Numerical simulation of boiling temperature dynamics in two-phase refrigerant flow in horizontal pipes

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Abstract. The transient dynamic characteristics of shell-and-tube multi-pass heat exchanger-evaporator of vapor compression cooling system are studied. Based on the balance of thermal power and the criterial heat transfer equations the mathematical model of the physical processes taking place in the heat exchanger-evaporator is compiled. The change in the coefficient of heat transfer due to a change in the two-phase regime of the working refrigerant flow is taken into account, as well as the change in time of the mass and vapor content of the working refrigerant in the evaporator over time. The change in time of the mass flow of the working refrigerant at the start of the cooling system from the standby mode is determined using the specified function of changing shaft speed of the compressor in time. The simplifying assumptions accepted for compilation of the mathematical model are listed. The differential equation of evaporation temperature dynamics is solved by the fourth-order Runge-Kutta method with a fixed step. The change in evaporation temperature in time at two values of the ambient temperature is studied. The existence of abrupt drop zone of evaporation temperature on the transient characteristics within 0.1 seconds after the system start from standby is revealed and explained.

1. Introduction

Processes of heat transfer during the phase transition of working refrigerants are widely used in industry and at home. Cooling, ventilation and air conditioning systems, refrigeration equipment and heat pumps represent small, well-known sphere of practical application of the reverse Carnot cycle underlying these vapor compression machines. For effective control of heat flow in heat exchangers (condensers, evaporators) there are needed the algorithms taking into consideration dynamics of phase transition processes when the environment, heat load and compressor parameters are changing [1]. For this purpose, active studies of the transient dynamic characteristics of thermal processes in vapor compression machines are conducted, and their results are presented in the works of many authors.

The authors of [2] proposed the dynamic model of compression cooling systems, based on the laws of conservation of mass and energy. As a result of the numerical study, some characteristics, in particular the cooling capacity and the coefficient of cold transformation, are obtained. In the conclusions of the work it was noted that the speed of the compressor shaft and the mass flow of the refrigerant quickly reach their steady values.

The article [3] is devoted to the development of a dynamic model of a heat pump with a spiral compressor. It’s authors proposed a model, similar to work [2], based on the energy balance equations and the criterial heat transfer equations for two-phase flows. The numerical solution of the equations
of the mathematical model was carried out by the Runge-Kutta method of the fourth order. As a result, the transient characteristics of the temperatures during condensation and evaporation of the working refrigerant in heat exchanger-condenser and heat exchanger-evaporator were obtained.

From the results of study of heat transfer during phase transitions, it is known that in process of condensation inside round tubes, the flow of coolant has a strongly marked stratified nature of the flow and the heat transfer is described by one criterial equation [4].

Thereby the boiling of the two-phase flow in the heat exchanger-evaporator is of particular interest. In this case the characteristics of the phase transition process depend on the vapor content, and the heat transfer is described by several criterial equations, according to the flow regime map [5].

Accounting for modeling the effect of various factors and internal processes has a significant influence on the accuracy of the obtained results. The mathematical models presented in literature do not take into account the change in the vapor content in evaporator, which, depending on the rate of pressure increase in the compressor, can vary significantly in the time interval from the start-up of the system until it reaches a steady state. In addition, the numerical value of heat transfer at boiling point for each time point is determined using the criterial equation only for wet flow regimes, which may have an effect on the accuracy of calculating the transient characteristics of the vapor compression cooling system due to low vapor content at the initial time point.

![Figure 1](image1.png)

**Figure 1.** Diagram of the structures of two-phase refrigerant flows in horizontal evaporator pipes: I – stratified with a smooth interface, II – stratified with a wave interface, III – shell structure, IV – dispersed structure, V – dispersed-ring structure, VI – ring structure, points – experimental data, lines – approximating functions (1).

