EXPERIMENTAL AND NUMERICAL INVESTIGATIONS OF HEAT REGENERATION PROCESS EFFICIENCY IN A HEAT PUMP WITH A MIXTURE OF REFRIGERANTS

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This article presents the results of investigations of new heat pump refrigerants that provide a high-efficiency working cycle of heat pump. These refrigerants are mixtures of a few cooling agents, and they are more environmentally friendly than the traditional ones. The mixture components used are C₃F₈ and SF₆ (R218/R846). The specific differences in these refrigerants used in the heat pump thermodynamic cycle are shown, and their benefits are presented. The influence of the recuperative heat exchanger characteristics on the efficiency of heat pump operation was investigated by numerical and physical modeling. Investigations were carried out on the experimental facility of the Department of Theoretical Basics of Heat Engineering at the National Research University “Moscow Power Engineering Institute”. The experimental facility was modernized by replacing the tubular (double-pipe) heat exchanger by a plate heat exchanger. This upgrading was made to decrease the flow resistance and increase the heating temperature at the compressor inlet for the refrigerant mixtures described above. Thermal testing of the plate heat exchanger was conducted. The efficiency of the plate heat exchanger as compared to the tubular one is shown.

KEY WORDS: heat pump, coolant, recuperator, heat exchanger, energy efficiency, coefficient of performance

1. INTRODUCTION

The restriction on the use of Freon as a refrigerant, which is dangerous for ozone, can be considered as the factor constraining the distribution of heat pumps (HP). This problem can be solved by using less dangerous substances as a cooling agent in HP. They can be new, alternative refrigerants of synthetic production, or natural substances (carbon dioxide, hydrocarbons, etc.). The authors of this paper suppose that a mixture of refrigerants consisting of fluorocarbon working substances may ensure high power efficiency and environmental safety on application in HP technologies. Moreover, these refrigerants are capable of providing new quality of technologies. For example, they allow application of HP in higher temperature ranges than the traditional HP.
The problem of Freons impact on the ozone layer and their influence on the greenhouse effect are determined in the Montreal and Kyoto protocols. That fact led to multibillion losses in the refrigerating industry all over the world. So, the necessity of solving the HP ecological problem has given impetus to the scientific community to develop new working substances for them, both of natural and synthetic origin.

During the last ten years the National Research University "MPEI" carried out a cycle of research works on fluorocarbon compositions as HP refrigerants. For the first time an azeotropic mixture of octafluoropropane, C₃F₈ (R218), and hexafluoride of sulfur, SF₆ (R846), was suggested by Mazurin (1996) for being used as a HP refrigerant. The mixture consists of 95 wt.% R218 and 5 wt.% R846. This mixture has azeotropic properties in the range of temperatures from –17 to –42°C. The refrigerant is designated by R510 and sometimes was called "Khladon-M" in publications. This refrigerant is safe for ozone, but possesses the essential potential of global warming. It is compatible with oils HF12–16 and HF 22s–16. That fact allows substituting Freons R12 and R22 with R510 in heat pumps and other devices without oil changeover. Device designers recommend this refrigerant for replacing R12 in home refrigerators and trade sealed refrigerators at temperatures below –20°C. According to the designers, in case the refrigerant is used in piston compressors this will allow saving the electric power by about 10–14%.

As is shown in (Antanenkova and Sukhikh, 2013), an increase in the efficiency of Freon HP can be reached by applying a nonazeotropic mixture as a working substance. In such a case, the local temperature difference almost does not change along the heat-exchanging devices. It provides a decrease in the processes irreversibility.

The heat conversion coefficient of the HP thermodynamic cycle is increased when using a nonazeotropic mixture of fluorocarbons and sulfur hexafluoride components as a refrigerant instead of R22. The research results of (Antanenkova and Sukhikh, 2013; Antanenkova et al., 2014) demonstrate the increase in the heat conversion coefficient from 3.63, with the use of R22, to 3.87 with the use of a mixed refrigerant RC318/R846 with the same parameters of the thermodynamic cycle. The greatest value of the heat conversion coefficient was found in the cycle with heat regeneration using a mixture of R31-10/R846 (90/10). Its value reached 4.02 with compressor temperatures of 50°C at the inlet and 90°C at the outlet.

