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OPTIMIZATION ANALYSIS AND EXPERIMENTAL INVESTIGATION ON HIGH TEMPERATURE CASCADE HEAT PUMP WITH VAPOR INJECTION

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ABSTRACT

High temperature heat pump has great development potential for industrial heat because it is energy-saving and environmentally friendly. This paper focuses on the optimization analysis and experimental investigation of an air source cascade heat pump (ASCHP) to realize 130℃ heat supply under the ambient temperature of 20℃, with refrigerant R245fa and R410A in the two stages respectively. The high stage is designed with vapor injection (VI). The optimal configuration of the intermediate temperature and VI branch temperature is obtained by optimization analysis. An ASCHP unit is developed and the experimental results match the optimized results well. The coefficient of the VI model is deduced according to the experimental result. The coefficient of performance (COP) of the developed heat pump unit is 1.39.

Keywords: air source cascade heat pump, vapor injection, optimization analysis

NOMENCLATURE

Abbreviations

ASCHP	air source cascade heat pump
VI	vapor injection
EEV	electronic expansion valve
HP	heat pump
COP	coefficient of performance
COP_s	COP of the system
COP_h	COP of the high stage
COP_l	COP of the low stage

Symbols

Q_h	heating capacity of high stage(kW)
Q_l	heating capacity of low stage(kW)
P_h	input power of high stage compressor (kW)
P_l	input power of low stage compressor (kW)
T	temperature of working fluid (℃)
P	absolute pressure of working fluid (MPa)
κ	the adiabatic index
V_1	the chamber volume when the suction port is fully closed(m^3)
V_2	the chamber volume when the injection port is fully closed(m^3)
T_c	condensing temperature of high stage(℃)
T_e	evaporating temperature of low stage(℃)
T_m	evaporating temperature of high stage(℃)
T_i	vapor injection temperature(℃)
P_m	evaporating pressure of high stage(℃)
P_i	vapor injection pressure(℃)
α	VI ratio
m_{inj}	mass flow rate of VI branch(kg/h)
m_{suc}	mass flow rate of main branch(kg/h)
V_h	high stage compressor volume(cc/r)
V_l	low stage compressor volume(cc/r)
ΔT_i	minimum temperature difference of injection heat exchanger(℃)
ΔT_{set}	designing temperature difference of injection heat exchanger(℃)
ΔT_h	superheating degree(℃)
ΔT_c	subcooling degree(℃)

1. INTRODUCTION

Heat pump has been put into practice of heating for decades due to its high efficiency [1]. Recently, the demand for high temperature heat pump has become more and more urgent in different kinds of industrial applications like electroplating, printing and textile for efficient energy utilization and environmental protection [2]. Liu et al. [3] built a single stage heat pump using a new refrigerant HTR01 and obtained 90℃ water based on the geothermal source of 50℃ water. This is the maximum reported temperature of single stage heat pump. Because the compression efficiency and discharge temperature will get worse with the increase of compression ratio, it is very difficult for a single stage heat pump to realize higher temperature difference.

To get higher supply heat temperature, cascade heat pump is an effective system. It is a combination of two

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heat pump (HP) cycles with an intermediate heat exchanger absorbing the heat released from the low stage cycle. On the other hand, vapor injection (VI) is an effective way for high temperature difference heat pump to increase the heating capacity and decrease the compressor discharge temperature [4-6]. Therefore, cascade heat pump (CHP) with vapor injection (VI) is very prospective for high temperature heat supply. However, because this system is coupled by two branches, its heating performance is influenced by the working conditions of branches greatly.

In this study, an optimal design program for cascade heat pump with vapor injection is set up and the heating performance of an air source heat pump (ASCHP) with VI is analyzed under different intermediate temperature and VI branch temperature. An ASCHP unit is developed based on the optimal configuration and experimentally researched aiming at presenting effective theoretical support for the application of ASCHP with VI.

2. THEORETICAL ANALYSIS

2.1 Principle of system

Fig 1 shows the system diagram of the ASCHP with VI. The system contains an oil cycle for heat supply, a R245fa cycle with VI and a R410A cycle. The oil is circulated by a pump and works as the heating consumer.

