
Characteristics of Vapor-Compression Cooling System on Multicomponent Working Medium

¹Anna V. Pleshka, ²Arthur R. Valinurov

^{1,2}Naberezhnye Chelny Institute, Kazan Federal University, Naberezhnye Chelny, Republic of Tatarstan, Russian Federation

Email: anya.pleshka@mail.ru

Received: 20th August 2019, Accepted: 30th September 2019, Published: 31st October 2019

Abstract

The article presents the structural diagram of the static mathematical model for vapor compression cooling system (VC CS), which is the sequence of units with known transfer functions of thermal processes and the static mathematical model that allows to calculate the static characteristics of the considered CS in the form of cooling capacity dependence on the ambient temperature and the mass flow rate of cooling water.

The non-azeotropic mixture R423A was chosen as the working agent for static characteristics development, as the most environmentally safe among single-component refrigerants, and with greater energy efficiency for reverse cycles of heat engines. The evaluation criterion was the condition of lower energy consumption for a working agent compression in the compressor at a sufficiently high value of the refrigeration coefficient.

The calculation result showed that with the cooling water temperature decrease at the inlet to the condenser t_{w1} with a constant flow of cooling water G_w , the heat load of the evaporator Q_{ev} will increase. A sharp increase of the evaporator heat load Q_{ev} was also observed with flow rate increase G_w to 0.4 kg/s, and with a further increase of cooling water flow rate, the heat load of the evaporator gradually increases. The dependence of the heat load of the evaporator Q_{ev} on the temperature of the cooling water t_{w1} is linear.

Keywords

Steam Compression Cooling System, Static Mathematical Model, Freon, Compressor, Boiling, Evaporation, Temperature Head.

Introduction

Steam compression chillers are the most common for artificial cooling in the moderate cold area [1]. They are widely used in cryogenic and refrigeration equipment for cooling, freezing and storage due to their energy efficiency (lower energy consumption as compared to other machines) and less environmental hazard. Currently, refrigeration equipment does not have one-component working substances with simultaneously high thermodynamic, thermophysical and environmental indicators, and in this regard, they started to apply mixtures consisting of one-component substances authorized for production and use by the Montreal Protocol [2].

In particular, non-azeotropic refrigerants have such advantages that during the circulation of the working fluid along the circuit of the refrigeration system, its composition changes, which can lead to cooling capacity and refrigeration coefficient increase as compared with the characteristics for pure refrigerants [3]. Thus, changing the composition of non-azeotropic mixtures over a wide range, it is possible to obtain such properties of the working fluid that will provide the greatest (in specific conditions) efficiency of the refrigeration machine, which is estimated by the refrigeration coefficient. When they use non-azeotropic refrigerants, the aim is to increase refrigeration capacity, reduce the temperature of the end of compression, improve the circulation conditions of oil in the system, expand the range of applications for boiling and condensation temperatures, and provide the desired combination of refrigeration capacity with pressure levels. In some cases, the use of non-azeotropic mixtures can reduce the influence of the finite temperature difference on the efficiency of the thermodynamic cycle by performing cycles with variable boiling and condensation temperatures. At that, during vapor compression cooling system design operating on these mixtures, it is necessary to take into account such features as evaporation with increasing temperature, condensation with decreasing temperature, the influence of mixture component concentration on heat transfer coefficients, and other factors.

Besides, in the absence of control actions, the system can switch to a new operating mode with a possible deterioration in performance, which necessitates automatic control [4].

Therefore, the aim of this work is to develop a static mathematical model and calculate the static characteristics of a vapor compression cooling system on a multicomponent working agent.

Methods

Instead of refrigerants prohibited by the Montreal Protocol, it is proposed to use alternative refrigerants that comply with international environmental agreements [5]. Compression chillers use not only pure refrigerants and azeotropic mixtures, but also non-azeotropic mixtures, the distinguishing feature of which is the difference in the equilibrium concentrations of the components in the liquid and gas phases [6]. Changing the composition of non-azeotropic mixtures over a wide range, it is possible to obtain such properties of the working fluid that will provide the greatest

(in specific conditions) efficiency of a refrigeration machine. Given this, initially one-component refrigerants were selected that are acceptable in terms of environmental performance, toxicity and flammability standards. Therefore, the calculations were performed for non-azeotropic mixtures containing these components. Further selection of refrigerants was made according to the main energy indicators of the cooling system, based on the conditions of energy cost reduction for the working agent compression in the compressor and the refrigeration coefficient increase. Energy performance was estimated using data interpolation from the tables of thermodynamic properties of multicomponent refrigerants [7]. During the analysis of the calculation results, the non-azeotropic refrigerant R423A was selected for further research.

