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# Research on heat transfer capacity of evaporator under different degrees of superheat

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***Abstract***
*The superheat degree at the outlet of the evaporator is conducive to the full gasification of the refrigerant, and its control is one of the key conditions for the smooth operation of the refrigeration system. In this paper, the heat transfer capacity of the evaporator is studied through simulation experiments, and the optimal adjustment method is found by comparing the outdoor temperature of -16°C and -6°C.*
***Keywords****: superheat degree; refrigerant; outlet state; heat exchange*

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## I INTRODUCTION

The evaporator, as an important part of the refrigeration system is a heat exchange device for heat exchange with the low temperature refrigerant. It works by vaporising the low temperature and low pressure refrigerant liquid flowing through the expansion valve, absorbing and releasing the heat and then transporting the refrigerant gas to the compressor, thereby achieving the purpose of refrigeration. In order to improve the efficiency of the refrigeration cycle, the improvement of the heat exchange capacity of the evaporator is an important goal of various scholars' research. Liang Li[1]analysed the flow path arrangement inside the evaporator and compared the heat transfer capacity of the evaporator under different piping conditions; Tao Hong[2]designed a thermal induction sensor for measuring the refrigerant state at the evaporator outlet and compared the heat transfer efficiency of the evaporator under different states.

In order to fully vaporise the refrigerant in the evaporator, a certain level of superheat is required at the outlet. An inappropriate superheat level can cause "liquid shock" to the compressor, affecting efficiency and reducing the life of the machine. The control of the evaporator superheat is therefore a key condition for the smooth operation of the entire refrigeration system and has been studied by various scholars using simulation software. Yin Youjun[3] used fuzzy control to optimise the control logic of the evaporator superheat electronic expansion valve; Liu Chunlei[4] established a set of control algorithms to achieve accurate control of the superheat.

This paper takes superheat as the main control object and studies the effect of different states of refrigerant at the evaporator outlet on the heat transfer capacity under the condition of superheat at the evaporator outlet.

## II RESEARCH MODEL BUILDING

Refrigerant in the evaporator mainly according to the phase change for heat exchange in the zone called two-phase zone; in the evaporator because the temperature is higher than the corresponding pressure saturation temperature, with a certain degree of superheat in the zone called superheat zone. Air and refrigerant in these two zones of heat exchange temperature changes as shown in Figure 1.1:

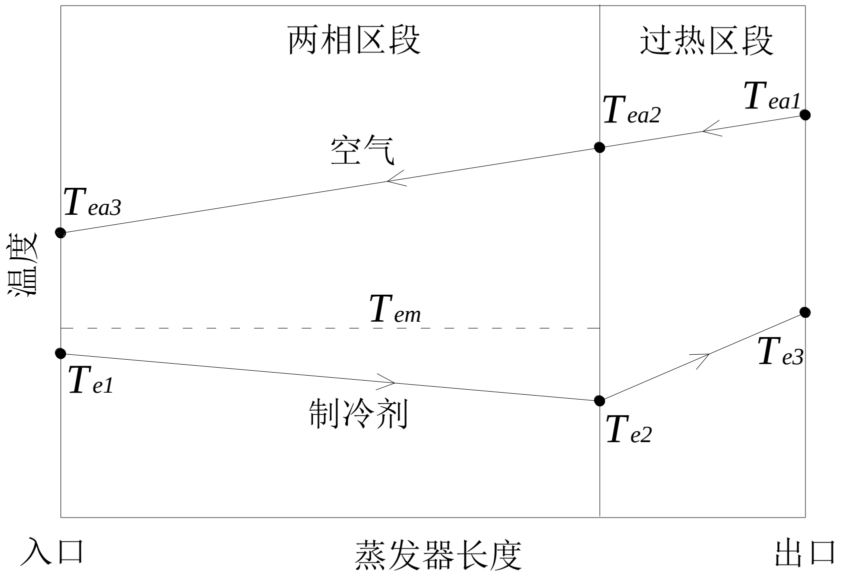


**Figure 1.1 Diagram of the heat transfer zone in the evaporator**

In the evaporator, the refrigerant passes through the two-phase section and the superheat section in turn, and the temperature rises from $T_{e1}$ to $T_{e3}$; the air passes through the superheat section in turn, and the two-phase section undergoes heat exchange, and the temperature drops from $T_{ea1}$ to $T_{ea3}$.

