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Investigation on Start-up and Reliability during Unsteady Process of Low Temperature Air Source Heat Pump

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ABSTRACT

In this paper the air source heat pump variable refrigerant flow (VRF) of two-stage vapor injection system were analyzed, to investigate the compressor start-up performance at low ambient temperature and the system reliability during unsteady operating process. The start-up control logic of VRF were designed, which combined the control of the electronic expansion valves (EXV), the oil switch valve (SV) and the elevator rate of compressor frequency, to achieve the unit steadily started up at -35℃. In the study on the reliability during unsteady process of the system, the phenomena of the refluence vapor and the enthalpy increasing by vapor injection flowing with liquid, and the trend of the system parameter at their critical state were researched. Besides, the relation of the compression ratio at low stage and the volume ratio of high stage to low stage was gained at refluence state, and it was validated with the experiment in the unit. The experimental results of the VRF showed that the critical compression ratio of low stage at refluence state was close to the reciprocal of the volume ratio, and the refluence didn't happen when the volume ratio was 1.0.

1. INTRODUCTION

Conventional air source heat pump (ASHP) have existed poor coefficient of performance (COP), and reliability problems at severe cold area, which limited the popularization of heat pump technology[1,2]. The researchers have proved two-stage vapor compression with vapor injection technique of low temperature air source heat pump to solve the applicability problem of ASHP in cold climate region, which include the one and twice throttle systems[3~5]. The two-stage compression with twice throttle system requires more rigid reliability of control, especially for the two-stage vapor injection ASHP with only one compressor, because it hardly monitors the mixture temperature of the injection vapor and low-stage cylinder in middle cavity of the two-stage compressor, so that it is not ensured wet compression in the high-stage cylinder or not.

In this paper the ASHP VRF of two-stage vapor injection compression cycle with flash tank was analyzed, the start-up control logic of the system was established to solve the difficult start-up at low temperature. Moreover, the key technology on controlling vapor injection of two-stage throttle cycle during unsteady process was putted forward, which could avoid the refluence vapor and vapor injection with liquid.

2. EXPERIMENTAL PRINCIPLE ANALYSIS

The schematic of the ASHP VRF of two-stage vapor injection compression cycle with flash tank is shown in Figure 1. The outdoor unit adopts pipe-fin heat exchanger and variable speed triple-cylinder two-stage rotary compressor with two cylinders in low-stage cylinder and one cylinder in high stage**[6]**. The two-stage compressor includes a three-cylinder operation mode and a two-cylinder operation mode, when the three-cylinder operation mode is stated, the volume ratio of high-stage cylinder to low-stage cylinder is 0.6, and when the two-cylinder operation mode is stated, the volume ratio is 1.0. The indoor units are equipped with five pipe-fin type air ducts (the nominal total heating capacity is 17.5 kW, and the heating capacity of three units is respectively 2.5 kW, the rest two units is 5.0 kW respectively). The refrigeration is R-410A.