There are additional advantages in the use of the mixture of these refrigerants (fluorocarbons with sulfur hexafluoride). The fluorocarbons C₃F₈, C₄F₈, and C₄F₁₀ practically do not dissolve the oil, so that the quantity of oil can be strictly regulated, its properties can be optimized by adding sulfur hexafluoride SF₆. Controversy, the R134a Freon and other mixtures like it require the use of expensive synthetic oils.

In the thermodynamic cycles with the positive inclination of condensation curve on the TS chart, the process of heat regeneration from a condensate flow (at the throttle device inlet) to steam flow (at the evaporator outlet) exerts a substantial influence on the cycle efficiency. Including the regenerative heat exchanger in the HP design is a
necessary condition for using a fluorocarbon mixture as a refrigerant, since otherwise, the compression process of superheated steam in the compressor cannot be provided. Due to the specifics of the cycle configuration on "pure" fluorocarbon's group and the mixture of their refrigerants, the temperature of the working substance after the compressor is low. This circumstance makes it possible to significantly increase the regeneration heat share in the cycles for increasing the heat conversion factor.

In (Antanenkova, 2013), the characteristics of the "double-pipe" heat recuperator were studied on an experimental facility of HP of power 8 kW. The facility construction is modular. It allows easy changeover of the refrigerant. The use of the recuperator as a heat regeneration device in a HP led to the increase in the irreversible losses in the heat exchanging process because of the friction and hydraulic resistance. That lowered the efficiency of the entire installation in comparison with the HP using R22. It was concluded in (Antanenkova, 2013) that it was necessary to upgrade the existing facility by replacing the "double-pipe" recuperator by a plate heat exchanger.

The upgrading procedure was performed and the heat engineering tests were carried out. The technical details of that plate heat exchanger are given in (Podlevskikh et al., 2015; Antanenkova, 2014).

2. EXPERIMENTAL SETUP AND MEASUREMENT TECHNIQUE

An experimental modular-type model of a HP of power 8 kW (HPM) was used for determining the facility power efficiency with application of different working substances. Its construction provides easy changeover of refrigerants and also the use of any modular units. The general view of the experimental HPM with a recuperative heat exchanger is presented in Fig. 1.

The following main devices constitute the hydraulic circuit of the installation: evaporator, condenser, recuperator, and compressor. Also the following accessory nodes are the parts of the circuit: filter dehumidifier, electromagnetic valve, and temperature-controlled valve.

The experimental stand is equipped with systems of pressure and temperature measurements at the inlet and outlet of each main device, of evacuation and filling with a working substance, of measuring the water flow rate in the heat supply pipe and low-potential heat pipe, of regulating the frequency of electric current supply and power measurement of the electromotor.

The external systems of the facility are: a system for hot water supply consisting of a condenser, a tank for hot water accumulation, a circulation pump, and open system of low-potential heat extraction.

The piston compressor is used for compression of the working substance in the experimental HPM. The compressor model is MT28JE4AVE of Maneurop (France). It is supplied with a three-phase electromotor. The compressor has a volume flow rate of 8.4 m³/h and electromotor rotating speed of 2900 rpm with a network frequency
of 50 Hz. To control the compressor, an Emotron FDU 2.0 (Emotron AB, Sweden) frequency converter is used. The installation is equipped with an A10-405 liquid separator (Alco Controls, Germany) for compressor protection against hydraulic impact (from drops of oil and refrigerant). A DCL 304S (Danfos) filter dehumidifier protects the facility from moisture, acids, and solid inclusions.

The hot water supply system is equipped with a Grundfos UPS 25–120 circulation pump with an adjustable productivity. The system also contains a Logalux SU300/1 (Buderus) vertical tank for hot water accumulation. The tank volume is 0.3 m³.

The condenser and the evaporator have an identical design that differs only in length. The patent (Sukhikh and Antanenkova, 2008) protects the construction of the
devices. The technical details of the condenser and evaporator design are described in (Antanenkova, 2013), some technological features are described in (Zoubkov and Ovtchinnikov, 1998). Construction type is a tube bundle. The working surface of the evaporator is slightly larger than that of condenser. It allows one to decrease the temperature differences in the device. This also allows one to achieve a lower value of low-potential water flow cooling. Cooling values are in the range of several degrees (3–5°C).

