STUDY ON HEAT PUMP CYCLE FOR COLD REGIONS WITH TWO-STAGE COMPRESSION IN ONE SCROLL COMPRESSOR

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ABSTRACT

A traditional heat pump with one-stage compression can not meet the high pressure ratio caused by the very low outdoor temperature. A new heat pump cycle is development for cold regions with two-stage compression in one scroll compressor with a supplementary circuit. Both he calculation and the experiment results show that the new heat pump can not only realize enough heating capacity with high EER but also improve the heat pump cycle by decreasing the discharge temperature even under the low evaporating temperature of -25^{o}C. The developed heat pump system can be used for heating in cold regions.

1. INTRODUCTION

With improvement of the people’s living standard and development of national economy, heat pump for air conditioning has been widely used. As the ambient temperature goes down, the heat capacity and the energy efficiency also decrease. Especially when the ambient temperature is very low in cold regions, the difference between condensing pressure and evaporating pressure is very high. One result caused by the low ambient temperature is that the heat capacity and EER is very low and the heat pump can not provide enough heat with high efficiency. Another result caused by the high pressure-ratio is that the discharging temperature is very high can the heat pump will be closed to prevent the compressor from being destroyed [1].

Two-stage compression is a quite well known alternative to improve refrigeration cycles [2]. It is actually currently used for big installations all over the world. In this case it usually uses two different compressors adapted to operate together at generally one working point. The goal is optimize the cycle efficiency, lower the compressor exhaust temperature and to boost application of heat pumps ([3],[4],[5],[6]) A few papers available in the open scientific literature treat two stage compression in one and only reciprocating [6] or scroll ([7][8][9]) compressor. As a result, they could lower exhaust temperature from 10 to 20^{o}C, and improve the COP up to 10%[7].

Most of the two-stage compression in one compressor has two evaporators as shown in Figure 1. There is still one problem how evaporator 1 can get heat to evaporate the refrigerant in it and there needs an additional heater such as a gas or oil burner ([10],[11]).

A new two-stage compression in one scroll compressor with an economizer is presented in this paper. The economizer is used as an evaporator before the injection point and also as a subcooler after the condenser.

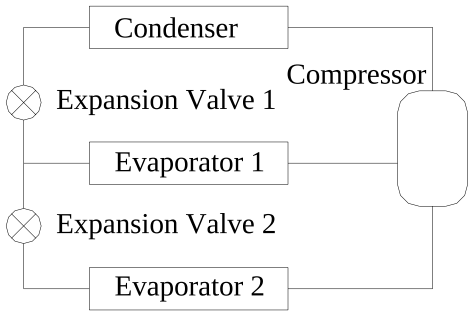

Figure 1 Current two-stage compression in one compressor

2. NEW HEAT PUMP CYCLE WITH ECONOMIZER

The working principle of the heat pump cycle for air conditioning in cold regions is shown in Figure 2. The superheated refrigerant vapor (point 3') with high pressure and temperature discharged by scroll compressor with supplementary inlets becomes subcooled liquid (point 4) when it flows through condenser. The liquid with high pressure from the condenser is divided two parts. One part is depressurized (point 4') by electronic expansion valve A, evaporates (pint 6) in the subcooler and flows back to the compressor through the supplementary pipe. The other part is far subcooled (point 5) in the subcooler and depressurized (point 5') by electronic expansion valve B, then it evaporates (point 1) in the evaporator, flows back to the compressor and behind and is primarily pressed (point 2) by the compressor. Then the two parts of refrigerant mix (point 6) in the compressor and is pressed to the vapor with high pressure and temperature (point 3') in the compressor. If there is not the refrigerant from the supplementary inlet, the refrigerant will be press to point 3 where the refrigerant temperature is higher. The thermodynamic cycle of the refrigeration system with supplementary pipe is shown in

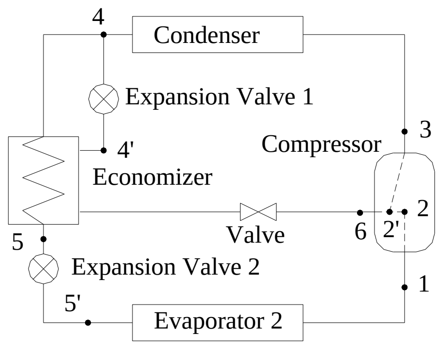

Figure 2 New two-stage compression in one compressor

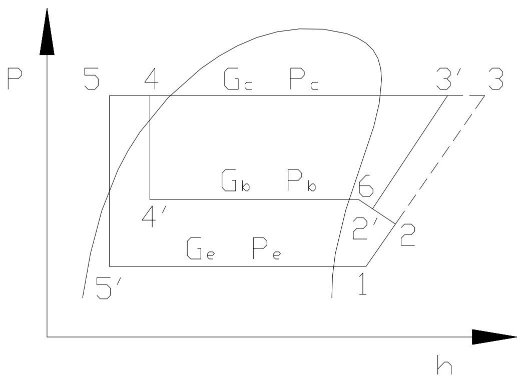

Figure 3 The cycle in *lg*p~h diagram

pressure-enthalpy diagram Figure 3.