Obviously, to calculate the heat transfer coefficient from the working refrigerant in the heat exchanger-evaporator it is necessary to consider the two-phase flow regime over the whole range of vapor content values. For this purpose, the authors of the work carried out an approximation of the diagram of the structures of two-phase refrigerant flows obtained by A.V. Gordienko (figure 1) [6], which resulted in the following expressions for approximating functions:
here $Re_v'$ and $Re_l'$ – reduced to the total diameter of the pipe Reynolds number of vapor and fluid, respectively; $\sigma_l$ and $\sigma_{H2O}$ – the coefficients of the surface tension of the refrigerant and water at 0°C, respectively.

It should be noted that the criterial equations (4), (5) and (6) for calculating the Nusselt number differ for the following extended regions of two-phase flow regimes in the diagram: (I + II), III, (IV + V + VI). Therefore an estimate of the relative mean square approximation errors was made only at the boundaries of the indicated combined zones and the errors were from 1.44% to 4.32%.

2. Materials and methods

2.1. Mathematical model of evaporation temperature dynamics

The heat exchanger-evaporator has a shell-and-tube multi-pass construction (figure 2). The working refrigerant (Freon-132b) flows inside the pipes and the coolant (antifreeze Tosol-65) flows around outside the bundle of pipes.

![Figure 2. Heat flows in the heat exchanger-evaporator.](image)

The balance of the heat output of the heat exchanger-evaporator is:

$$Q_e = Q_{h.t.e} - Q_{comp.e}.$$

(2)

On the assumption of the balance of the heat output (2) the equation of evaporation temperature dynamics is obtained in the following form:

$$m_e(t) \cdot \left( C_p_l' + C_p_g' \right) \cdot \frac{dT_e}{dt} = F_e \cdot \frac{1}{\alpha_l} + \frac{\delta_{w,e}}{\alpha_e(t)} + \frac{1}{\alpha_e(t)} \cdot \ln \left( \frac{T_e' - T_e}{T_e'' - T_e} \right) - (i_1 - i_2) \cdot G_{comp}(t),$$

(3)

here $C_p_l'$ and $C_p_g'$ – isobaric heat capacities of the liquid and gaseous phases of the working refrigerant, respectively, kJ/(kg·K); $m_e$ – mass of the working refrigerant in the heat exchanger-evaporator, kg; $i_1$ и $i_2$ – specific enthalpies at operating points (figure 2), kJ/kg; $F_e$ – the area of heat transfer surface of the heat exchanger-evaporator, m²; $\delta_{w,e}$ – the wall thickness of the heat exchanger-evaporator pipe, m; $T_e$ – the evaporation temperature of the working refrigerant in the heat exchanger-evaporator, °C; $T_e'$ and $T_e''$ – temperatures of the coolant at the inlet and outlet of the heat exchanger-evaporator, respectively.

Heat transfer coefficient at boiling of working refrigerant in round pipes [6]:

$$\frac{\dot{Q}}{A} = \frac{\dot{m} \cdot \Delta H}{d^2} \cdot \left( \frac{1}{Re_l} + \frac{1}{Re_v} \right),$$

here $\dot{Q}$ – the heat flux transferred by the liquid, W; $\dot{m}$ – mass flow rate of the liquid, kg/s; $\Delta H$ – heat of condensation of the working refrigerant, kJ/kg; $d$ – diameter of the round pipe, m; $Re_l$ and $Re_v$ – Reynolds numbers of liquid and vapor phases, respectively.
- for stratified modes (stratified with a smooth and with a wave interface of two-phase flow)
\[
\alpha_e(t) = 0.28 \cdot 10^{-5} \cdot \frac{\lambda_e(t)}{d_{e,\text{int}}} \left[ \frac{\nu''_e(t) \cdot d_{e,\text{int}}}{v''_e(t)} \right]^{0.19} \left[ \frac{\nu'_e(t) \cdot d_{e,\text{int}}}{v'_e(t)} \right]^{0.66} \left[ \frac{\nu'_e(t) \cdot d_{e,\text{int}}}{v'_e(t)} \right]^{0.33} \left[ \frac{L_e}{d_{e,\text{int}}} \right]^{0.66}; \quad (4)
\]