Some technical data of the condenser and evaporator are as follows:

- Both devices:
  - finned surface extension ratio of pipes inside devices $\zeta = 7.0–7.5$;
  - casing is a smooth pipe;
  - material — steel;
  - diameter of the casing 59.4 mm.

- Condenser:
  - length 1100 mm;
  - the length of a working (finned) surface $L_c = 1065$ mm;
  - the working surface area $F_c = 0.3044$ m$^2$.

- Evaporator:
  - length 1200 mm;
  - the length of a working (finned) surface $L_{ev} = 1165$ mm;
  - the working surface area $F_{ev} = 0.3327$ m$^2$.

Water of the external circuit flows in the internal pipes of the devices; refrigerant condenses or boils on the finned surface in the interpipe space.

The measuring system of the experimental facility is described in detail in (Antanenkova et al., 2014; Antanenkova, 2013; Podlevskikh et al., 2015).

The measurement tools accuracy is given below:

- The refrigerant pressure at the HPM evaporator and recuperator inlet and outlet is measured by MO 11202 pressure gauges with an upper limit of pressure measurement of 10 kgs/cm$^2$, deviation is ±0.04 kgs/cm$^2$.
- The refrigerant pressure at the inlet and outlet of the condenser is measured by the same gauges with an upper limit of pressure measurement of 25 kgs/cm$^2$, deviation is ±0.1 kgs/cm$^2$.
- The temperatures at the characteristic points of the experimental HPM (Fig. 1) are taken by a Termoizmeritel TM-12.2 12-channel precision measuring module (the producer is the industrial-ecological enterprise "SIBEKOPRIBOR", Novosibirsk, Russian Federation). Temperature measurement is performed by using individual static characteristics (ISC) of temperature sensors. The temperature range is from –50°C to 200°C. Deviations in the temperature range
from 0°C to 100°C is ±0.05°C, in the ranges below 0°C and above 100°C does not exceed ±0.1°C.

- The water flow rates relative deviation is ±2–5% in the range of the minimum.

The data for a plate heat exchanger (recuperator) is given in Table 1. The heat exchange surface of the plate heat exchanger is almost two times less than the surface of the previously used "double-pipe" tubular recuperator.

The intensity of the water heating process in a HPM boiler depends on the temperature of the low-potential heat carrier. During water heating, visual control of all working parameters is made. In a cold season (in winter) the temperature of water at the evaporator inlet becomes low (about 6°C). So, mixing of cold water with hot water is carried out. If needed, full switching to hot water supply is made.

The water heating process in the condenser slows down as the water temperature increases. To achieve the mode close to a stationary one, the rate of the electromotor rotation is decreased by using a frequency converter. After stabilization of hot water temperature in the range of 50–65°C within 7–10 min all parameters of the HPM specified in Fig. 1 are fixed. The barometric pressure is recorded by a mercury barometer at the same time also.

3. EXPERIMENTAL RESULTS: PROCESSING AND ANALYSIS

Experimental investigations of the processes proceeding in the HPM were carried out with a mixture of R218 and R846 refrigerants in the mass ratio of 91.83% and 8.17%, respectively. The thermodynamic cycle is shown in Fig. 2.

The line connection by points 1–10–2–9–11–3–4 in Fig. 2 forms a refrigerant cycle; arrow path 5–6 describes water heating in the HWS system; arrow path 7–8 pres-

### Table 1: Constructive characteristics of HPM regenerator

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Designation, dimensionality</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of plates</td>
<td>N</td>
<td>14</td>
</tr>
<tr>
<td>Number of effective plates</td>
<td>Ne</td>
<td>12</td>
</tr>
<tr>
<td>Each plate area</td>
<td>( f, \text{ m}^2 )</td>
<td>0.014</td>
</tr>
<tr>
<td>Size of one plate</td>
<td>( a_1 \times b_1 \times \delta, \text{ mm} )</td>
<td>200 × 70 × 0.3</td>
</tr>
<tr>
<td>Heat exchanger dimensions</td>
<td>( a \times b \times c, \text{ mm} )</td>
<td>208 × 78 × 40</td>
</tr>
<tr>
<td>Distance between plates</td>
<td>( \gamma, \text{ mm} )</td>
<td>2.8</td>
</tr>
<tr>
<td>Equivalent diameter</td>
<td>( d_{eq}, \text{ mm} )</td>
<td>5.6</td>
</tr>
<tr>
<td>Area of the passage section</td>
<td>( f_1, \text{ m}^2 )</td>
<td>5.6×10^{-4}</td>
</tr>
<tr>
<td>Heat exchange surface</td>
<td>( S, \text{ m}^2 )</td>
<td>0.168</td>
</tr>
</tbody>
</table>

