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Study on the Control Characteristics of Variable Frequency Refrigeration Unit under Multiple Intermittent Loads

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Abstract
Research on the operation control characteristics of variable frequency refrigeration units under multi intermittent loads energy is the basis for the survival and development of modern society and the driving force of the world economy. Economic growth is always accompanied by energy consumption, especially in the past 100 years, global energy consumption has increased at an average rate of 3% per year. Refrigeration energy consumption accounts for an important part of energy consumption. Every 25% - 30% of the world's electricity is used for refrigeration. Frequency conversion technology is an important means to reduce the energy consumption, save electricity and improve the operation performance of the refrigeration system. It has been widely used in household air conditioners, but its development in the refrigeration and refrigeration industry is far behind. The lack of relevant control strategies and operation and maintenance experience may be the bottleneck restricting the application of frequency conversion technology in domestic and foreign refrigerators. Therefore, this topic mainly focuses on the variable frequency refrigeration system in the high temperature cold storage, and studies the operation control strategy of the variable frequency refrigeration system carrying multiple intermittent loads and the operation characteristics of the variable frequency unit on the premise of ensuring the demand for refrigeration capacity.
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Ⅰ.RESULT AND DISCUSSION

When carrying two intermittent air coolers, the variable frequency refrigeration unit can control the outlet pressure of the evaporator of the refrigeration system to realize the adjustment of turning off one air cooler and operating only one air cooler under the same outdoor temperature and different temperatures in the refrigerator test box[1]; When the outdoor temperature changes in a certain range, it is necessary to adjust the evaporator outlet pressure control of different refrigeration systems to realize the adjustment of turning off one air cooler and only one air cooler[2]. In order to further comprehensively study the operation and control characteristics of the variable frequency refrigeration unit when using this control method to carry multiple intermittent loads on the variable frequency refrigeration system, this chapter uses Matlab software to model the various components of the variable frequency refrigeration system, and through the connection between the input and output parameters of each component and the parameters of other components, it can be coupled into a complete refrigeration system, It is the basis of multi load simulation analysis of variable frequency refrigeration unit.

In the simulation calculation process, it is necessary to iterate the frequency of the compressor to make the cooling capacity of the system reach the desired cooling capacity[3]. When the compressor frequency changes, the refrigerant flow of the whole system also changes, so the refrigerant flow is an important output parameter of the compressor model. In addition, as the largest energy consuming component in the whole refrigeration system[4], the compressor affects the performance coefficient of the whole system, so the actual power consumption of the compressor is also an important parameter of the model. In terms of system structure, the compressor sucks refrigerant gas of low temperature and low pressure from the evaporator[5], so the suction temperature and suction pressure of the compressor are the input parameters of the compressor, and the refrigerant enthalpy value and exhaust temperature at the compressor exhaust port need to be input into the condenser model, which are the input parameters of the condenser, thus affecting the performance and calculation of the entire condenser device. Therefore, the exhaust temperature The outlet enthalpy is also an important output parameter of the compressor. In addition, in the whole refrigeration system model, it is necessary to compare the mass flow through the compressor with that of the thermal expansion valve[6]. To sum up, in the variable frequency compressor model, the input parameters need to include the suction temperature, suction pressure, compressor frequency, compressor speed, as well as the internal structure parameters of the compressor. Finally, the output parameters are calculated to include the refrigerant mass flow, the actual power consumption of the compressor, the outlet enthalpy value, and the exhaust temperature.

Ⅱ.RESULT AND DISCUSSION

2.1Calculation of refrigerant mass flow

For compressor mass flow m_ Com refers to the current commonly used compressor flow calculation model:

$$m_{com} = \lambda \frac{V_{th}}{v_{suc}} \quad (2.1)$$

Where, m_ Com - refrigerant mass flow, kg/s

λ-Gas transmission coefficient

V_{Th}—theoretical volume gas transmission capacity of compressor, m3/s

υ_{Suc}—specific volume of refrigerant gas at suction port, m3/kg

V—The calculation formula of this :

$$V_{th} = 60n\pi P(P-2t)(2N-1)h \quad (2.2)$$

Where, n - speed of variable frequency compressor, r/min
P - Vortex pitch, m
T - wall thickness of vortex body, m
N - Number of compression chambers
H - height of vortex body：

$$n = \frac{120f(1-s)}{Pd} \quad (2.3)$$

Where,
f -motor power supply frequency, Hz
S-motor slip
Pd - Number of magnetic poles of motor

According to the formula, in order to calculate the theoretical gas throughput of the scroll compressor, it is necessary to know some important structural parameters inside the compressor, including the wall thickness, pitch, height of the scroll, and relevant parameters of the motor. However, because these parameters involve the trade secrets of the manufacturer, the specific internal structure parameters are not provided, but the theoretical volume gas transmission capacity and the exhaust capacity of the compressor at different frequencies are directly provided, as shown in Table 4.1. During the calculation, the data in Table 4.1 is directly used to obtain the theoretical gas transmission capacity of the compression mechanism.

