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Efficiency of Operation of Air Heat Pumps with Evaporators of Various Designs

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Abstract

Resume. *The development of the heat pump market ensures the growth of energy prices, the fight against global warming and the stimulation of the growth rates of countries in transition to "clean energy sources". the most environmentally friendly and energy efficient, taking into account the cost of their installation and the impact of air source heat pumps compared to geothermal ones. However, their use in countries with cold climates is limited by the formation of frost on the evaporator unit heat exchanger, which significantly increases their heating capacity and conversion efficiency.*

Purpose of research. *The purpose of this work is to compare the efficiency of heat pumps with an evaporative circuit heat exchanger manufactured by MITSUI (Japan) and a heat exchanger with a MOVEBIT defroster manufactured by ALTEC (Russia).*

Methods. *The research methodology is based on large laboratory tests of an evaporator with an oscillatory circuit of the MOVEBIT system (ALTEK - Russia) and an industrial heat exchanger MITSUI (Japan), a graphical analysis of the operation of a heat pump with common evaporators, and a theoretical substantiation of the transformation ratio. To compare the main standardized performance indicators, the transformation coefficient (COP – coefficient of performance) is used, showing the ratio of received energy to applied work.*

Results. *The analysis of the results of the heat pump operation showed that while maintaining the humidity at a constant level of φ ≈ 65% and lowering the temperature from +10°C to +5°C, a decrease in the heat output of the unit is observed, regardless of the type of MOVEBIT and MITSUI heat exchanger, which is explained by the formation of condensate. With a further decrease in temperature to zero and below, ice forms on the evaporator of the heat exchanger. Taking into account the cost of removing ice on the MITSUI heat exchanger, the transformation ratio of the heat pump unit (HPU) is 2.08. The removal of ice by the MOVEBIT system practically does not require additional costs and the transformation ratio of the installation is 4.45, which is similar to the operation of an air source heat pump at positive temperatures.*

Conclusion. *Our studies show that the best prospect for de-icing is the application of mechanical oscillations with the use of magneto-constrictive transducers. The COP of the air source heat pump with the MOVEBIT system exceeds the COP of the heat pump with the MITSUI standard evaporator by 2 times at a temperature of 0°C and below.*

Keywords: *Heat pump; evaporator; ice; transformation coefficient; energy consumption; heating capacity.*

Conflict of interest. *The authors declare the absence of obvious and potential conflicts of interest related to the publication of this article.*

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Эффективность работы воздушных тепловых насосов с испарителями различных конструкций

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Резюме

Развитие рынка тепловых насосов обусловлено ростом цен на энергоресурсы, борьбой с глобальным потеплением и стимулированием правительствами развитых стран перехода на «чистые источники энергии». Наиболее экологически безопасными и энергоэффективными с учетом затрат на их прокладку и эксплуатацию являются воздушные тепловые насосы по сравнению с геотермальными. Однако их применение в странах с холодным климатом ограничено образованием наледи на теплообменнике испарительного блока, что значительно снижает их теплопроизводительность и коэффициент преобразования.

***Цель исследования.** Целью настоящей работы является сравнение эффективности работы тепловых насосов с теплообменником испарительного контура производства MITSUI (Япония) и теплообменника с антиобледенителем MOVEBIT производства АЛТЕК (Россия).*

***Методы.** Методология исследования базируется на проведении лабораторных испытаний испарителя с колебательным контуром системы MOVEBIT (АЛТЭК – Россия) и теплообменника промышленного производства MITSUI (Япония), графоаналитического анализа работы теплового насоса с различными испарителями, теоретического обоснования полученных значений коэффициента трансформации. Для сравнения основных стандартизированных эксплуатационных показателей используется коэффициент трансформации (СОР), показывающий отношение полученной энергии к затраченной работе.*

***Результаты.** Анализ результатов работы теплового насоса показал, что при сохранении влажности на постоянном уровне $\varphi \approx 65\%$ и понижении температуры с +10°С до -5°С наблюдается снижение теплопроизводительности установки, независимо от вида теплообменника MOVEBIT и MITSUI. Однако при использовании теплообменника MOVEBIT теплопроизводительность установки на 10,1% выше, чем с теплообменником MITSUI. Это объясняется наличием на испарителе теплообменника MITSUI наледи, уменьшающей его теплопередающие способности.*