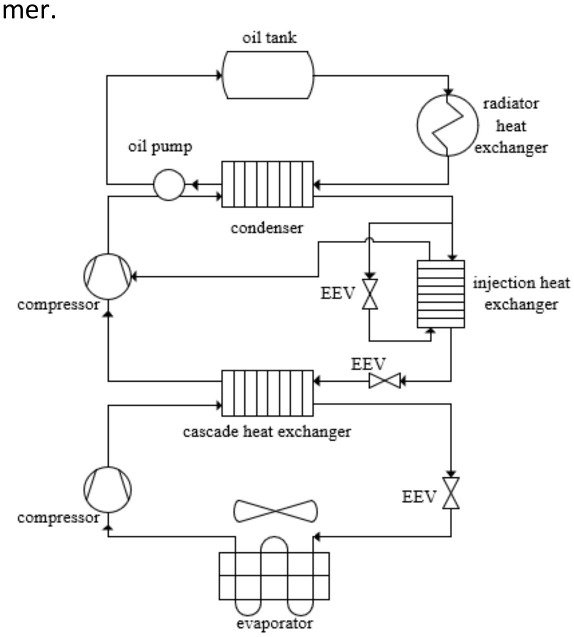


Fig 1 system diagram of ASCHP

The basic principle of the ASCHP system is presented in the Temperature-Entropy diagram in Fig 2. The evaporating temperature of high stage is defined as the intermediate temperature of system and expressed as T_m. The temperature difference between the evaporating and condensing temperature is divided into three parts by the intermediate temperature T_m and the VI temperature T_i in the high stage. The temperature difference ratios and system parameters are defined as follows.

$$r_1 = \Delta T_1 / \Delta T \tag{1}$$

$$r_2 = \Delta T_2 / \Delta T_1 \tag{2}$$

$$COP_s = Q_h / (P_l + P_h) \tag{3}$$

$$COP_h = Q_h / P_h \tag{4}$$

$$COP_l = Q_l / P_l \tag{5}$$

$$\alpha = m_{inj} / m_{suc} \tag{6}$$

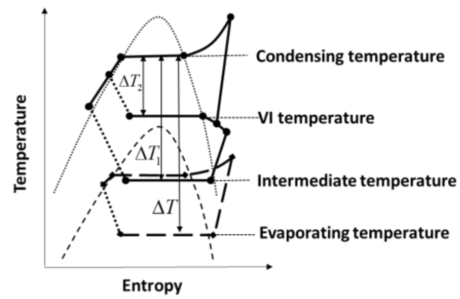


Fig 2 Temperature elevation of system

2.2 Programming flow chart

The main task of the optimization analysis is aimed to allocate the temperature difference ΔT_1 and ΔT_2 properly under the designed temperature elevation.

An optimization analysis modeling is set up and programming based on MATLAB as shown in Fig 3. The HP cycle are calculated at the constant subcooling and superheating degree. For the high stage, the VI model is quoted from the model of Han et al [6], as Fig 4 shows. The dynamic process of vapor injection is simplified to transient process, including three stages: the first stage compression process 1-2, the injection process 2, 5-3 and the second stage compression process 3-4. The first and second stage compression process could be considered as adiabatic compression. The injection process is assumed as an adiabatic, isovolumetric and transient mixing process. Based on the first law of thermal dynamics, the VI ratio is deduced as Eq(7)-Eq(9). For a special compressor, the ratio of V_2 to V_1 is constant, and then VI ratio α is relevant to the states of injection port point 5 and suction point 1.

$$\alpha = k_1 \, p_5 / p_1 - k_2 \tag{7}$$

Where,

$$k_1 = \frac{P_3}{P_5} \frac{V_2}{V_1} \frac{T_1}{\kappa T_5} \tag{8}$$

$$k_2 = \left(\frac{V_2}{V_1}\right)^{1-\kappa} \frac{T_1}{\kappa T_5} \tag{9}$$

Based on the VI compressor model above, the analysis of R245fa cycle are determined by the superheating degree, subcooling degree and the given r_1 and r_2. The mass flow rate of high stage is determined by the heating demand, and the mass flow rate of low stage is constrained by the heating capacity that matches with the cooling capacity of high stage. Then, the correspnding compressors suited for the optimal refrigerant flow rates are chosen.

