Thermal calculation and experimental corroboration of counter-flow tube-type evaporators and condensers of heat pumps operating on zeotropic mixtures

A P Usachev¹, A V Rulev¹, E Yu Usacheva¹ and A L Shurayts²

¹ Saratov State Technical University named after Gagarin Yu.A., 77 Polytechnischeskaya street, Saratov, 410054, Russian Federation
² AO “Gipronigaz”, 54 prospect im. Kirova S.M., Saratov, 410012, Russian Federation

E-mail: ritamitrofanova@yandex.ru

Abstract. When using renewable heat sources that have limited volumetric heat capacity, for example, air, water, gases, the temperature of which changes significantly during cooling in the evaporator and heating in the condenser, execution of cycles for heat pumps operating on zeotropic mixtures, the temperature of which also changes, makes it possible to increase their energy efficiency. The paper proposes a technique for thermal calculation of counter-flow tube-type evaporators and heat pump condensers which use media with limited volumetric heat capacity as heat sources, and zeotropic mixtures as working substances. We carried out experimental corroboration of the proposed technique for thermal calculation of counter-flow tube-type evaporators and heat pump condensers, which confirms the validity of the developed mathematical equations of heat transfer in the “medium with limited volumetric heat capacity, i.e. propane-butane zeotropic mixture” system. Average deviation of experimental data from theoretical dependencies with a confidence level of 95% is 27.6%, which makes it possible to recommend them for use in design and operational practice.

1. Introduction
When using renewable heat sources with limited volumetric heat capacity (VHC), for example, air, water, gases, the temperature of which changes significantly during cooling in the evaporator and heating in the condenser of the heat pump (HP), the use of substances with constant temperatures of boiling and condensation as working agents is characterized by the decrease in the energy efficiency of the HP use [1–3]. At the same time, the implementation of a cycle with variable temperatures of heat sources (air, water, various gases) and working agents in the evaporator and condenser makes it possible to increase energy efficiency of the heat pumps use [4–6]. The use of working agents consisting of zeotropic mixtures with variable boiling points in the evaporator and condenser makes it possible to increase energy efficiency of the heat pumps use [7–9]. The use of working agents consisting of zeotropic mixtures with variable boiling points in the evaporator and condenser of the heat pump is the subject of the works written by A.A. Sukhikh, K.S. Generalov, I.A. Akimov [3], Bukin V.G., Kuzmin Yu.A. [4, 5], Kim M., Ogurechnikov L.A., Mezentseva N.N. [10, 11] and a number of other foreign authors [6–9]. In [10], the author concluded that zeotropic mixture composition is significantly influenced by condensation temperatures in the condenser and boiling temperatures in the HP evaporator, which in their turn depend on the changing temperatures of heated and cooled media with VHC.
Analysis of the use of working substances as components of a zeotropic mixture in heat pumps showed that mixtures of R22/R142b, R290/600 (propane and n-butane), R600a/R601 (iso-butane and n-pentane), R290/R601a (propane and iso-pentane), R600a/R601b (iso-butane and n-pentane) meet the requirements [12, 13] in terms of the degree of destruction of the Earth's ozone layer and production of the greenhouse effect and are considered completely safe in this respect.

In [14-17], the authors proposed to achieve the minimum value of difference between average condensation and boiling temperatures by selecting each of the two zeotropic mixture components with close physical properties and molar concentration of its low-boiling component \( \psi_c \) based on the dependence of its saturation temperatures in the condenser and the evaporator of the heat pump on the relative amount of the boiled mixture.

It has been proved that the temperatures of heated and cooled media with limited heat capacity have a significant influence on the choice of zeotropic mixture components and molar concentration value of the liquid phase of its low-boiling component \( \psi_c \), at which the minimum value of the difference between the average condensation and boiling temperatures and the maximum HP energy efficiency are achieved. With frequent use of heat pumps in systems for heating the external inflowing air, the final temperature of the air heated in the condenser is taken as \( t = 20^\circ C \). In this case, the maximum energy efficiency is achieved when using heat pumps with zeotropic mixture “R290 (propane) - R600 (n-butane)” with molar concentration value of the low-boiling component R290 (propane) in the mixture equal to \( \psi_c = 42 \text{ mol} \% \) [16, 17].

