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A new design structure of large temperature difference heating system based on electric drive heat pump unit

Si Liu[1], Ming Huang[2]

[1]*Henan Polytechnic University, Civil Engineering College,*
Shanyang District, Jiaozuo City, Henan Province
[2]Henan Polytechnic University, Civil Engineering College,
Shanyang District, Jiaozuo City, Henan Province

Abstract: This paper presents a new design structure of large temperature difference heating system based on electric drive heat pump unit. In terms of electric heat pump unit, the evaporation temperature and evaporation pressure of the conventional single-stage electric drive heat pump unit are low, resulting in a low energy efficiency coefficient of the electric heat pump. Therefore, this paper proposes a large temperature difference heating electric heat pump driven by the compressor with two-stage gas-liquid separator. Through theoretical analysis, it presents that the heat pump unit has a higher energy efficiency coefficient than the conventional single-stage heat pump unit under the same working conditions. In the same working condition, the performance improvement rate of the heat pump system with the large temperature difference taking the thermoelectric drive heat pump unit can reach 12.8%.

Keywords: large temperature difference central heating system; Two-stage gas-liquid separator; Ejector; Energy efficiency coefficient

1. Introduction

With the acceleration of urbanization construction, living standards are also being improved and people expect a more comfortable indoor environment. The proportion of building energy consumption in our energy consumption structure continues to increase [1]. While developing new energy sources as a heat source, we should also pay attention to improving the utilization rate of heat energy. Heat pump heating technology is an energy-efficient technology that is widely used in production.

In earlier 1984, Japanese researcher studied a variable capacity heat pump system to save energy of air conditioner. They used a gas injection and a release compressor, and the heat pump had an energy savings of 15% compared with conventional system.[2] Ma et al. [3] researched the heat pump system coupled with scroll compressor with supplementary inlets theoretically, and found that the heating/cooling performance of the heat pump system could be even more improved by optimizing the design parameters. Eric et al. [4] developed a simplified simulation model in order to study the scroll compressor with supplementary inlets. Liu et al. [5] found a bypass mechanism mathematical model in scroll compressor and integrated into a simulation package to predict its performance.

The single-stage compression air-source heat pump cannot be operated efficiently and steadily for long periods in cold regions, in order to solve this problem, much research has been conducted [6].With the many possible solutions, there is a large temperature difference central heating system based on absorption heat pump unit. The design idea of this system is to use the absorption heat pump heat exchange unit in the secondary heat exchange station firstly, so that it could greatly reduce the return water temperature of the first-stage pipe network (to about 20°C).After the first-stage pipe network returns to the first station of heat exchange, the residual heat of the circulating cooling water is directly absorbed by the heat exchanger, and then the residual heat of the cooling water is multi-stage absorption through the multi-stage absorption heat pump, and finally the water is returned through the steam-water heat exchanger. The water is supplied after the temperature raise to the temperature of the water supply of first-stage pipe network. The above systems have the following technical defects in engineering applications:

(1) It not directly absorb the heat in the circulating cooling water through the conventional plate heat exchange equipment, because the primary network has a high return water temperature. It is difficult to greatly utilize the residual heat of the circulating cooling water of the power plant [7].

(2) What's more, there are many problems that the life of the absorption heat pump is low, the cost is high and the performance is unstable.

In order to solve the problems of absorption heat pumps, the research team has proposed a technical solution to replace absorption heat pumps with electric heat pumps in secondary heat exchange stations. A technical scheme of two-stage heat pumps with different evaporation and condensing pressures for heat extraction was proposed [8]. Nevertheless, the heat pump also has one defect: It will reduce the heat supply of two-stage heat pump units and the flexibility of optimal energy efficiency adjustment [9].