***Заключение.** Проведенные нами исследования показали, что наиболее перспективным способом удаления наледи является применение механических колебаний с помощью магнитострикционных излучателей. СОР воздушного теплового насоса с системой MOVEBIT превышает СОР теплового насоса со стандартным испарителем MITSUI в 2 раза при температуре 0°С и ниже.*

***Ключевые слова:** тепловой насос; испаритель; наледь; коэффициент трансформации; энергопотребление; теплопроизводительность.*

***Конфликт интересов:** Авторы декларируют отсутствие явных и потенциальных конфликтов интересов, связанных с публикацией настоящей статьи.*

***Финансирование.** Данная работа проводилась в рамках реализации программы поддержки вузов "Приоритет 2030" на базе Белгородского государственного технологического университета им. В.Г. Шухова с использованием оборудования Центра высоких технологий при БГТУ им. В.Г. Шухова.*

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Introduction

The development of the heat pump market is resulted from the rising of energy prices, the fight against global warming and stimulating the transition to "clean energy sources" by the governments of developed countries. The Russian government, like other advanced countries of the world, has focused on the influx of new promising projects [1]. The general global trend of transition of heat supply of engineering systems to the use of renewable energy sources should be noted [2-4], as well as secondary thermal emissions using heat pumps [5-7]. All types of air source heat pump evaporators carry out heat exchange between air and the refrigerant passing through the vapor-liquid phase. The efficiency of heat transfer is determined by factors such as the speed of the refrigerant, its quantity, temperature difference, the material of the heat exchanger, the shape, the presence of various deposits in the form of dust, ice. The existence of various types of evaporators is motivated by the design requirements and technical conditions for a heat pump [8-12]. The most environmentally friendly and energy efficient, taking into account the cost of their installation and operation, are air source heat pumps compared to geothermal ones [13]. However, their use in countries with cold climates is limited by the formation of frost on the heat exchanger of the evaporator unit, which significantly reduces their heating capacity and conversion efficiency. Our studies have shown that the most promising way to remove frost is the use of mechanical oscillations with the use of magneto-constrictive transducers [14].

The purpose of this work is to compare the efficiency of heat pumps with an evaporative circuit heat exchanger manufactured by MITSUI (Japan) and a heat exchanger with a MOVEBIT defroster manufactured by ALTEC (Russia) [15]. To compare the main standardized performance indicators, COP is used – the coefficient of performance, which shows the ratio of the received energy Q_r to the energy loss for its transformation Q_l, which includes the costs of the compressor, fans and consumption of various electronic equipment responsible for the operation of the heat pump.

Materials and methods

Due to the of lack of mass-produced air source heat pumps capable of operating with zero overheating on the market , a number of changes were made to the equipment used in the tests: an after-boil receiver tank was additionally installed on the suction line, and the throttle device was equipped with a stepper motor with electronic control of the refrigerant supply, for visual control, sight glasses were installed in the liquid and gas lines. In laboratory tests, a $9m^2$ refrigerator, an ultrasonic air humidifier in the refrigerator, an air heat pump with electronic throttling control, a heat exchanger manufactured by ALTEK (Russia), a heat exchanger manufactured by MITSUI (Japan), and measuring equipment TESTO (Germany) were used. As part of laboratory tests, it was decided to take into account the cost of electricity for the operation of the compressor, respectively Q_l, watt (W):

$$Q_l = L + Q_f - Q_v, \quad (1)$$

notation Q_f – the energy costs for frost control, W;

L – the indicated output, W;

Q_v – the expenditure of energy consumed by EEV (electronic expansion valve), fans of the outdoor and indoor units, Wt.

The evaporator cooling capacity Q_u, which is designed for laboratory testing purposes, is determined by equation, W:

$$Q_u = Q_K - Q_l. \quad (2)$$

notation Q_k – the total refrigerant heat rejection effect, W;

Q_l – the heating capacity of the compressor, *W*.