However, the existing works do not touch upon issues of thermal calculation and its experimental corroboration for evaporators and condensers of heat pumps operating on zeotropic mixtures.

2. Development of thermal calculation technique to determine the length of counter-flow tube-type evaporators and condensers of heat pumps operating on zeotropic mixtures of optimal chemical composition

As a rule, horizontal and low-inclined pipes or bundles of parallel-connected horizontal pipes are used in the heat pump technology for evaporation and condensation of zeotropic mixtures intended for heating and cooling media with VHC. It is known that the processes of evaporation and condensation in horizontal pipes proceed in the reverse sequence and are described by the same heat transfer equations, and also have \( z \) enlarged flow regimes of a vapor-liquid zeotropic mixture (Figure 1) inside the pipe [18], where \( z \) is the number and name of the flow regime: \( z = 1 \) – stratified-cork; \( z = 2 \) – ring-wave; \( z = 3 \) – dispersed. Thus, when liquid evaporates in a horizontal pipe (\( z = 1; 2; 3 \)), first there is a stratified-cork flow regime (\( z = 1 \), section 3a and 3b, Figure 1), which, with an increase in the amount of evaporated liquid, turns into a ring-wave flow regime (\( z = 2 \), section 2a and 2b, Figure 1), and upon further boiling it turns into a dispersed (\( z = 3 \), section 1, Figure 1) flow regime.

At the same time, during vapor phase condensation in a horizontal pipe (\( z = 3; 2; 1 \)), on the contrary, first there is a dispersed flow regime (\( z = 3 \), section I in Figure 1), which, with an increase in the amount of condensed vapor, transforms into a ring-wave flow regime (\( z = 2 \), sections 2a and 2b, Figure 1), and upon further condensation it transforms into a stratified-cork flow regime (\( z = 1 \), sections 3a and 3b, Figure 1). The reverse, but strictly defined sequence of changing the flow regimes of the vapor-liquid mixture made it possible to develop the generalized equation according to which the calculated length of horizontal evaporators and condensers of heat pumps is determined on the basis of the heat balance equation as the sum of the lengths of individual sections in stratified-cork \( L_{z=1} \), ring-wave \( L_{z=2} \) and dispersed \( L_{z=3} \) flow regimes according to the formula:

\[
\sum_{z=1}^{3} L_z = L_{z=1} + L_{z=2} + L_{z=3}.
\]
3

Figure 1. Design diagram for the technique for determining the length of the evaporative and condenser sections of the pipe heat exchanger of heat pumps operating on zeotropic mixtures of working agents.

The length of an individual $z$-th section of the counter-flow type evaporator with complete evaporation of the zeotropic mixture inside the pipe and the condenser with its complete condensation is determined by the formula:

$$L_z = \frac{G}{\pi \cdot d_{\text{in}} \cdot k_{z,\text{ev}}} \left\{ \frac{r_{z,\text{av}}}{t_{e,z} - t_{h,z}} \int_{t_{e,z}}^{t_{h,z}} \frac{dX_z}{dr_z} \int_{t_{e,z}}^{t_{h,z}} \frac{dX_z}{dr_z} \right\} \cdot \frac{dr_z}{t_{e,z} - t_{h,z}} \cdot \frac{dX_z}{t_{e,z} - t_{h,z}},$$

(2)

