT_6} \qquad (1)$$

Assuming that compressing process 1-2 is isentropic, the refrigerant enthalpy in state 2 is

$$h_2 = h_1 + w_{iq} \qquad (2)$$

Supposing the supplementary suction process is adiabatic, the refrigerant enthalpy in state 2’ is approximately

$$h_{2'} = \alpha h_6 + (1-\alpha) h_2 \qquad (3)$$

Assuming the compressing process 2′-3 is also isentropic, the refrigerant enthalpy in state 3 is

$$h_3 = h_{2'} + w_{ih} \qquad (4)$$

If the heat exchange process in the economizer is adiabatic, the refrigerant enthalpy in state 5 is

$$h_5 = h_4 - \alpha(h_6 - h_4)/(1-\alpha) \qquad (5)$$

On the refrigerant enthalpy in each state point of the cycle, the main performances of the improved ASHP can be easily determined as following:

cooling capacity

$$Q_k = q_{mk}\left(h_3 - h_4\right) \qquad (6)$$

heating capacity

$$Q_0 = q_{m0}\left(h_1 - h_5\right) \qquad (7)$$

compressing work

$$W_i = q_{mk} h_3 - q_{mb} h_6 - q_{m0} h_1 \qquad (8)$$

energy efficiency ratio for heating

$$EER_h = \eta_m \eta_{mo} Q_k / W_i \qquad (9)$$

energy efficiency ratio for cooling

$$EER_r = \eta_m \eta_{mo} Q_0 / W_i \qquad (10)$$

Experimental Apparatus

On this theory, the prototype ASHP were developed, and the dynamic performance of the ASHP were tested in the standard fixed differential enthalpy laboratory where each of the operating parameters such as the condensing pressure and the evaporating pressure can be easily controlled.

All the electric parameters of the ASHP such as the input voltage、 currency、and power were recorded by a data logger. Each temperature and pressure sensor was calibrated to reduce experimental uncertainties and was connected to a HP data logger. Table 2 shows the accuracy and tolerance of each sensor for measuring data. The uncertainties of the heating capacity and COP estimated by the analysis were approximately 2.6%.

Relevant parameters were recorded only if the fluctuation of every parameter was within 1% and the operating mode was stable after 1 hour. The ultimate parameters were the average statistics for the whole 10-minute period.

Table 2 :the accuracy and tolerance of each sensor for measure

Sensor	Number	Accuracy	Tolerance	Full scale	Model
temperature	15	±0.2℃	±0.2℃	—	T-type

					thermocouple
Pressure transducer	8	±0.2% of full scale	±0.2% of full scale	3447KPa	Model:
Currency acquisition unit	1	±0.2% of full scale	±0.2% of full scale	—	DZFC-1
Power acquisition unit	1	±0.2% of full scale	±0.2% of full scale	—	DZFC-1
Data logger	1	—	—	—	HP 34970A

Experiment Results and Discussions

Heating Capacity

The variations of heating capacity (Q_h) with the evaporating temperature (T_e) are shown in Fig. 3 when the condensing temperatures (T_c) were 45℃ and 48℃ respectively. The heating capacity decreased linearly with the decrease of the evaporating temperature, but rate of decrease was much slower than that of the conventional ASHP system. The heating capacity of the improved ASHP was approximately 5.5kW when the condensing temperature was 45℃and the evaporating temperature was –25℃, and this capacity would satisfy the heat requirement in North China where the lowest ambient temperature is around –15℃.

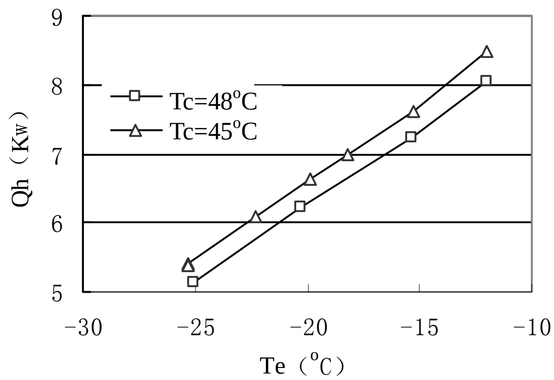

Fig.3 Heating Capacity Curves

The fact that the heating capacity decreased with the increase in the condensing temperature leads to the conclusion that reasonable decrease in supply water temperature can improve the performance of the improved ASHP.