The static mathematical model for the study was compiled according to the works [8, 9], while the basis for its development was the structural diagram of the simulated object, which is the sequence of units with known static characteristics of elementary thermal processes (Fig. 1) [10]. For the model of the vapor compression cooling system under consideration, the determining influences are the mass flow rate of cooling water G_w through the heat exchanger-condenser and its temperature t_{w1} , and it is necessary to take into account the effect of the flow rate G_w on the heat transfer coefficient of the heat exchanger-condenser k_k tubes.

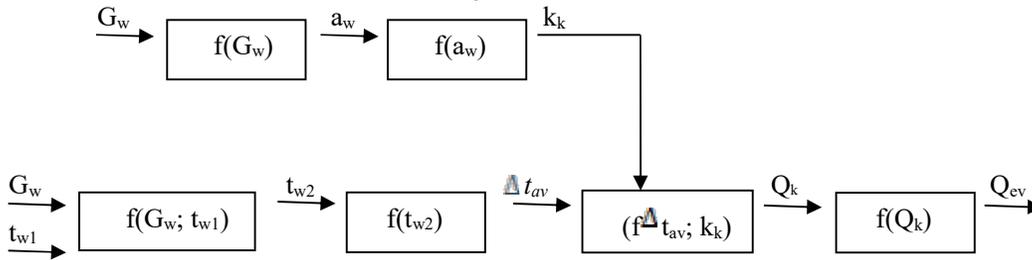


Figure 1: Block Diagram of the Vapor Compression Cooling System

In order to determine the first function $t_{w2} = f(G_w, t_{w1})$ let's put down the heat balance equation for the heat exchanger-condenser, in which the heat from the hot working agent is removed by cooling water, while the working agent condenses and completely passes from the gaseous to the liquid state. The equation of the condenser heat balance has the following form, kJ/s:

$$Q_k = G_a \cdot q_k = G_w \cdot c_{pw} \cdot (t_{w2} - t_{w1}) \tag{1}$$

where Q_k is the thermal load on the capacitor, kW; G_a - mass flow rate of the working substance, kg/s; $q_k = i_2 - i_3$ - the specific heat consumption per unit flow rate of the refrigerant during condensation, kJ/kg, here i_2 and i_3 is the refrigerant enthalpy at the operating points of the cycle at the inlet and outlet of the heat exchanger-condenser; t_{w1} and t_{w2} - water temperature at the inlet and outlet of the heat exchanger-condenser, °C; c_{pw} - specific mass isobaric heat capacity of water, kJ/(kg·K).

A stable thermal regime of the evaporator of the vapor compression cooling system during change of cooling water qualitative and quantitative parameters (temperature t_{w1} and mass flow rate G_w) circulating through the heat exchanger-condenser can be achieved by excluding the water temperature t_{w2} at the outlet of the condenser from the heat balance equation (1), based on the condition that it is not necessary to control its value, then we obtain the temperature of outlet water from the heat balance equation (1):

$$t_{w2} = t_{w1} + \frac{G_a \cdot q_k}{G_w \cdot c_{pw}} \tag{2}$$

To determine the second function $\Delta t_{av} = f(t_{w2})$ (the average temperature head in the heat exchanger-condenser) and simplify the mathematical transformation, we can use the arithmetic expression used for small values of the ratio $\Delta T_b / \Delta T_m < 2$ of the coolant temperature differences circulating on both sides of the heat transfer surface, °C [6]. Considering the average temperature head as the arithmetic mean and taking into account the expression (2):

$$\Delta t_{av} = \frac{(t_k' - t_{w1}) + (t_k'' - t_{w2})}{2} = \left[\frac{t_k'}{2} - \frac{t_{w1}}{2} + \frac{t_k''}{2} - \frac{1}{2} \cdot \left(t_{w1} + \frac{G_a \cdot q_k}{G_w \cdot c_{pw}} \right) \right] \tag{3}$$

$$\Delta t_{av} = \left(\frac{t_k'}{2} - \frac{t_{w1}}{2} + \frac{t_k''}{2} - \frac{t_{w1}}{2} - \frac{G_a \cdot q_k}{2 \cdot G_w \cdot c_{pw}} \right) = \left(\frac{t_k' + t_k''}{2} - t_{w1} - \frac{G_a \cdot q_k}{2 \cdot G_w \cdot c_{pw}} \right) \tag{4}$$

where t_k' and t_k'' - are the condensation temperatures of the refrigerant at the condensation pressure on the saturated liquid line and the saturated vapor line, respectively, °C.