The modelling work in this paper was carried out using MATLAB software in combination with calls to REFPROP for the determination and calculation of the individual parameters of the refrigerant. The evaporator modelling assumptions are as follows:

（I）Neglecting the pressure drop of the refrigerant in the superheated and two-phase zones;

（II）One-dimensional flow of air outside the tube for heat transfer;

（III）Counter-flow heat transfer of air to the refrigerant inside the tube.

The heat transfer area of the two-phase section and the superheated section of the evaporator is calculated by the following equation:

$$A_{e\_tp} = m_{er}\left(h_{e2} - h_{e1}\right) / \left(K_{e\_tp}\Delta T_{e\_tp}\right)$$
$$A_{e\_sh} = m_{er}\left(h_{e3} - h_{e2}\right) / \left(K_{e\_sh}\Delta T_{e\_sh}\right)$$

$A_{e\_tp}$, $A_{e\_sh}$ -Evaporator two-phase section, superheated section heat transfer area

$h_{e1}$, $h_{e2}$, $h_{e3}$ -Refrigerant inlet and outlet enthalpy of the two-phase section of the evaporator, and outlet enthalpy of the superheated section

$m_{er}$ -evaporator refrigerant flow rate

$K_{e\_tp}$, $K_{e\_sh}$ -Total heat transfer coefficient in the two-phase section of the evaporator, superheated section

$\Delta T_{e\_tp}$, $\Delta T_{e\_sh}$ -Two-phase section, log-average temperature difference in the superheated section

The logarithmic mean temperature difference between the two heat transfer zones can be calculated by the following equation:

$$\Delta t_{e\_tp} = \left[\left(T_{ea2} - T_{e2}\right) - \left(T_{ea3} - T_{e1}\right)\right] / In\left[\left(T_{ea2} - T_{e2}\right) / \left(T_{ea3} - T_{e1}\right)\right]$$
$$\Delta t_{e\_sh} = \left[\left(T_{ea2} - T_{e2}\right) - \left(T_{ea1} - T_{e3}\right)\right] / In\left[\left(T_{ea2} - T_{e2}\right) / \left(T_{ea1} - T_{e3}\right)\right]$$

$T_{e1}$, $T_{e2}$, $T_{e3}$ -Inlet and outlet temperatures of the refrigerant in the two-phase section of the evaporator, in the superheat section outlet temperature in the superheated zone

$T_{ea1}$, $T_{ea2}$, $T_{ea3}$ -Air inlet and outlet temperatures in the superheated section of the evaporator and outlet temperature in the two-phase section

The heat transfer coefficient[5] can be calculated by the following equation:

$$\frac{1}{K}=\frac{1}{\alpha_i}+R_w+\frac{A_i}{\alpha_o A_o}$$

$K$ -Total heat transfer coefficient of a single heat transfer zone

$\alpha_o$ -Heat transfer coefficient on the air side

$\alpha_i$ -Heat transfer coefficient on the refrigerant side

$R_w$ -Tube wall thermal resistance (with fins)

$A_o$ -Heat transfer area on the outer side of the heat exchanger tube

$A_i$ -Heat transfer area on the inner side of the heat exchanger tube

As the heat transfer process in the superheated section is dry in the evaporator model, the heat transfer coefficient on the air side of the superheated section is calculated by the following equation:

$$\alpha_o=\alpha_{of}\frac{a_f\eta_f+a_b}{a_f+a_b}$$

$$\eta_f=\frac{th(mh')}{mh'}$$

$$h'=\frac{d_o}{2}\left(\frac{s_1}{d_o}-1\right)\left[1-0.35\ln\left(1.063\frac{s_1}{d_o}\right)\right]$$