Table 2.1 Theoretical Displacement of Compressor

model	volume （ cm3/rev ）	Exhaust volume			
		30rps(m3/h)	50rps(m3/h)	60rps(m3/h)	100rps(m3/h)
VLZ028	27.8	3.0	5.0	6.0	10.0

In addition, gas transmission coefficient λ Calculation formula:

$$\lambda = \lambda_V \lambda_P \lambda_T \lambda_D \quad (2.4)$$

Where,

λ_V—Volume coefficient

λ_P—pressure loss coefficient

λ_D—leakage coefficient

λ_T—temperature coefficient

The gas transmission coefficient of scroll compressor has the following characteristics:

1.The clearance volume of scroll compressor is very small. The exhaust hole in the innermost chamber is not connected with the suction chamber, and there is no expansion process of clearance volume gas, so the clearance volume coefficient λ_V=1；

2. The scroll compressor has no suction valve, the suction is suction type, and the suction pressure loss is very small, so the pressure loss coefficient λP=1；

3. The suction chamber of the scroll compressor is at the outermost side, the suction heating is not large, and the

temperature coefficient λT=1 ;

4. The leakage coefficient is the only factor affecting the volumetric efficiency of the scroll compressor λ_D。 The pressure difference between the compression chambers of each ring of the compressor is small, and the leakage is small and internal leakage. The leakage is smaller when the seal is perfect, so the leakage coefficient of the scroll compressor is large. This paper takes λ_D=0.95。

Ⅲ.Compressor power calculation

Theoretical power of compressor$N_{tℎ}$:

$$N_{tℎ} = V_{th}\lambda \frac{p_e k}{k-1}[\left(\frac{p_c}{p_e}\right)^{\frac{k-1}{k}} - 1] \quad (3.5)$$

Where, p_E—evaporation pressure, Pa—

p_C—Condensation pressure, Pa

K-isentropic index, taking 1.125 for R404A

Actual input power of motorN_{in} :

$$N_{in} = N_{tℎ}/\eta_{el} \quad (3.6)$$

Where, η_ El is the electrical efficiency. For scroll compressor, it is usually taken as 0.6~0.75. In this paper, it is taken as 0.68

3.1.4 Calculation of outlet enthalpy and exhaust temperature

Indicative power of variable frequency compressor N_ I Calculation formula:

$$N_i = N_{tℎ}/\eta_i \quad (3.7)$$

Where,

η_I—indicating efficiency, taken as 0.95 in this paper

Refrigerant enthalpy at compressor outlet h_ 2 Calculation formula:

$$h_2 = h_1 + N_i/m_{com} \quad (3.8)$$

Where ,

h_1—compressor suction enthalpy, kJ/kg

After calculating the enthalpy of the refrigerant at the compressor outlet, call Refprop software to calculate the compressor exhaust temperature according to the enthalpy and exhaust pressure of the refrigerant at the compressor outlet.

Refprop is an internationally authoritative calculation software for the physical properties of working fluids developed by NIST (National Institute of Standards and Technology) of the United States. It can not only find single substances, but also various property parameters of mixed substances (such as R404A refrigerant used in this topic), such as temperature, pressure, density, steam density, liquid enthalpy, steam enthalpy and other physical parameters, Refprop is used as an auxiliary tool to query and call the physical parameters of R404A refrigerant and air.

Ⅳ. Condenser mathematical model

4.1 Condenser modeling simplification

The heat transfer process of the condenser is very complex, which is closely related to the structure of the condenser itself and the changes of the outdoor environment. Therefore, it is necessary to simplify the model when establishing it. On the premise of ensuring the accuracy and stability of the model, retain the factors that play a dominant role and ignore the factors that have less impact on the model calculation.

According to the research content of this topic, based on the above principles, the numerical heat transfer model of the condenser is simplified as follows:

(1) The flow pattern of refrigerant inside the condenser tube and air outside the tube can be considered as one-dimensional steady flow;

(2) The change of refrigerant pressure in condenser tube is not considered;

(3) The condenser is a countercurrent heat exchanger;

(4) Ignore the thermal resistance and axial heat conduction of condenser tube wall;

(5) Ignore heat leakage;

(6) The influence of any non condensable gas, air and lubricating oil in the pipe is not considered.