where $G$ is the estimated mass flow rate of the zeotropic mixture circulating in the evaporator and condenser, kg/h; $X_{z,e}$, $X_{z,b}$ are the final and initial values of the dryness degree of the vapor-liquid mixture, at which there is a transition from one flow regime to another, unit fraction; $d_{\text{in}}$ is the inner diameter of the flow-through tube-type evaporator and heat pump condenser, m; $k_{z}(t_z)$ is the value of the heat transfer coefficient, as a function of the heat transfer coefficient, characteristic of stratified-cork, ring-wave and dispersed flow regimes, W/(m$^2$ K); $t_h$ is the heat transfer agent temperature, °C; $t_z$ is the current temperature of the vapor-liquid mixture, changing in the temperature range from $t_{b,z}$ to $t_{e,z}$ in the evaporator and from $t_{e,z}$ to $t_{b,z}$ in the condenser, °C; $t_{e,z}$, $t_{b,z}$ are the initial and final temperatures of the evaporated or condensed vapor-liquid zeotropic mixture in the areas, respectively, with stratified-cork, ring-wave and dispersed flow regimes, °C; $r_{z,\text{av}}, c_{z,\text{av}}$ are, respectively, the average values of the latent heat of vaporization and specific heat capacity of the mixture in the intervals of its evaporation or condensation, respectively, in areas with stratified-cork, ring-wave and dispersed flow regimes, kJ/kg, kJ/kg K.

When obtaining formula (2), the zeotropic mixture entering the evaporator with molar content of the low-boiling component in the liquid phase $\Psi^l$, according to [15, 16], completely boils away in the temperature range from $t_{b,z,1}$ to $t_{e,z,1}$. In this case, for each of the indicated flow regimes, the mixture evaporation temperature changes in the following ranges: stratified-cork regime – $t_{e,z,1} \geq t_{e,z,2} \geq t_{h,z,1}$; ring-wave regime – $t_{e,z,2} \geq t_{e,z,3} \geq t_{h,z,2}$; dispersed regime – $t_{e,z,3} \geq t_{e,z,4} \geq t_{h,z,3}$. The supplied heat flux consists of both the heat of the mixture vaporization and the heat of its heating in the temperature range of its complete evaporation from $h$ to $t_e$. 
The zeotropic mixture entering the condenser with the molar content of the low-boiling component in the vapor phase $\psi_{z=3} = \psi_{z=1}$ completely condenses in the temperature range from $t_{z=3}$ to $t_{b,z=1}$. In this case, for each of the indicated flow regimes, the mixture condensation temperature changes in the following ranges: dispersed regime $- t_{z=3} \geq t_{z=3} \geq t_{b,z=1}$; ring-wave regime $- t_{z=3} \geq t_{z=3} \geq t_{b,z=2}$; stratified-cork regime $- t_{z=3} \geq t_{z=3} \geq t_{b,z=1}$. The removed heat flux arises both due to mixture condensation and cooling in the temperature range of its complete condensation from $t_{z=3}$ to $t_{b,z=1}$.

Specific heat capacities and latent heats of vaporization of the zeotropic mixture saturated vapor for the case when the heat transfer agent flows in the tubular annulus (a pipe with a thermal conductivity coefficient of the evaporating pipe coil, W/(m·K); $\alpha_{z, int}$ are, respectively, internal heat transfer coefficients for stratified-cork, ring-wave and dispersed flow regimes, determined according to [21], W/(m²·K); $d_o$, $d_{int}$ are, respectively, the outer and inner diameters of the evaporator and condenser, m, determined using the measurement data; $\lambda$ is the thermal conductivity coefficient of the evaporating pipe coil, W/(m·K); $\alpha_{z, o}$ is the external coefficient of heat transfer from the outer surface of the evaporator and condenser pipes to the heat transfer agent for the case of its forced flow in the tubular annulus during longitudinal washing, determined according to [20], for the case when the heat transfer agent flows in the tubular annulus (a pipe with a smaller diameter is located in a pipe of a larger diameter), W/(m²·K).

In the stratified-cork flow regime (with a predominantly clear contrast between the vapor and liquid phases of the propane and butane mixture), the heat exchange mechanism is similar to heat transfer during high-volume boiling of liquid [20]. Defining the average value of the heat transfer coefficient.
for the propane-butane mixture is based on [21], taking into account composition of the zetotropic mixture, according to [22, 23]:

\[
\alpha_{av,z=1} = 6.4 \left[ X_{av,z=1} / (1 - X_{av,z=1}) \right]^{0.15} \cdot q^{0.7} / \left[ (3.3 - 0.0115(t_{int} - 100)) \chi_{av,z=1} \right],
\]