The selection of the evaporator area F_u is carried out according to the formula [16], *W*:

$$F_u = \frac{Q_u}{K_u \cdot \Delta t_m}. \quad (3)$$

The mean temperature difference in the evaporator Δt_m,°C:

$$\Delta t_m = \frac{\Delta t_h - \Delta t_l}{ln(\Delta t_h / \Delta t_l)} \quad (4)$$

notation Δt_h и Δt_l – the highest or the lowest differential temperature between the heat-transfers at the inlet and outlet of the heat exchanger respectively, °C;

$$\Delta t_h = t_1 - t_2;$$

$$\Delta t_l = t_3 - t_2;$$

t_1 – the air inlet temperature of the evaporator, °C;

t_2 – the boiling point of the refrigerant, °C;

t_3 – the air downstream temperature of the evaporator, °C.

A new generation evaporator with an oscillatory circuit of the MOVEBIT system has passed a series of tests and has shown its promise. The surface of the heat exchanger is 6 m^2. Subtracting rounding angles and other elements that do not affect the heat transfer process, we take the surface F_t equal to 5.9 m^2.

The capacity of the (axial) fan is 700 m^3/h.

We find the heat output by the formula, *W*:

$$Q_m = p \cdot S \cdot V \cdot c \cdot (t_2 - t_1), \quad (5)$$

notation: p – the air density – 1.2041 kg/m^3 at t_1 = 20°C;

S – the area of the cross-section of the airway – 0.0177 m^2;

Q_m – the air flow rate, m/s [annex 15 table 1];

t_1 – the air inlet temperature of the condenser, °C [annex 11 table 1];

t_2 – the air downstream temperature of the condenser, °C [annex 12 table 1];

c – the heat capacity volume of the air – 1050 J/kg·K.

Calculation of the cost of electricity for defrosting the evaporator is based on the calculation of condensate on the evaporator and the cost of energy for its removal by means of electrical heating. The calculation is carried out according to the equation:

$$Q_r = p \cdot V \cdot \lambda \cdot m_{\kappa} + p \cdot V \cdot c \cdot m_{\kappa} \cdot \Delta t, \quad (6)$$

notation: λ– the specific melting heat of ice is taken – 340 KJ/kg;

$m_{к}$ – the specific ice mass (condensate), kg;

c – the specific heat (for ice 2.1 KJ/kg °C);

Δt – the difference between the initial ice temperature and the melting point °C;

p – the air density, kg/m^3;

V – the air mass passing through the evaporator, m^3.

The amount of condensate crystallizing on the surface of the evaporator during the passage of an air flow with a volume (V = 1 m^3) through the evaporator in the form of ice and frost is calculated by the formula:

$$m_l = \frac{d_n - d_i}{10^3}, \quad (7)$$

notation: d_n – the moisture content of the external air, g/kg;

d_i – the moisture content of the air at the boiling point of the refrigerant in the evaporator (the figure is taken according to the i-d diagram), g/kg.

Results and discussion

Tests of the heat pump unit (HPU) were carried out at outdoor temperatures of -5, 0, 5.10 °C. The results of testing HPU with an evaporator manufactured by MITSUI and MOVEBIT are presented in the table 1.

Based on the test results presented in the table, using equation (1), we can calculate the energy costs for the operation of the compressor:

$$Q_l = L + Q_f - Q_v = 700+168+10=878, W. (8)$$

Using formula (2), we determine the cooling capacity of the evaporator Qi intended for laboratory testing:

$$Q_u = Q_{\kappa} - Q_l = 3000 - 700 = 2300, \text{Wt}. \quad (9)$$

The calculated heat transfer surface, for test conditions, is calculated according to equation (4).

The heat transfer coefficient K is calculated according to the formulas [17, 18], but in this case it is determined with a high degree of accuracy from the tabular values K=23.8 W/m^2·K.

