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Efficiency of heat pump systems of air conditioning for removing excessive moisture

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Abstract The conditions of efficient use of heat pumps in air conditioning systems are considered in order to ensure the established temperature and relative humidity of air in premises with the removal of excess moisture in the warm or hot periods of the year. For this purpose, a thermodynamic analysis of heat pump air conditioning schemes with exhaust air recirculation through a condenser and through a heat pump evaporator has been carried out. To determine the potential capabilities of such schemes to maintain comfortable conditions in the production room, a numerical analysis of their operating parameters, depending on the temperature and relative humidity of external atmospheric air, was performed. It has been shown that recirculation of exhaust air through the heat pump evaporator allows to maintain the given conditions in the room in a wider range of parameters of external atmospheric air. In addition, it has been shown that such a scheme requires less specific energy consumption for the operation of heat pump, which means that it is more efficient.

Keywords: Heat pump; Air conditioning systems; Comfortable conditions; Production facilities with excess moisture

Nomenclature

d	–	moisture content of air, $\mathrm{kg}_{moisture}/\mathrm{kg}_{dry\,air}$
G	–	mass flow, kg/s

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h	–	specific enthalpy of the air, kJ/kg
K_0	–	relative flow of external air
L	–	power of the heat pump compressor, kW
Q	–	thermal power, kW
Q_{cool}	–	cooling capacity, kW
t	–	temperature, °C

Greek symbols

ε_{HP}	–	efficiency of the operation of the heat pump
εsh	–	cooling coefficient of the whole scheme
φ	–	relative humidity

Subscripts

0	–	external atmospheric air
1	–	entrance to the room
2	–	exit of the room
ev	–	outlet of the heat pump evaporator
c	–	outlet of the heat pump condenser
mix	–	the air flow mixture parameters
rec	–	the recirculation flow
ger	–	the total flow of air through the room
m	–	moisture removed to drainage
r	–	room

Abbreviations

C_{HP}	–	heat pump condenser
E_{HP}	–	heat pump evaporator
HP	–	heat pump
HPS	–	heat pump system
MC	–	mixing chamber
OC	–	object of air conditioning

1 Introduction

At present, heat pump technology is becoming increasingly widespread. The most common systems are for ventilation, heating and air conditioning, maintaining a certain (comfortable) indoor temperature, and are simple systems that use standard heat pumps that work in the mode of heating or cooling air [1–3]. There are quite a number of works devoted to the study of the performance of such simple heat pump systems [4,5].

More complex schemes have been developed to improve energy efficiency of such systems [6–8]. In this case, the schemes are supplemented by additional elements, for example, heat exchangers for the use of heat or cold exhaust air. However, in a number of premises, both in public and industry, it is necessary to maintain a certain level of not only temperature, but also

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relative humidity, which requires the development of effective systems for drainage or humidification of air. Indeed, uncontrolled change in humidity affects not only the health and well-being of people, but also the conditions of storage of raw materials and products, and may be the reason for damage to many technological processes. Therefore, it is very important to maintain the humidity in such areas at an optimal level. There are different ways to solve this problem. One of them is the use of additional adsorption dehumidifiers or the development of combined heat pumps using elements for absorbing moisture [9]. Such systems, however require additional energy consumption.

Alternatively, heat pump systems are used to maintain a temperature and relative humidity at a given level. It is known that increasing energy efficiency of such systems is possible due to the recirculation of part of the exhaust air. Recirculation of exhaust air allows us to substantially reduce energy costs for the preparation of the tidal air. However, since recirculation affects the air quality in the room, its value should be agreed, on the one hand, with the air exchange rate in accordance with the recommendations for a particular type of room, and on the other hand, with the parameters of the outside air. In order to determine the required coefficient of recirculation in these conditions, it is necessary to conduct a thermodynamic analysis of the corresponding heat pump systems. However, such works are practically absent from known literature sources to date. Existing methods of calculating ventilation systems [10], in which the coefficient of recirculation is determined for the calculated parameters of the external atmospheric air and is taken constant, regardless of weather conditions, do not provide a given temperature-humidity regime in the room in a wide range of changes in atmospheric air parameters.

In this connection, in the framework of this article, the operating modes of the heat pump air conditioning systems were studied in conditions of the variable recirculation coefficient, which depends on the parameters of the external atmospheric air. For this purpose, two heat pump schemes were analyzed: with the recirculation of exhaust air through the condenser and through the evaporator of the heat pump. A comparative thermodynamic analysis of the work of such schemes was carried out, on the basis of which their necessary working parameters were determined for maintaining the temperature and humidity in the room at a given level, as well as conclusions were made regarding the efficiency of the schemes.