where \( \alpha_{av,z=1} \) is the heat transfer coefficient for stratified-cork flow regime with a clear contrast between the vapor and liquid phases of the zeotropic propane-butane mixture, \( W/(m^2 \cdot K) \); \( X_{av,z=1} \) is the average value of the dryness degree of the vapor-liquid propane-butane mixture in the pipe section with stratified-cork flow regime, equal to half-sum of the initial \( X_{b,z=1} \) and final \( X_{e,z=1} \) values; \( q \) - specific heat flux, \( W/m^2 \); \( t_{int} \) is the saturated liquid temperature, °C; \( \chi_{av,z} \) is the average value of the parameter that determines the effect of the propane content in the vapor-liquid zeotropic mixture in the stratified-cork (\( z = 1 \)) and ring-wave flow regimes (\( z = 2 \)); \( \psi_{av.z}^v, \psi_{av.z}^l \) are, respectively, the average equilibrium propane content in the vapor and liquid phases of the zeotropic mixture in the middle of the section with the stratified-cork flow regime, mol %.

The final degree of dryness corresponding to transition boundary of stratified-cork flow regime into ring-wave flow regime was determined on the basis of the Baker diagram [18, 24] and was \( X_{e,z=1} = 0.3 \).

The effect of the heat flux on the heat transfer coefficient in the ring-wave flow regime is manifested to a lesser extent, in contrast to the conditions of the stratified-cork flow regime [21]:

\[
\alpha_{av,z=2} = \left( \lambda / d_{int} \right) \cdot 0.6 \cdot Re^{0.8} \cdot Pr^{0.4} \left[ X_{av,z=2} / (1 - X_{av,z=2}) \right]^{0.13} \cdot \left( q / r \cdot 1 \right)^{0.2} / \chi_{av,z},
\]

where \( Re, Pr \) are Reynolds and Prandtl criteria corresponding to saturation temperature of the liquid; \( X_{av,z=2} \) are the average values of dryness degree of the vapor-liquid mixture in the pipe sections with the ring-wave flow regime, determined according to formula (5); \( I \) – is the specific mass flow rate, kg/m²·h.

According to the studies [24], starting from the time when \( X = X_{e,z=2} = 0.981 \), all liquid droplets are carried away from the boiling film located on the inner surface of the pipe into the vapor core of the liquid flow, while the temperature of the pipe wall rises rapidly, which indicates that the inner surface of the pipe is washed only by the vapor phase of the propane and butane mixture.

Moreover, when \( X = X_{e,z=2} = 0.981 \), the value of the heat transfer coefficient begins to fall rapidly. At the same time, the temperature of the liquid-vapor flow in the center of the pipe, up to the value of \( X = 1 \), remains almost constant, indicating the presence of evaporating droplets of the liquid phase in the center of the flow.

The average heat transfer coefficient \( \alpha_{av,z=3} \) remains constant in the dispersed flow of the vapor-liquid mixture in the form of fog, which is characteristic of a single-phase vapor flow and, according to [20], is determined similarly as for the case of single-phase flow of dry saturated vapor:

\[
\alpha_{av,z=3} = 0.021 \frac{\lambda}{d_{int}} Re^{0.8} \cdot Pr^{0.4}.
\]
3. Experimental corroboration of the thermal calculation technique for determining the length of counter-flow tube-type evaporators and condensers of heat pumps operating on zeotropic mixtures of optimal chemical composition

In order to corroborate the validity of equations (1) – (7) for determining the length \( \sum_{z=1}^{3} L_z \) of the counter-flow tube-type evaporator and condenser of heat pumps operating on the zeotropic mixture, we carried out relevant experimental studies. The diagram of the pilot-plant equipment is shown in Figure 2.