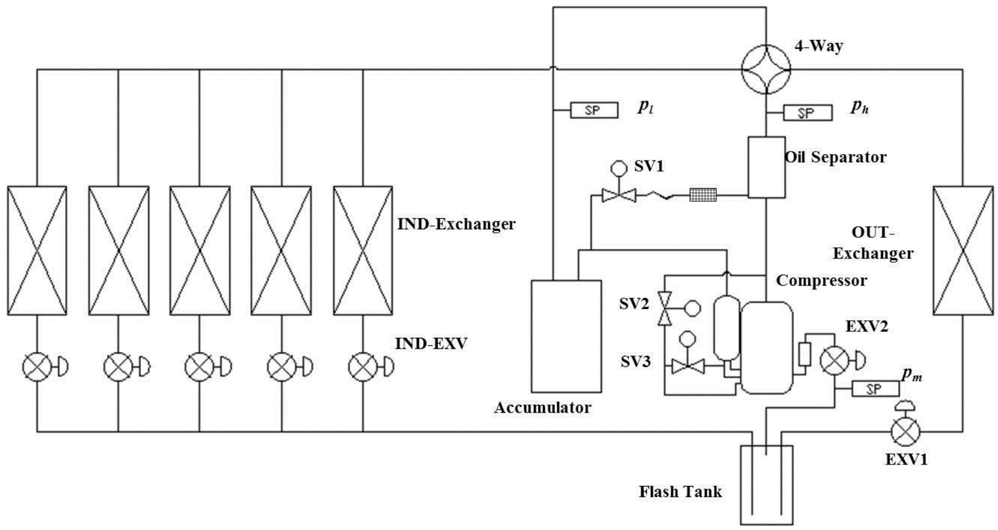

Figure 1: VRF of two-stage vapor injection system

SV1: Oil Solenoid Valve; SV2: Variable Cylinder Solenoid Valve1; SV3: Variable Cylinder Solenoid Valve2; EXV1: Main Load Electric Expansion Valve; EXV2: Vapor Injection Electric Expansion Valve; IND-EXV: Indoor Unit Electric Expansion Valve; P_h: High Pressure Sensor; P_m: Medium Pressure Sensor; P_l : Low Pressure Sensor

2.1 The Start-up Control Logic of VRF

When the ambient temperature is under -25℃, in the initial stage of the unit's start-up (about 90s), the compressor suction pressure is close to the protection shutdown value. Usually, the suction pressure protection is ceased to be effective, to insure the compressor could start up with low suction pressure until entering normal operation at extremely low ambient temperature. The suction pressure of the low-stage cylinder is influenced by the control of each system component. If the system is not started properly, after it enters the normal control, the suction pressure is too low, which will cause the compressor to reduce rotate speed because low suction pressure limit. In severe cases, the suction pressure protection shutdown malfunction will occur directly, causing the unit to fail to start at low temperature.

The suction pressure control at low-temperature start-up involved several components, such as the oil return solenoid valve, the compressor frequency, and the flow area of the initial electronic expansion valve for the indoor and outdoor machines. After the compressor frequency reaching the proper setpoint, the pressure difference is established and then the 4-Way can be turned on. When the unit is stopped for a long time at low temperature, the outdoor temperature of the external refrigerant is low, and the flash tank has much liquid refrigerant due to the migration of the refrigerant. In order to avoid the high-stage cylinder inhaling the liquid refrigerant at the compressor start-up, the vapor injection value EXV2 is closed, and it is opened after the compressor is stable.

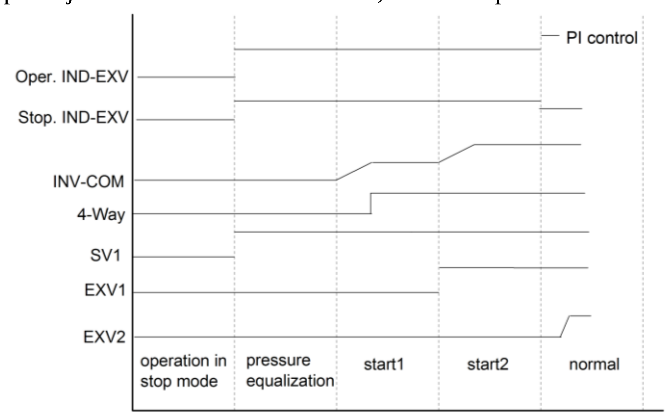

Figure 2: Heating start-up control timing chart

In summary, during the unit's start-up stage at low temperature, the suction pressure protection control logic is shielded. The control timing chart is formulated as shown in Figure 2, and it was verified when the outdoor temperature is -35 ℃ and the indoor temperature is 20 ℃. The experimental test results will be showed in the third section.

2.2 The Effect of the Vapor Injection Rate of the Compressor on the System State

For the ASHP VRF of two-stage vapor injection system, the refrigerant is in a gas-liquid mixed state after the first-stage throttle, and then enters flash tank to separate out the saturated liquid and saturated gaseous refrigerants. It is assumed that the saturated vapor in flash tank is fully separated from the saturated liquid, the volume ratio of high-stage cylinder to low-stage cylinder, the outlet refrigerant state of evaporator and condenser are constant. When the medium pressure in the system changes from high to low by controlling the first-stage throttled electron expansion valve, the system will appear three phenomena: vapor injection with liquid, normal vapor injection and refluence state.
The dryness after the first-stage throttle is defined *x*.

$$x = \frac{M_{i,g}}{M_c} \tag{1}$$

For easy to analyse, ω is defined for the medium vapor injection rate of the two-stage compressor.

$$\omega = \frac{M_i}{M_c} \tag{2}$$

$$\omega = \frac{\eta_{v,H} \cdot f_H \cdot V_H \cdot \rho_{s,H} - \eta_{v,L} \cdot f_L \cdot V_L \cdot \rho_{s,L}}{\eta_{v,H} \cdot f_H \cdot V_H \cdot \rho_{s,H}} \tag{3}$$

