Development of operation principles and calculation methods for compression heat pumps using zeotropic mixtures as working fluids

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Abstract. The article presents operation principles and calculation methods for compression heat pumps using zeotropic mixtures as working fluids. It proposes the principle and methodological provisions for achieving the minimum value of mean condensation and boiling temperatures difference by selecting the molar concentration value for the low-boiling component of the zeotropic mixture out of two components that are close in their physical properties, and taking into account saturation temperature change in the condenser and evaporator of the heat pump, depending on the relative amount of the boiled out mixture and limited heat capacity heated and cooled mediums temperature changes. The authors developed a thermal calculation model for flow tubular condensers and evaporators of the zeotropic mixture, which allows to take into account the intensity of heat exchange changes and temperature conditions depending on the continuously changing composition and flow regimes of the vapor-liquid mixture taking place in the following sequence: dispersed, circular wave and layered-plug flow modes.

1. Introduction

When using renewable heat sources with limited volumetric heat capacity (HC), for example that of air, the temperature of which, when cooled in the evaporator and heated in the condenser of the heat pump (HP), changes significantly, the use of substances with constant boiling and condensation temperatures as a working fluid is characterized by a decrease in the energy efficiency of its use [5, 13, 18]. At the same time, implementation of a cycle with variable temperatures of both HC mediums and working fluids in this case allows to increase energy efficiency of the heat pumps use [1, 2, 17]. Scientific works written by such researchers as Sukhikh A.A., Generalova K.S., Akimova I.A. [13], Bukina V.G., Kuzmina Yu.A. [1, 2], Kutepova A.M. [4], Kim M. and a number of other international authors [14-17], Ogurechnikova L.A., Mezentseva N.N. [8], works of Moscow Power Engineering Institute, the Institute of Thermophysics named after S.S. Kutateladze Siberian Branch of the Russian Academy of Sciences, Astrakhan State Technical University, and other scientific institutions, are dedicated to using working fluids made of zeotropic mixtures with variable temperatures in the evaporator and condenser of the heat pump. Work [8] comes to the conclusion regarding significant influence of the condensation temperatures in the condenser and boiling temperature in the heat pump evaporator on the composition of the zeotropic mixture, which in their turn depend on the changing temperatures of the heated and cooled mediums with limited heat capacity.
However, these scientific works do not touch upon issues regarding the choice of the best combinations of working fluids mixtures and their composition in the compression heat pumps for heating and cooling the HC mediums that provide maximum energy efficiency.

Analysis of the possibility of using certain zeotropic mixtures as working fluids in the heat pumps, i.e. R22/R142b, R32/R134a, R32/R152a showed that these substances have low ozone-depleting activity, and mixtures of the saturated hydrocarbons R290 / 600 (propane and butane), R600a/R601 (isobutane and n-pentane), R290/R601a (propane and isopentane), R600a/R601b (iso-butane and n-pentane) are completely safe in terms of the degree of their activity related to ozone layer destruction [7]. These gases do not cause noticeable greenhouse effect, do not influence climate change negatively, and do not have negative impact on human health [3]. Saturated hydrocarbon mixtures have a significantly lower price in comparison with other working fluids.

2. Development of functioning principles, devoted to providing minimum value of mean condensation and boiling zeotropic mixture temperatures difference between condenser and evaporator of the heat pump

The scheme of the heat pump, performing the cycle with variable temperatures of zeotropic hydrocarbonic mixture, used as a working fluid, and environments with restricted thermal capacity, used as a heated source and outlet with opposite flow directions, is given in figure 1 [5].

![Diagram of a heat pump](image)

Figure 1. Scheme of a heat pump, working on zeotropic mixtures with countercurrent flow of mixture and environments with HC in the evaporator and the condenser.