![Figure 2. Experimental equipment diagram:](image)

1 – sample pressure gauge; 2 – heat transfer agent flow meters; 3 – three-way valve; 4 – shut-off and control valve; 5 – thermal converter-sensor for determining the temperature of zeotropic mixture at the inlet and outlet of the evaporator pipe; 6 – prototype for determining the evaporator pipe length; 7 – outer case with heat transfer agent for evaporator pipe 6; 8 – thermal converter-sensor for determining the temperature of heat transfer agent at the inlet and outlet of case 7 of evaporator pipe 6; 9 – prototype for determining the condenser pipe length; 10 – outer case with heat transfer agent for condenser pipe 6; 11 – thermal converter-sensors for determining zeotropic mixture temperature in the evaporator pipe; 12 – thermal converter-sensors for determining the heat transfer agent temperature in case 7 of the evaporator pipe; 13 – multichannel temperature meters; 14 – laptop; 15 – thermal converter-sensors for determining zeotropic mixture temperature in the condenser pipe; 16 – thermal converter-sensors for determining the heat transfer agent temperature in case 10 of the condenser pipe; 17 – thermal valve; 18 – compressor with electric drive; 19 – turbulator and vortex flow meter of the zeotropic mixture vapor phase; 20 – signal processing unit; 21 – recorder of measured values; 22 – interface.
Evaporative and condensing heat exchangers here are made of straight steel pipes with the inner diameter of 6.0 mm and length of 5500 mm. To reduce heat loss, the outer surface of pipes 7 and 10 was covered with thermal insulation.

The stand has a system for monitoring the equipment parameters in the process of conducting experiments and automatically taking readings from sensors placed on the experimental equipment.

Primary sensors provided the following measurement accuracy:
- temperature – 0.1 °C;
- pressure – not less than 0.0025 MPa;
- consumption – not less than 1.5% of the maximum value.

Heat exchangers of the “pipe-in-pipe” type are installed on the stand. Considering higher intensity of heat transfer from boiling or condensed zeotropic mixture as compared to cooled or heated water, tube-type evaporators and condensers on the side of the outer surfaces had evenly placed pins obtained by deforming cutting technology. This form of finning allows implementing the principle of counter-flow flow of the zeotropic mixture and hot water in the “pipe-in-pipe” heat exchanger.

For experimental determination of the length of the evaporator and condenser sections $L_{e,z}$ and $L_{c,z}$, we used the temperature distribution analysis in the evaporator and condenser pipelines during boiling and condensation of zeotropic mixture. The obtained temperature distribution of the vapor-liquid mixture was used to determine the boundary of the evaporation and condenser sections for each of the regimes with any predetermined accuracy.

Zeotropic mixture boiling in the flow-through system occurs within temperature range starting at $t_{b,z=1}$ and ending at $t_{e,z=3}$ and, vice versa, condensation begins at $t_{e,z=3}$ and ends at $t_{b,z=1}$. To determine the boundaries of the evaporating and condensing sections of each of the regimes, it is necessary to find points on the pipeline where the temperature values of the vapor-liquid mixture $t_z$ can be found: $t_{b,z=1}$; $t_{e,z=1}$; $t_{e,z=2}$; $t_{e,z=3}$.

The values of the initial, current and final temperatures of condensation $t_z$ in the condenser in the range $t_{zh}^c + t_{zh}^e$, as well as the initial, current and final values of the boiling points $t_z$ in the evaporator $t_{zh}^e + t_{zh}^c$, depending on the relative amount (degree of dryness) $X_z$ of the boiled off or condensed zeotropic mixture at a given value of the molar concentration low-boiling component $\Psi_1^z$, are found according to the first law of Konovalov, taking into account the laws of Raoul and Dalton and the Antoine formula [15, 16]:

$$X_z = P \left( \frac{\Psi_z}{P - 10^A \cdot 10^{-C_1} t_z} + \frac{1 - \Psi_z}{P - 10^A \cdot 10^{-C_1} t_z} \right), \quad (8)$$

where $P$ is the absolute pressure of the zeotropic mixture in the evaporator or condenser of the heat pump, Pa $10^5$; $A_1$, $B_1$, $C_1$ are coefficients characteristic of the component with the lower boiling and condensation temperature at the mixture pressure $P$ within certain temperature range $t_z$; $A_2$, $B_2$, $C_2$ are coefficients characteristic of the component with higher boiling and condensation temperature at mixture pressure $P$ within certain temperature range $t_z$.