An analysis of the results of the heat pump operation showed that while maintaining the humidity at a constant level $\varphi \approx$ 65% and lowering the temperature from +10°C to +5°C, a decrease in the heat output of the unit is observed, regardless of the type of heat exchanger, MOVEBIT and MITSUI. However, when using the MOVEBIT heat exchanger, the heating capacity of the installation is 10.1% higher than with the MITSUI heat exchanger. This is due to the presence of frost on the evaporator of the MITSUI heat exchanger, which reduces its heat transfer capabilities.

Table 1. Test results of HPU at temperatures before the evaporator unit -5, 0, 5, 10°C

№	Measurable values	Units of measurement	The evaporator manufactured by MITSUI				The evaporator manufactured by MOVEBIT			
			Temperature							
			10 °C	5 °C	0°C	-5°C	10°C	5°C	0°C	-5°C
1	The low pressure	bar	6.56	5.52	3.81	3.35	5.40	4.07	3.55	3.02
2	The temperature of the refrigerant gas	°C	-1.6	-6.1	-14.9	-17.6	-6.7	-13.4	-16.4	-19.7
3	The temperature of the gas pipeline	°C	6.4	1.8	-8.1	-10.7	0.6	-8.3	-7.0	-9.1
4	The overheating	K	8.0	7.9	6.8	6.9	7.3	5.2	9.4	10.7
5	The air inlet temperature of the evaporator	°C	10.0	5.0	0.0	-5.0	10.0	5.0	0.0	-5.0
6	The air downstream temperature of the evaporator	°C	6.8	1.7	-4.0	-8.6	6.7	1.3	-3.4	-8.0
7	The high pressure	bar	22.23	21.44	20.07	20.20	21.80	20.90	20.20	19.70
8	The gas temperature	°C	38.3	36.9	34.3	34.5	37.5	35.9	34.6	33.6
9	The temperature of the liquid pipeline	°C	28.2	26.9	25.9	28.0	29.3	26.7	26.1	25.2
10	The over cooling	K	10.1	9.9	8.5	6.5	8.2	9.2	8.5	8.3
11	The air inlet temperature of the condenser	°C	21.6	21.4	21.4	22.6	21.5	21.7	21.4	21.0
12	The air downstream temperature of the condenser	°C	34.7	32.7	33.0	32.5	35.1	35.2	33.4	32.4
13	The electricity costs to defrost the evaporator actual /calculated	W/h	0 22	158 196	529 492	402 350	0	12	12	12
14	The humidity inside the chamber	%	64.3	65.5	66.4	63.3	67	64.1	63.2	65.9
15	The air flow rate downstream of the capacitor	m/s	10.50	10.71	10.11	10.46	10.39	10.7	10.39	9.94
16	The active power (the wattage)	W	637	692	643	670	643	653	602	557
17	The heating capacity	W	2927	2602	2502	2204	3012	3003	2721	2417
18	COP with no de-icing costs / taking into account the costs of de-icing	COP	4.60 4.60	4.05 3.00	3.58 2.10	3.29 2.05	4.70 4.70	4.60 4.50	4.52 4.40	4.34 4.20

The second factor in reducing productivity is lowering the temperature of the low-grade heat source. The calculations assume a decrease in the transformation ratio by 1% with a decrease in the temperature of the source of low-grade heat (air) by 1°C. Accordingly, a decrease in COP is a natural test result.

When measuring the parameters of the heat pump, the costs of electricity for the operation of the defrosting system were highlighted in a separate column, item 13 of the table. Defrosting requires the maximum amount of energy at t_1 = 0°C and is Q = 529 W, which exceeds the calculated amount of electricity by 7%. This is due to an increase in energy consumption for heating not only ice, but also the heating of structural and heat transfer elements that make up the evaporator, as well as the air surrounding the evaporator.

A decrease in the ambient temperature does not lead to an increase in the energy consumption for defrosting the evaporator. As the experiments showed, with a decrease in the temperature in the chamber by 5°C, the energy consumption decreased by 24%, exceeding the calculated values by 4%. This is an acceptable indicator that does not distort the overall dynamics of energy consumption for defrosting a standard evaporator using the electric heating method [19, 20].