- for intermittent modes (shell, shell-and-ring two-phase flow)
\[
\alpha_e(t) = 0.65 \cdot 10^3 \cdot \frac{\lambda_e(t)}{d_{e,\text{int}}} \left[ \frac{\nu''_e(t) \cdot d_{e,\text{int}}}{v''_e(t)} \right]^{0.73} \left[ \frac{\nu'_e(t) \cdot d_{e,\text{int}}}{v'_e(t)} \right]^{0.73} \left[ \frac{L_e}{d_{e,\text{int}}} \right]^{-1.69}; \quad (5)
\]

- for dispersed modes (dispersed, dispersed-ring and ring two-phase flow)
\[
\alpha_e(t) = 0.018 \cdot \frac{\lambda_e(t)}{d_{e,\text{int}}} \left[ \frac{\nu''_e(t) \cdot d_{e,\text{int}}}{v''_e(t)} \right]^{0.19} \left[ \frac{\nu'_e(t) \cdot d_{e,\text{int}}}{v'_e(t)} \right]^{-0.3} \left[ \frac{L_e}{d_{e,\text{int}}} \right]^{-0.3}; \quad (6)
\]

here in equations (4)-(6) \( \nu''_e \) и \( \nu'_e \) – reduced speeds of the gaseous and liquid phases of the working refrigerant, m/s, calculated by the following relation:
\[
\nu''_e(t) = \frac{G_{\text{comp}}(t) \cdot x_e(t)}{\rho'_e(t) \cdot S_{\text{pipe}} \cdot z_e} ; \quad (7)
\]
\[
\nu'_e(t) = \frac{G_{\text{comp}}(t) \cdot [1 - x_e(t)]}{\rho'_e(t) \cdot S_{\text{pipe}} \cdot z_e} ; \quad (8)
\]

here \( G_{\text{comp}} \) – mass flow of the working refrigerant, determined by the operation of compressor according to equation (12); \( z_e \) – number of parallel pipes in the heat exchanger-evaporator; \( x_e \) – vapor content of the working refrigerant at the inlet to the heat exchanger-evaporator:
\[
x_e(t) = \frac{i_x(t) - i''_e(t)}{i'_e(t) - i''_e(t)} . \quad (9)
\]

The coefficient of heat transfer of coolant for the shell-and-tube multi-pass heat exchanger-evaporator was determined by the criterial equation for the flow around a bundle of pipes with a staggered arrangement [7]:
\[
\alpha_{\text{T,e}} = 0.41 \cdot \frac{\lambda_{\text{T,e}}}{d_{e,\text{ext}}} \left[ \frac{\nu_{\text{T,e}} \cdot d_{e,\text{ext}}}{\nu_{\text{T,e}}} \right]^{0.6} \left[ \frac{\nu_{\text{T,e}}}{a_{T,e}} \right]^{0.33} . \quad (10)
\]

In this equation outer diameter \( d_{e,\text{ext}} \), m, is taken as the determining size and maximum velocity \( \nu_{\text{T,e}} \) of coolant in the narrowest section of the bundle is taken as the determining velocity.

Mass of the working refrigerant \( m_e \) in the heat exchanger-evaporator at the nominal operating mode, taking into account the vapor content, is determined from the expression:
\[
m_e = \frac{V_e}{L_e} \left[ \int_0^{L_e} \rho''_e + \left[ \rho'_e - \rho''_e \right] \left[ 1 + \frac{1}{\frac{1-x}{L_x} + \frac{1}{L_e}} \cdot \frac{\rho''_e}{\rho'_e} \right] \cdot \mu \right] \, dl , \quad (11)
\]

here \( (1-x) \cdot l/L_e + x \) is the function of flow vapor content \( X \) along \( L_e \) of the heat exchanger-evaporator (at \( l=0 \) vapor content is equal to its current value \( X = x \), at \( l = L_e \) vapor content is \( X = 1 \) (figure 3).
2.2. Accepted assumptions and calculation method

Compiling a mathematical model of dynamics of evaporation temperature in heat exchanger-evaporator following assumptions were made to simplify the calculation:

- in the differential equation (3) of the heat balance the calculation of the heat transfer coefficient ignored thermal resistance of contaminants and vapor film;
- the temperature at the boundary of the inner wall of the heat exchanger pipes and the vapor film was assumed to be equal to the evaporation temperature \( T_e \);
- input and output temperatures and mass flow of the heat carrier were considered constant;
- leakage of the working refrigerant through the compressor was considered to be equal to zero.

