Calculated and experimental evaluation heat pump distiller on pentane as working substance.

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ABSTRACT

One of the most common and reliable methods of water treatment is the method of thermal distillation. Despite the reliability of the method, its application is constrained by high energy intensity. The most effective way to reduce the cost of production of distillate is the use of thermal transformers, providing regenerate and heat recovery phase transformations of the distillate.

The use of working fluid with the most favorable thermodynamic properties is of paramount importance for the creation of high efficiency thermotransformers. The work is considered working fluid for high-temperature heat pumps and the results of the calculation-experimental study of high-temperature vapor compression heat-pumping distiller on natural working substance n-pentane.

Keywords: high-temperature heat pump, heat pump distiller, distillation, water of injection quality, n-pentane, hermetic refrigeration compressor

INTRODUCTION

Presently, the authors of the article are developing low productivity heat-pumping distillers (HPD) based on vapor compression heat pumps (HP) in order to drastically reduce the energy intensity of the distillation process in the mobile technology. A patented technical solution for mobile HPD for the production of water of injection quality is proposed and protected. (Malafeev et al., 2017). Reducing the cost of producing distillate is provided by the recuperation and regeneration of the heat of the phase transformations of water by means of the reverse vapor compression thermodynamic cycle of the HP functioning on the low-pressure working fluid (w.f.). The reduction in energy consumption in comparison with distillers based on heating elements is proportional to the conversion factor of the heating elements and in the limit can reach 40 times (Kalnin et al., 2010).

The use in high-temperature HPD w.f. with a lower normal boiling point than water allows you to generate almost any thermal power using the basic equipment used in refrigeration (primarily compressors).
Most of the HP manufactured and operated today belong to the category of medium-temperature units that heat the coolant to temperatures of 40 ÷ 70 °C. Nevertheless, in a number of industries such as food, chemical, pulp and paper and textile, for the implementation of the technological process, a large amount of heat must be supplied with a temperature level of 70 ÷ 150 °C (Wolf et al., 2012). The development and implementation of high-temperature HP is constrained by the difficulty of choosing a working substance with the necessary properties from the list of legal acts allowed for use.

Working fluid suitable for use in HPD should refer to the low-pressure group and have a normal boiling point ($t_s$) above minus 10 and a sufficiently high critical temperature for the implementation of a subcritical vapor compression cycle, the minimum value of which is determined by the required temperatures of the hot coolant.

As a device for compressing and moving w.f. vapor in heat pumps use a small capacity compressor (CM) volumetric the operation principle, designed for operation at the condensation temperature in the range of 60 ... 90 °C. For example, the Copeland ZH series compressor for recovery systems has a passport maximum condensation temperature when operating on R134a freon is 85 °C (Talyzin, 2017). Viking Heat Engines manufactures high-temperature heat pumps HeatBooster (Nilsson, 2017), operating on R1336mzz (Z) and R245fa freons, with a boiling point of 30 to 110 °C, and condensation 90-160 °C. The piston compressor HBC511 is used as the heart of the heat pump.

As a rule, the possibility of failure-free operation of a refrigerated hermetic compressor is limited by the crankcase temperature (120 ... 140 °C). First of all, this is determined by the maximum permissible temperature of the motor windings, depending on the quality of insulation (heat resistance classes and their corresponding temperatures are determined in accordance with GOST 8865 – 93 (ГОСТ 8865-93). At elevated temperatures, the physicochemical properties of the insulation in the environment of the working fluid and oil deteriorate. When heated, the destruction of oil is possible with the formation of fatty acids that corrode the insulation of wires (Babakin and Vygodin, 1998).

**MAIN SECTION**

In order to rationale the choice of the working fluid in the composition of the HPD, a thermodynamic analysis of the low-pressure pump cycle was carried out: boiling point - 90 °C, condensation - 110 °C, overheating and supercooling - 10 °C, isentropic compressor efficiency - 0.63.