When the outdoor temperature rises above 0°C, the energy consumption for defrosting is reduced. At a temperature of +10°C, its theoretical consumption is 22 W. Practical tests have shown that the accumulation of condensate in the form of frost occurs only in places that do not fall under the air flows created by the fan (on the evaporator coils and in non-insulated sections of the cold pipes).

In the case of the tested heat exchanger equipped with the oscillatory circuit of the MOVEBIT system, the energy consumption was 12 W, regardless of the ambient temperature. A graphical representation of the efficiency of an air source heat pump against ambient temperature using a standard MITSUI evaporator and a MOVEBIT oscillating evaporator is shown in (fig. 1).

To compare the efficiency of heat exchangers (fig. 1) shows the operation of an air source heat pump with a MITSUBISHI heat exchanger, which, according to some estimates, is considered to be the best on the HTH market [20-22]. The graph shows a similar transformation ratio with a laboratory heat pump, a small difference in values is due to the peculiarity of the ZUBADAN technology. This is due to more stable operation by controlling the compressor discharge temperature through injection and increasing the refrigerant superheat in the POWER RECEIVER.

A sharp decrease in the performance of an air source heat pump at low temperatures is explained by the operation of the defrosting system. This process clearly demonstrates a graph of changes in the transformation ratio over time (fig. 2), which does not take into account the cost of electricity for defrosting.