1 – countercurrent pump evaporator; 2 – inter-tube space of the evaporator 1; 3 – pipeline of zeotropic mixture’s vapour phase for connection with the evaporator outlet 1; 4 – compressor; 5 – pipeline for zeotropic mixture vapour phase for connection with the evaporator outlet 4; 6 – countercurrent pipe condenser; 7 – inter-pipe space of the condenser 6; 8 – pipeline for zeotropic mixture liquid phase for connection with countercurrent pipe condenser outlet 6; 9 – regulator for decreasing temperature of zeotropic mixture saturated liquid phase; 10 – pipeline for zeotropic mixture liquid phase for connecting regulator 9 and pump evaporator inlet 1; \( t_{\text{cond} \text{m.p.}}, t_{\text{cond} \text{m.t.}} \) – primary and terminal temperatures of condensation of zeotropic mixture vapour stage in the condenser, °C; \( t_{\text{evap} \text{m.p.}}, t_{\text{evap} \text{m.t.}} \) – primary and terminal boiling temperatures of the liquid phase of mixture, °C. Notations of zeotropic mixture processes: \( a-b \) – vapor phase contraction in the compressor 4; \( b-c \) – vapor phase condensation in the condenser 6; \( c-d \) – decrease of saturated liquid phase temperature in the regulator 9; \( d-a \) – evaporation of saturated liquid phase in the evaporator 1.
Maximum energy efficiency is achieved via decreasing the difference of temperatures between working fluid from zeotropic mixture in the evaporator and the condenser of a heat pump down to minimum value. Novelty of the proposed method is that molar concentration value of low-boiling component of the zeotropic mixture $\Psi_i$ is chosen in such a way that it achieves the minimum value of average temperature difference of condensation and boiling for countercurrent condenser and evaporator through expression-based fitting [6]:

$$t_{\text{cond}} - t_{\text{evap}} = \frac{(t_{\text{cond},m.m} - t_{\text{evap},m.m}) - (t_{\text{cond},m.t.} - t_{\text{evap},m.t.})}{\ln(t_{\text{cond},m.m} - t_{\text{evap},m.m}) / (t_{\text{cond},m.t.} - t_{\text{evap},m.t.})} = \text{min},$$

(1)

if temperature difference $t_{\text{cond},m.p.} - t_{\text{evap},m.p.}$ is bigger than $t_{\text{cond},m.t.} - t_{\text{evap},m.t.};$

$$t_{\text{cond}} - t_{\text{evap}} = \frac{(t_{\text{cond},m.m} - t_{\text{evap},m.m}) - (t_{\text{cond},m.p.} - t_{\text{evap},m.p.})}{\ln(t_{\text{cond},m.m} - t_{\text{evap},m.m}) / (t_{\text{cond},m.p.} - t_{\text{evap},m.p.})} = \text{min},$$

(2)

if temperature difference $t_{\text{cond},m.p.} - t_{\text{evap},m.p.}$ is bigger than $t_{\text{cond},m.t.} - t_{\text{evap},m.t.};$

where $\Psi_i - i$-th value of molar condensation of the low-boiling component in the zeotropic mixture made up of two components that are close in their physical properties, where $\Psi_i = \Psi_1, \Psi_2, \Psi_3, \ldots$ $\Psi_t$, mol.\%.

For determination of values of primary, current and terminal temperatures of condensation in the condenser during the interval $t_{\text{cond},m.p.} \div t_{\text{cond},m.t.}$, as well as primary, current and terminal temperature values of boiling in the evaporator $t_{\text{evap},m.p.} \div t_{\text{evap},m.t.}$ depending on relative quantity (degree of dryness) $X$ of boiled out or condensed zeotropic mixture with given value of molar concentration of low-boiling component $\Psi_i$, we offer dependence, obtained according to the first Konovalov’s Law, taking into account Raul’s and Dalton’s Laws, and Antoine’s equation [10,11,12]:

$$X = P \left( \frac{\Psi_i}{A_i \cdot C_i \cdot 10^{B_i / C_i}} + \frac{1 - \Psi_i}{P \cdot 10^{A_i \cdot C_i / B_i}} \right),$$