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ents the cooling process of the low-potential heat carrier (tap water). In Fig. 2, the substantial temperature glide is observed (nonisothermal process during the boiling and condensation of refrigerant): 10–2t shows the theoretical reversible process of pressurization in the compressor, 10–2, the real process of compression. Path 2–9–11 describes the processes of cooling and condensation of mixture refrigerant; path 11–3 is the cooling process of the condensate in the recuperator. The refrigerant boiling process is presented by path 4–1; and heating process of refrigerant vapors in the recuperator is described by path 1–10.

It was found that in a stationary mode of low-potential heat carrier temperature, the results of measurements in different operating modes of HPM are very close to each other. So, in this paper we present and analyze the results of only one characteristic operating mode of HPM. The results of the experimental mode of HPM operation with a plate heat exchanger as the recuperator and the R218/R846 (91.83/8.17 wt.%) mixture as the working substance are listed in Table 2.

The measured values from Table 2 can be used for determining all the parameters of the modular heat pump facility.

Heat fluxes released in the condenser $Q_c$ and consumed in the evaporator $Q_{ev}$ are determined as

$$Q_c = G_w^c c_w (t_6 - t_5),$$  \hspace{1cm} (1)

$$Q_{ev} = G_w^{ev} c_w (t_7 - t_8),$$  \hspace{1cm} (2)

where $c_w$ is the water heat capacity, kJ/(kg·K).

Equations of heat balance for the condenser and evaporator, the flow rate values of the refrigerant mixture $m_{cc}$ in the HPM circuit can be calculated subject to (1) and (2) as
\[ m_{cc}^c = \frac{Q_c + Q_{\text{loss}}^c}{(h_2 - h_{11})}, \quad (3) \]

\[ m_{cc}^{ev} = \frac{Q_{ev} - Q_{\text{loss}}^{ev}}{(h_1 - h_4)}, \quad (4) \]

where \( h_i \) is the enthalpy at the characteristic point \( i \) of a cycle (Fig. 2).

The mixture flow rate can be determined by using the ratio

\[ \frac{m_{cc}^{compr}}{h_2} \frac{h_1}{h_0}. \quad (5) \]

The following numerical values were obtained in the calculations:

\[ m_{cc}^{ev} = 0.094 \text{ kg/s}; \quad m_{cc}^c = 0.092 \text{ kg/s}; \quad m_{cc}^{compr} = 0.091 \text{ kg/s}. \]

Besides, the compressor efficiency value can be defined as

\[ \eta_{0i}^{compr} = \frac{h_2 - h_{10}}{h_2 - h_{10}}. \quad (6) \]

The compressor efficiency value is \( \eta_{0i}^{compr} = 0.85 \).

The heat flux transferred by the recuperator can be calculated on the heating side as

\[ Q_{\text{rec}} = m_{cc}^{am} (h_{11} - h_3) \quad (7) \]

and on the heated side as

\[ Q_{\text{rec}} = m_{cc}^{am} (h_{10} - h_1), \quad (8) \]

where \( m_{cc}^{am} \) is the average value calculated by using the results of Eqs. (3)–(5).

The thermophysical properties of the R218/R846 (91.83/8.17 wt.%) mixture for using in Eqs. (3)–(8) were determined based on the data of Table 2 in the NIST REFPROP 9.0 program (2007).