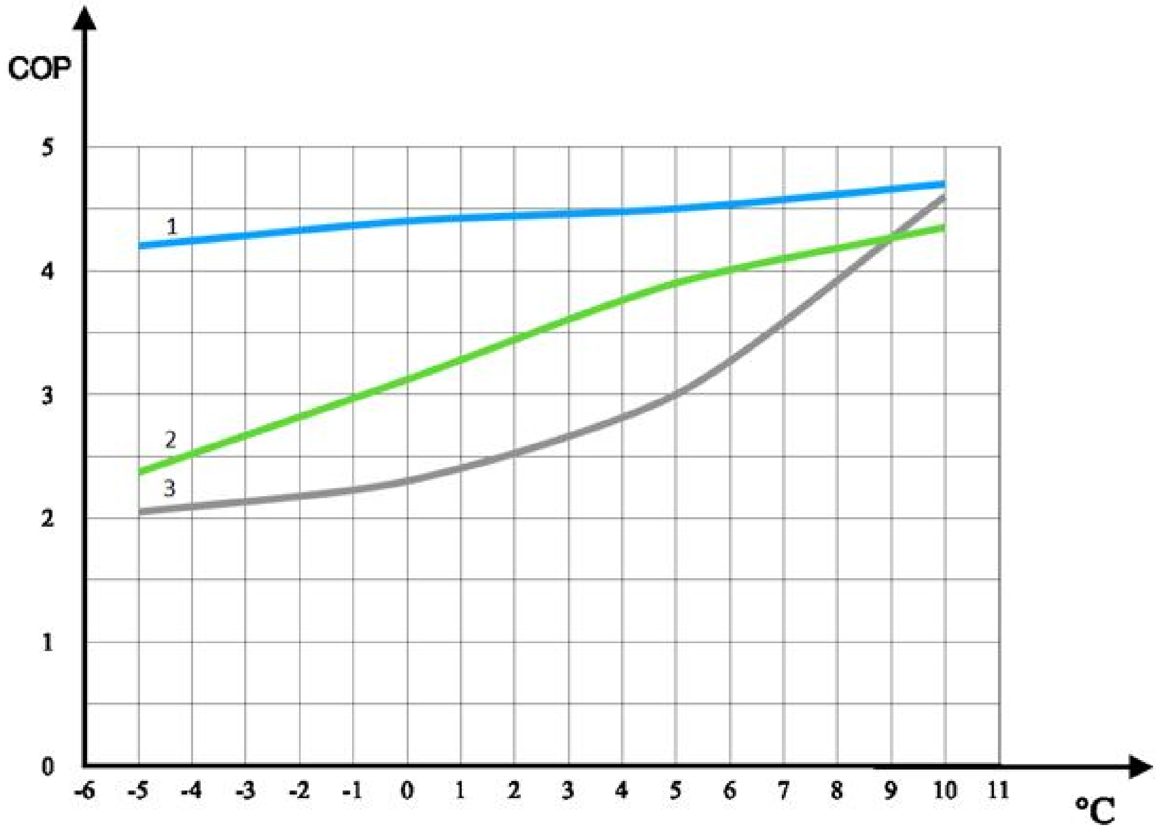

Fig.1. Dependence of the HPU transformation coefficient on the outside air temperature when using heat exchangers of various manufacturers in the evaporative circuit: **1** – MOVEBIT, **2** – MITSUBISHI, **3** – MITSUI

Line 1 shows the change in the COP of the heat pump with the MOVEBIT evaporator at 0°C for 120 minutes. The oscillation amplitude has a very small interval along the y-axis, and the resulting direct COP has higher values than curve 2 of the MITSUI evaporator heat pump. Fluctuations have large COP values from 0 to 4.6, therefore, the resulting MITSUI 3 straight line has lower values than that of the MOVEBIT heat exchanger (fig. 2).

In more detail, the process of ASHP operation and its performance is explained in (fig. 3). The segment bounded by points A-B has the maximum COP, because the evaporator has the maximum heat transfer capacity. After passing point "B", the process of ice formation begins with a faster intensity (segment B-C). At a certain point, the superheat begins to decrease, approaching zero degrees, which requires urgent cleaning of the evaporator or turning off the compressor. At point "C" the compressor is switched off and the evaporator goes into defrosting mode "C-C/". Points "C/ -d/" – evaporator defrosting mode time, "d/ -d" – end of defrosting mode. The compressor is switched on at point "d". In the "d-A" section, the compressor begins to pump refrigerant through the system. At point "A/" the heat pump reaches its maximum heat output.

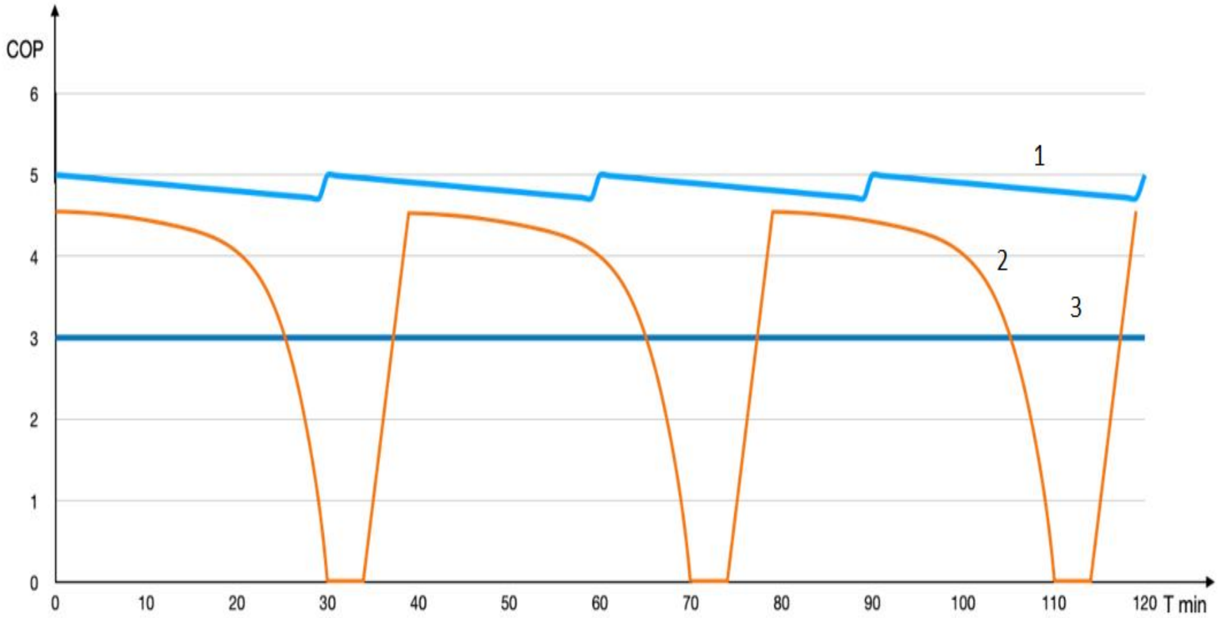

Fig. 2. Change in the transformation coefficient of TNU (COP) at a temperature of 0 ° C for 120 min. with evaporators: **1** – MOVEBIT, **2** – MITSUI, **3** – resulting direct efficiency of the MITSUI heat exchanger