(3)

where $P$ is absolute pressure of zeotropic mixture in the evaporator or the condenser of the heat pump, Pa $10^5$; $A_i, B_i, C_i$ are coefficients that are specific for a component with lower boiling temperature and condensation when pressure of mixture equals $P$ within determined range of temperature $t$; $A_1, B_1, C_1$ are coefficients that are specific for a component with higher boiling temperature and condensation when pressure of mixture equals $P$ within determined range of temperature $t$ [10,11].

Change of temperature of boiled out or condensed mixture within the interval of degree of dryness change in $X$ range, determined from 0 to 1,0, in its turn, conditions change of temperature of heated or cooled environment with HC, for example, air depending on temperature change, degree of dryness and zeotropic mixture consumption, for countercurrent heat exchanger it is determined via equalizing the balance between working fluid and environment with HC in the evaporator and the condenser of the heat pump.

The scheme of heat pump cycle, working on zeotropic mixtures, with countercurrent mixture flow and environments with HC in the evaporator and the condenser is given in figure 2.

Thus, according to the proposed method, we can choose concentration of zeotropic mixture low-boiling component, which provides minimum value of temperatures difference $(t_{\text{cond},m.m} - t_{\text{evap},m.p})=\text{min}$, and thus maximum energetic efficiency of heat pump.
Figure 2. Graph of zeotropic mixture and limited HC environment temperature change, allowing to estimate the minimum value of mean temperatures difference in the counter-flow evaporator and condenser of the heat pump.

$\Psi_i$ selection is carried out based on the example of two zeotropic mixtures, each consisting of two components close in their physical properties:

- zeotropic mixture «R600a (iso-butane) – R601 (n-pentane)» with concentration of low-boiling component R600a (iso-butane), taken within the range of $\Psi_i=0,0\text{ mol }%$ with the step of 2,0 mol %.
- zeotropic mixture «R290 (propane) – R600 (n-butane)» with concentration of low-boiling component R290 (propane), taken within the range $\Psi_i=0,0\text{ mol }%$ with the step of 2,0 mol %.

Calculation results show that minimum value $(t_{\text{cond}}^\text{m,m} - t_{\text{evap}}^\text{m,m}) = 18°C = \text{min}$, according to expression (3), is achieved for zeotropic mixture “R600a (iso-butane) – R601 (n-pentane)” when molar concentration of low-boiling component R600a (iso-butane) in the mixture equals $\Psi_i=45$ mol %.

Calculation results also show that when values of molar concentration of low-boiling component of $\Psi_i=0,0\text{ mol }%$ and $\Psi_i=100\text{ mol }%$ are achieved, the mixture transforms into pure agent with constant vapor phase condensation temperatures $t_{\text{cond}}^\text{m,}\Psi=0,0\text{%} = \text{const}$, $t_{\text{cond}}^\text{m,}\Psi=100\text{%} = \text{const}$ in the condenser and with constant liquid phase evaporation temperatures $t_{\text{evap}}^\text{m,}\Psi=0,0\text{%} = \text{const}$, $t_{\text{evap}}^\text{m,}\Psi=100\text{%} = \text{const}$ in the evaporator. At that, the value of difference of average condensation temperature in the condenser and evaporation temperature in the evaporator, according to expression (1), increases up to maximum values, that equal $t_{\text{cond}}^\text{m,}\Psi=0,0\text{%} - t_{\text{evap}}^\text{m,}\Psi=0,0\text{%} = 36°C$, $t_{\text{cond}}^\text{m,}\Psi=100\text{%} - t_{\text{evap}}^\text{m,}\Psi=100\text{%} = 36°C$, while energy efficiency of the heat pump, on the contrary, decreases down to minimum values. The heat pump cycle where $\Psi_i=0,0\text{ mol }%$ and $\Psi_i=100\text{ mol }%$, (Fig. 2) is depicted as $a'$-$b'$-$c'$-$d'$-$a'$. Calculation results show that temperature values for cooled and heated environments with HC exert an important effect on selecting the value of zeotropic mixture component concentration. Change of temperature values of the environments with HC in condenser outlet $t_{i,t}^\text{cond}$ and evaporator outlet $t_{i,t}^\text{evap}$ leads to picking other makes of mixtures and other values of their low-boiling component proportion.
Selection results are shown in the Diagram “Temperature – Entropy” (figure 2). The results of the developed method can be found in Patent № RU 2658414 U1 [10], registered on 21.06.2018.