The heat conversion coefficient of the HPM was calculated by two different methods: as the ratio of the condenser heat flux (1) and the measured power consumed by the compressor given by the formula

\[ \mu = \frac{Q_c}{N_{\text{compr}}}, \quad (9) \]

and also by dividing the condenser specific heat capacity \( q_c = h_2 - h_9 \) by the real thermodynamic work \( I_{\text{compr}} = h_2 - h_1 \) made by the compressor:
**TABLE 2:** Results of HPM working parameter measurements

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature, °C</td>
<td></td>
</tr>
<tr>
<td>Mixture temperature at the evaporator outlet $t_1$</td>
<td>9.4</td>
</tr>
<tr>
<td>Mixture temperature at the compressor outlet $t_2$</td>
<td>78.0</td>
</tr>
<tr>
<td>Mixture temperature at the recuperator outlet (before a throttle) $t_3$</td>
<td>43.7</td>
</tr>
<tr>
<td>Mixture temperature at the evaporator inlet $t_4$</td>
<td>10.8</td>
</tr>
<tr>
<td>Water temperature at the condenser inlet $t_5$</td>
<td>55.7</td>
</tr>
<tr>
<td>Water temperature at the condenser outlet $t_6$</td>
<td>59.1</td>
</tr>
<tr>
<td>Water temperature at the evaporator inlet $t_7$</td>
<td>16.8</td>
</tr>
<tr>
<td>Water temperature at the evaporator outlet $t_8$</td>
<td>15.2</td>
</tr>
<tr>
<td>Mixture temperature at the condenser outlet $t_9$</td>
<td>59.7</td>
</tr>
<tr>
<td>Mixture temperature at the compressor inlet $t_{10}$</td>
<td>35.5</td>
</tr>
<tr>
<td>Mixture temperature at the recuperator inlet (after the condenser) $t_{11}$</td>
<td>59.2</td>
</tr>
<tr>
<td>Air temperature $t_a$</td>
<td>21.0</td>
</tr>
<tr>
<td>Total pressure, kPa</td>
<td></td>
</tr>
<tr>
<td>Mixture pressure at the evaporator inlet $p_{in}^{ev}$</td>
<td>642.2</td>
</tr>
<tr>
<td>Mixture pressure at the evaporator outlet $p_{ev}^{out}$</td>
<td>603.0</td>
</tr>
<tr>
<td>Mixture pressure at the compressor inlet $p_{in}^{c}$ ($p_{out}^{rec}$)</td>
<td>548.0</td>
</tr>
<tr>
<td>Mixture pressure at the condenser inlet $p_{in}^{ev}$</td>
<td>2298.0</td>
</tr>
<tr>
<td>Mixture pressure at the outlet of a recuperator in the area of the condenser ($p_{out}^{rec}$)</td>
<td>2229.0</td>
</tr>
<tr>
<td>Water flow rate, kg/s</td>
<td></td>
</tr>
<tr>
<td>in the evaporator $G_w^{ev}$</td>
<td>0.608</td>
</tr>
<tr>
<td>in the condenser $G_w^{c}$</td>
<td>0.410</td>
</tr>
<tr>
<td>Compressor power $N_c$, kW</td>
<td>1.79</td>
</tr>
</tbody>
</table>

\[ \mu_h = \frac{q_c}{I_{compr}}. \]  (10)

The numerical values of the heat conversion coefficient calculated by two methods were equal to 3.26 and 3.23, respectively.

Heat losses at the condenser $Q_{loss}^c$ and evaporator $Q_{loss}^{ev}$ were estimated in calculations. Their values did not exceed 1.3% and 0.8%, respectively, in relation to the heat...
fluxes in those devices, i.e., they were negligible, and in further calculations were not considered.

The deviation of the condenser heat flux \( Q_c \) from the amount \( (Q_{ev} + N_c) \) did not exceed 0.5% in relation to \( Q_c \), i.e., a heat balance convergence takes place.

The deviations of the mixture flow rate calculated by Eqs. (3)–(5) did not exceed 1.8% in relation to their average value. This is the evidence of the heat balance coherence that confirms the reliability of the results obtained.

The numerical values of the heat conversion coefficient of HPM calculated by two independent methods practically coincide (3.26 and 3.23) as shown above.

All the results allow considering the R218/R846 mixture as well as RC318/R846 and R31-10/R846 mixture like perspective working substances for heat pumps. However, there is a necessity to define the most optimum working substance parts ratio and to find design decisions allowing one to provide maximum HPM efficiency for using these mixture refrigerants.