The boiling point of the zeotropic mixture in the evaporator is determined in degrees Celsius according to the formula (8), when current temperature $t_z$ equals $t_{zh}^e$, that is, $t_z = t_{zh}^e$, the value of the relative amount of the boiled-off zeotropic mixture is $X_z = 0$, and the end boiling point is $t_z = t_{zh}^e$ at the value of $X_z = 1$. The temperature of the beginning of the zeotropic mixture condensation in the condenser $t_z = t_{zh}^c$ is determined in degrees Celsius according to the formula (8), when the value of the relative amount of the condensed zeotropic mixture is $X_z = 1$, and the temperature of the end of condensation is $t_z = t_{zh}^c$ at the value of $X_z = 0$. Finding $t_z$ at given $X_z$ is carried out according to the formula (8) using selection method.
It should be noted that the discrepancy associated with the use of the calculation formula (8), based on the laws of Dalton, Raoult and Antoine’s correlation, using the measurement data is 4.5% [15, 16] for zeotropic mixtures, and for saturated hydrocarbons with the absolute pressure of up to 1.0 MPa in particular.

The equipment we used todetermine zeotropic mixture temperatures \( t_{b=1}; t_{e=1}; t_{c=2}; t_{e=2} \) in the tube-type evaporator and condenser, included the following elements (Figure 2):

– two bunches, each of which consists of 23 thermocouples, for measuring zeotropic mixture temperatures along the length of the evaporator and condenser pipes;

– four multichannel meters TM-5122;

– a portable laptop.

The working ends of thermocouples with sensitive junctions were placed on the evaporator and condenser pipes with a pitch of 0.25 m along the entire length of the heat exchangers. The length of the evaporator and condenser sections is found by analysing temperature distribution of the propane butane mixture along its flow in the pipelines. For example, the end boiling point \( t_{e=3}^{ev} \) is indicated by the thermocouple \( n = 21 \) (Figure 2). This means that the total length of the evaporator section is

\[
L_{e=3}^{ev} = (n-1) \cdot 0.25 = (21-1) \cdot 0.25 = 5.0 \text{ m}.
\]

If the temperature \( t_{e=3}^{ev} \) is between adjacent thermocouple electrodes, then the length of the heat pump evaporator is determined by interpolation with an accuracy of 0.1 °C according to the formula:

\[
\frac{t_{e=3}^{ev} - t'}{t'' - t'} = \frac{L_{e=3}^{ev} - L'}{L'' - L'},
\]

where \( t' \) and \( t'' \) are the boiling propane-butane mixture temperatures, measured by the first and second adjacent thermoelectric converters, counted along the flow of the vapor-liquid mixture, °C; \( L', L'' \) are distances from the beginning of the evaporation pipeline to the first and second adjacent thermocouple electrodes, counted along the flow of the vapor-liquid mixture, m.

Theoretical values of the lengths of evaporation and condenser sections of the zeotropic mixture (butane-propane mixture) for each of the regimes of the vapor-liquid flow, calculated by formulas (1) – (7), i.e. \( L_{e=1}, L_{c=2}, L_{c=3} \), are shown in the diagram (Figure 3) as a solid line.

The total length of the flow-through evaporator and condenser with boiling and condensation of propane-butane mixtures in pipes is determined based on the equation (1). Analysis of the distribution of the total length \( \sum_{z=1}^{z=3} L_{z} \) in separate areas with stratified-cork, ring-wave and dispersed flow of the vapor-liquid zeotropic mixture is shown in the diagram (Figure 3). The relative values of the lengths (fraction of the total length) on the diagram \( L_{e=1}; L_{c=2}; L_{c=3} \) in the section, respectively, with the stratified-cork, ring-wave and dispersed flow of the zeotropic mixture, %, are determined as:

\[
L_{e=1} = \frac{L_{e=1}}{\sum_{z=1}^{z=3}} \cdot 100\%, \quad L_{c=2} = \frac{L_{c=2}}{\sum_{z=1}^{z=3}} \cdot 100\%, \quad L_{c=3} = \frac{L_{c=3}}{\sum_{z=1}^{z=3}} \cdot 100\%.
\]

In order to obtain a unified theoretical line for the HP evaporator and condenser, we carried out the calculations at the same temperature differences between water and zeotropic mixture.