$$m=\sqrt{\frac{2\xi\alpha_{of}}{\lambda_L\delta_f}}$$

$a_f$ -Side area of fins per meter of tube length

$a_b$ -Tube area between fins per meter of tube length

$\eta_f$ -Fin efficiency

$h'$ -Fin Fold Height

$s_1$ -Pipe spacing

$m$ -Fin Specification

$\lambda_f$ -Thermal conductivity of fins

$\delta_f$ -Fin thickness

Evaporator two-phase section of the heat transfer process for wet conditions, the air has water analysis out, two-phase area air side heat transfer coefficient need to consider the precipitation coefficient, the calculation formula is as follows:

$$\alpha_o=\xi\alpha_{of}\frac{a_f\eta_f+a_b}{a_f+a_b}$$

$$\xi=\frac{h_{a2}-h_{a3}}{C_{pa}\left(T_{a2}-T_{a3}\right)}$$

$$m=\sqrt{\frac{2\xi\alpha_{of}}{\lambda_L\delta_f}}$$

$\xi$ -Coefficient of moisture dispersion

$h_{a2}$ , $h_{a3}$ -Inlet and outlet enthalpy of wet air in the two-phase section of the evaporator

For the superheated section, the refrigerant-side heat transfer coefficient is calculated from the Dittus-Boeler equation [7]correlation:

$$\alpha_i=\frac{\lambda N_u}{d_i}$$

$$N_u=0.023R_e^{0.8}\,\mathrm{P}_r^{0.3}$$

$$R_e=\frac{G_i d_i}{\mu}$$

$$P_r = \frac{\mu C_p}{\lambda}$$

$R_e$ -Reynolds number

$P_r$ -Prandtl number

$N_u$ -Refrigerant Nussle number

$G_i$ -Mass flow rate

$d_i$ -Inner diameter of tube

$\mu$ -Power Viscosity

$C_p$ -Constant pressure specific heat capacity

$\lambda$ -Thermal conductivity

The heat transfer coefficient of the refrigerant side of the superheated section of the evaporator is calculated by the following formula, and the heat transfer coefficient of the refrigerant side of the two-phase section of the evaporator is calculated by Kandlikar correlation formula:

$$\frac{\alpha_{tp}}{\alpha_l} = C_1 \left(C_0\right)^{C_2} \left(25F_{rl}\right)^{C_5} + C_3 \left(B_0\right)^{C_4} F_{fl}$$

$$\alpha_l = 0.023 \left( \frac{g\left(1-x\right)d_i}{\mu_l} \right)^{0.8} \frac{{P_{rl}}^{0.4} \lambda_l}{d_i}$$

$$C_0 = \left( \frac{1-x}{x} \right)^{0.8} \left( \frac{\rho_g}{\rho_l} \right)^{0.5}$$

$$B_0 = \frac{q}{gr}$$

$$F_{rl} = \frac{g^2}{9.8 {\rho_l}^2 d_i}$$

$\alpha_{tp}$ -Heat transfer coefficient on the refrigerant side of the two-phase section

$\alpha_l$ -Heat transfer coefficient on the refrigerant side of the single-phase section

$C_0$ -Convective characteristic number

$B_0$ -Boiling characteristic number

$F_{rl}$ -Liquid phase Froude number

$g$ -Mass flow rate

$x$ -Two-phase zone dryness

$d_i$ -Inner diameter of tube

$\mu_l$ -Liquid phase kinetic viscosity

$\lambda_l$ -Liquid phase thermal conductivity

$P_{rl}$ -Liquid phase Prandtl number

$\rho_g$ , $\rho_l$ -Gas and liquid phase densities

$q$ -Heat flow density

$r$ -Latent heat of vaporization

The pressure drop in the two-phase zone of the evaporator refrigerant is calculated by the following equation:

$$\Delta P_{tp} = \Delta P_f + \Delta P_d + \Delta P_g$$

$\Delta P_f$ -Frictional pressure drop

$\Delta P_d$ -Accelerated pressure drop

$\Delta P_g$ -Gravitational pressure drop

Since the heat exchanger is placed horizontally, the $\Delta P_g$ is 0; the acceleration pressure drop $\Delta P_d$ is calculated as follows:

$$\Delta P_d = G^2 \Delta \upsilon$$

$G$ **-**Refrigerant mass flow rate

$\Delta\upsilon$ -Specific inlet/outlet capacity difference between the two phases of the refrigerant