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2 Purpose of the work

The purpose of this article is to study the working parameters of the heat pump air conditioning systems, which ensure the maintenance of specified temperature and humidity conditions in the middle of the room when the temperature and humidity of the external atmospheric air undergo changes. The aim of the work is also to study the effectiveness of these schemes, which is characterized by the cold productivity of the corresponding scheme per unit of the work of the compressor of the heat pump, the determination of a more efficient scheme and the maximum parameters of the outside air, within which the system can provide specified conditions in the middle of the room.

3 Description of air conditioning heat pipe systems with recirculation of exhaust air through the condenser and evaporator of the heat pump

Figure 1a depicts heat pipe systems (HPS) air conditioning with exhaust air recirculation through the HP condenser and mixing its part in the mixing chamber (MC) with fresh air dried in the evaporator HP. The other part of the airflow after the condenser is released into the environment. After mixing the airflow in the MC, it is fed to the entrance of the production premises.

Figure 1b shows heat pipe systems air conditioning with recirculation of exhaust air through heat pump a evaporator and mixing it in a mixing chamber with fresh air. After mixing, the air flow is sent to a condenser, where it is heated to a predetermined temperature and from the intermediate point of the condenser a portion of the flow in quantity G_{ger} is fed to the entrance of the room. The other part of the air is mixed with fresh air in the same amount and after the condenser is released into the environment.

The working processes of heat and humidity preparation of air in the air conditioning system for the two described schemes are shown in the h-d diagram in Fig. 2.

Figure 2a illustrates the working process of changing the air condition in the air conditioning system of the object of conditioning with the recycling of exhaust air through the HP condenser. In the stationary mode of the installation operation, the flow of air with mass flow G_{ger} and parame-

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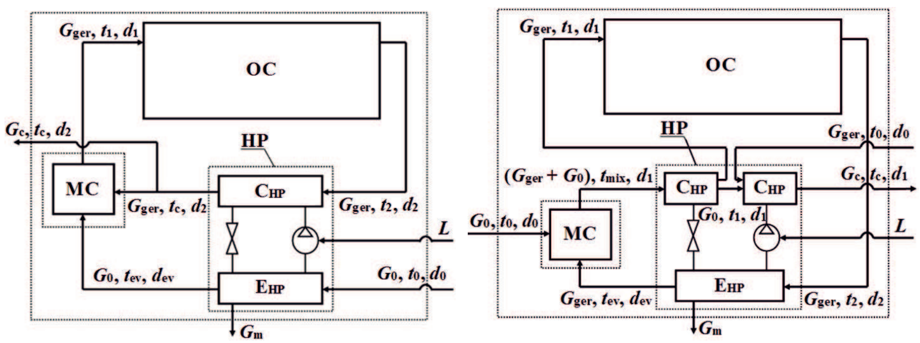
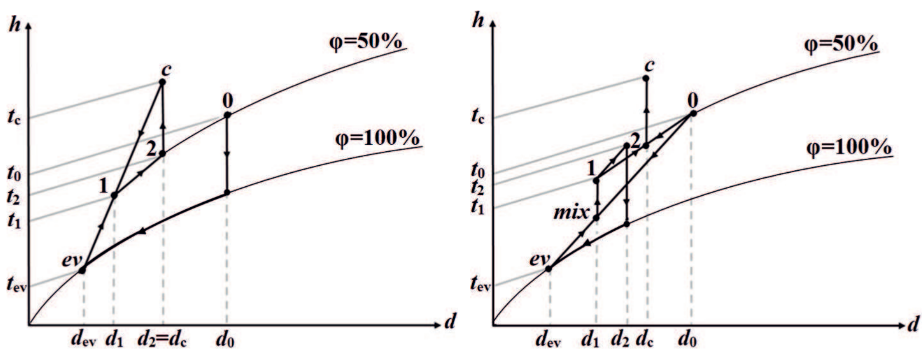


a) b)

Figure 1: HPS air conditioning systems with recirculation of exhaust air through: a) HP condenser; b) HP evaporator: OC – object of conditioning; HP – heat pump; C_{HP} – HP condenser; E_{HP} – HP evaporator; MC – mixing chamber; L – drive power of the heat pump compressor.



a) b)

Figure 2: The working process of heat and moisture preparation of air in the air conditioning system with recirculation of exhaust air through, in the *h-d* diagram: a) HP condenser, b) the HP evaporator.

ters t_1, d_1 after passing the production room with parameters t_2, d_2 enters the condenser HP. External air with parameters t_0, d_0 and mass ratio G_0 enters the evaporator of the heat pump, where the moisture content G_m is removed due to its cooling and drainage to the parameters t_{ev}, d_{ev} (point B). The part of the exhaust air heated in the condenser HP, with a mass flow rate G_{rec} and parameters t_c, d_2 enters the mixing chamber, where it mixes with the flow of chilled and drained in the evaporator atmospheric air. The obtained mixture with the given parameters enters the air condi-

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tioned room. The other part of the air with mass flow rate G_c is removed from the condenser into atmosphere.