Where, $M_{i,g}$——the mass flow of saturated vapor after the first-stage throttling, the unit is kg/s;
M_c——the mass flow of condenser, the unit is kg/s;
M_i——the mass flow of vapor injection, the unit is kg/s;
M_e——the mass flow of evaporator, the unit is kg/s;
$\eta_{v,L}$, $\eta_{v,H}$——the compression volume efficiency of low-stage cylinder and high-stage cylinder;
$\rho_{s,L}$, $\rho_{s,H}$——the suction density of low-stage cylinder and high-stage cylinder, the unit is kg/cm^3;
Used the integrated two-stage compressor, the ω is

$$\omega = 1 - \frac{\eta_{v,L} \cdot \rho_{s,L}}{\lambda \cdot \eta_{v,H} \cdot \rho_{s,H}} \tag{4}$$

Where, λ—the volume ratio of high-stage cylinder and low-stage cylinder.
Assuming the compression volume efficiency of low-stage cylinder and high-stage cylinder is equal, which can be further simplified as:

$$\omega = 1 - \frac{\rho_{s,L}}{\lambda \cdot \rho_{s,H}} \tag{5}$$

（1）Normal vapor injection （$0<\omega\leq x$）
When the section of the first-stage throttle electronic expansion valve is moderate, the level of the refrigerant in flash tank will be between the liquid pipe outlet and the vapor injection pipe nozzle, and close to the liquid pipe outlet, then the saturated gaseous refrigerant enters the compressor medium cavity through the vapor injection pipe, the saturated liquid refrigerant via the outlet pipe and the two-stage throttle electronic expansion valve enters the evaporator to evaporate. At this time, $x=\omega$, the refrigerant state in flash tank and p-h diagram are shown in Figure 3.
When the section of the first-stage throttle electronic expansion valve is small, the mass flow rate of the medium vapor injection is less than the saturated gaseous mass flow rate after being first-stage throttled, the liquid level of refrigerant will be the same height with the liquid pipe outlet in flash tank. Some of the saturated gas after the first-stage throttling enters the compressor medium cavity through the vapor injection pipe, and the other part is mixed with the liquid refrigerant, entering the two-stage throttle electronic expansion valve through the liquid outlet pipe.

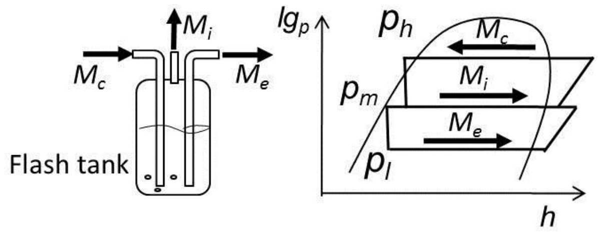

Figure 3: Refrigerant state in flash tank and p-h diagram at $x=\omega$

（2）Refluence state（ω<0）

When the flow area of the first-stage throttling electronic expansion valve is too small to make the medium pressure p_{FT} being lower than a certain pressure value, the inspiratory mass flow rate of the high-stage cylinder M_c will decrease and the exhausting mass flow rate of the low-stage cylinder M_c will not change, resulting M_c is less than M_e. Then the phenomenon of refrigerant backflow occurs in the vapor injection pipe, that is, the part of superheated gas from the low-stage cylinder exhaust flows into flash tank through the vapor injection pipe. At this time, the refrigerant state in flash tank and the p-h diagram are shown Figure 4.

The characteristic of the system at the refluence state is that the medium pressure is low, and part of the low-stage cylinder exhaust flows into flash tank through the vapor injection pipe to make the superheat of “injection vapor” being greater than zero.

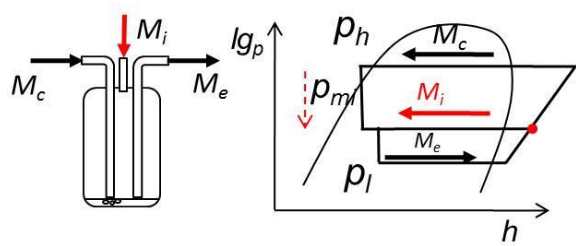

Figure 4: Refrigerant state in flash tank and p-h diagram at ω<0

（3）Vapor injection with liquid（ω>x）

When the section of the first-stage throttling expansion valve is too large, the medium pressure rises, then the inspiratory mass flow rate of the high-stage cylinder M_c will increase, and the exhaust mass flow rate of the low-stage cylinder M_e will not change, so the injection vapor mass flow rate M_i will increase. And when the medium pressure rises close to the condensing pressure, it will cause the saturated gas mass flow $M_{i,g}$ being less than M_i. On the other hand, with the section of the first-stage throttling electronic expansion valve increasing, the refrigerant subcooling of the condenser reduces even to the two-phase state, the refrigerant mass distribution in the system changes, and the liquid level of flash tank is too high even to reach the same height with the vapor injection pipe nozzle.