4.2Condenser mathematical model

The condenser used in the refrigeration system studied in this paper is a finned tube heat exchanger. Its working principle is: the refrigerant condenses in the pipe and releases heat, transferring heat to the pipe wall and the scale layer. After the pipe wall and the scale layer absorb heat, they transfer heat to the fins and then convection to the air, causing the air temperature to rise. The air is sent to the outdoor environment through the condensing fan. The refrigerant mainly depends on the latent heat in the whole heat exchange process of the condenser. As the condenser sucks in the superheated gas discharged from the compressor, the dryness decreases continuously during the heat release process until the supercooled liquid. The refrigerant heat transfer process can be divided into gas superheating zone, gas-liquid two-phase zone and liquid supercooling zone. During modeling, modeling and analysis are carried out from the refrigerant side and air side in turn. According to the different flow state of refrigerant in the condenser tube, the heat transfer coefficient of the refrigerant side surface and the air side in the condenser can be calculated from the two-phase area and single-phase area respectively.

1. Calculation formula for each heat transfer coefficient of condenser

(1) Refrigerant side heat transfer coefficient

For single zone (supercooled zone, overheated zone):

$$Nu = 0.023Re^{0.8}Pr^{0.3} \quad (4.1)$$

Two-phase region：

$$\alpha_{TP} = \alpha_l \left[(1-\chi)^{0.8} + \frac{3.8\chi^{0.76}(1-\chi)^{0.04}}{{Pr_r}^{0.38}}\right] \quad (4.2)$$

$$\alpha_l = 0.023\frac{\lambda_l}{d_i}\left(\frac{Re_l}{1-\chi}\right)^{0.8} {Pr_l}^{0.4} \quad (4.3)$$

Where, χ— Refrigerant dryness

Pr_R—Prandtl number of refrigerant

α_L—Heat transfer coefficient of liquid refrigerant, W/(m2 · K)

λ_L—Thermal conductivity of refrigerant liquid phase, W/(m · K)

d_I—Inner diameter of condenser pipe, mm

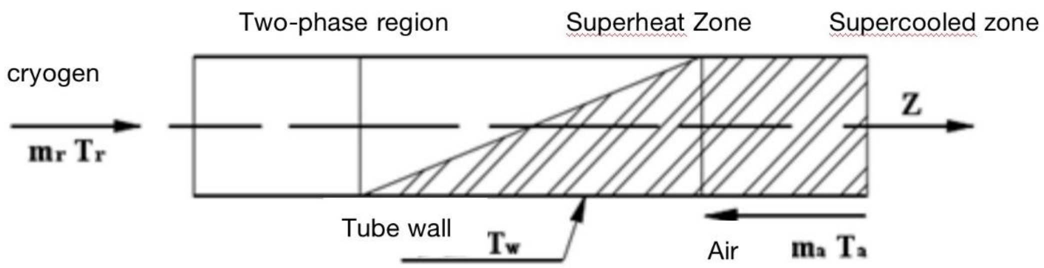
(2) Air side heat transfer coefficient

$$\alpha_a = C\psi\frac{\lambda_a}{d_e}Re_a^n\left(\frac{b}{d_e}\right)^m \quad (4.4)$$

2. Numerical model equation

According to the simplified heat transfer conditions of condenser, the model is established as shown in Figure

Fig. Heat Transfer Model of Condenser



In Figure , for each micro segment, the refrigerant condenses and releases heat, and the air absorbs the heat released by the refrigerant. When the temperature rises, the heat released by the refrigerant is equal to the heat absorbed by the air. Assume that the condenser is countercurrent, so during iteration, the initial state of the refrigerant can be known for the initial micro segment, but the final state of the air is not known. At this time, it is necessary to assume the final state of the air outlet. After iteration, compare the calculated air outlet temperature with the assumed air outlet temperature to see if it is less than the error (the error is 0.1 ℃ in this paper). If it is less than the error, the assumed outlet temperature is correct, Otherwise, re assume the air outlet temperature for calculation.

In the heat transfer process of each micro segment, the energy conservation between the refrigerant side and the air side shall be met. In the heat transfer process, the heat transfer between the refrigerant and the tube wall shall be equal to that between the air and the fins.