Comparing with the conventional two-stage compression heat pump system, the new heat pump cycle for air conditioning in cold regions has remarkable characteristics as follows.

1)There is a supplementary pipe connected parallelly between condenser and compressor. The refrigerant pressure in the subcooler is between evaporation pressure (Pe) and condensation pressure (Pc), and is called supplementary pressure.

2)The scroll compressor has a supplementary inlet, and there is heat exchange between the two parts of the refrigerant. The heat exchanger evaporates the refrigerant in thesupplementary pipe and subcools the refrigerant out of the condensor. And there is an electronic expansion valve to control the mass flow rate of refrigerant in the supplementary pipe by the superheat degree at the compressor supplementary inlet.

3) The prototype can operate as conventional heat pump by turning off supplementary circuit with the valve. If the supplementary circuit turns on, the heat pump cycle operate according to economizer cycle. But if the economizer circuit turns off, the heat pump cycle operates in common operating condition. This valve enlarges operating range of heat pump in low outside temperature.

4) The system does not need additional heat source comparing with troditional two-stage compression in one compressor which needs a burner or electric heater.

3. ANALYSIS OF THE CYCLE

The heat pump cycle is shown in Figure 2. Process 1-2-2′-3’ is the supplementary and compression process in compressor. Process 1-2 is the compression process of refrigerant from evaporator before supplementary, then it is mixed with the refrigerant in state point 6 from supplementary pipe, the state of point 2’ is the state after supplementary. Process 2′-3’ is compression process after supplementary. Process 3-4 is condensing process, and process 4-4′ is throttling process in supplementary pipe. Process 4-6’ is the evaporating process of refrigerant in supplentary pipe and process 4′-6 is subcooling process of refrigerant in main cycle in subcooler. Process 5-5′ and 5′-1 are the throttling and evaporating processes of refrigerant in main cycle.

Because the supplementary process is quick, it can be regarded as adiabatic and isochoric. The supplementary rate α can be expressed as equation (1).

$$\alpha = \frac{G_b}{G_c} = \frac{G_c - G_e}{G_c} = \frac{\xi v_2\left(P_b - P_2\right)}{R\,k\,T_6} \qquad (1)$$

where

G_b——mass flow rate of refrigerant in supplementary pipe (kg/s）

G_c——mass flow rate of refrigerant in condenser（kg/s）
G_e——mass flow rate of refrigerant in evaporator（kg/s）
ξ——losing rate of pressure by supplementary
v_2——specific volume of refrigerant on point 2（m^3/kg）
P_b——supplementary pressure（kPa）
P_2——refrigerant pressure on point 2（kPa）
R——gas constant of refrigerant [kJ/（kg·K）]
k——adiabatic parameter of the refrigerant
T_6——refrigerant temperature on point 6（K）

Take process 1-2 as isentropic compression, the refrigerant enthalpy on point 2 is

$$h_2 = h_1 + w_{iq} \tag{2}$$

where

w_{iq}——isentropic compression work of process 1-2（kJ/kg）

If the supplementary process is adiabatic, the refrigerant enthalpy on point 2' is

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

take process 2′-3 as isentropic compression also, the refrigerant enthalpy on point 3 is

$$h_3 = h_{2'} + w_{ih} \tag{4}$$

where

w_{ih}——isentropic compression work of process 2′-3（kJ/kg）

If the heat exchanged process in subcooler is adiabatic from the environment, the refrigerant enthalpy on point 5 is

$$h_5 = h_4 - \alpha(h_6 - h_4)/(1-\alpha) \tag{5}$$

Based on the refrigerant enthalpy in each point of the cycle, the parameter of the system be expressed as

heating capacity: $$Q_h = G_c(h_{3'} - h_4) \tag{6}$$

cooling capacity: $$Q_k = G_e(h_1 - h_{5'}) \tag{7}$$

compression work: $$W_i = G_c h_{3'} - G_b h_6 - G_e h_1 \tag{8}$$

energy efficiency rate of heat pump mode: $EER_h = Q_h / W_i$ （9）
energy efficiency rate of refrigeration mode: $EER_r = Q_k / W_i$ （10）

4. ANALYSIS OF THE SUPPLENTARY PRESSURE

The discharge temperature is shown in Figure 5 when the superheat degree at the compressor inlet is keep at 5°C. The discharge temperature rises as the rising of the condensing temperature and the falling of the evaporating temperature. When the discharge temperature is higher than 130°C, the compressor must be shut down. From Figure 4 we can find that the discharge temperature is often higher than 130°C when the outdoor temperature is lower than –15°C in the cold regions.