The refrigerant state in flash tank and the p-h diagram are shown Figure 5. The vapor injection with much liquid refrigerant enter into the compressor, which will cause the compressor exhausting superheat low.

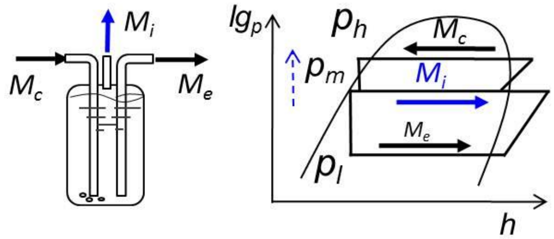

Figure 5: Refrigerant state in flash tank and p-h diagram at ω>x

2.3 The Relationship between the Vapor Injection Rate, the Compression Ratio of Low-stage Cylinder and Subcooling

It is assumed that the condensing temperature is 40 ℃, the low pressure is 0.5 MPa, the isentropic efficiency of high-stage cylinder and low-stage cylinder is 1.0, the suction superheat of low-stage cylinder is 3 ℃, the compression ratio is 0.4-1.0. Calculate the vapor injection rate at different compression ratio of low-stage cylinder, and the result is shown in Figure 6.

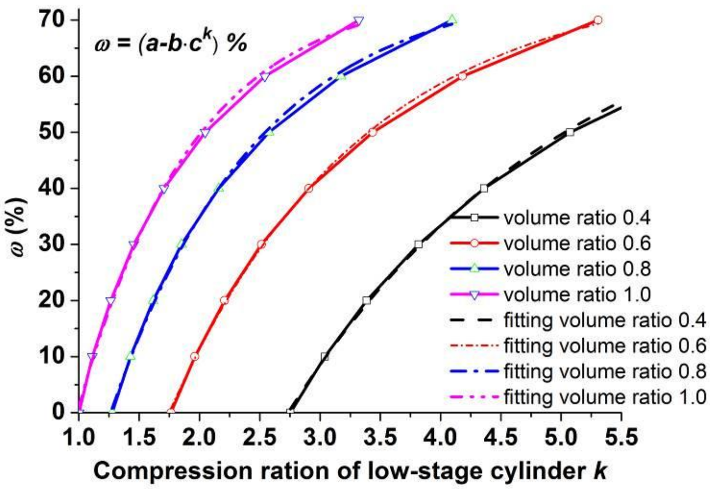

Figure 6: The relation of the vapor injection rate with different compression ratio of low-stage cylinder

It can be seen from Figure 6 that at the same compression ratio, the larger with the volume ratio, the greater with the vapor injection rate; at a certain volume ratio, the vapor injection rate increases as the compression ratio increasing. Through the fitting calculation, the correlation between the compression ratio of low-stage cylinder and the vapor injection rate under the different volume ratio can be obtained as follows. According to this correlation, the vapor injection rate can be controlled by the compression ratio of low-stage cylinder.

$$\omega = \left(a - b \cdot c^{k}\right)\ \% \tag{6}$$

The relationship between *a, b, c* coefficient and the volume ratio *λ*, as is shown in Table 1.

Table 1: The relation between the *a*, *b*, *c* coefficient and the volume ratio

λ	*a*	*b*	*c*
0.4	80.65	255.39	0.66
0.6	77.83	231.03	0.54
0.8	76.77	216.81	0.44
1	75.69	212.88	0.35

Refluence critical point: When the vapor injection rate is 0, the compression ratio of the low-stage cylinder is defined critical compression ratio k_{cr}. It is assumed that: the condensing temperature is 40 ℃, the isentropic efficiency is 1.0, the suction superheat is 3 ℃, the volume ratio is 0.4~1.0. Calculate the critical compression ratio of the low-stage cylinder k_{cr} under the different volume ratio. The result is shown in Figure 7. At the same volume ratio, the critical compression ratio changes little. After the volume ratio exceeds 0.6, the critical compression ratio deviates from 1/*λ* very little. Based on the comprehensive error analysis, the critical compression ratio of the low-stage cylinder can be approximated to 1/*λ*. When the volume ratio is 1.0, there is no refluence. When the volume ratio is 0.6, the critical compression ratio of the low-stage cylinder is 1.74 (±0.03). The backflow are related to compression ratio and the volume ratio, but independent of the exhaust pressure.