V. Mathematical model of evaporator

5.1 Evaporator modeling simplification

Because the frequency of the compressor is not fixed, but changes with the outdoor temperature and the temperature in the warehouse, and the state of the refrigerant in the evaporator also changes with the change of working conditions, the entire refrigeration system is unstable in the actual control process, and the internal work of the evaporator is also a complex and unstable process. However, this topic mainly studies the operating characteristics of the variable frequency refrigeration unit when it reaches the steady state when the outdoor temperature and the temperature in the warehouse reach a certain working condition and multiple loads are started and stopped under this working condition. Therefore, combined with the main research contents of this paper, the evaporator is modeled using the steady state distribution method to simulate the operation of the evaporator when the outdoor temperature and the temperature in the warehouse reach a certain working condition and the refrigeration system operates stably, The evaporator is simplified as follows.

(1) It is considered that the working medium in the evaporator is a homogeneous saturation model, the working medium is incompressible, and the device is a pure working medium with little change in the filling amount of the working medium.

(2) The evaporator is a countercurrent heat exchanger;

(3) Both the refrigerant inside the tube and the air outside the tube are considered as one-dimensional flow and heat transfer;

(4) Refrigerant flows stably in the pipe;

(5) The radial temperature of the pipe wall is consistent, and the thermal resistance of the pipe wall and the axial heat conduction are not considered;

(6) Ignore the pressure drop in the overheat zone.

5.2 Mathematical model of evaporator

There are two phase regions of refrigerant in evaporator: gas-liquid two-phase region and gas superheating region. The heat transfer coefficients on the surface of the refrigerant side and on the air side of the evaporator are calculated.

1. Calculation formula for each heat transfer coefficient of evaporator

(1) Refrigerant side heat transfer coefficient

Overheating zone:

$$Nu = 0.023Re^{0.8}Pr^{0.3} \quad (5.1)$$

Two-phase region：

$$\alpha_{TP} = \alpha_l[C_1(C_0)^{C_2}(25F_{r1})^{C_5} + C_3(B_0)^{C_4}F_{fl}] \quad (5.2)$$

$$\alpha_l = 0.023(\frac{g(1-\chi)D}{\mu_l})^{0.8}\frac{Pr_l^{0.4}\lambda_l}{D_i} \quad (5.3)$$

1. Mathematical model equation

The heat transfer model of evaporator is shown in Figure .

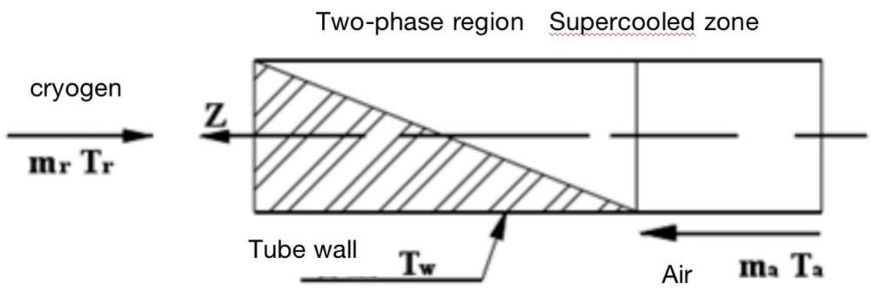


Fig. Heat transfer model of evaporator

As shown in Figure 4.4, the models of evaporator and condenser are similar, both of which are countercurrent, so the air outlet parameters should be assumed for iteration during calculation. Different from the condenser, during the whole heat exchange process of the evaporator, the refrigerant evaporates to absorb heat and the air cools to cool down.

(1) Two phase numerical model

a. Refrigerant side energy equation

$$\dot{m}_r \frac{d\bar{h}_r}{dz} = -\pi D_i \alpha_i (T_w - T_r) \quad (5.1)$$

$$\bar{h}_r = \chi h_v + (1-\chi) h_l \quad (5.2)$$

Where, h R - enthalpy of gas-liquid two-phase refrigerant mixture, kJ/kg

χRefrigerant dryness

αI - Surface heat transfer coefficient of refrigerant side, W/(m2 · K)

DI - Inner diameter of evaporator coil, m

b. Air side energy equation

$$\dot{m}_a \frac{dh_a}{dz} = -\pi D_o \alpha_o \beta \xi (T_w - T_a) \quad (5.3)$$

Where, β— Air side heat transfer ratio

ξ—Moisture desorption coefficient at air side

The evaporator and condenser are both tube fin heat exchangers, which are modeled using the steady-state lumped parameter method. Therefore, the general process is similar to that of the condenser. See Figure for the specific process. The evaporator program needs to input the following parameters: structural parameters of the evaporator, such as the structural form and actual pipe length, the refrigerant inlet temperature, enthalpy value, pressure, air volume, wind speed, temperature, humidity and other state parameters at the air inlet, and the refrigerant mass flow. The calculation differences between evaporator and condenser are as follows:

1. The refrigerant side of the evaporator evaporates and absorbs heat from the air side through the tube wall. The air temperature decreases. The tube wall temperature of each unit is higher than the refrigerant temperature but lower than the air temperature. In the whole process, the enthalpy of refrigerant increases and the enthalpy of air decreases. The refrigerant in the condenser condenses and releases heat, and the air absorbs the heat released by the refrigerant. The tube wall temperature of each unit is higher than the air temperature but lower than the refrigerant temperature. In the whole process, the enthalpy of the refrigerant decreases and the air enthalpy increases.
2. The refrigerant at the evaporator inlet is gas-liquid two-phase fluid, and the refrigerant dryness in the evaporator gradually increases to become superheated gas. Therefore, the heat transfer model of the evaporator mainly includes two calculation areas: the superheated area and the two-phase area. The condenser model calculation mainly includes three calculation areas, including the superheat area, two-phase area and supercooled area.
3. In the evaporator, the coil temperature is lower than the air temperature outside the tube, and there is moisture separation during the heat exchange at the air side of the evaporator. In the condenser, the coil temperature is higher than the air temperature outside the tube, and the heat exchange at the air side is under dry working conditions.
4. The condenser studied in this paper uses a DC variable frequency fan, which will change the air volume of the condenser at a certain condensing temperature and ambient temperature, while the air volume in the evaporator is a certain value.
5. During the modeling of this topic, in order to improve the calculation speed, the pressure drop loss in the condenser was ignored, but in order to control the outlet pressure of the evaporator, the pressure drop in the evaporator cannot be ignored. Therefore, the pressure drop calculation was added in the iteration of the evaporator model. The literature [42] pointed out that the pressure drop loss in the superheated area was one order of magnitude smaller than that in the two-phase area. Therefore, the pressure drop loss in the superheated area was ignored in this modeling, The pressure drop in the two-phase zone is calculated only.

Ⅵ. CONCLUSION

In this chapter, under the MATLAB language environment, a numerical model is established for the variable frequency refrigeration system with multiple intermittent loads, including the mathematical models of variable frequency compressor, condenser, thermal expansion valve and multiple evaporators. And the thermodynamic performance parameters of refrigerant can be obtained by calling Refprop software under the MATLAB language environment, so that simulation analysis can becarried out.

(1) The compressor is the key of the whole refrigeration system. By using the lumped parameter method to establish the model, the relationship between the theoretical gas delivery capacity of the variable frequency compressor and the compressor mass flow rate is obtained, and the exhaust temperature, theoretical power consumption and actual power consumption of the compressor are calculated, providing conditions for the calculation of the condenser model.

(2) The condenser is used as the high temperature heat exchanger in the refrigeration system. This chapter establishes a mathematical model for the heat transfer of the variable air volume condenser by using the steady-state distributed parameter method, obtains the outlet enthalpy value of the condenser and the outlet

temperature of the refrigerant, and then inputs the outlet enthalpy value and temperature of the refrigerant in the condenser into the thermal expansion valve model as the inlet parameters for the expansion valve calculation.

(3) The action response of the expansion valve is fast, and the flow channel is complex. This chapter uses the steady state method to model and analyze it. The process of refrigerant flowing through the thermal expansion valve is a process of throttling and pressure reduction, and the enthalpy value remains unchanged. Using this model to calculate the mass flow through the thermal expansion valve, and compare the sum of the mass flow of each expansion valve with the mass flow of the compressor, which is the key to the whole system simulation.

(4) The evaporator and condenser are both tube fin heat exchangers, so they are similar to the condenser in the calculation process. However, in the evaporator, as a frequency conversion refrigeration system that controls the outlet pressure of the evaporator, the pressure drop in the evaporator coil is an important parameter that must be considered. The pressure drop in the evaporator needs to be calculated. The pressure drop in the superheated area is one order of magnitude smaller than that in the two-phase area. When modeling the evaporator, the pressure drop in the superheated area is ignored.

(5) Finally, the whole variable frequency refrigeration system is coupled according to the relationship between the components, the quantitative and variable in the variable frequency refrigeration system are analyzed and determined, and the evaporation temperature, condensation temperature and compressor frequency are iterated in three layers according to the evaporator outlet pressure, superheat, and refrigerant flow, laying the foundation for the subsequent simulation analysis.

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