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-pentan.

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 refers 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, isoentropic compressor efficiency - 0.63

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

Fluid name	Coefficient of performance	Pressure ratio	Pressure difference, bar	Discharge temp, °C	Thermodynamic efficiency
R152a	9.90	1.47	13.65	128.06	0.52
R13I1	10.31	1.44	9.58	128.33	0.54

CYCLOPRO	10.46	1.42	12.84	126.46	0.55
R717	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
PROPYNE	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
IBUTENE	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_{crit}, °C	P_{crit}, 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 tк = 150 ÷ 160 °C and evaporation to = 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.

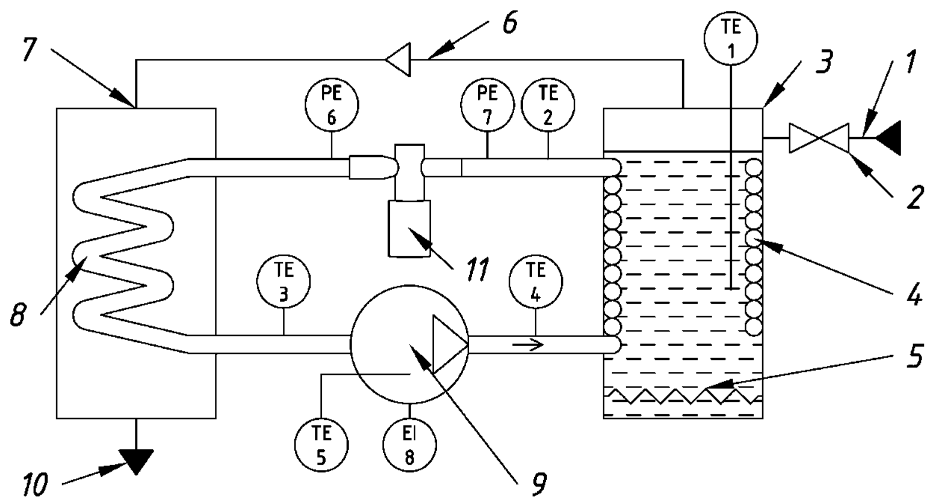

Figure 1. Schematic diagram of the model HPD: 1 – inlet pipe, 2 - shut-off valve, 3 - steam generator, 4 – HP condenser, 5 - heating element, 6 - steam pipe, 7 – water condenser, 8 – HP evaporator, 9 – compressor, 10 - outlet pipe distillate, 11 - electronic control valve (ERV), TE1, TE2, TE3, TE4, TE5 - 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-DI6-111-1.0) 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

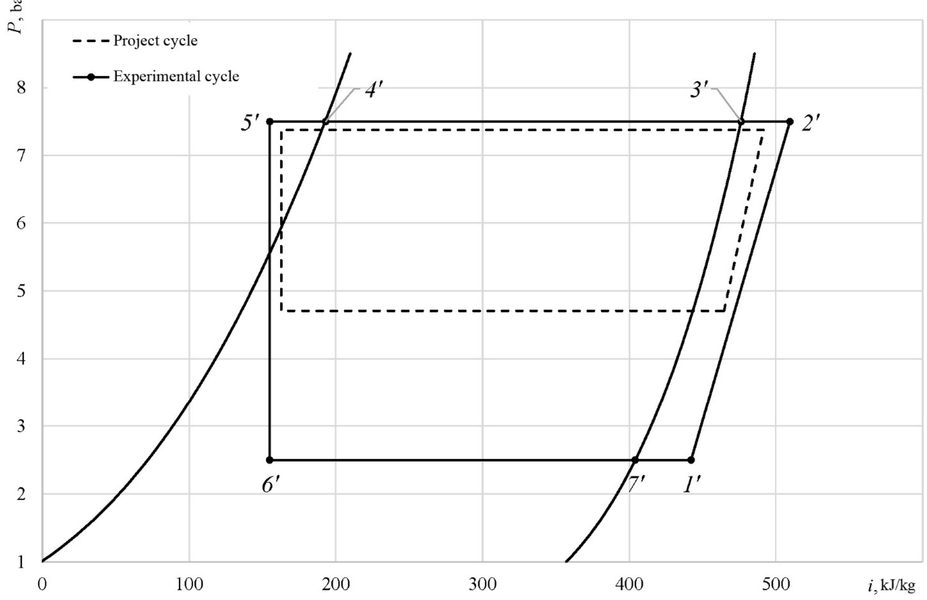

Figure 2. Thermodynamic cycle of TND

Value	The parameters of the project cycle	The parameters of a 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, π_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, η_i (Быков et al., 1992)	0.7	0.62
The conversion factor excluding mechanical and electrical losses in compressor, μ_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:

$$m_{w.f.} = \frac{L_w * m_w}{q_c} = 0.0017705\ kg/s \qquad \text{Eq (1)}$$

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

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

$$m_{w.f.c.} = V_{w.f.} * \rho_{11} = 0.0017692\ \ kg/s \qquad \text{Eq (2)}$$

where $V_{w.f.} = V_{cyl} * n_{net.} * \lambda = 271.17\ \ cm^3/s$ – volume flow w.f., $V_{cyl} = 6.08\ cm^3$ – cylinder volume, $n_{net.} = 50\ HZ$ – 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 $m_{w.f.} = 0.00177\ \ kg/s$, 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
Compressor efficiency, $\eta_{compressor}$	0.63	0.43
Coefficient of performance, μ	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 ccoefficient 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

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NOMENCLATURE

q_0	Specific cooling capacity (kJ/kg)	m_w	Distillate mass flow (ks/s)
q_c	Specific heating capacity (kJ/kg)	$m_{w.f.c.}$	Mass flow rate of the working substance based on the parameters of the compressor (ks/s)
$p_к$	Pressure ratio	$V_{w.f..}$	Volumetric flow rate working fluid(m^3/s)
l_s	Specific isentropic compression work (kJ/kg)	$V_{cyl.}$	Compressor cylinder capacity (m^3)
l_i	Specific indicator compression work (kJ/kg)	n_{net}	Motor shaft speed (Hz)
h_i	Indicator compressor efficiency	λ	Total coefficient feed rate of compressor
μ_c	The conversion factor excluding mechanical and electrical losses in compressor,	$m_{w.f}$	Mass flow rate of refrigerant in cycle (kg/s)
$m_{w.f.}$	Mass flow working fluid (kg/s)	Q_0	Cooling capacity (W)
L_w	Heat of water vaporization (kJ/kg)	$Q_к$	Heating capacity (W)
t_s	Normal boiling point (°C)	W	Power consumption (Вт)
HPD	Heat pump distiller	ERV	Electronic regulatory valve
HP	Heat pump	w.f.	Working fluid

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