For further analysis about the influence of subcooling on dryness and vapor injection rate, it is assumed that: the condensing temperature is 40 ℃, the low pressure is 0.5MPa, the isentropic efficiency is 0.7, the suction superheat is 3 ℃ and the volume ratio is 0.6. Calculate the dryness and vapor injection rate corresponding to different medium pressure when the condenser outlet temperatures is 28~40 ℃, and the result is shown in Figure 8. When the vapor injection rate ω=x, it can be seen from the figure that as the vapor injection rate increasing, the condenser outlet temperature and the medium pressure increase synchronously. At the left side of the vapor injection rate ω curve (including the vapor injection rate ω curve), the relationship $x \geq \omega$ is established, that is, the area is the normal vapor injection area; the right side of the ω curve is $x < \omega$, that means, the area is the vapor injection with liquid area.

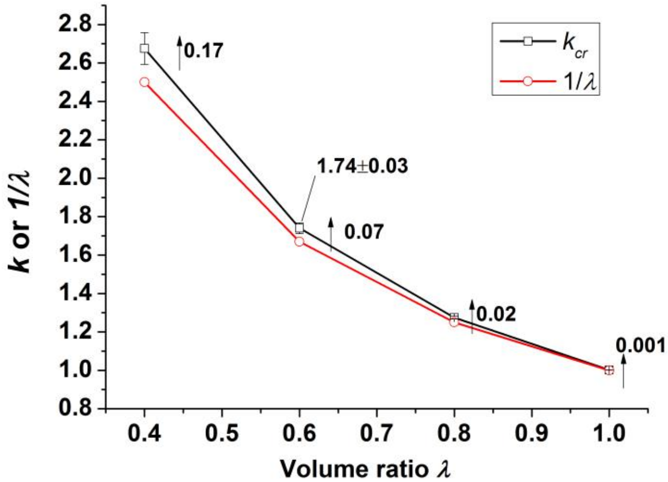

Figure 7: The relation of the critical compression ratio k_{cr} and the volume ratio

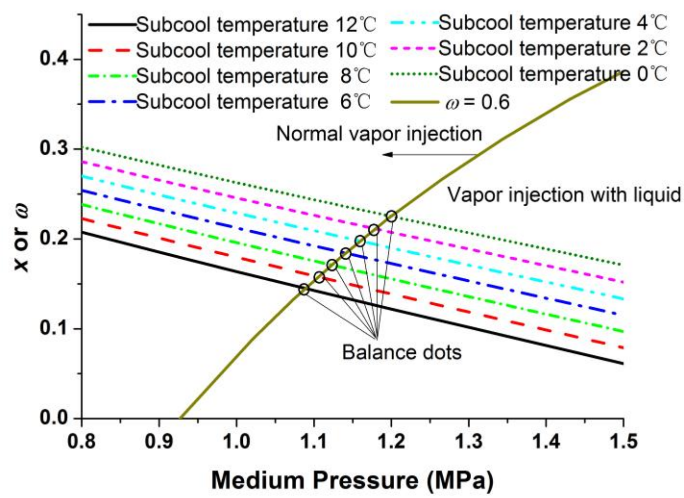

Figure 8: The dryness and vapor injection rate at different medium pressure

The vapor injection with liquid area can be subdivided into two cases. One is that the vapor injection gas-liquid mixture is mixed with the first-stage exhausting, and then the two-stage suction superheat is greater than or equal to 0; the other case is the suction of high-stage cylinder carrying liquid because of vapor injection with excessive fluidity, which maybe cause liquid hitting the high-stage cylinder. Then, the high-stage inhalation being the saturated gas is regard as the high-stage wet compression critical point, and the corresponding first-stage throttling dryness is defined as the critical dryness x_{cr}.