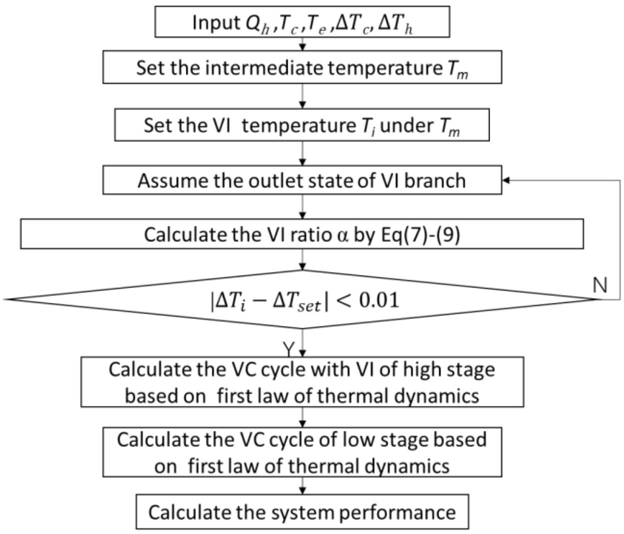


Fig 3 Programming diagram

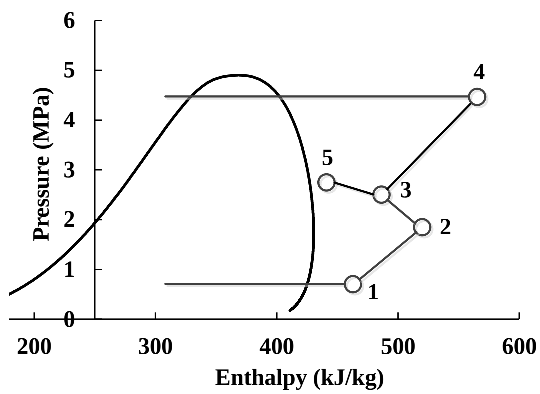
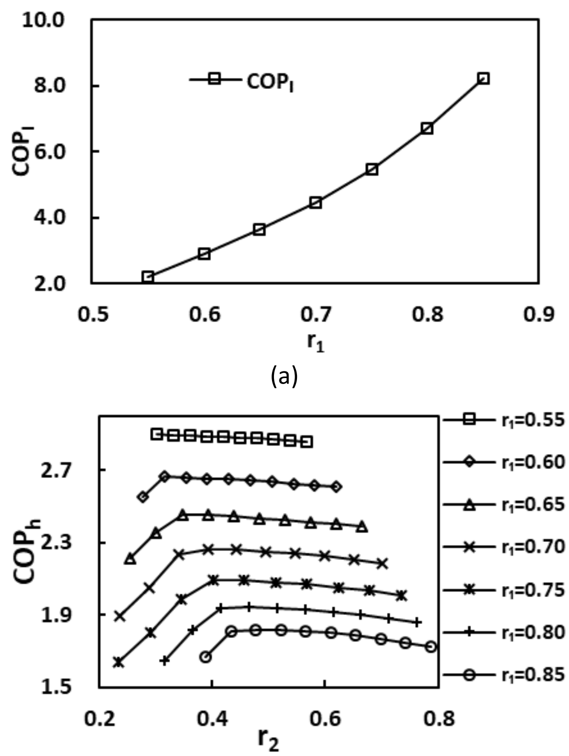


Fig 4 The diagram of simplified compression process with VI

2.3 Optimization analysis

The system applies an R245fa cycle with VI as the high stage and an R410A cycle as the low stage. The condensation temperature of the high stage and the evaporation temperature of the low stage are designed at 130℃ and 10℃, respectively. The efficiencies of the compressors at different evaporating/condensing temperature are quoted from the operating data of actual commercial scroll compressor and VI compressor under different condition. The heating performance of the two stages as well as the system can be analyzed by this program.

Fig 5(a) shows that the COP_l increases with r_1 as the decreased pressure difference saves the compressor power consumption. Fig 5(b) shows the changes of COP_h for various temperature ratio r_1 and r_2. The reason for the decreased tendency of COP_h with increasing r_1 seems to be related to the increased pressure difference between the evaporating pressure and condensing pressure of high stage. The increased pressure difference rises the compressor power consumption, so the COP_h decreases. Under the constant r_1, the pressure difference keeps constant and there exists an optimal r_2 because the VI saves the compressing work, but the compression efficiency deteriorates with increased α. For a small r_1, both the temperature difference of high stage and VI ratio are small and the compression efficiency is not too bad, then the change of COP_h is monotonous.