A large-temperature difference heat and electricity drive heat pump unit system is proposed. The equipment is with single-stage gas-liquid separator. Energy analysis based on the second law of thermo dynamic scan be used to evaluate the distribution of energy losses [9].This paper discusses an energy analysis of the heat pump system with ejector based on theoretical arithmetic. The energy losses of the components in the heat pump system with the injector, especially the ejector, which is an essential part that crucially affects the performance of the whole system, were investigated in detail in comparison to the heat pump [10].This work is expected to promote the further development and commercial application of the conventional heat pump system [11].

2. System model

2.1. Workflow

In this system, the gaseous refrigerant from the high-pressure compressor is condensed into a liquid refrigerant through a condenser. After expanding through the throttle valve 1, it enters the gas-liquid separator 1. The saturated liquid refrigerant separated by the gas-liquid separator 1 enters the nozzle as the primary fluid of the ejector.

The refrigerant becomes a high-speed and low-pressure gas-liquid mixed state through the ejector, and ejects the gaseous refrigerant from the outlet of the evaporator to perform mixing and boosting inside the ejector. The gas-liquid mixture finally formed flows out from the outlet of the expansion chamber and enters the gas-liquid separator 2.The saturated liquid refrigerant separated by the gas-liquid separator 2 is converted into a gas-liquid mixture through the throttle valve 2 and returned to the evaporator, where it is ejected into the ejector by a fluid.

Thegaseous refrigerant enters the low-pressure stage compressor. After mixing with the saturated vapor from the gas-liquid separator 1, it enters the high-pressure stage compressor, and then enters the condenser after pressurization to complete the cycle of the system.

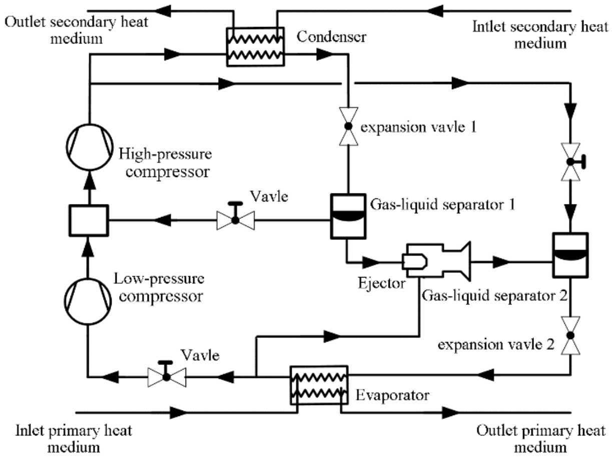

Figure 1: Unit working principle diagram

The primary heat medium releases heat to reach about 50℃through two-stage board. Then it enters the low-pressure stage evaporator to release heat to about 5°C and then return to the first station of heat exchange. The temperature of the primary heat medium between the two-stage evaporator is determined according to the principle of energy conservation and overall efficiency.

The secondary heat medium (about 45°C) return to the secondary heat exchange station. It enters the low-pressure stage condenser to absorb heat and heat up, then enters the high-pressure stage condenser and further absorbs heat to about 60°C.Next, it enters the two-stage plate to absorb heat for further and the temperature raises to about 70° C to supply. The temperature of the secondary heat medium between the two-stage condenser can also be determined according to the principle of energy conservation and overall efficiency.

2.2. System model establishment

In order to simplify the model, the following assumptions are made when modeling the system:

(1) The system is in a state of thermal equilibrium in stable operation and is treated in a one-dimensional process;

(2) The pressure loss of the working fluid flowing in the pipeline and the evaporator is negligible. The heat transfer process in each heat exchanger is set as a constant pressure process;

(3) Adiabatic non-isentropic compression is performed in the compressor.

(4) The compressor model only considers the state of intake and exhaust of the compressor, regardless of the actual actual compression process in the middle.

Based on the above assumptions, the calculation model is established. The basic thermal cycle analysis of the heat pump system is not repeated here. Only the main parameters reflecting the performance of the system are analyzed.