It is assumed that the condensing temperature is 40 ℃, the low pressure is 0.5 MPa, the isentropic efficiency is 0.7, the suction superheat is 3 ℃ and the volumetric ratio is 0.6. Calculate the vapor injection rate and two-stage wet compression critical dryness x_{cr} respectively at the different medium pressure on the vapor injection with liquid, and the result is shown in Figure 9. As the vapor injection rate increasing, the critical dryness gradually increases. When ω-x<0.1, the superheat of the high-stage suction is positive. The two-stage wet compression zone is below the critical dryness x_{cr} zone. Therefore, in order to avoid the high-stage wet compression, the maximum of vapor injection rate should be smaller than the sum of the critical dryness x_{cr} and 0.1.

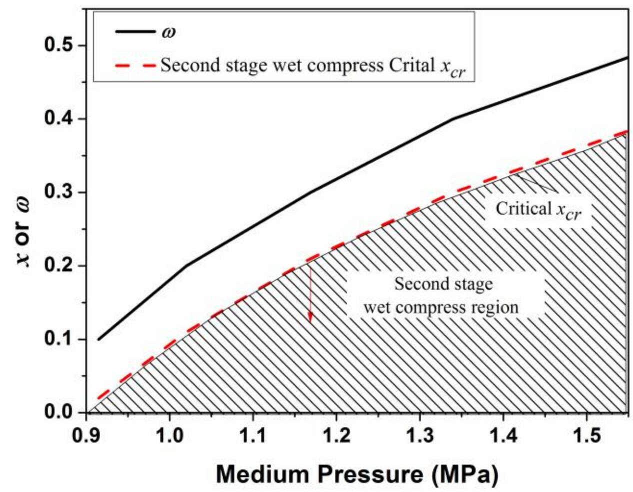

Figure 9: The vapor injection rate and high-stage wet compression critical dryness x_{cr} at different medium pressure

3. EXPERIMENTAL RESULTS AND ANALYSIS

3.1 The Verification and Analysis of Start-up Control Logic at Low Temperature

In order to verify the unit start-up reliability at low temperature, when the ambient temperature is -35 ℃ and the indoor temperature is 20 ℃, the indoor machines were all set to operate at 30 ℃. The changes about the high-stage exhaust pressure, medium pressure and low-stage suction pressure within 120 seconds after the unit start-up are shown in Figure 10.

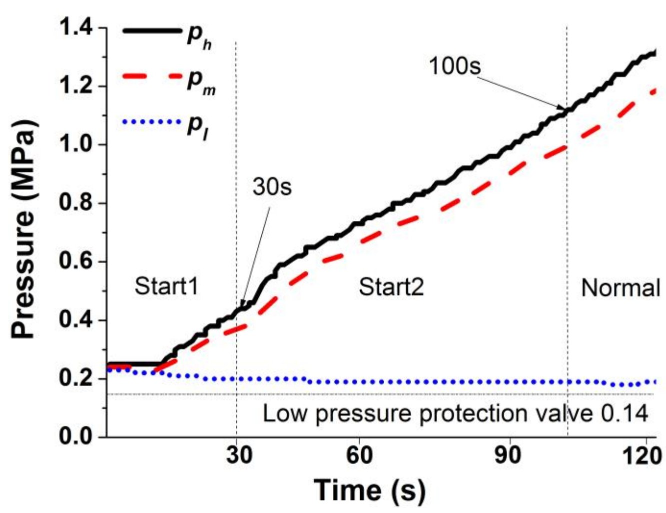

Figure 10: Different pressures within 120 seconds after unit start-up at ambient temperature -35 ℃

In the start1 phase, the compressor frequency rises smoothly, and the 4-WAY normally works after the suction and exhaust pressure difference being established. Then the low pressure gradually reduces to 0.2 MPa, which takes 30 seconds. In the start2 phase, during the compressor frequency accelerated, the exhaust pressure significantly increases, and the suction pressure slowly reduces to 0.19 MPa and maintain. After cutting into the normal operation, the suction pressure dropped to 0.18 MPa, but it did not reach the shutdown protection value, and there is no large fluctuation.

Figure 11 is a plot of the unit pressure and compressor power curve within 15 minutes after the unit start-up. When the compressor frequency reaches a higher value, in order to avoid the suction pressure being too low to result in shutdown protection, the strategy of slowing down the frequency is taken. After 4 minutes, the augment of the compressor power significantly slows down. When the condition for vapor injection enthalpy increasing is satisfied, the vapor injection valve gradually opens at the 6th minute, the medium pressure begins to gradually decrease, and tends to be stable after the 10th minute; meanwhile, the compressor power increases smoothly and the suction pressure only drops 0.01MPa. After the vapor injection control was completed, the unit enters the steady stage, and there is no large fluctuation in the power and pressure of the unit.