The equation of temperature dynamics in the heat exchanger-evaporator was solved by the fourth-order Runge-Kutta method.

At the same time, values of evaporation heat transfer coefficient \( \alpha_e \) in two-phase region were determined at each moment of time, depending on mass flow of the working refrigerant \( G_{\text{comp}}(t) \) (12), which, in turn, depended on shaft speed of the compressor \( n_{\text{comp}} \). The process of determining shaft speed of the compressor in time was specified by the exponential equation (13).

\[
G_{\text{comp}}(t) = \frac{1.664 \cdot \pi \cdot R_{\text{comp}}^3}{\nu} n_{\text{comp}}(t), \quad (12)
\]

\[
n_{\text{comp}}(t) = n_{\text{nom}} \left[ 1 - \exp \left( \frac{-7 \cdot t}{t_{\text{nom}}} \right) \right], \quad (13)
\]

here \( t_{\text{nom}} \) – compressor’s time of reaching the nominal operating mode, s; \( n_{\text{nom}} \) – nominal shaft speed, s\(^{-1}\).

3. Results and discussion

As a result of the performed calculations, the transient characteristics of the temperature of the working refrigerant in the heat exchanger-evaporator were obtained after the system start from the standby mode at ambient temperature of plus 25°C and 50°C (figure 4).

A slight decrease in evaporation temperature of the working refrigerant was observed within 0.1 seconds after the system start at an ambient temperature of 50°C (figure 4). Abrupt evaporation temperature drop observed after that is caused by a change in the mode of the two-phase flow from stratified to intermittent (the first fracture of the characteristic). In the transition to the disperse mode (the second fracture of the characteristic), a steadily increasing evaporation temperature until the steady-state value is observed. The reason for the negative dynamics of evaporation temperature, according to equation (3), is the small value of the heat transfer coefficient, caused by a relatively
small logarithmic temperature head and low reduced velocities $v''_e$, $v'_e$ of the motion of phases of the working refrigerant with a much larger amount of heat removed by the compressor.

![Figure 4](image)

**Figure 4.** Transient characteristics of temperature in the heat exchanger-evaporator when system starts from standby mode: a – at ambient temperature of plus 25°C; b – at ambient temperature of plus 50°C.

Obviously, one of the factors influencing the time of stabilization of the parameters of the cooling system, according to equation (3), is the value of the heat content of the working refrigerant, determined by its mass contained in the heat exchanger-evaporator. The mass $m_e$ in accordance with the expression (11) essentially depends on the volumes of the flow cavities of the heat exchanger and, consequently, on the areas of the heat transfer surfaces. These parameters in the engineering calculations of the vapor compression cooling systems are determined using static criterial equations for the nominal operating mode, at which the vapor content $X$ of the working refrigerant at the inlet of the heat exchanger-evaporator is 20-30%. Such vapor content corresponds to a shell, shell-and-ring two-phase flow modes [6], in which the heat transfer is times-higher than for a stratified flow and stratified flow with a smooth and wave interface between phases.

4. Conclusion

The presence of abrupt evaporation temperature drop on the transient characteristics within 0.1 seconds after the system start from standby mode and the influence of the initial values of the ambient temperature on the transient response have been revealed and explained. Simulation results analysis shows that to study the transient dynamic characteristics of vapor compression cooling systems, it is not enough to use the criterial equations obtained for any two-phase flow regime and providing satisfying convergence of calculated and experimental data at the nominal operating mode of the system.

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