Method of heat calculation is proposed for determination of value of heat exchanging surface of evaporators and condensers of heat pumps, working on zeotropic mixtures of optimal composition, as described above.

3. Development of the thermal design model for tubular evaporators and condensers of heat pumps working on zeotropic mixtures of optimal composition

Horizontal or slightly inclined pipes or bundles of paralleled pipes are normally used for working fluids evaporation and condensation in heat pumps.

Here \( t_{\text{cond}}^m, t_{\text{cond}}^n \) – are constant condensation temperatures of the pure agents’ vapour phase in the condenser, \( t_{\text{evap}}^m, t_{\text{evap}}^n \) – are constant boiling temperatures of the pure agents’ liquid stage in the evaporator, \( t_{\text{a}}, t_{\text{so}}, t_{\text{d}}, t_{\text{cd}} \) – are primary and terminal temperatures of the environment with HC, e.g. air in the inter-tubular space of the evaporator, \( t_{\text{t}}, t_{\text{t}} \) – are terminal and primary temperatures of the environment with HC, e.g. air in the condenser, \( \Delta t_{\text{inh}}, \Delta t_{\text{e}} \) – is a temperature drop between heat exchange flows in the counter-flow condenser and evaporator correspondingly, \( t_{\text{a}}, t_{\text{so}}, t_{\text{d}}, t_{\text{cd}} \) – are mean condensation and boiling temperatures for the zeotropic mixture in the condenser and evaporator correspondingly, \( t_{\text{a}}, t_{\text{so}}, t_{\text{d}}, t_{\text{cd}} \).

Notations of pure agents’ processes: \( a \) – contraction of the vapour phase in the compressor; \( b \) – condensation of the vapour phase in the condenser; \( c \) – temperature decrease of the saturated liquid phase in the regulator; \( d \) – evaporation of the saturated liquid phase in the evaporator.

It is known that processes of evaporation and condensation in horizontal tubes take place in the reverse order, they are described by the same heat exchange equations and have \( z \) enlarged flow modes of the vapour-liquid zeotropic mixture inside the pipe (Figure 3), where \( z \) – is the number and title of the flow mode: \( z=1 \) – layered plug mode; \( z=2 \) – circular wave mode; \( z=3 \) – dispersed mode. Thus, when liquid evaporates in the horizontal pipe (\( z=1, 2, 3 \)), the layered plug flow mode takes place first (\( z=1 \)), as the amount of evaporated liquid increases, it transforms into circular wave flow mode (\( z=2 \)), as it proceeds to boil out, it transforms into dispersed flow mode (\( z=3 \)). At the same time, in the course of steam phase condensation in the horizontal pipe (\( z=3, 2, 1 \)), the order is reverse with dispersed flow mode taking place first (\( z=3 \)), then, as the amount of condensed steam increases, it transforms into circular wave flow mode (\( z=2 \)), and consequently in the course of condensation, it transforms into layered plug flow mode (\( z=1 \)). This reverse but strictly defined sequence of flow mode changes of the vapour-liquid mixture allows us to outline the prerequisites for the development of a generalized model for thermal calculation of horizontal tube evaporators and condensers of heat pumps.