Given the known heat-transfer surface area F_k of the heat exchanger-condenser tubes and the heat transfer coefficient k_k , in accordance with the Fourier law, we write the equation for the third function $Q_k = f(\Delta t_{cp})$ in the

structural diagram (Fig. 1), which determines the value of the amount of heat transferred through the wall of the heat exchanger-condenser kJ/s:

$$Q_k = k_k \cdot F_k \cdot \Delta t_{av} \quad (5)$$

where Δt_{av} is the temperature head in the heat exchanger-condenser, °C. The amount of heat transferred by the working agent to the cooling water through the wall of the heat exchanger-condenser in the process of its complete condensation is obtained by solving the system of three equations: the temperature of the cooling water at the outlet of the heat exchanger-condenser (2), the arithmetic mean temperature head (4) and heat transfer through the tube wall (5). This solution has the following form:

$$Q_k = k_k \cdot F_k \cdot \left(\frac{t_k^i + t_k^o}{2} - t_{w1} - \frac{G_w \cdot q_k}{2 \cdot G_w \cdot c_{pw}} \right) \quad (6)$$

Heat transfer coefficient through the walls of the condenser tubes $k_k(G_w)$, W/m²*K:

$$k_k(G_w) = \frac{1}{\frac{1}{\alpha_w(G_w)} + R_{TC,k} + \frac{1}{\alpha_k}} \quad (7)$$

Heat transfer coefficient from the water in the condenser $\alpha_w(G_w)$, W/m²*K:

$$\alpha_w(G_w) = \frac{Nu_w \cdot \lambda_w}{d_{in}} = \frac{\lambda_w}{d_{in}} \cdot (0,021 \cdot Re_w^{0,8} \cdot Pr_w^{0,43} \cdot \varepsilon_{trns}) \quad (8)$$

where Nu_w is the Nusselt number; λ_w – the coefficient of thermal conductivity, W/(m*K); d_{in} – the inner diameter of the heat exchanger-condenser tubes; Re_w – Reynolds number; Pr_w – Prandtl criterion; $\varepsilon_{trns} = 1$ – transition mode ratio.

$$\alpha_w(G_w) = \frac{\lambda_w}{d_{in}} \cdot (0,021 \cdot \left(\frac{w \cdot d_{in}}{\nu_w} \right)^{0,8} \cdot Pr_w^{0,43} \cdot \varepsilon_{trns}) \quad (9)$$

where w is the water velocity in the apparatus, m/s; ν_w – kinematic viscosity of water, m²/s.

$$\alpha_w(G_w) = \frac{\lambda_w}{d_{in}} \cdot \left[0,021 \cdot \left[\left(\frac{4 \cdot G_w}{\pi \cdot \rho_w \cdot d_{in}^2 \cdot n_{1x}} \right) \cdot \frac{d_{in}}{\nu_w} \right]^{0,8} \cdot Pr_w^{0,43} \cdot \varepsilon_{trns} \right] \quad (10)$$

where ρ_w – water density, kg/m³; n_{1x} – the number of pipes in one stroke of the heat exchanger.

Let's put down the fourth equation for the function $Q_{ev} = f(Q_k)$ on the structural diagram (1), which determines the thermal load on the vapor compression cooling system, characterized by the amount of heat Q_{ev} removed from the heat carrier in the heat exchanger-evaporator. To do this, we compose the equation of material balance, based on the equality of the mass flow of the working agent circulating in the cooling system. We get the following equality:

$$Q_{ev} = Q_k = Q_{kom} \quad (11)$$

where $Q_k = G_k \cdot q_k$ is the thermal load on the capacitor; $Q_{ev} = G_{ev} \cdot q_{ev}$ is the heat load on the evaporator, here $q_{ev} = i_1 - i_4$ – the specific heat consumption per unit flow rate of the refrigerant during evaporation, kJ/kg, here i_1 and i_4 are the enthalpies of the refrigerant at the operating points of the cycle at the outlet and inlet of the heat exchanger-evaporator.