(a)

(b)

Fig 5 Performance of high stage and low stage

The system's heating COP varies as depicted in Fig 6. The change of r_2 has little influence on the COP when r_1 is less than 0.6. In this case, the high stage has already exhibited a satisfied performance at small temperature difference. The advantage of VI in high stage can't be fully utilized and the COP of system is mainly dominated by the low stage. For a larger r_1, the large temperature difference of high stage will exert the advantage of VI for the system. However, it seems that the large injection ratio can dominantly worsen the performance of system. Therefore, the well allocated temperature differences show the optimal COP of system. The best design COP is 1.57 when r_1 is 0.75 and r_2 is 0.62.

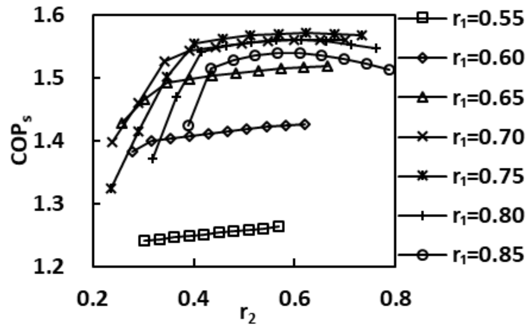
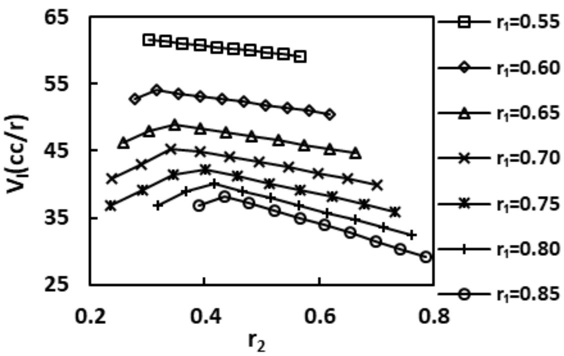
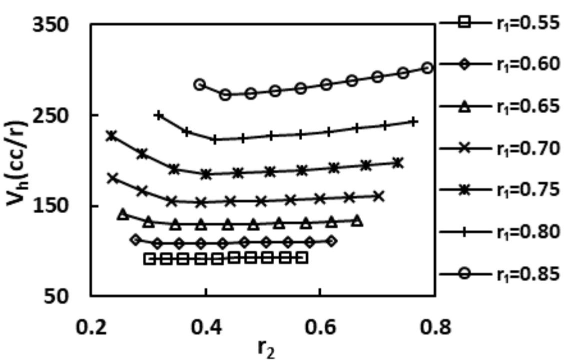


Fig 6 COP of the system

The corresponding volumes of compressors under different conditions are shown in Fig.7.



(a)



(b)

Fig 7 Required volumes of compressors

The reason for the decreased volume of low stage with the increased r_1 is due to the increased COP_l. The tendency of V_l under constant r_1 can be explained by the heating capacity of low stage matched with the cooling capacity of high stage, because VI is beneficial to enhance not only the heating capacity but also cooling capacity. The corresponding volume of high stage compressor almost shows a reverse tendency with the COP_h, which results from the constraint heating capacity. A larger volume of compressor is required to satisfy the heating capacity as the suction density decreases with increased r_1. According the optimal r_1 and r_2, a 200cc/R VI compressor of R245fa and a 40cc/R compressor of R410A are chosen to meet the requiring heating capacity.

3. EXPERIMENTAL RESULT AND DISCUSSION

3.1 Experimental bench

Fig 8 Prototype of ASCHP unit

The prototype designed according to the optimal parameters is shown in Fig 8. The oil is heated to 130℃ from 100℃ and the rated heating capacity of the ASCHP unit is 13 kW. Transducers are applied in the prototype to gain the operating state of system. Data acquisition is detected by the Agilent instrument. The uncertainties of all the instruments are within 0.5%. The test conditions are listed in the Tab 1.

Tab 1 Test conditions

Outdoor dry bulb temperature/℃	20
Oil inlet temperature/℃	100
Oil outlet temperature/℃	130
Opening ratio of EEV1	70
Opening ratio of EEV2	30,35,40,45,50

3.2 Experimental results and discussion

Fig 9 shows the variations of r_2 and VI ratio according to the opening ratio of EEV2 in the experiment. The increase in the opening ratio of EEV2 leads to the increasement of the VI pressure/temperature. Then the VI ratio α increases and r_2 decreases with the opening ratio of EEV1 keeping constant.