The two-phase zone friction pressure is calculated by the Lockhart-Martinell correlation:

$$\Delta P_f = \varphi_l^2 \Delta P_l$$

$$\Delta P_f = \varphi_g^2 \Delta P_g$$

$$\Delta P_l = f_l \frac{L}{D_i} \times \frac{G^2 (1-x)^2}{2\rho_l}$$

$$\Delta P_g = f_g \frac{L}{D_i} \times \frac{G^2 x^2}{2\rho_g}$$

$$\phi_l = 1 + \frac{20}{X_{tt}} + \frac{1}{X_{tt}^{\ 2}} (R_e > 4000)$$

$$\phi_g = 1 + 20X_{tt} + X_{tt}^2 (R_e < 4000)$$

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_g}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_g}\right)^{0.1}$$

$\phi_l$ , $\phi_g$ -Reinforcement factors by refrigerant liquid and gas phase

$\Delta P_l$ , $\Delta P_g$ -Pressure drop in the liquid and gas phases

$L$ -Flow tube length of refrigerant in the two-phase zone

$f_l$ , $f_g$ -Friction coefficients calculated using liquid and gas phase mass flow densities

$X_{tt}$ -Martinelli parameters

$\mu_g$ -Gas phase kinetic viscosity of refrigerants

The $f_l$ and $f_g$ friction coefficients are calculated using the Blasius formula:

$$f = 0.3164 R_e^{-0.25}$$

$$R_{el} = \frac{G(1-x) d_i}{\mu_l}$$

$$R_{eg} = \frac{G x d_i}{\mu_g}$$

$R_{el}$ , $R_{eg}$ -Reynolds number calculated using liquid and gas phase mass flow densities

## III SIMULATION EXPERIMENT PROCESS

In this paper, the model is used to simulate the evaporator outlet heat exchange change pattern under the test conditions, comparing the simulated outdoor temperature of -16°C with -6°C. The process of the simulation experiment is shown in the following figure:

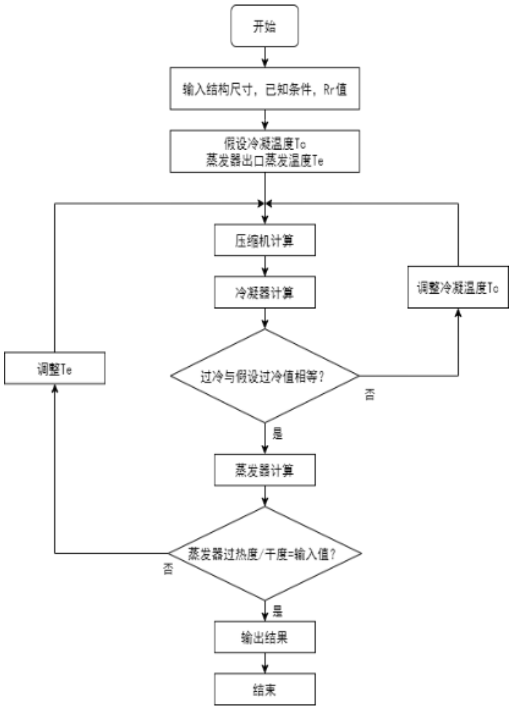


**Figure 2.1 Flow chart of the simulation experiment**

## IV CONCLUSION

Figures 3.1 and 3.2 show the change in heat exchange for outdoor temperatures of -6°C and -16°C respectively. When the outdoor ambient temperature drops from -6°C to -16°C, the heat exchange of the evaporator decreases by approximately 24%. The maximum value of heat exchange in the overall heat exchange process for different operating conditions occurs at a refrigerant dryness of 0.96. Comparing the heat exchange of the evaporator at a superheat of 7°C with the maximum value, it can be found that the heat exchange at the evaporator outlet with liquid under both operating conditions is 21% and 24% higher than that at the outlet superheat respectively.