The heat flux transferred by the recuperator, calculated by Eqs. (1) and (2), was 2.1 kW. The overheating factor of the working substance at the compressor inlet was \( t_{10} - t_1 = 26^\circ \text{C} \). This points to the construction optimality of the plate heat exchanger for heat regeneration needs in HPM. Besides, the pressure losses on the gas side in the recuperator were 55 kPa (9% of the inlet pressure). It is three times less than the losses of pressure in the evaporator area in the recuperator of "double-pipe" type (27%) (Antanenkova, 2013).

Thus, despite the substantial lowering of the heat exchange surface in comparison with a tubular recuperator, all the technological parameters of the facility were provided at the required level and the water temperature at the condenser outlet reached 26\(^{\circ}\)C.

The heat transfer coefficient of the recuperator [W/(m\(^2\)·K)] was calculated by using the heat flux transferred:

\[
k_{\text{rec}} = \frac{Q_{\text{rec}}}{F_{\text{rec}} \Delta t_{\text{rec}}}, \tag{11}
\]

where \( F_{\text{rec}} \) is the heat exchange surface of the recuperator [m\(^2\)] and \( \Delta t \) is the logarithmic temperature difference [\(^{\circ}\)C]:

\[
\Delta t_{\text{rec}} = \frac{(t_3 - t_1) - (t_{11} - t_{10})}{\ln \left( \frac{t_3 - t_1}{(t_{11} - t_{10})} \right)}. \tag{12}
\]

The quantity \( \Delta t \) was determined from the real temperature data taken at the inlet and outlet of the recuperator, it was equal to 28.7\(^{\circ}\)C. The integral heat transfer coefficient calculated by using the known area of the heat exchange surface (Table 1) was 435 W/(m\(^2\)·K).

The working substance temperature at the compressor outlet \( t_2 \) in the experiment was 78\(^{\circ}\)C. It is not possible to achieve maximum temperature due to the limits on the compressor operation. Therefore, it is possible to increase the heat flux transferred.
by the recuperator in the case of compressor upgrading (Antanenkova and Sukhikh, 2013; Antanenkova, 2014).

4. NUMERICAL INVESTIGATIONS

Numerical investigations of the influence of heat flux transferred by the recuperator on the HPM efficiency were carried out using the experimental data discussed above. During the numerical investigations, the initial data were fixed and the values were taken from the experimental results. The recuperator heat flux values define the temperature $t_{10}$ at the outlet. Thus, the temperature was iterated from 15°C to 50°C with a step of 5°C. The enthalpy values at the characteristic points of the thermodynamic cycle were calculated with the help of the REFPROP 9.0 program.

Hydraulic losses in heat-exchanging devices during calculations are not considered. It was suggested that the working substance at the evaporator outlet is in a state of dry saturated steam. That assumption was confirmed with the HPM test results.

The initial data for numerical investigations were the following:

• Mixture pressure at the evaporator outlet $p_{out}^{ev} = 603$ kPa.
• Mixture pressure at the condenser inlet $p_{in}^{cp} = 2298$ kPa.
• Compressor efficiency value $\eta_{i}^{compr} = 0.85$.

As the results of numerical research, the following data were obtained:

• Mass-specific heat flux released in the condenser $q_{c} = h_{2} - h_{9}$.
• Mass-specific heat flux consumed in the evaporator $q_{ev} = h_{1} - h_{4}$.
• Mass-specific heat flux transferred in the recuperator $q_{rec} = h_{10} - h_{1}$.
• Mass-specific thermodynamic work of compressor $l_{k} = h_{2} - h_{10}$.
• Mixture temperature $t_{2}$ at the compressor outlet.
• HPM heat conversion coefficient.

All data were determined as a function of various temperatures of the working substance overheating at the compressor inlet. The data of calculation are listed in Table 3.