Based on conservation of energy, the total heat absorbed by the heat medium through the high pressure stage condenser and the low pressure stage condenser is

$$Q_C = Q_2 + Q_6 \quad (1)$$

Compressor power consumption is

$$w_{s.n} = \frac{n}{n-1} RT_1[1-(\frac{P_2}{P_1})^{\frac{n-1}{n}}] \quad (2)$$

The heat absorbed by the secondary heat medium is

$$Q^{'} = c_p m^{'}(t_6 - t_2) \qquad (3)$$

By definition, the coefficient of performance of the heat pump system is

$$C_{COP} = Q_C / W \quad (4)$$

The performance improvement rate of the system relative to the traditional two-stage compression system (TCRS) is:

$$COP_i = \frac{COP - COP^{'}}{COP^{'}} \quad (5)$$

In this formula, *COP'*is the coefficient of performance of the two-stage compression system (*TCRS*).

3. Calculation and analysis

Comparing the large temperature difference thermoelectric drive heat pump unit system with the traditional two-stage compression system, the performance of the system can be better analyzed.

The main physical parameters of the working fluid in the calculation process are shown in Table 1.

Table 1: Calculation parameters of design condition of heat pump system

Parameter	Numerical value
Primary heat medium inlet and outlet /°C	50/5
Secondary heat medium inlet and outlet temperature/℃	45/70
Heat source water mass flow /kg·s^{-1}	10
evaporation temperature /℃	-20 ~ 0
condensation temperature /℃	30 ~ 50
Intermediate temperature /℃	0 ~ 35

3.1. The effect of intermediate temperature on system performance

In the two-stage compression system, the intermediate pressure has a significant impact on the system performance. This paper uses the maximum system performance coefficient as the principle to determine the intermediate pressure. The intermediate pressure at this time is the optimum intermediate pressure, and the temperature corresponding to this pressure is the optimum intermediate temperature.

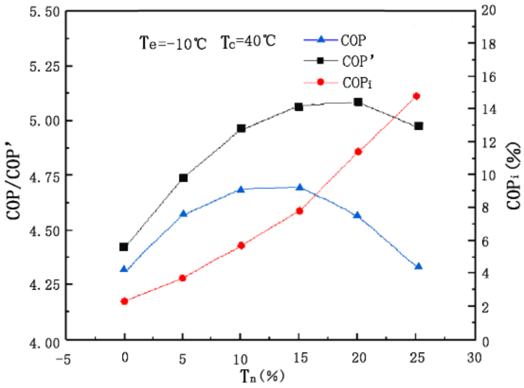

Figure 2: Variation of *COP*、*COP'* and COP_i with intermediate temperature at T_e= -10℃, T_c=40°C

Figure 2 shows the evaporation temperature T_e = -10 °C, the condensation temperature T_c = 40 °C, the system COP and *TCRS* system *COP'* and the system performance improvement rate COP_i with the intermediate temperature. It can be seen from the figure that as the intermediate temperature increases from -5°C to 25°C, both *COP* and *COP'* increase first and then decrease, and *COP* increases from 4.39 to 5.03 and then decreases to 4.94.The optimum intermediate temperature 20°C;The *COP'* increased from4.29 to 4.70 and then decreased to 4.28. The optimal intermediate temperature was 15 °C, and the *COP'* was reduced to a greater extent than *COP*. The *COP* is always large *COP'*, and the system performance coefficient improvement rate COP_i increases from 2.3%to 14.6%.Thisisbecause as the intermediate temperature increases, the compressor ratios of the two systems continue to increase, but in the new system, the injector also has a supercharging effect, so the boost ratio of the new compressor increases, and the compressor consumes less power. The rate of increase of the coefficient of performance of the system becomes larger. It can be seen that the injector-pressurized single-stage compression system has better economy.