Power Input

The variations of power input (P) with the evaporating temperature are shown in Fig. 4 when the condensing temperatures were 45℃ and 48℃. With the decrease of the evaporating temperature, the power input decreased very slowly, in contrast with a conventional ASHP, in which the power decreases linearly with the decrease in the evaporating temperature. For the conventional ASHP, decrease in the evaporating temperature means a reduction in the suction pressure of the main circuit, and the mass flow rate decreases so fast that compressor power

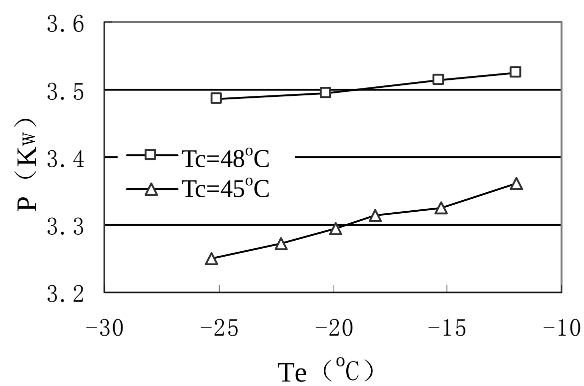

Fig. 4 Power Input Curves

consumption decreases greatly and causes a corresponding increase in the pressure ratio borne by the compressor. However, for the improved ASHP, though the flow rate through the main circuit decreased with the decrease in the ambient temperature, the flow rate through the supplementary circuit increased. Thus the total mass flow rate out of the compressor was kept nearly constant, and the corresponding power input decrease was slowed much slowly. The results showed that the ASHP with supplementary suction kept the power input nearly constant when it operated under conditions of low ambient temperature.

The power input also decreased more slowly with the increase of the condensing temperature. And the higher the condensing temperature is, the less the variation between the power input and the evaporating temperature.

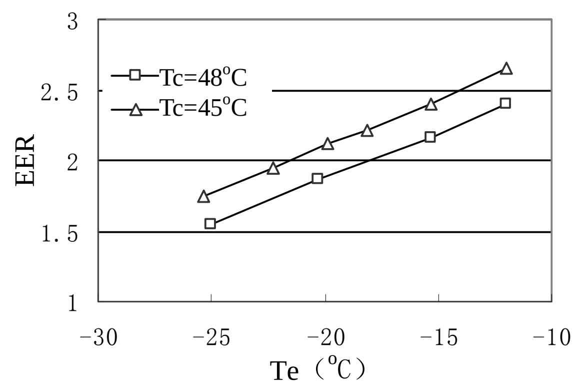

Fig. 5 EER Curves

Energy Efficiency Ratio

The variations of energy efficiency ratio (EER) with the evaporating temperature are shown in Fig. 5 when the condensing temperatures were 45℃ and 48℃.

The variation of EER with evaporating temperature was similar to the variation of heating capacity because the power input showed a gradual variation with the evaporating temperature.

Compressor Discharge Temperature

The variations of compressor discharge temperature (T_3) with the evaporating temperature are shown in Fig. 6 when the condensing temperatures were 45℃ and 48℃.

Compressor discharge temperature increases with the increase of condensing temperature and the decrease of the evaporating temperature, which is similar to that of a conventional ASHP. However, the discharge temperature was much steadier especially under the low ambient temperatures and was always less than 130°C, which greatly differs from a conventional ASHP. The discharge temperature of a conventional ASHP increases continually when the condensing temperature is equal to 45 ℃ and the evaporating temperature is lower than −15℃, which is the main reason why the compressor is often ruined when it continues running in the low ambient temperature.