It follows from the diagram (Figure 3) that the stratified-cork flow regime section accounts for \( L_{e=1} = 37.8\% \) of the total surface of the evaporator and condenser, where \( X_{e=1} = 30\% \) of the entire liquid phase boils away (condenses) (points 1-2-3, Figure 3).

The section with the ring-wave flow regime accounts for \( L_{c=2} = 45.8\% \) of the total length of the evaporator (condenser), where \( X_{c=2} = 68\% \) of the entire liquid phase boils away (condenses) (points 4 → 5 → 6, Figure 3).
In the section with the dispersed flow regime, only $X_{z=3} = 2\%$ of the entire liquid phase boils away (condenses) at the length equal to $L_{z=3} = 16.4\%$ (points 7 → 8 → 9, Figure 3).

It can be seen from the diagram (Figure 3) that the main amount of the liquid phase $X_{z=2} = 68\%$ (points 4 → 5 → 6) boils away (condenses) in the section with the ring-wave flow regime. In this case, only $L_{z=2} = 45.8\%$ (points 4 → 5 → 6) of the total length of the evaporator (condenser) falls on the section with the ring-wave flow regime. A decrease in the relative length in this section is explained by the increase in the heat transfer coefficient due to the onset of the wave-ring flow regime, when the liquid phase washes the entire inner surface of the evaporator (condenser).

![Figure 3. Changes in the degree of dryness X, %, with change in the surface of the vaporizer (condensation) section: — theoretical line; △ — experimental values.](image)

It also follows from the diagram (Figure 3) that in the section with the dispersed flow regime, only $X_{z=3} = 2\%$ of the liquid phase boils away (points 7 → 8 → 9). In this case, the section with the dispersed flow regime accounts for more than $L_{z=3} = 16.4\%$ (points 4 → 5 → 6) of the total length of the evaporator (condenser). A sharp increase in the relative length in this section is explained by the decrease in the heat transfer coefficient, due to the onset of the dispersed flow regime, when all liquid droplets are carried away from the boiling film located on the inner surface of the pipe into the vapor core of the liquid flow, and the inner surface of the pipe is washed only by the vapor phase of the propane and butane mixture.

For reference, the diagram shows the experimental values of the relative lengths of the evaporator and condenser. In order to obtain uniform experimental values for the HP evaporator and condenser, we carried out the experiments at the same temperature differences between water and zeotropic mixture. Each experimental point shown in the diagram is taken as common for the evaporator and condenser, i.e. averaged over four measurements ($n = 4$) for the evaporator and over four measurements ($n = 4$) for the condenser. The reliability of results is taken as $\alpha = 0.9$, and the value of
Student's coefficient is $t_{\alpha=0.9} = 2.35$. The average deviation of the calculated and experimental values \[ \sum_{z=1}^{3} L_z \] for the evaporator and condenser with a confidence level of 95% is 27.6%.

Thus, the obtained experimental data confirm analytical dependences (1) – (7) for determining the relative value of the lengths (fraction of the total length) in the sections, respectively, with the stratified-cork, ring-wave and dispersed flow of the zeotropic mixture and make it possible to recommend them for use in the design and operational practice.

4. Conclusion

1. We proposed a technique of thermal calculation (1) – (7) of counter-flow tube-type evaporators and condensers of heat pumps using media with a limited volumetric heat capacity (air, water, various gases) as renewable heat sources, the temperatures of which significantly change during cooling in the evaporator and heating in the condenser of the heat pump, and which use zeotropic mixtures as working substances, the temperature of which also changes.

2. We carried out experimental corroboration of the proposed technique of thermal calculation for counter-flow tube-type evaporators and heat pump condensers. We proved the validity of the corresponding mathematical equations of heat transfer (1) – (7) in the system “medium having a limited volumetric heat capacity – propane-butane zeotropic mixture”. The average deviation of experimental data from theoretical dependencies with a confidence level of 95% is 27.6%, which makes it possible to recommend them for use in design and operational practice.