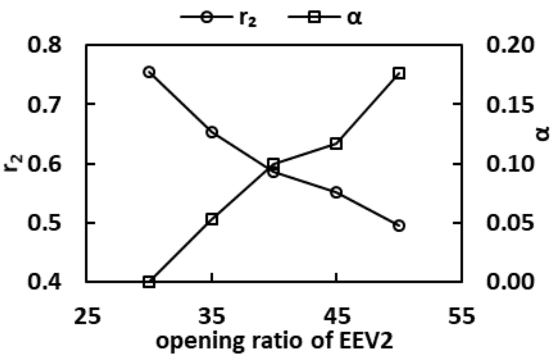


Fig 9 The changes of r_2 and α with opening ratio of EEV2

Fig 10 shows the p-h diagram of R245fa cycle under different r_2. From the purple line to black line, r_2 increases from 0.49 to 0.75.

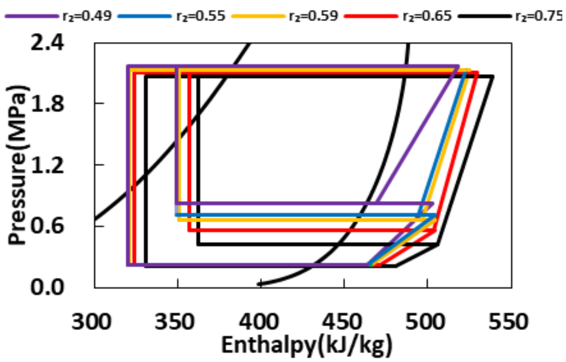
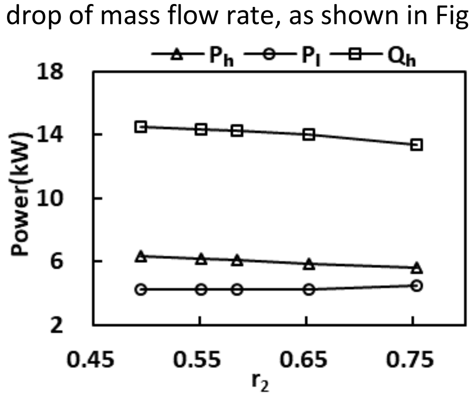


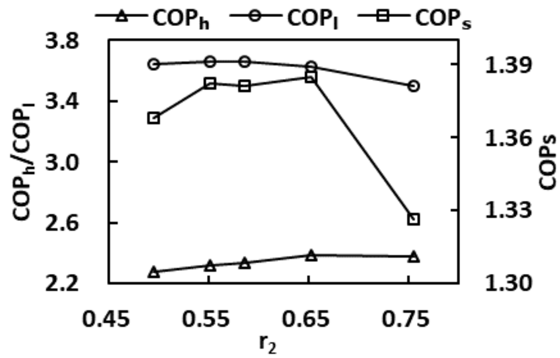
Fig 10 The p-h diagram of high stage under different r_2

The decrease of both suction temperature and superheating degree of injection leads to the decrease of discharge temperature, which efficiently avoids the compressor's overheating. The growing α results in a better cooling to the main branch, then the inlet subcooling degree of EEV1 increases and the enthalpy of evaporator's inlet decreases. It is important for the R245fa refrigerant to fully absorb the heat from low stage and cool down the R410a refrigerant in the condenser of low stage.

From Fig 11(a), the system heating capacity has a little decline with the increasing r_2. It can be explained by the drop of mass flow rate, as shown in Fig 12 (a).

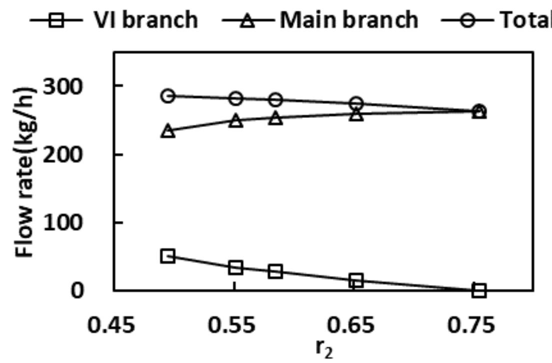


(a)