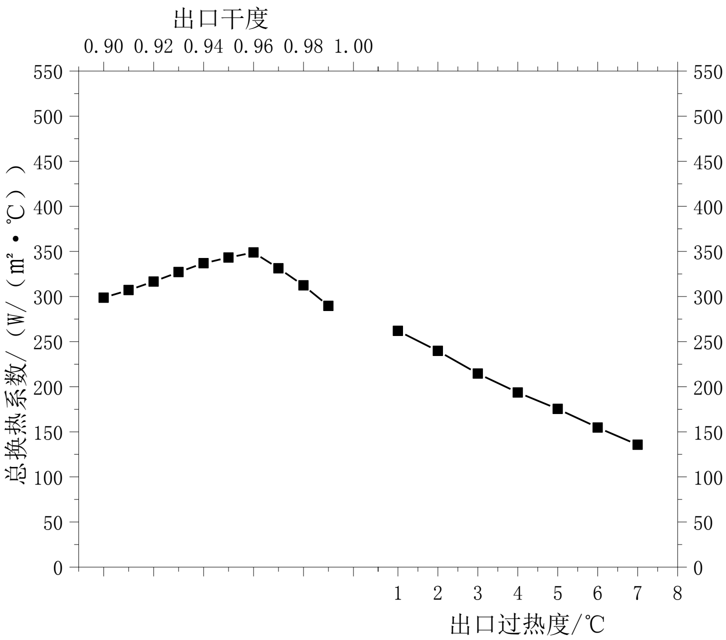


**Figure 3.1 Variation of heat exchange at -6°C with evaporator outlet state**

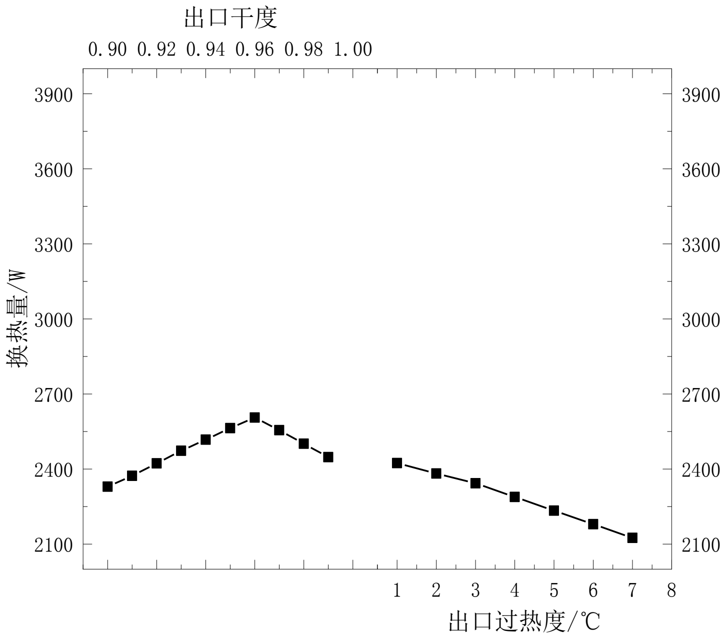


**Figure 3.2 Variation of heat exchange at -16°C with evaporator outlet state**

This phenomenon is due to the fact that when the dryness of the refrigerant reaches a certain value, the effect on the heat exchange temperature difference becomes gradually worse, and at this time the amount of liquid carried in the compressor Therefore, when the dryness of the refrigerant continues to fall, the heat exchange continues to fall. As shown in the graph, the heat exchange increases from 3351.39W to 3406.48W and then decreases to 3346.71W when the dryness changes from 0.95 to 0.97. In the superheat section, the heat exchange decreases with the increase of superheat, and the heat exchange changes monotonically and decreases more.