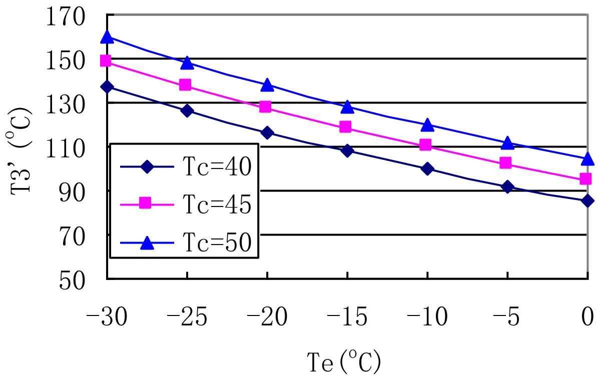

Figure 4 Discharge temperature curve

If the Electronic Expansion Valve A control the refrigerant flowrate to keep the compressor from wet compression, we can take the superheat degree at point 2' as 2°C. The influence of the supplementary pressure to the system is shown in Figure 2 with 5°C subcooling degree at outlet of the condenser (point 4). From Figure 5, we find that both the heating capacity and the discharge temperature drop as the rising of the supplementary pressure. So the supplementary must get an optimal range to keep the low discharge temperature and the high heating capacity. Compared Figure 5 with Figure 4, the results show that the discharge temperature can be decreased by 10~20°C at least.

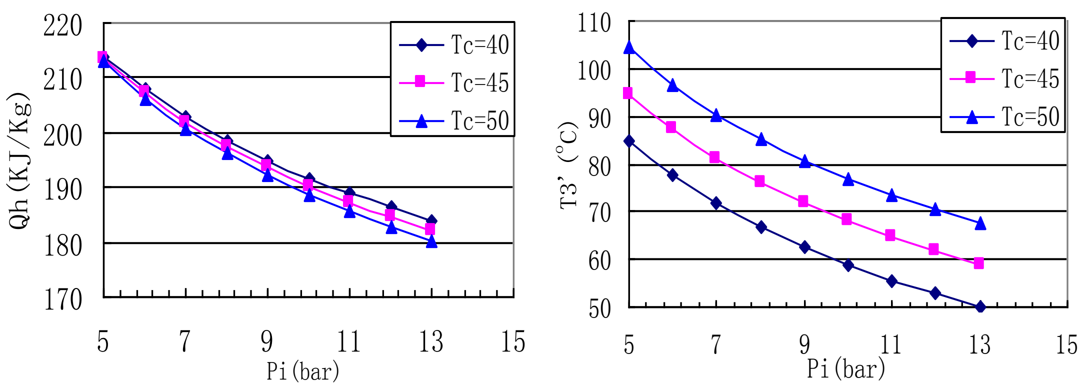

Figure 5 Influence of the supplementary pressure

4. EXPERIMENT RESULTS AND DISCUSSIONS

4.1 Heating Capacity

The variations of heating capacity (Q_h) with evaporation temperature (Te) are shown in Figure 6 when the condensation temperature (Tc) are 45°C and 48°C.

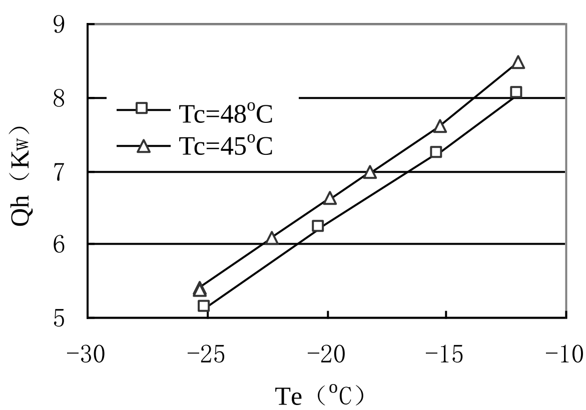

Figure 6 Heating Capacity Curves

The heating capacity decreases with the decreasing of evaporation temperature and the rising of condensation temperature. But even when the evaporation temperature is –25°C, the capacity is still higher than 5.0KW which is much better than the traditional air source heat pump systems.

4.2 Power Input

The variations of power input (P) with evaporation temperature are shown in Figure 5 when the condensing temperature is 45°C and 48°C.