Based on this description, the problem of heat exchange at evaporation and condensation of the zeotropic mixture in the horizontal pipe can be formulated as follows. The heat exchange device in the form of a bundle of straight pipes with internal diameter \( d \) (see figure 3) is supplied with saturated liquid or vapour phase of the optimal composition zeotropic mixture, which allows to achieve minimal temperature drop with mass flow rate \( G \). The outer surface of the heat exchange pipe receives or carries off heat flow \( q \) of constant intensity \( \alpha \).

In the course of complete evaporation, dryness factor \( X \) of the steam-liquid mixture increases from \( X_{z=1, p} \) to \( X_{z=1, f}=1,0 \) or, conversely, in the process of full condensation \( X \) decreases from \( X_{z=1, i}=1,0 \) to \( X_{z=1, p} \). At that the dryness degree changes in the evaporator for each of the flow modes within the following ranges: layered plug flow mode \( X_{z=1} \geq X_{z=2} \geq X_{z=3} \); circular wave mode \( X_{z=2} \geq X_{z=3} \); dispersed mode \( X_{z=3} \); circular wave mode \( X_{z=2} \); layered plug mode \( X_{z=1} \). Zeotropic mixture that enters the evaporator with molt content of the low-boiling component in the vapour phase \( \psi' \), completely boils out within the range of temperatures from \( t_p \) to \( t_i \). At that, for every flow mode, the evaporation temperature of the mixture changes within the following ranges: layered plug mode \( t_{z=1} \geq t_{z=2} \geq t_{z=3} \); circular wave mode \( t_{z=2} \geq t_{z=3} \); dispersed mode \( t_{z=3} \). The
supplied heat flow consists of both mixture evaporation heat and heat used for its warming up within temperature range for its complete evaporation from \( t_p \) to \( t_t \).

Zeotropic mixture that enters the condenser with molar content of the low-boiling component in the vapour phase \( \psi' \), is fully condensed within the range of temperatures from \( t_z \) to \( t_p \). At that, for every flow mode, the condensation temperature of the mixture changes within the following ranges: dispersed flow mode \( t_z = 3 > t_z = 3 > t_p = 3 \); circular wave mode \( t_z = 2 > t_z = 2 > t_p = 2 \); layered plug mode \( t_z = 1 = t_z = 1 = t_p = 1 \).

The carried off heat flow consists of mixture condensation and cooling within the range of temperatures of its complete condensation from \( t_t \) to \( t_p \).

The values of specific thermal capacity and latent heat of vapour formation for the saturated vapour phase of the zeotropic mixture, within the range of layered plug, circular wave and dispersed flow modes, are taken as constant values and equal their mean values, i.e.: \( c_{z1,mean} \), \( c_{z2,mean} \), \( c_{z3,mean} \) and \( r_{z1,mean} \), \( r_{z2,mean} \), \( r_{z3,mean} \).

The change of flow modes and content of the low-boiling component in the vapour \( \psi' \) and liquid \( \psi' \) phases of the vapour-liquid mixture leads to heat transfer coefficient change between the internal surface of the heat exchange pipe and the zeotropic mixture. At that, for each of the flow modes mentioned, the content of low-boiling component in liquid phase \( \psi' \) changes within the following ranges: layered plug flow mode \( \psi'_{z=1} \leq \psi'_{z=1} \leq \psi'_{z=1,p} \); circular wave flow mode \( \psi'_{z=2} \leq \psi'_{z=2} \leq \psi'_{z=2,p} \); dispersed flow mode \( \psi'_{z=3} \leq \psi'_{z=3} \leq \psi'_{z=3,p} \).

Heat transfer coefficient \( k_z(\alpha_z) \) for the evaporator and condenser changes within similar ranges: layered plug flow mode \( k_{z=1} = k_{z=1} < k_{z=1} < k_{z=1} = k_{z=1} = k_{z=1} \); circular wave flow mode \( k_{z=2} = k_{z=2} = k_{z=2} = k_{z=2} = k_{z=2} \); dispersed flow mode \( k_{z=3} = k_{z=3} = const \), but taking into account various intensity of heat transfer during evaporation and condensation, they are calculated according to their characteristic dependences.