Nomenclature w.f. for the conditions under consideration is extremely limited. (table 1)

| Fluid name | Efficiency of performance | Pressure ratio | Pressure difference, bar | Charge temp, °C | Dynamic efficiency |
|------------|----------------------------|----------------|--------------------------|-----------------|--------------------|
| R152a      | 9.90                       | 1.47           | 13.65                    | 128.06          | 0.52               |
| R1311      | 10.31                      | 1.44           | 9.58                     | 128.33          | 0.54               |
| Working Fluid | μ (m/s) | ν (m/s^2) | Δh (kPa) | ΔP (kPa) | T (°C) |
|---------------|---------|-----------|---------|--------|------|
| CYCLOPRO      | 10.46   | 1.42      | 12.84   | 126.46 | 0.55 |
| R217          | 10.48   | 1.48      | 24.62   | 141.81 | 0.55 |
| RC318         | 10.54   | 1.50      | 8.36    | 118.12 | 0.55 |
| R3110         | 10.57   | 1.51      | 7.34    | 116.27 | 0.55 |
| RE170         | 10.59   | 1.45      | 12.26   | 127.24 | 0.55 |
| PROPANE       | 10.65   | 1.46      | 12.68   | 128.70 | 0.56 |
| R236fa        | 11.00   | 1.52      | 8.14    | 120.27 | 0.57 |
| R600a         | 11.16   | 1.45      | 7.41    | 120.14 | 0.58 |
| R764          | 11.21   | 1.51      | 11.45   | 143.91 | 0.59 |
| 1BUTENE       | 11.42   | 1.47      | 7.01    | 121.52 | 0.60 |
| 1BUTENE       | 11.44   | 1.47      | 6.85    | 122.01 | 0.60 |
| R245mc        | 11.45   | 1.52      | 6.27    | 118.34 | 0.60 |
| R236ea        | 11.54   | 1.54      | 6.77    | 119.05 | 0.60 |
| R600          | 11.65   | 1.48      | 5.96    | 119.85 | 0.61 |
| T2BUTENE      | 11.68   | 1.49      | 5.97    | 121.95 | 0.61 |
| C2BUTENE      | 11.75   | 1.49      | 5.68    | 122.89 | 0.61 |
| R245fa        | 11.84   | 1.56      | 5.64    | 119.89 | 0.62 |
| NEOPENTN      | 11.88   | 1.49      | 4.43    | 116.16 | 0.62 |
| R1233zd       | 11.93   | 1.55      | 4.55    | 121.36 | 0.62 |
| METHANOL      | 12.00   | 1.88      | 2.24    | 154.03 | 0.63 |
| R4112         | 12.02   | 1.58      | 3.54    | 111.66 | 0.63 |
| R245ca        | 12.11   | 1.58      | 4.29    | 118.86 | 0.63 |
| R245mf        | 12.14   | 1.60      | 3.87    | 117.52 | 0.63 |
| R347mcc       | 12.18   | 1.59      | 3.18    | 114.23 | 0.64 |
| ETHANOL       | 12.19   | 1.99      | 1.55    | 143.83 | 0.64 |
| R601a         | 12.19   | 1.54      | 3.12    | 116.74 | 0.64 |
| R610          | 12.25   | 1.58      | 3.00    | 118.30 | 0.64 |
| R601          | 12.27   | 1.57      | 2.67    | 117.15 | 0.64 |
| ACETONE       | 12.27   | 1.67      | 1.92    | 129.28 | 0.64 |
| R365mfc       | 12.28   | 1.62      | 2.84    | 116.35 | 0.64 |
| NOVEC649      | 12.33   | 1.64      | 2.21    | 110.40 | 0.64 |
| CYCLOPEN      | 12.34   | 1.61      | 1.97    | 122.06 | 0.64 |
| R602a         | 12.44   | 1.64      | 1.52    | 115.23 | 0.65 |
| R602          | 12.47   | 1.67      | 1.27    | 115.87 | 0.65 |

Table 1. Project cycle parameters for different working fluids
Hydrocarbons seem to be one of the promising w.f. for high temperature HP (Kalnin and Malafeev, 2014). These refrigerants have relatively good thermodynamic and thermophysical properties, do not affect the ozone layer and do not create a greenhouse effect, are non-toxic, cheap and available, mix with mineral oils. The properties of the most available hydrocarbons are presented in table 2.

| Refrigerant | t_s, °C | T_sat, °C | P_sat, bar | Lower flammability mixed with air, % vol. | Upper flammability mixed with air, % vol. | Auto ignition temperature, °C | Ignition energy, MJ |
|-------------|---------|-----------|------------|------------------------------------------|------------------------------------------|-------------------------------|-------------------|
| R290        | -42.1   | 96.7      | 42.5       | 2.3                                      | 9.4                                      | 470                           | 0.25              |
| R600        | -0.5    | 152.0     | 301.9      | 1.8                                      | 9.1                                      | 405                           | 0.25              |
| R600a       | -11.8   | 134.7     | 36.3       | 1.8                                      | 8.4                                      | 460                           | 0.376             |
| R601        | 36.1    | 196.6     | 33.7       | 1.5                                      | 7.8                                      | 286                           | 0.22              |
| R601a       | 27.8    | 187.2     | 33.8       | 1.4                                      | 7.9                                      | 430                           | 0.28              |
| RC270       | -31.5   | 125.1     | 55.8       | 2.4                                      | 10.3                                     | 500                           | 0.17              |
| R1270       | -47.6   | 91.1      | 45.6       | 2.4                                      | 11                                       | 455                           | 0.24              |