The change of heat exchange in 16℃working condition can be seen: in the two-phase section, although the heat exchange of the evaporator in the suitable dryness range of 0.95~0.97 does not change much, however, when the amount of liquid carried is too large, the heat exchange still decreases significantly, at the dryness of 0.96, the maximum heat exchange is 2608.96W, at the dryness of 0.90 the heat exchange is 2329.48W, the heat exchange decreases by 10.73%, the reduction of heat exchange amounted to 279.48W, while at this time the heat exchange in the state of superheat degree of 7℃was 480.81W less than the maximum heat exchange, and the heat exchange in the state of dry degree of 0.90 was only 201.52W higher than that in the state of superheat degree of 7℃, although the improvement of heat exchange by liquid at the exit of evaporator still existed at this time, however, the effect was not obvious.

Through the two working conditions of data comparison can be concluded: evaporator outlet reasonable with liquid under the circumstances of its heat exchange is much higher than the conventional outlet 5 ℃~ 7 ℃ superheat heat exchange, if the evaporator outlet dryness control in the range of 0.95 ~ 0.97 can realize the evaporator high efficiency heat exchange.

## REFERENCES

[1]. LIANG Li,WANG Wei,LIU Jiying,ZHANG Linhua. Simulation study on the effect of superheat on the performance of finned tube evaporators[J]. Building Energy Conservation,2017,45(02):105-108.

[2]. TAO Hong,TAO Leren,ZHENG Zhigao,ZHANG Lijun. Experimental study of thermal measurement of evaporator outlet state[J]. Journal of Engineering Thermophysics,2009,30(11):1920-1922.

[3]. Yin Youjun,Zhang Baohuai. Simulation and analysis of fuzzy control of evaporator electronic expansion valve superheat degree[J]. Energy Research and Utilization,2005(02):15-18.

[4]. Liu Chunlei,Bai Xue,Liu Haiwei,Wang Ruxin. Research on control algorithm of evaporator superheat degree and software implementation[J]. Journal of Hebei College of Construction Engineering,2017,35(02):76-79+89.

[5]. Zhang Chunlu. Principles and techniques of refrigeration and air conditioning system simulation [M]. Beijing:Chemical Industry Press,2012:77-107.

[6]. Ding Guoliang, Zhang Chunlu. Simulation and optimization of refrigeration and air conditioning units [M]. Beijing: Science Press, 2001.

[7]. Liang Caihua,Zhang Xiaosong,Xu Guoying. Simulation and experimental study of the effect of superheat variation on the performance of refrigeration systems[J]. Fluid Mechanics,2005(09):43-47.

[8]. Shi W. X., Tian C. Q., Wang P. L., eds. Refrigeration technology for air conditioning 5th edition [M]. Beijing: China Construction Industry Press,2016.05.

[9]. Mohamed R. Salem. Performance enhancement of a vapor compression refrigeration system using R134a/MWCNT-oil mixture and liquid-suction heat exchanger equipped with twisted tape turbulator[J]. International Journal of Refrigeration,2020,120.

[10]. S.A. Klein,D.T. Reindl,K. Brownell. Refrigeration system performance using liquid-suction heat exchangers[J]. International Journal of Refrigeration,2000,23(8).

[11]. Le Cao,Jinping Liu,Rixin Li,Shizhe Huang,Fayong Zhang,Xiongwen Xu. Experimental study on the mixed refrigerant heat transfer performance in a plate-fin heat exchanger during a single-stage cryogenic cycle[J]. Applied Thermal Engineering,2016,93(C).
[12]. Rajvir Singh Doohan,Pradeep Kumar Kush,Govind Maheshwari. Exergy based optimization and experimental evaluation of plate fin heat exchanger[J]. Applied Thermal Engineering,2016,102.
[13]. Weiqiang Pang,Jinping Liu,Jiaji He,Xiongwen Xu. Thermal performance of brazed plate heat exchangers for a mixed-refrigerant Joule–Thomson cooler[J]. International Journal of Refrigeration,2016,61.
[14]. Mirza M. Shah. A method for predicting heat transfer during boiling of mixtures in plain tubes[J]. Applied Thermal Engineering,2015,89.
[15]. Weiqiang Pang,Jinping Liu,Xiongwen Xu. A strategy to optimize the charge amount of the mixed refrigerant for the Joule–Thomson cooler[J]. International Journal of Refrigeration,2016,69.