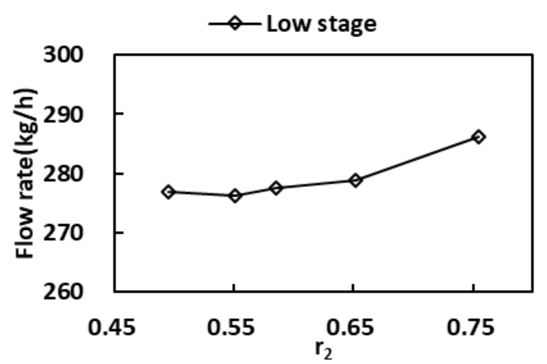


(b)

Fig 11 Heating performance under different r_2



(a) High stage



(b) Low stage

Fig 12 Mass flow rate changes with r_2

The high stage performance rises as Fig 11(b) shows, because the increasement of r_2 causes the improvement of compressor efficiency, as shown in Fig 14. Due to the balance between the decreased compressing work and the deteriorated compressor efficiency, the improvement of COP_h is not so dramatic, but the high stage shows a slightly better performance at a larger r_2.

The performance of low stage shows a reverse trend with the high stage. It seems that the increasing suction temperature of high stage can't lower the discharge temperature of low stage any more with increasing r_2, then the efficiency of low compressor gets worse and the mass flow rate increases as a compensation for the heating capacity matched with the cooling capacity of high stage. When r_2 is 0.75, the VI ratio α is approximately zero, or in other words, the prototype works more like a cascade system without VI, and the COP_s is worse than the other VI conditions.

The mass flow rate of main branch increases and the VI branch decreases as Fig 12(b) shows, as a result, the total mass flow of high stage drops a little. This is because the flow resistance of R245fa increases with the decreasing opening ratio of EEV2.

As shown in Fig 13, the injection ratio α is linear to the pressure ratio P_i / P_m and the coefficient k_1 and k_2 can be fitted according to the experimental data. In this study, k_1 and k_2 are 0.1023 and -0.2068. Then the injection ratio α can be calculated by the measured pressure P_i and P_m in the process of compressor control.

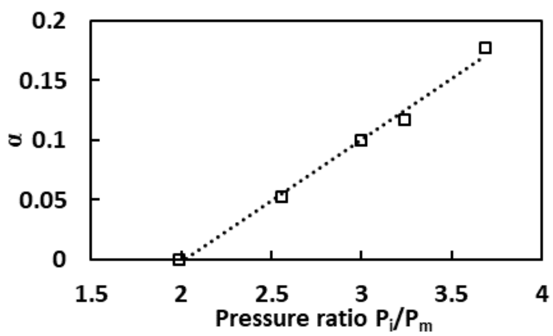


Fig 13 The relationship between injection ratio and pressure ratio

The best heating performance is 1.39 in the experimental test when r_1 is 0.78 and r_2 is 0.65, which shows a good accuracy of the optimal designing parameters. The difference between the best COP of design and experiment seems that leakage and heat loss are not considered in the design process

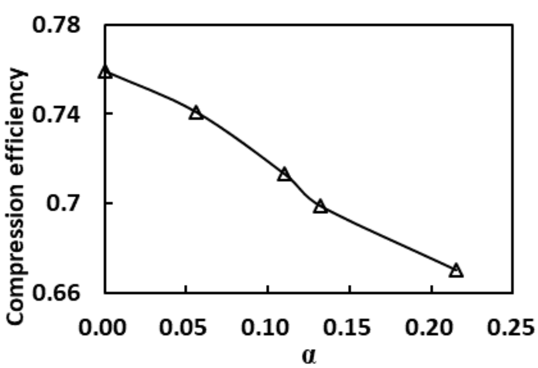


Fig 14 Compression efficiency of high stage

4. CONCLUSIONS

In this study, an air source cascade heat pump (ASCHP) with VI system is presented for high temperature heat supply. The system performance under different intermediate temperature and VI temperature is analyzed to get optimal stage configuration. An ASCHP unit is developed and researched experimentally to analyze the heating performance under different VI conditions. The main conclusions go as follows:

(1) The optimal temperature allocation of intermediate temperature and VI temperature is obtained, and the design value of r_1 and r_2 are 0.75 and 0.62.

(2) The coefficient of the relationship between injection ratio and pressure ratio is obtained based on the experimental results, which can be used in the compressor control strategy.

(3) The best operating parameters show a good accuracy of the optimal design and the best performance is up to 1.39 on the condition that r_1 and r_2 are 0.78 and 0.65 respectively. The application of VI has an obvious improvement on the performance of low stage, which effectively increases the COP_s and heating capacity at the cost of the compression efficiency of high stage.

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