Table 2. Hydrocarbon properties

N-pentane (R601) was chosen as the working fluid for the experimental model of HPD. It is worth noting that the Japanese company Mayekawa more than 5 years ago introduced VT for operation in the condensation temperature range of more than 150 °C. N-Pentane has also been used as the most suitable refrigerant. As a lubricating oil, PAG (polyalkylene glycol) was tested and selected for its heat resistance and for its sufficient viscosity at temperatures up to 180 °C. In tests carried out at condensation temperatures t_s = 150 ÷ 160 °C and evaporation t_o = 70 ÷ 80 °C, the Coefficient of performance COP = 3 was confirmed (“Mayekawa develops high temperature hydrocarbon steam heat pump,”).

In order to experimentally confirm the operability of the developed technical solution for mobile HPD and determine the possibility of using a standard hermetic refrigeration compressor at non-standard temperature mode and w.f., an experimental model of high-temperature HPD was first developed and created (Malafeev et al., 2019).

The schematic diagram of the HPD model is shown in Fig. 1. Installation works as follows. The source water is poured into the steam generator 3 through the inlet pipe 1. The shut-off valve 2 is closed. By heating element 5, the water is heated to saturation temperature, after which a control signal is supplied to open the electronic regulatory valve (ERV) 11 and start compressor 9 HP.

A standard thermodynamic cycle HP is realized: the w.f. is compressed by the compressor HP and is pumped into the HP condenser 4, in which it condenses due to the removal of the heat of the phase transition to the source water, after which the supercooled liquid is throttled by means of an ERV and enters the HP evaporator 8, where it boils when supplied heat condensation vapors of the distillate.

An open water cycle is a sequence of steps: the source water is heated to a saturation temperature in the steam generator 3 due to the supply of condensation heat w.f. HP, the resulting pure water vapor due to the difference in densities of
saturated and superheated steam through the steam pipe 6 enters the condenser 7, where it undergoes a phase transformation and turns into a liquid state, and the heat of condensation of water returns back to the w.f. HP through the walls of the HP evaporator 8. After the unit reaches the steady state, heating element 5 is turned off.

**Figure 1.** Schematic diagram of the model HPD: 1 – inlet pipe, 2 - shut-off valve, 3 - steam generator, 4 – HP condenser, 5 - heating element, 6 - steam pipe, 7 – water condenser, 8 – HP evaporator, 9 – compressor, 10 - outlet pipe distillate, 11 - electronic control valve (ERV), TE1, TE2, TE3, TE4, TE5 - electronic temperature sensor, PE6, PE7 - electronic pressure sensor, EI8 - digital multimeter

When creating the first experimental installation, for the simplicity of production and minimizing the cost, the heat-exchangers HPD made in the laboratory of the department are made of twisted copper tube. ALCO EX-4 electronic expansion valve (Emerson climate technologies, USA), designed to operate in the temperature range from -50 to +100 °C, was used as a throttle.

In the experimental TND, the Atlant SK-140-N5-02 hermetic piston refrigeration refrigerator KM (Atlant CJSC, Belarus) was installed with a cylinder volume of 6.08 cm³, designed to operate on R12 freon (previously, the authors of the work (Naberezhnykh and Demenev, 2013) described successful tests of the refrigerating compressor for household appliances at a constant temperature of the motor windings of the order of 110 °C). The declared maximum operating temperature for this compressor is 120 °C, which corresponds to the heat resistance
class of insulation E. Moreover, the flash point of mineral oil in most cases is not lower than 160 ... 180 °C (Babakin and Vygodin, 1998).

The possibility of using this refrigerating compressor at temperatures exceeding the certified values also depends on the compliance of the required and installed power of the electric motor. According to the calculations, the maximum power consumption of the compressor model Atlant SK-140-H5-02 during TND operation in the boiling / condensing temperature range 90/110 °C should not be more than 80 W, which is more than one and a half times lower than declared by the manufacturer (137 W).