Effective length of the counter-flow tubular evaporator and condenser \( \sum_{z=1}^{z=3} L_z \) is determined based on the heat balance equation, as the sum of surfaces of separate sections at layered plug \( L_{z=1} \), circular wave \( L_{z=2} \) and dispersed \( L_{z=3} \) flow modes, according to the formula:

\[
\sum_{z=1}^{z=3} L_z = L_{z=1} + L_{z=2} + L_{z=3},
\]
The length of individual \( z \)-th section of the counter-flow tubular evaporator at full evaporation of the zeotropic mixture inside the pipe and condenser at its full condensation is determined according to the formula:

\[
L_z = \frac{G (x_{zt} - x_{zp})}{\pi d_z k_{z,\text{evap}}(\alpha_{z,\text{evap}})} \left( r_{z,\text{mean}} \int_{t_{p,z}}^{t_{z}} \frac{dx_z}{dt_z} dt_z + c_{z,\text{mean}} \int_{t_{p,z}}^{t_{z}} \frac{dt_z}{t_1 - t_z} dt_z \right),
\]

where \( G \) is the calculated mass flow rate of the zeotropic mixture which is circulating in the evaporator and condenser, kg/hour; \( X_{zt}, X_{zp} \) – terminal and primary values of the dryness degree of vapour-liquid mixture, at which we can observe transition from one flow mode into another, in unit fractions; \( d_z \) – diameter of the flow type tubular evaporator and condenser of the heat pump, m; \( k_z(\alpha_z) \) – value of the heat transfer coefficient, as heat transfer coefficient function, which is characteristic of layered plug, circular wave and dispersed flow modes, Watt/(m\(^2\)-K); \( t_1 \) – heat carrying agent temperature, \(^\circ\)C; \( t_z \) – current temperature of the vapour-liquid mixture, changing within the range of temperatures from \( t_{p,z} \) to \( t_{z} \), in the evaporator and from \( t_{z} \) to \( t_{p,z} \) in the condenser, \(^\circ\)C; \( t_{p,z}, t_{z} \) – primary and terminal temperature of the evaporated and condensed vapour-liquid zeotropic mixture at various sections, with layered plug, circular wave and dispersed flow modes correspondingly in the flow through tubular evaporator or condenser at corresponding dryness degree \( X_{zt} \) and \( X_{zp} \), \(^\circ\)C; \( r_{z,\text{mean}}, c_{z,\text{mean}} \) – corresponding, mean values of the latent heat of vaporization and specific heat capacity of the mixture within ranges of its evaporation or condensation, correspondingly, at sections with layered plug, circular wave and dispersed flow modes, kj/kg, kj/kg·K.

Inserting equation (5) to determine the length of \( z \)-th sections \( L_{z=1}, L_{z=2}, L_{z=3} \) for layered plug, circular wave and dispersed flow modes into formula (4), we determine the value of the total heat exchange length \( \sum_{z=1}^{3} L_z \) of the flow through evaporator and condenser of the heat pump.

Thus, the article presents the development of functioning principles and thermal design model for compression heat pumps working on zeotropic mixtures.

4. Conclusions

1. Analysis of the possibility of using working fluids as components of zeotropic mixture in heat pumps showed that mixtures R22/R142b, R32/R134a, R32/R152a have low ozone-depleting activity, and mixtures of saturated hydrocarbons R290/600 (propane and butane), R600a/R601 (iso-butane and n-pentane), R290/R601a (propane and iso-pentane), R600a/R601b (iso-butane and n-pentane) are considered absolutely safe in terms of the degree of their activity related to ozone layer destruction. These gases do not cause noticeable greenhouse effect, do not influence climate change negatively, and do not have negative impact on human health. Saturated hydrocarbon mixtures have a significantly lower price in comparison with other working fluids.