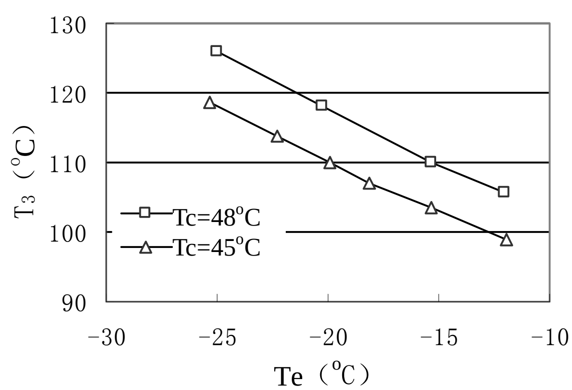

Fig. 6 Discharge Temperature Curves

Performance Comparison with Conventional System

If the solenoid valve A in the supplementary circuit was cut off, the prototype cycle degenerate into a conventional ASHP system. The dynamic performance comparison of the prototype ASHP operating under these two circumstances are shown in Table 3.

Compared with a conventional ASHP system, the heating capacity, power input and energy efficiency of the improved ASHP with the supplementary circuit increased 8.6%, 2.5% and 6.0% respectively when the condensing temperature was 45℃ and the evaporating temperature was –15°C. But these values became 5.5%, 1.6% and 3.7% respectively when the condensing temperature is 45°C and the evaporating temperature was –12°C. This demonstrates that the improvement of heating capacity and EER becomes inconspicuous with the increase of the evaporating temperature. The improvement is negligible when the evaporating temperature is higher than –10℃.

Compared with a conventional ASHP system, the discharge temperatures of the improved ASHP with supplementary circuit decreased 5°C and 6°C respectively under the operating conditions listed in Table 3. This demonstrates that the improved ASHP system prototype with the supplementary circuit on can noticeably decrease the discharge temperature. A low discharge temperature is vital to the compressor's reliability.

Table 3 Performances Comparison between Prototype and Conventional Heap Pump

Operating condition	Performance comparison	Q_h (W)	P (W)	*EER*	T_3 (℃)
T_e =–12℃ T_C =45℃	System 1	8038	3308	2.43	106
	System 2	8484	3360	2.52	101
	Absolute difference	446	63	0.09	－5
	Relative difference	5.5%	1.6%	3.7%	—
T_e =–15℃ T_C =45℃	System 1	6666	3045	2.19	116
	System 2	7239	3121	2.32	110
	Absolute difference	573	76	0.13	－6
	Relative difference	8.6%	2.5%	6.0%	—

Where, System 1----- the improved ASHP system with supplementary circuit off ;

System 2-------the improved ASHP system with supplementary circuit on;

Absolute difference----(value of system 2)-(value of system 1);

Relative difference----(absolute difference)/(value of system 2)

Conclusions

An improved ASHP was developed for cold regions. The dynamic performances of the prototype ASHP were tested in a laboratory test that could control all the parameters. Results show the following:

1)When the ambient temperature was near－15℃, the prototype ASHP ran smoothly for a long time with enough heat to satisfy the heating requirements in cold regions.

2) The heating capacity and power input of the ASHP will increase if the supplementary circuit is in use in low ambient temperatures, but the increase in the heating capacity is larger than that of the power input, so the heating EER is improved. The improvements become very small with increase in the evaporating temperature, and the effect can be negligible when the evaporating temperature is higher than –10℃.

3) The improved ASHP has a lower discharge temperature, and the discharge temperature is steady and remains below 130℃ for low ambient temperatures which is vital to the feasibility of the

ASHP used in the cold regions.

4) As is known, during the winter in the cold regions such as North China, most of the time the ambient temperature is round the 0℃,though sometime the ambient temperature maybe as low as -15 ℃,the whole sustaining time is very short . it is natural that the ideal ASHP developed for the cold regions should guarantee the EER of the regular working condition and can still satisfy the need for heating when the ambient temperature is very low, the improved ASHP can satisfy all these needs finely.

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