The power input has a little decreasing as the decreasing of evaporation temperature. Because there is a supplementary pipe in the new system, the refrigerant volume through the compressor has a little change when the difference between the condensation temperature and the evaporation temperature.

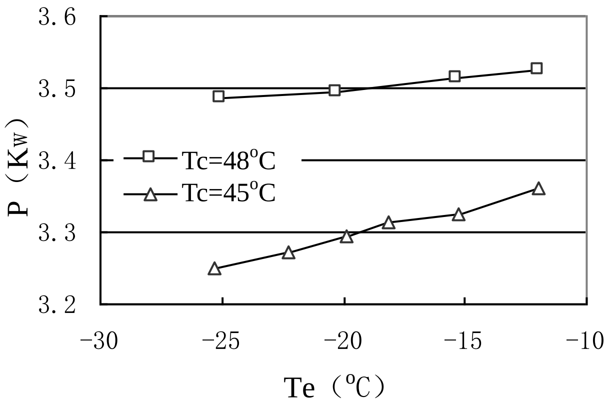

Figure 7 Power Input Curves

Moreover, the power input decreases a little as the rising of condensation temperature. And the higher is the condensation temperature, the less the power input varies as the evaporation temperature changes.

4.3 Energy Efficiency Ratio

The variations of Energy Efficiency Ratio (EER) with evaporation temperature are shown in Figure 8 when the condensing temperature is 45°C and 48°C.

Because the power input varies a little as the evaporation temperature changes, the variation of EER is same as that of heating capacity.

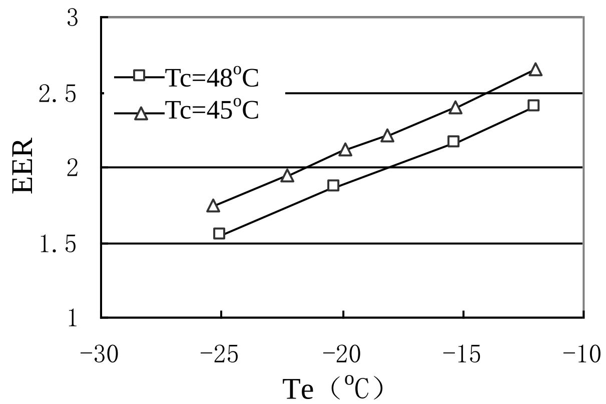

Figure 8 EER Curves

4.4 Compressor Discharge Temperature

The variations of compressor discharge temperature (T_3) with evaporation temperature are shown in Figure 9 when the condensing temperature is 45°C and 48°C.

Compressor discharge temperature rises as the rising of condensation temperature and the falling of evaporation temperature. Compressor discharge temperature is stable at each operating condition and less than 130^{o}C. But in the traditional heat pump systems, compressor discharge temperature is not stable and rises continually which is the main reason for spoiling of the compressor.

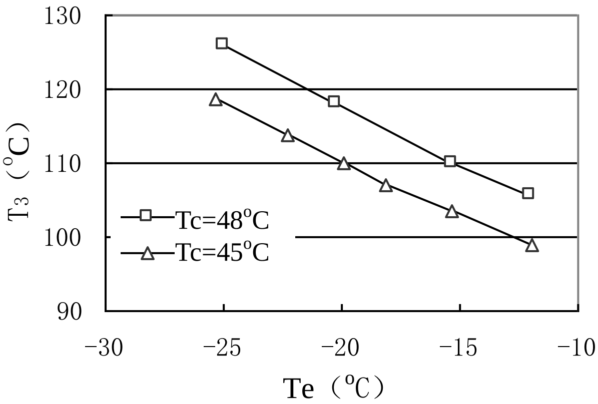

Figure 9 Discharge Temperature Curves

5. DISCUSSIONS

The compressor discharge temperature in the experimental data is higher than that in the calculation. This is mainly influenced by the compressor efficiency. A part of input work to the compressor is turned into heat directly which leads to the rising of the compressor temperature and the rising of the compressor discharge temperature.

5. CONCULSIONS

A new heat pump cycle with two-stage compression in one stroll compressor by a supplementary pipe is presented in this work. From the experiment results and analysis, we can find that:

1) The new heat pump can operate steadily and supply enough heat at the outdoor temperature is –15~-20^{o}C at which condition traditional system cannot work normally. the compressor discharge temperature obviously, which can provide the system operating steadily.

2) The two-stage compression in one stroll compressor increases the heating capacity and decreases the power input, so energy efficiency ratio rises. However the contribution is more and more inconspicuous as the rising of evaporation temperature.

3) The supplementary circuit uses the subcooling heat behind the condenser to evaporate the refrigerant in supplementary pipe, which makes the additional heater such as a burner or electric heater is unnecessary.

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