Figure 2b depicts the working process of changing the air condition in the air-conditioning system with recirculation of exhaust air through the HP evaporator. In the stationary mode of operation of the installation, the flow of air with mass flow rate G_{ger} and parameters t_1, d_1 after passing the air conditioned space with the parameters t_2, d_2 goes to the evaporator HP. The evaporator cools and drains the air to the parameters t_{ev}, d_{ev} (point B). After the HP evaporator, the air with the parameters t_{ev}, d_{ev} enters the mixing chamber, where it mixes with the ambient air flow. The resulting mixture with corresponding parameters enters the HP condenser. After heating the air in the condenser to the set temperature t_1, part of the air with mass flow G_{ger} and set parameters t_1, d_1 goes into the room. The other part of the air is mixed with the ambient air (G_{ger}) with the parameters t_0, d_0 and after passing the condenser is removed with a temperature t_c into atmosphere.

4 Thermodynamic analysis of schemes

4.1 HPS air conditioning with exhaust air recirculation via HP condenser

The thermodynamic state of the system under consideration, in accordance with the accepted conditions for the given parameters of air in the room, depends on the parameters of the external atmospheric air (temperature and relative humidity). This condition can be determined by unknown air parameters at the nodal points of the air conditioning system. These parameters include the proportion of fresh air flowing into the system, air parameters after the evaporator HP, air parameters flowing into the atmosphere. All these values can be determined by solving the system of equations of thermal and material balance as separate elements of the scheme, and the scheme as a whole.

In the general case, the proportion of ambient air flowing into the room through the HP evaporator can be represented as

$$K_0 = G_0/G_{ger} \ , \tag{1}$$

where G_0, G_{ger} are the mass consumption of atmospheric and general (through the room) flows of air.

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From the material balance of the scheme as a whole (Fig.1a)

$$G_{ger}(d_2 - d_1) + G_0 d_0 = G_c d_2 + G_0(d_0 - d_{ev}) \tag{2}$$

we obtain, after a series of mathematical transformations, an expression that characterizes the relative flow of external air

$$K_0 = \frac{d_2 - d_1}{d_2 - d_{ev}} , \tag{3}$$

where d_1 is the moisture content of air supplied to the room, d_2 is the moisture content of air leaving the room, and d_{ev} is the moisture content of air supplied to the mixing chamber after the HP evaporator. The latter value can be determined by using the h-d of the wet air diagram on the line $\varphi = 100\%$, or from the interpolation equation [11]

$$d_{ev} = 4.42 \times 10^{-3} e^{0.0596 t_{ev}} , \tag{4}$$

where t_{ev} is the air temperature after the HP evaporator, which can be determined depending on the enthalpy of the same air at the outlet of the evaporator, according to the following interpolation equation [11]:

$$t_{ev} = -2.1 \times 10^{-3} h_{ev}^2 + 0.552 h_{ev} - 4.58 . \tag{5}$$

In turn, the enthalpy of the air flow at the outlet of the evaporator can be determined on the basis of the thermal balance of the mixing chamber

$$Q_{ev} + Q_{rec} = Q_1 , \tag{6}$$

where $Q_{ev} = G_0 h_{ev}$ is the amount of heat entering the mixing chamber with the air flow after the evaporator, $Q_{rec} = G_{rec} h_c$ is the amount of heat coming from the recirculation flow after the condenser, and $Q_1 = G_{ger} h_1$ is the amount of heat entering the room. So we can get

$$h_{ev} = \frac{h_1 - (1 - K_0) h_c}{K_0} . \tag{7}$$

An important task in analyzing the system is to determine the air enthalpy (with further temperature determination of t_c) after cooling the heat pump condenser. The equation for determination h_c can be obtained from the heat balance of a heat pump

$$Q_{ev} + L = Q_c . \tag{8}$$

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The work of the HP compressor drive is expressed as

$$L = Q_{ev}/\varepsilon_{HP} \ , \tag{9}$$

the amount of heat removed in the evaporator HP

$$Q_{ev} = G_0(h_0 - h_{ev}) \ , \tag{10}$$

the amount of heat removed from the condenser, respectively

$$Q_c = G_{ger}(h_c - h_2) \ . \tag{11}$$

Substituting Eqs. (9)–(11) into Eq. (8), we obtain an expression for determining enthalpy

$$h_c = h_2 + K_0(h_0 - h_{ev})\left(\frac{\varepsilon_{HP} + 1}{\varepsilon_{HP}}\right) \ . \tag{12}$$