To fix the operating parameters of the TND model, the following instruments were used: measuring device-regulator TRM-138, multimeter KMS-F1 and pressure sensors (PD100-D16-111-1.0) and temperature (Pt1000), all manufactured by OVEN LLC (Russia).

During the experiment, the TND model was brought to a steady state, as a result of which the productivity for distillate was mw = 1 l / h. Parameters of points of design and experimental thermodynamic cycles of TND are given respectively in table 3 and table 4. Comparison of specific parameters of two cycles calculated using the CoolProp database (Bell et al., 2014), given in Fig. 2 and in table 5.

| Point number | Pressure, bar | Temperature, °C | Enthalpy, kJ/kg | Entropy, kJ/(kg*K) | Specific volume, m3/kg |
|--------------|---------------|-----------------|-----------------|-------------------|-----------------------|
| 1'           | 4.70          | 100.00          | 464.67          | 1.31              | 0.0808                |
| 2'           | 7.37          | 117.34          | 491.87          | 1.333             | 0.0510                |
| 3'           | 7.37          | 110.00          | 475.09          | 1.290             | 0.0492                |
| 4'           | 7.37          | 110.00          | 190.80          | 0.548             | 0.0019                |
| 5'           | 7.37          | 100.00          | 162.95          | 0.474             | 0.0018                |
| 6'           | 4.70          | 90.00           | 162.95          | -                 | -                     |
| 7'           | 4.70          | 90.00           | 443.41          | 1.249             | 0.0774                |

Table 3. The parameters of the project cycle

| Point number | Pressure, bar | Temperature, °C | Enthalpy, kJ/kg | Entropy, kJ/(kg*K) | Specific volume, m3/kg |
|--------------|---------------|-----------------|-----------------|-------------------|-----------------------|
| 1            | 2.5           | 84.7            | 442.13          | 1.311             | 0.1538                |
| 2            | 7.5           | 125.2           | 509.50          | 1.376             | 0.0518                |
| 3            | 7.5           | 110.8           | 476.35          | 1.291             | 0.0484                |
| 4            | 7.5           | 110.8           | 193.08          | 1.291             | 0.0019                |
| 5            | 7.5           | 97.1            | 154.92          | 0.453             | 0.0018                |
| 6            | 2.5           | 65.4            | 154.92          | -                 | -                     |
| 7            | 2.5           | 65.4            | 404.06          | 1.202             | 0.1430                |

Table 4. Parameters of an experimental cycle
Figure 2. Thermodynamic cycle of TND

| Value                                    | The parameters of the project cycle | The parameters of an experimental cycle |
|------------------------------------------|-------------------------------------|----------------------------------------|
| Specific cooling capacity, $q_0$, kJ/kg  | 301.7                               | 287.2                                  |
| Specific heating capacity, $q_k$, kJ/kg  | 328.9                               | 354.8                                  |
| Pressure ratio, $\pi_k$                  | 1.6                                 | 3                                      |
| Specific isentropic work, $l_s$, kJ/kg   | 16.87                               | 41.82                                  |
| Specific indicator work, $l_i$, kJ/kg    | 27.2                                | 67.3                                   |
| Indicator compressor efficiency, $\eta_i$ (Быков et al., 1992) | 0.7                               | 0.62                                   |
| The conversion factor excluding mechanical and electrical losses in compressor, $\mu_c$ | 12.27                             | 3.95                                   |

Table 5. Specific parameters of the thermodynamic cycle TND

Steady state was significantly different from the design of the boiling temperature. It is assumed that the discrepancy occurred due to errors in estimating the area of the heat exchange surface and the non-optimal design of the devices.
Due to the TND operation in off-design mode, the consumed power of the compressor electric motor was twice as high as the certified one (280 W versus 137 W). At the same time, under such severe operating conditions, compressor continued to function. An unambiguous statement about its applicability as part of a high-temperature HP can be given only after lengthy tests.

One of the defining parameters of the operation of the compressor and the heat pump installation is the mass flow of w.f. in the system. To verify the adequacy of the experimental data, the mass flow rate was calculated in two ways.

Determination of mass flow w.f. based on the performance of the HPD:

\[
m_{w.f.} = \frac{L_w \cdot m_w}{q_c} = 0.0017705 \text{ kg/s} \quad \text{Eq (1)}
\]

where \(L_w = 2260 \text{ kJ/kg}\) – heat of water vaporization;
\(m_w = 1 \text{ l/hour}\) – distillate mass flow;
\(q_c = 354.8 \text{ kJ/kg}\) – specific heating capacity.