Then the heat load of the heat exchanger-evaporator is the following:

$$Q_{ev} = G_{ev} \cdot q_{ev} = G_k \cdot q_{ev} = \frac{Q_k}{q_k} \cdot q_{ev} = \frac{q_{ev}}{q_k} \cdot Q_k \quad (12)$$

Thus, after the conversion, we finally obtain a static mathematical model of a vapor compression cooling system on a multicomponent working agent, which will be represented by the following equation:

$$Q_{ev}(G_w, t_{w1}) = \frac{q_{ev}}{q_k} \cdot k_k(G_w) \cdot F_k \cdot \left(\frac{t_k^i + t_k^o}{2} - t_{w1} - \frac{G_w \cdot q_k}{2 \cdot G_w \cdot c_{pw}} \right) \quad (13)$$

Results and Discussion

The model allows you to determine the amount of heat removed in the evaporator from the coolant, depending on changes of cooling water temperature t_{w1} and the mass flow rate G_w of cooling water (Figure 2, Figure 3).

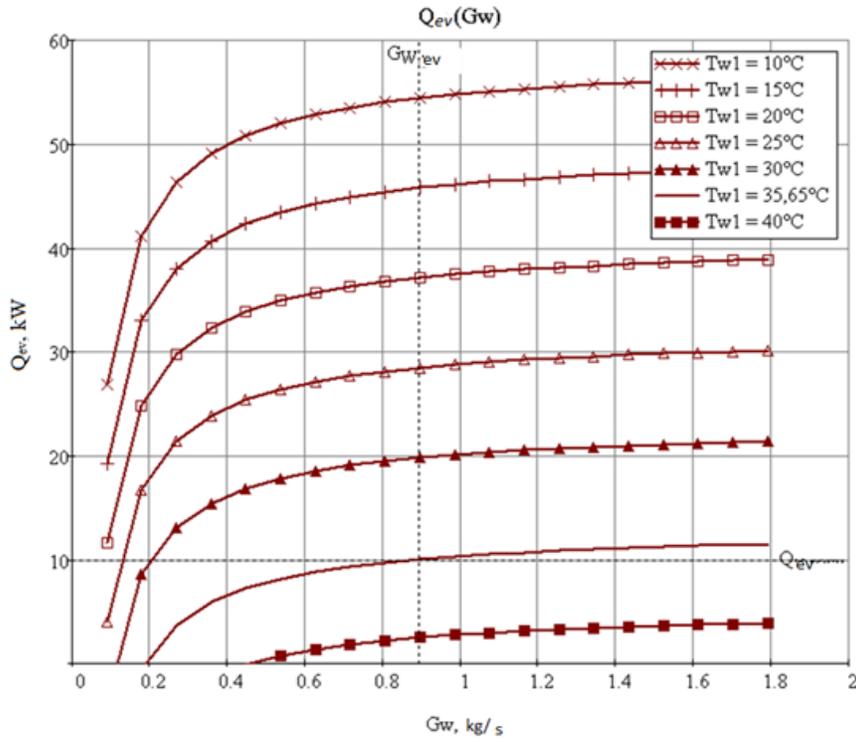


Figure 2: Dependence of the Evaporator Heat Load on the Flow of Cooling Water at Different Water Temperatures at the Inlet to the Condenser