References

[1] Martynovsky V S 1979 Cycles, schemes and characteristics of thermotransformers (Moscow: Energiya) p 285
[2] Usacheva E Yu, Shuraits A L and Rulev A V 2015 Research of temperature conditions in the evaporator and condenser of heat pump dryers operating on non-azeotropic hydrocarbon mixtures Bulletin of SSTU 3 57–61
[3] Sukhoi A A, Generalov K S and Akimov I A 2000 Testing of a heat pump for individual home heating Trudy mgue: Technique of low temperatures in the service of ecology 49–53
[4] Bukin V G and Kuzmin A Yu 1996 Experimental study of small refrigerating machines on a mixture of R22/R142b Refrigerating equipment 5 12–14
[5] Bukin V G and Kuzmin A Yu 2007 Refrigerating machines working on non-azeotropic mixtures of refrigerants (Astrakhan: publishing house of AGTU) p 156
[6] Kim T S, Shin T Y and Kim M 1994 Cycle analysis and heat transfer characteristics of heat pump using R22/ R142b refrigeration Sand Ro S.T. 17 391–99
[7] Kim M, Kim M S and Kim Y 2004 Experimental study on the performance of heat pump system nith refrigerant mixtures composition change Energy 24 1053–68
[8] Ho-Saeng Lee, Hyeon-Ju Kim, Dong-gyu Kang and Djingsoo 2012 Jung Thermodynamic performance of R32/R152a mixtruru fjr water source heat pumps Energy 40 251–57
[9] Jianyong C and Janlin Yu 2008 Of new refrigeretion cycle using mixture R32/R134a for resintial air conditijner applications Energy and Buildings 40 171–79
[10] Ogurechnikov L A 2011 Condensation R32/R134a in the technology of heat pump heat supply Refrigerating equipment 2 46–8
[11] Ogurechnikov L A and Mezentseva N N 2008 Analysis of the efficiency of using mixtures of ozone-safe refrigerants in steam-compression heat pumps Power engineering and heat engineering 12 57–66
[12] 2020 Montreal Protocol on Substances that Deplete the Ozone URL https://unep.ch/ozone/pdf/Montreal-Protocol2000.pdf
[13] 2020 Kyoto Protocol to the UN Framework Convention on Climate Change URL https://treaties.un.org/doc/source/RecentTexts/kyoto-en.htm/
[14] Shurayts A L, Rulev A V and Usacheva E Yu 2016 Assessing Energy Efficiency of Compression Heat Pumps in Drying Processes when Zeotropic Hydrocarbon Mixtures are Used as Working Agents MATEC Web Conf. 73 1–9

[15] Shuraits A L, Rulev A V and Usacheva E Yu 2017 Selection of mixtures of working agents and their composition in compression heat pumps of heat and gas supply and ventilation systems for heating and cooling environments with limited heat capacity Construction and architecture 4(48) 47–57

[16] Usacheva E Yu, Shuraits A L and Rulev A V 2018 Choice of mixtures of agents in heat pumps for heating and cooling media with limited capacity issue Russian Journal of Building Construction and Architecture 1 (37) 53–66

[17] Usachev A P, Rulev AV and Usacheva E Yu 2018 Method for obtaining a working agent in a compression heat pump Patent No 2658414 18

[18] Butterworth D and Hewitt G. 1977 Two-phase flow and heat transfer Chemistry p 328

[19] Usachev A P and Rulev A V 2013 Development of methodological provisions for the thermal calculation of industrial pipe vaporizers of liquefied petroleum gas mixtures Thermal processes in engineering 5343–53

[20] Mikheev M A and Mikheeva I M 1973 Fundamentals of heat transfer (Moscow: Energiya) p 320

[21] Yushida H and Yamaguchi S 1970 Heat Transfer in two-phase flow of freon 12 in horizontal pipes (Moscow: Mir) pp 252–72

[22] Kutepov A M, Sterman L S and Styushin N G 1977 Hydrodynamics and heat transfer during evaporation (Moscow: Higher school) p 352

[23] Preobrazhenskiy N I 1975 Liquefied gases (Leningrad: Nedra) p 227

[24] Usachev A P and Rulev A V 2013 Determining the boundaries of flow modes of vapor-liquid propane-butane mixture in flow-through tube evaporators Electronic scientific journal "Oil and Gas business" 1 547–54