Determination of the mass flow rate of w.f. based on the parameters of the compressor:

\[
m_{w.f.c} = V_{w.f} \cdot \rho_{11} = 0.0017692 \text{ kg/s} \quad \text{Eq (2)}
\]

where \(V_{w.f} = V_{cyl} \cdot n_{net} \cdot \lambda = 271.17 \text{ cm}^3/\text{s}\) – volume flow w.f., \(V_{cyl} = 6.08 \text{ cm}^3\) – cylinder volume, \(n_{net} = 50 \text{ HZ}\) – motor shaft speed, \(\lambda = 0.895\) – total coefficient feed rate, calculated by the method (Bykov et al., 1992).

The discrepancy in determining the mass flow rate of the working substance was 0.07%.

For the true value of the mass flow rate w.f. accepted arithmetic mean of the above values \(m_{w.f} = 0.00177 \text{ kg/s}\), and with this in mind, the characteristics of the work of the HPD in absolute values are calculated (table 6).

| Value                              | Design specifications | Experimental specifications |
|------------------------------------|-----------------------|----------------------------|
| Cooling capacity, \(Q_0\), W       | 534                   | 508                        |
| Heating capacity, \(Q_c\), W       | 582                   | 627                        |
| Power consumption, W, W            | 76.5                  | 279.5                      |
| \(\eta_{compressor}\) efficiency  | 0.63                  | 0.43                       |
| Coefficient of performance, \(\mu\)| 7.6                   | 2.43                       |

*Table 6. Characteristics of the experimental model of TND*
CONCLUSION

Evaluation of working substances for use in high-temperature heat pumps shows that one of the most promising working substances for low-tonnage systems is n-pentane.

The experiments carried out on the model of a mobile high-temperature heat pump distiller on natural working substance n-pentane based on a hermetic piston refrigeration compressor confirmed the efficiency of the proposed technical solution.

The energy efficiency of the experimental TND was significantly lower than expected, which is explained by the operation in a mode different from the calculated due to not optimal selection of components. The value of the real coefficient of performance of the heat pump was only 2.43 instead of the expected value of 7.6.

Currently, work is underway to manufacture a new prototype of the TND. The obtained experience gives grounds to count on significantly higher specific energy efficiency indicators of the newly created installation.

ACKNOWLEDGEMENTS

The reported study was funded by RFBR, project number 19-38-90308

NOMENCLATURE

$q_0$ \hspace{1cm} \text{Specific cooling capacity (kJ/kg)}

$q_c$ \hspace{1cm} \text{Specific heating capacity (kJ/kg)}

$p_c$ \hspace{1cm} \text{Pressure ratio}

$l_s$ \hspace{1cm} \text{Specific isentropic compression work (kJ/kg)}

$l_i$ \hspace{1cm} \text{Specific indicator compression work (kJ/kg)}

$h_i$ \hspace{1cm} \text{Indicator compressor efficiency}

$l_{ic}$ \hspace{1cm} \text{The conversion factor excluding mechanical and electrical losses in compressor,}

$m_{w.f.}$ \hspace{1cm} \text{Mass flow working fluid (kg/s)}

$m_{w}$ \hspace{1cm} \text{Distillate mass flow (ks/s)}

$m_{w.f.c.}$ \hspace{1cm} \text{Mass flow rate of the working substance based on the parameters of the compressor (ks/s)}

$V_{w.f.}$ \hspace{1cm} \text{Volumetric flow rate working fluid (m$^3$/s)}

$V_{cyl.}$ \hspace{1cm} \text{Compressor cylinder capacity (m$^3$)}

$n_{net}$ \hspace{1cm} \text{Motor shaft speed (Hz)}

$\lambda$ \hspace{1cm} \text{Total coefficient feed rate of compressor}

$m_{wef}$ \hspace{1cm} \text{Mass flow rate of refrigerant in cycle (kg/s)}

$Q_0$ \hspace{1cm} \text{Cooling capacity (W)}

$Q_c$ \hspace{1cm} \text{Heating capacity (W)}

$W$ \hspace{1cm} \text{Power consumption (Br)}

$W_{\text{HPD}}$ \hspace{1cm} \text{Heat pump distiller}

$W_{\text{HP}}$ \hspace{1cm} \text{Heat pump}

$W_{\text{ERV}}$ \hspace{1cm} \text{Electronic regulatory valve}

$w.f.$ \hspace{1cm} \text{Working fluid}
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