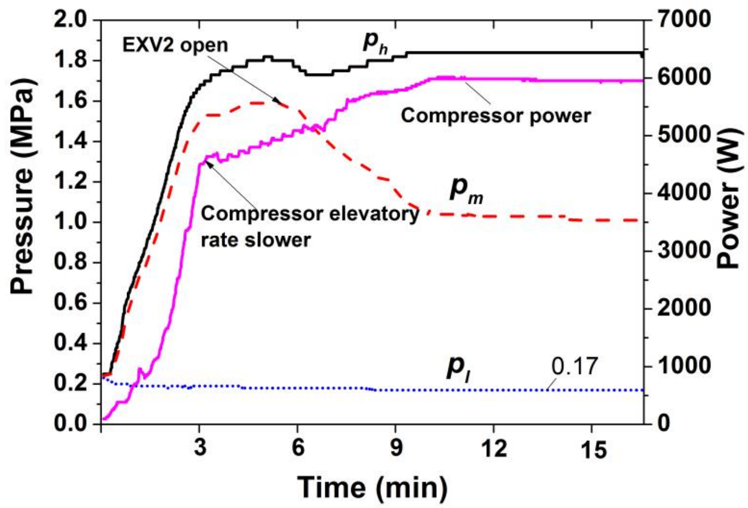

Figure 11: The curve of the unit pressures and compressor power with operating time at ambient temperature -35℃

From the result of the experiment, the low-temperature start-up control logic proposed in this paper is reliable, which effectively avoids the occurrence of the suction pressure protection in the low-temperature start-up stage. The compressor power does not fluctuate greatly, and the system operates stably in 10 minutes, which can improve the user's heating comfort at low-temperature.

3.2 The Verification and Analysis of Refluence and Vapor Injection with Liquid

In order to verify the critical point of refluence when the volume ratio is 0.6, all the indoor units operates cooling when the indoor temperature is 27/19℃ and the outdoor temperature is 43℃ and 35℃ respectively, and the compressor operates at triple-cylinder mode in the same frequency. The target suction superheat is 3℃, which is controlled by the second-stage throttling electronic expansion valve. Refluence is controlled by the first-stage throttling electronic expansion valve. The judgment condition of refluence is: ΔT_m >5℃ and ΔT_m is equal to the value that the vapor injection temperature minus the medium pressure saturation temperature. The test results are shown in Table 2.

Table 2: The data of system state with first-stage throttle EXV1 at different temperatures

T_{out}(℃)	EXV1(pulse)	T_m(℃)	T_s(℃)	p_l(MPa)	p_m(MPa)	p_h(MPa)	ΔT_m(℃)
43	140	52.7	20	1.39	2.40	3.33	12.99
43	160	52.67	20	1.39	2.43	2.97	12.47
43	180	41.3	20	1.36	2.47	2.9	0.50
43	200	41.32	19	1.3	2.50	2.85	0.08
35	150	52.2	21	1.37	2.36	2.65	13.13
35	170	51.3	22	1.37	2.40	2.65	11.51
35	200	39.6	19	1.35	2.38	2.65	0.10

Not：T_{out} ambient temperature，T_m medium temperature，T_s suction temperature.

With the increasing of the EXV1 flow area, the vapor injection changes from the backflow to the normal vapor injection, and the injection vapor temperature drops significantly, and is close to the saturation temperature corresponding to the medium pressure. The relationship between the flow area of the first-stage throttle electronic expansion valve and the compression ratio *k* of the low-stage cylinder is shown in Figure 12. With the increasing of the EXV1 flow area, the low-stage cylinder compression ratio *k* increases gradually; when the value of *k* is greater than the critical point k_{cr}, the vapor injection changes from the backflow to the normal vapor injection. This shows that when the volume ratio is 0.6, the critical compression ratio of the low-stage cylinder is close to the reciprocal of the volume ratio.