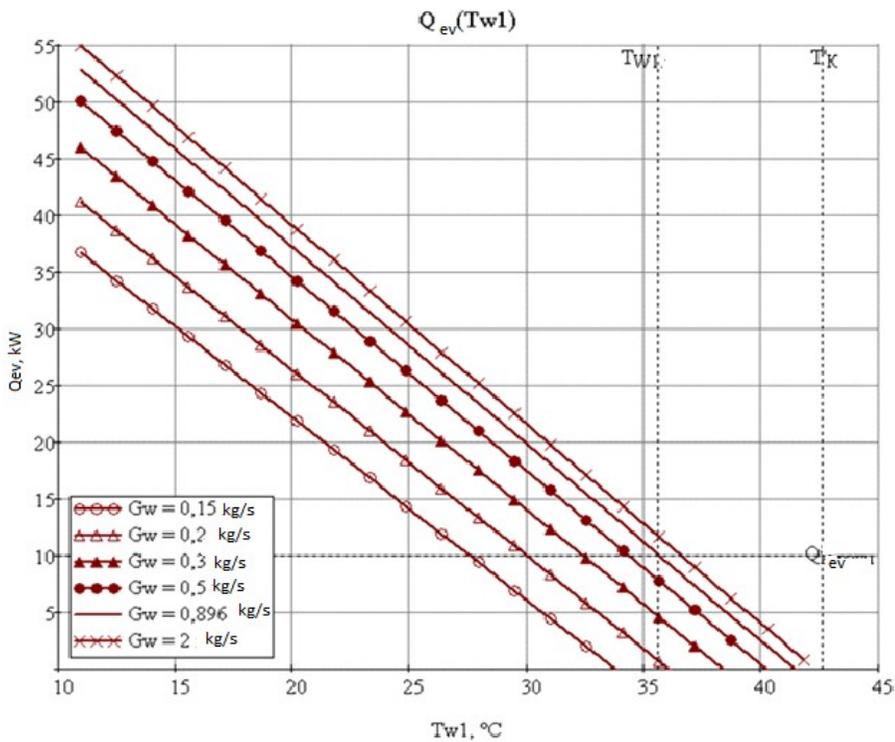


Figure 3: Dependence of the Evaporator Heat Load on the Temperature of Cooling Water at the Inlet to the Condenser at Different Flow Rates of Cooling Water

According to these graphs, it can be seen that with cooling water temperature decrease at the inlet to the condenser t_{w1} and its constant mass flow G_w , the heat load of the evaporator Q_u will increase. Figure 2 shows a sharp increase of the evaporator heat load with the flow rate increase of cooling water G_w to 0.4 kg/s. With a further increase of

cooling water flow rate, the heat load of the evaporator increases smoothly. The dependence of the evaporator heat load on the temperature of the cooling water (Fig. 3) is linear.

Conclusions

Thus, the mathematical model was compiled and the static characteristics of the vapor compression refrigeration machine on a multicomponent working agent were calculated. They considered the dependence of the heat load of the evaporator on the flow of cooling water with a changing temperature of the water at the inlet to the condenser and the dependence of the heat load of the evaporator on the temperature of the cooling water at the inlet of the condenser with a changing flow of cooling water. The obtained static model makes it possible to evaluate the performance parameters of the steam compression cooling system under consideration: mass flow of the working agent, the amount of heat Q_H , the heat transfer coefficient in the condenser k_K and the heat transfer surface area F_K of the condenser.

Acknowledgements

The work is performed according to the Russian Government Program of Competitive Growth of Kazan Federal University.

References

1. Baranenko A.V. Refrigerators: the textbook for students of technical colleges studying "Engineering and Physics of Low Temperatures" / A.V. Baranenko, N.N. Bukharin, V.I. Pekarev, I.A. Sakun, L.S. Timofeevsky; Ed. by L.S. Timofeevsky. - St. Petersburg: Polytechnic, 1997. - 992 p.
2. Morozyuk T.V. Theory of refrigeration machines and heat pumps / T.V. Morozyuk. - Odessa: Studio "Negociant", 2006. - 712 p.
3. Babakin B.S. Alternative refrigerants and refrigeration system service based on them / B.S. Babakin, V.I. Stefanchuk, E.E. M.: Kolos, 2000. - 160 p.
4. Tatarenko Yu.V. Automation of the main elements of refrigeration machines: Textbook. / Yu.V. Tatarenko - St. Petersburg: ITMO University, 2015. - 43 p.
5. Tsvetkov, O.B., Laptev, Yu.A. Refrigerants and environment // Journal of Physics: Conference Series. 2017. Vol. 891. Article number 012218. pp. 1-3.
6. Mogorychny, V.I. Boiling of multicomponent working fluids used in refrigeration and cryogenic systems / V.I. Mogorychny, A.S. Dolzhikov // IOP Conf. Series: Journal of Physics: Conf. Series. 2017. Vol. 891. Article number 012022. pp. 1-13.
7. PC software: Reference Fluid Thermodynamic and Transport Properties v9. – Access mode: <https://www.nist.gov/srd/refprop>.
8. Karelin D.L., Gureev V.M., Muljukin V.L. Mathematical Model Of Liquid Vapor Compression System For Multicircuit Cooling Systems Of High Mobile Platform // International Journal of Applied Engineering Research. 2015. - №24. - pp. 45150-45155.
9. Karelin D.L., Boldyrev A.V., Boldyrev S.V., Belousov A.M. Modeling of dynamics of vapor compression cooling system // IOP Conference Series: Materials Science and Engineering. - 2018. - Vol.412, Is.1. - Art. № 012032. - pp. 1–4.
10. Ahmet K., Onder K., Ali K.Y. Performance and exergetic analysis of vapor compression refrigeration system with an internal heat exchanger using a hydrocarbon, isobutane (R600a). Int. J. Energy Res. 2008; 32:824–836.