3.2. Influence of evaporation temperature on system performance

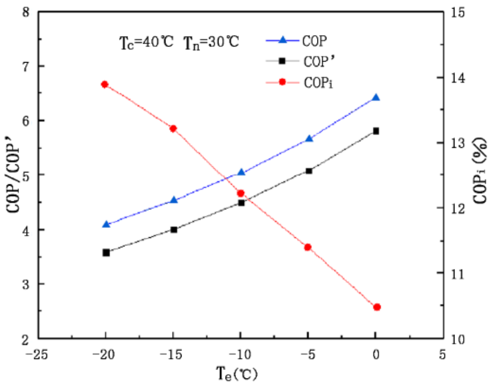

Figure 3: Variation of COP and COPi with evaporating temperature

Figure 3 shows the condensing temperature T_c = 40°C, the intermediate temperature T_n = 30°C, the system *COP* and *TCRS* system *COP'* and system performance improvement rate COP_i with evaporation temperature. As the evaporation temperature increased from -20°C to 0 °C, both *COP* and *COP'* increased gradually, *COP* increased from 4.1 to 6.43, *COP'* increased from 3.6 to 5.82, and COP was always greater than COP'. The system performance coefficient improvement rate COP_i decreased from 13.9% to 10.5%. This is because when the condensation temperature is constant, the higher the evaporation temperature, the smaller the temperature difference between the two, the smaller the expansion work recovered by the injector, and the lower the coefficient of performance of the system.

3.3. Influence of condensation temperature on system performance

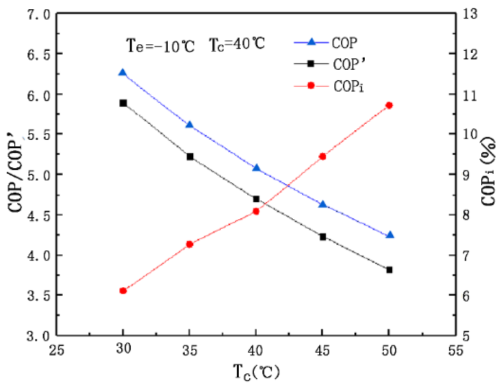

Figure 4: Variation of COP and COPi with condensing temperature

Figure 4 shows the *COP* and TCRS system *COP'* and the system performance improvement rate COP_i as a function of condensing temperature for the evaporation temperature T_e=-10°C and the intermediate temperature T_n=20°C. As the condensing temperature increased from 35 °C to 50 °C, both *COP* and *COP'* gradually decreased, *COP* decreased from 6.18 to 4.27, *COP'* decreased from 5.9 to 3.79, and *COP'* decreased to a greater extent than *COP*. The system performance coefficient improvement rate COP_i increased from 6.1% to 10.73%. This is because when the evaporation temperature is constant, the condensation temperature increases, the injector boost ratio becomes larger, the power consumption of the compressor decreases, and the coefficient of performance of the system is more obvious than the conventional two-stage compression.

4. Conclusion

This paper proposes a "large temperature difference thermo electric drive heat pump unit system "for "single stage compressor configuration two-stage evaporation and condensation equipment". A theoretical model was established based on the above system to analyze its thermodynamic properties.

(1) The single-compressor drive double-evaporator and double-condenser structure, the energy efficiency of the unit will be greatly improved compared with the conventional unit. The effects of intermediate temperature, evaporation temperature and condensation temperature on system performance were investigated. When the intermediate temperature rises, the *COP* of the heat pump system first increases and then decreases, and there is an optimal intermediate temperature.

(2) The ejector is used to reduce the condensing pressure of the low-pressure stage condenser. The evaporation pressure of the low-pressure stage evaporator is increased. The intermediate temperature of the primary heat medium and the secondary heat medium can be flexibly adjusted to improve the overall efficiency of the heat pump unit. When the evaporation temperature increases, the heat pump system *COP* increases and the injector boost ratio decreases. When the condensation temperature increases, the heat pump system *COP* decreases and the injector boost ratio increases.

(3) Achieving high efficiency and large temperature difference to take heat, greatly reducing the temperature of the primary heat medium return water. It is beneficial to the efficient and full utilization of the waste heat of

the power plant. In the same working condition, the performance improvement rate of the heat pump system with the large temperature difference taking the thermoelectric drive heat pump unit can reach 12.8%.

5. References

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