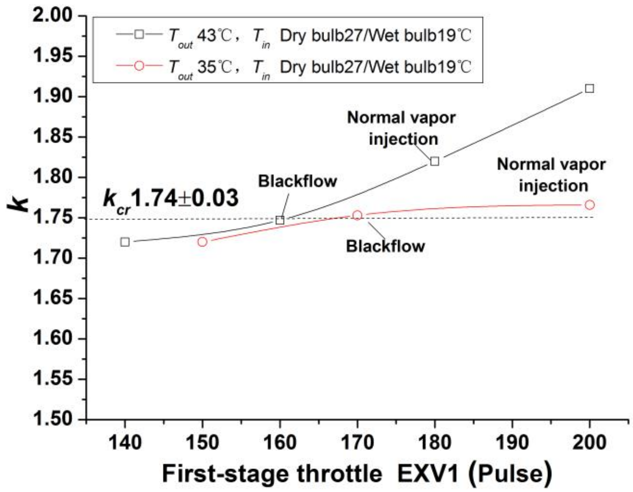

Fig. 12: The *k* with first-stage throttle EXV1 at different temperatures

Figure 13 is the plot of the relationship between subcooling and the liquid level of flash tank, and exhausting superheat. It shows that with the subcooling increasing, the high-stage discharge superheat increases gradually, and the liquid level of flash tank decreases in the opposite direction. When the liquid level rose, the part of the liquid refrigerant enters the high-stage cylinder along with the gaseous refrigerant, resulting in the decrease of the exhaust superheat. Therefore, whether the unit is vapor injection with liquid or not can be judged by the high-stage discharge superheat. The high-stage discharge superheat should be over 10℃ to avoid vapor injection with liquid.

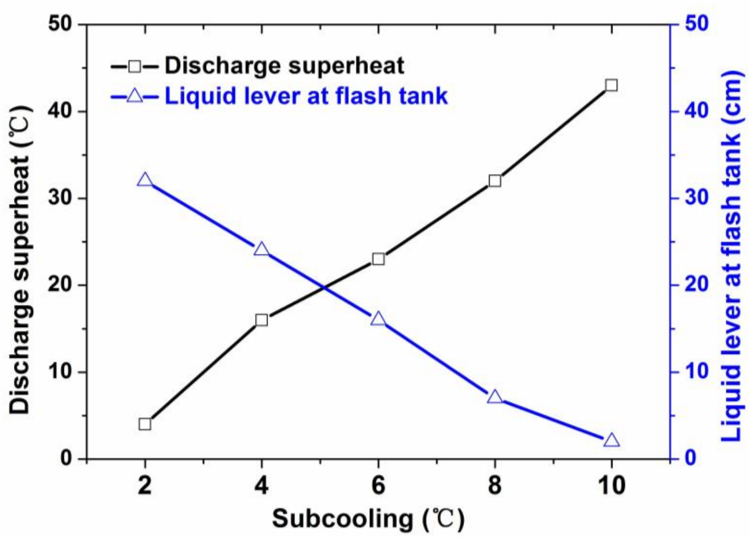

Fig. 13: Discharge superheat and liquid level at flash tank with subcooling

4. CONCLUSIONS

For the start-up and operational reliability issues of the two-stage vapor injection ASHP VRF, it can get the following conclusions by the thermodynamic theoretical analysis and experiments verification.
(1) The complete start-up control logic at low temperature of the ASHP VRF is established, and it can solve the problem of low pressure protection after the unit starting at low temperature -35 ℃, which combined with the control of the indoor and outdoor electronic expansion valves, the oil return solenoid valve and the elevator rate of compressor frequency.
(2) In the reliability study of the system during unsteady process of operation, for the specific refluence vapor of the two-stage throttling imperfect cooling system, the compressor vapor injection rate has been put forward to analyze the vapor injection state of the system and experiments have been carried out to verify that the critical compression ratio at the low-stage cylinder is close to the reciprocal of the volume ratio when the refluence occurs, and that there is no refluence when the volume ratio is 1.0.
(3) From the two-stage compression system reliability point of view, the vapor injection with liquid could be subdivided into two cases, which borderline was the high-stage inhalation being the saturated gas. To ensure vapor injection without liquid, the high-stage discharge superheat should be over 10℃.

NOMENCLATURE

f	frequency	k	compression ratio
M	mass flow rate	**Subscript**	
IND	indoor	c	condenser or condensation
SV	solenoid valve	i	injection
EXV	electron expansion valve	e	evaporator or evaporation
p	pressure	s	suction
λ	volume ratio	g	gas
T	temperature	v	specific volume
V	volume	L	low-stage or low-stage cylinder
x	dryness	H	high-stage or high-stage cylinder
η	efficiency	m	medium
ρ	density	cr	critical
ω	vapor injection rate	out	outdoor

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