2. The article proposes principles and methodological provisions for achieving minimum value of difference of average condensation and boiling temperatures through selecting the molar concentration value of the zeotropic mixture low-boiling component consisting of two components that are close in their physical properties. They take into consideration saturation temperature changes in the condenser and evaporator of the heat pump depending on relative amount of boiled out mixture and temperature change in heated and cooled environments with restricted thermal capacity.

3. The authors of the article developed a thermal design model for tubular evaporators and condensers of heat pumps working on zeotropic mixtures. It allows to take into consideration change in the intensity of heat exchange and temperature conditions depending on constantly changing composition and flow modes of the vapor-liquid mixture taking place in the following sequence: dispersed, circular wave, and layered-plug flow modes.
References

[1] Bukin V G and Kuzmin A Yu 2007 Refrigerators operating on azeotropic mixtures of refrigerants (Astrakhan: ASTU Press) p 156

[2] Bukin V G and Kuzmin A Yu 1996 Experimental study of small refrigerating machines on a mixture of R22 / R142b Kholodilnaya tekhnika 5 12–4

[3] Kyoto Protocol to the United Nations Framework Convention on Climate Change http://bellona.ru/2007/05/08/kioitkij-protokol-k-ramonnoj-konvent (date of access 05/04/2017)

[4] Kutepov A M, Sterman L S and Styushin N G 1977 Hydrodynamics and heat transfer in vaporization: a textbook for universities (Moscow: High School) p 352

[5] Martynovskiy V S 1979 Cycles, circuits and characteristics of thermotransformers (Moscow: Energiya) p 320

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

[7] Montreal Protocol on Substances that Deplete the Ozone Layer http://www.un.org/ru/documents/decl_conv/conventions/pdf/montreal.pdf (date of access 05/04/2018)

[8] Ogurechnikov L A 2011 Condensation of R32 / R134a in heat pump heat supply Kholodilnaya tekhnika 2 46–8

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

[10] Usachev A P, Rulev A V and Usacheva E Yu Utility model patent № RU 2658414 U1 Method for selecting working agents in a compression heat pump Registered on 21/06/2018

[11] Rulev A V, Usachev A P, Shurayts A L et al 2010 System research on increase of heat exchange intensity of liquefied petroleum gas regasificators (Saratov: Saratov State Technical University) p 244

[12] Staskevich N L and Vigdorchik D Ya 1986 Handbook of liquefied petroleum gases (Leningrad: Nedra) p 543

[13] Sukhikh A A, Generalov K S and Akimov I A 2000 Heat pump tests for individual house heat supply Works of MSUEE: Low temperature technology for ecology (Moscow: MSUEE)

[14] Ho-Saeng Lu, Hyeon-Ju Kim, Dong-gyu Kang and Djingsoo Jung 2012 Thermodynamic performance of R32/R152a mixture for water source heat pumps Enje 40 251–7

[15] Jianyong C, Janlin Yu 2008 Of new refrigeration cycle using mixture R32/R134a for resitential-air conditioner applications Energy and Buildings 40 171–9

[16] Kim M, Kim M S and Kim Y 2004 Experimental study on the performance of heat pump system nith refrigranten mixtures composition change Energy 24 1053–68

[17] 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 No.6. 391–9

[18] Shurayts A L, Rulev A V and Usacheva E Yu 2016 Assessing Energy Efficiency of Compression Heat Pumps in Drying Processes when Zetotropic Hydrocarbon Mixtures are Used as Working Agents. MATEC Web Conf. Volume 73, (2016) 02015 XV International Conference “Topical Problems of Architecture, Civil Engineering, Energy Efficiency and Ecology – 2016” 1–9 http://dx.doi.org/10.1051/matecconf/20167302015