The transformation ratio of a heat pump is limited by the area S_o:

$$S_o = S_t - S_p , \tag{10}$$

notations: S_t – the area, which is bounded by the points OABC и dA$^/$ O$^/$;

S_p – the area, which is bounded by the points Cc$^/$d$^/$d;

S_o – the area, which is bounded by the points OEFO$^/$.

The average transformation ratio of the heat pump will correspond to a straight line passing through the points E-F (fig. 3a).

In (fig. 3b), the operation of a heat pump with a MOVEBIT evaporator is detected by a graphical method. The curve bounded by points A-B shows the operation of the heat pump with the presence of dense crystallized condensate in the form of frost 0.1-0.4 mm. Provided with such thickness of thermal insulation properties, there is no observation of the heat transfer of the evaporator and the COP of the entire heat pump. In the "B" setting, the oscillatory circuit is turned on, the exposure time is 0.2-1.5 seconds (section "B-A$^/$"). As a result of resonance, the evaporator is cleaned of frost and ice. Stopping the compressor is not required. The resulting direct COP for the MOVEBIT evaporator, limited by the EF points, is significantly higher than in option "a". The energy consumption for the operation of the oscillatory circuit is 12–14 W/h, therefore, any change in the energy consumption required for the operation of the heat pump installation does not depend [23].

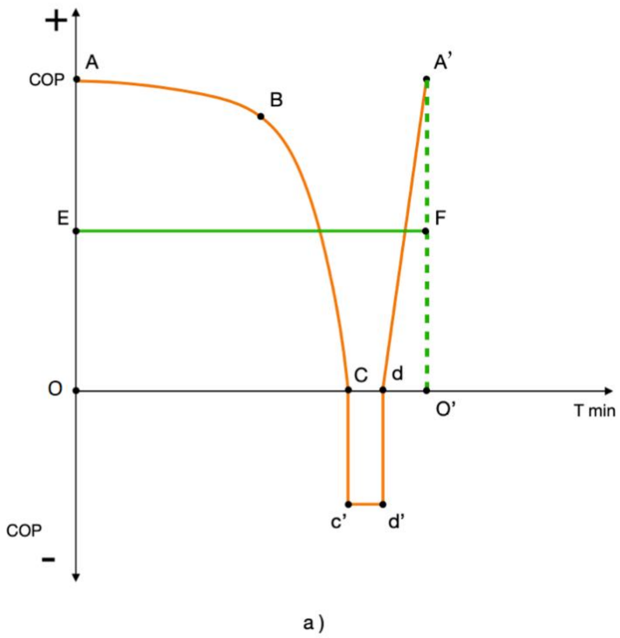

Fig. 3. The effect of ice on the efficiency of HPU: **a** – with the MITSUI evaporator, **б** – with the MOVEBIT evaporator

Conclusion

The conducted studies have shown the high efficiency of the MOVEBIT oscillating circuit evaporator. The coefficient of transformation (COP) of an air source heat pump with the MOVEBIT system exceeds the COP of a heat pump with a standard MITSUI evaporator by 2 times at a temperature of 0°C and below. Graph-analytical and theoretical calculations confirmed the increase in the transformation ratio and productivity of the heat pump by a factor of two in the temperature range from 0°C to -5°C. The integration of outdoor units into existing air source heat pumps cuts energy consumption in half, which opens up great prospects for using a heat pump installation over a wide temperature range. Measures for the transition of standard evaporators to MOVEBIT evaporators do not require significant changes in the design of the air source heat pump.

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