<table>
<thead>
<tr>
<th>$t_{10}$, °C</th>
<th>$q_{c}$, kJ/kg</th>
<th>$q_{ev}$, kJ/kg</th>
<th>$Q_{rec}$, kJ/kg</th>
<th>$L_{c}$, kJ/kg</th>
<th>$t_{2}$, °C</th>
<th>$\mu$</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>40.46</td>
<td>25.25</td>
<td>4.95</td>
<td>15.21</td>
<td>61.8</td>
<td>2.66</td>
</tr>
<tr>
<td>20</td>
<td>45.54</td>
<td>29.55</td>
<td>9.25</td>
<td>15.99</td>
<td>64.7</td>
<td>2.85</td>
</tr>
<tr>
<td>25</td>
<td>50.55</td>
<td>33.83</td>
<td>13.53</td>
<td>16.72</td>
<td>68.0</td>
<td>3.02</td>
</tr>
<tr>
<td>30</td>
<td>55.49</td>
<td>38.11</td>
<td>17.81</td>
<td>17.38</td>
<td>71.7</td>
<td>3.19</td>
</tr>
<tr>
<td>35</td>
<td>60.42</td>
<td>42.38</td>
<td>18.04</td>
<td>18.04</td>
<td>75.7</td>
<td>3.35</td>
</tr>
<tr>
<td>40</td>
<td>65.32</td>
<td>46.66</td>
<td>22.08</td>
<td>18.66</td>
<td>79.9</td>
<td>3.50</td>
</tr>
<tr>
<td>45</td>
<td>70.21</td>
<td>50.95</td>
<td>26.36</td>
<td>19.26</td>
<td>84.2</td>
<td>3.64</td>
</tr>
<tr>
<td>50</td>
<td>75.07</td>
<td>55.25</td>
<td>34.95</td>
<td>19.82</td>
<td>88.7</td>
<td>3.77</td>
</tr>
</tbody>
</table>
It was confirmed by numerical investigations that in the case of an increase in the heat flux transferred in the recuperator, the value of the thermodynamic work of compressor grows slightly. At the same time the amount of heat produced by the HPM increases significantly. Thus, the HPM heat conversion coefficient increases. This dependence is shown in Fig. 3.

5. CONCLUSIONS

Fluorocarbons have some differences in comparison with the widespread refrigerants of synthetic and natural origin. Primary benefits and features of the mixture of fluorocarbon refrigerants for being used in a heat pump are the following:

- The positive inclination of the condensation curve requires including a recuperative heat exchanger in the circuit. Only then pressurization in the compressor proceeds in the field of superheated steam. An increase in the heat regeneration fraction leads to a slight growth of the thermodynamic work of compressor while the amount of heat produced by the heat pump increases significantly. Thus, the heat conversion coefficient increases too.
- The temperature glide is present in the mixture of refrigerants cycle. Its value depends on the addition of the second component for the working substance of fluorocarbon mixture. This addition allows regulation and, therefore, optimization of the temperature difference in the heat pump's evaporator and condenser.

FIG. 3: Dependence of the heat conversion coefficient $\mu$ on the recuperator outlet temperature $t_{10}$
Heat Regeneration Process Efficiency

- Low temperatures at the compressor outlet in comparison with the traditional refrigerants ensure its reliable operation.
- The insignificant percent of the addition of sulfur hexafluoride component gives the possibility for regulation and optimization of the oil amount transferred.
- Mixture refrigerants of such type are environmentally friendly and do not destruct the ozone shield.
- The nonazeotropic character of mixture boiling and condensation causes strict requirements to the stability of composition and impermeability of the circuit.

Experimental and numerical investigations of the parameters of the heat pump model that operates with a mixture of fluorocarbon refrigerants are carried out.

The new experimental and numerical data were obtained on the upgraded HPM after replacing the "double-pipe"-type recuperator with a plate heat exchanger. The results of the tests with R218/R846 mixtures (91.83/8.17 wt.%) confirmed the validity of the choice of the plate heat exchanger instead of tubular analogs.

It is experimentally shown that the essential nonisothermicity of condensation and boiling processes allows one to reduce the temperature difference in the condenser and evaporator. Due to this, the "irreversibility" of the heat exchanging processes is reduced, and, therefore, the efficiency of the entire HPM is increased.

The increase in the recuperator heat fraction can be recommended for the solution of the special technological problems regarding to heat pump operation in a raised temperature range. In this case, perspective and reasonable is the use the fluorocarbon mixture refrigerants as a working substance.

The analysis of the research results allows the conclusion that the heat regeneration process is a necessary condition of the heat pump thermodynamic cycle implementation, and also gives additional possibilities for achieving high performance by operation on mixture refrigerants.

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