The efficiency of operation of the heat pump, which works in the air conditioning mode as a refrigerating machine, can in this case be estimated by the value of the refrigeration coefficient for the ideal Carnot reverse cycle, taking into account the non-convergence of the heat transfer processes in the evaporator and the condenser, which, according to the recommendation of work [12] for air-air HP, can be accepted at the temperature level of 10 °C

$$\varepsilon_{HP} = \frac{1}{\dfrac{273 + t_c + 10}{273 + t_{ev} - 10} - 1} \ . \tag{13}$$

The HP refrigeration factor, as well as coefficient of performance (COP) of the HP, depends only on the operating conditions of the HP itself, i.e. the temperature level of the heat transfer processes carried out in the evaporator and the HP condenser. Therefore, in order to characterize the efficiency of the whole heat pump system, it is more appropriate to use a complex indicator (cooling coefficient of the whole scheme), which can be represented as follows:

$$\varepsilon_{sh} = \frac{Q_{cool}}{L} \ , \tag{14}$$

where Q_{cool} is the refrigeration capacity at the entrance to the room and is determined as

$$Q_{cool} = G_{ger}(h_0 - h_1) \ . \tag{15}$$

Taking into account Eqs. (9), (10), and (15), the expression (14) can be written as follows:

$$\varepsilon_{sh} = \frac{\varepsilon_{HP}(h_0 - h_1)}{K_0(h_0 - h_{ev})} \ . \tag{16}$$

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4.2 HPS air conditioning with exhaust air recirculation via HP evaporator

As for the previous scheme, unknown parameters of air at the nodal points of the circuit will be determined by the solution of the system of equations of thermal and material balance as separate elements of the scheme and the scheme as a whole.

In the general case, the relative fraction of the ambient air flow can be determined from Eq. (1) of the material balance of moisture for the whole circuit or the balance of the mixing chamber. From the equation of the material balance of the mixing chamber

$$G_{ger}d_{ev} + G_0 d_0 = (G_{ger} + G_0)d_1 \tag{17}$$

we can obtain expression

$$K_0 = \frac{d_1 - d_{ev}}{d_0 - d_1} , \tag{18}$$

where d_{ev} is the moisture content of air supplied to the mixing chamber after the evaporator TH, d_1 is moisture content of air supplied to the room, and d_0 is the moisture content of the external air entering the mixing chamber. The value d_{ev} can be determined by using the h-d of the humid air diagram on the line $\varphi = 100\%$, or from the interpolation Eq. (3), in which the air temperature after the evaporator HP is determined by the interpolation Eq. (4).

In turn, the required enthalpy of the air flow at the outlet of the evaporator can be determined on the basis of the thermal balance of HP (8). In this case, the heat flux from the air in the evaporator

$$Q_{ev} = G_{ger}(h_2 - h_{ev}) , \tag{19}$$

and the heat flux removed from the condenser respectively

$$Q_c = (G_{ger} + G_0)(h_1 - h_{mix}) + G_{ger}(h_c - h_0) + G_0(h_c - h_1) , \tag{20}$$

where h_{mix} is determined from the thermal balance of the mixing chamber and is equal to

$$h_{mix} = \frac{h_{ev} + K_0 h_0}{1 + K_0} . \tag{21}$$

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Thus, substituting the relations (9), (19), (20), and (21) into Eq. (8), we obtain

$$h_{ev} = h_2 - \left((1+K_0)(h_1 - h_{mix}) + (h_c - h_0) + K_0(h_c - h_1)\right) \times \left(\varepsilon_{HP}/(\varepsilon_{HP}+1)\right) . \tag{22}$$

The enthalpy of air after the cooling of the HP condenser can be determined from the thermal equation of the whole circuit

$$L + Q_0 + Q_{ger} + Q_r = Q_c + Q_m , \tag{23}$$

where L is the power of HP compressor drive, Q_0 is the heat flux coming from the ambient air to the mixing chamber, Q_{ger} is the heat flux coming from the ambient air to the HP condenser, Q_r is the heat flux generated in the room, Q_c is the heat flux that is removed from the condenser with air flow into the environment, and Q_m is the amount of heat that is removed in drainage with condensate from the evaporator (in the calculations it is possible to neglect). The components of the thermal balance of the circuit can be expressed due to the corresponding mass flow rate and enthalpy of air. Then, after a series of mathematical transformations, we obtain an expression for determining the enthalpy of air after the condenser HP

$$h_c = h_0 + \frac{h_2 - h_{ev}}{\varepsilon_{HP}(K_0+1)} + \frac{h_2 - h_1}{K_0 + 1} . \tag{24}$$

The efficiency of the operation of the heat pump can be estimated by the value of the cooling coefficient by Eq. (13), and the cooling coefficient of the whole scheme can be represented by the dependence (14) taking into account Eqs. (9), (15), and (19). Then we will get

$$\varepsilon_{sh} = \frac{(h_0 - h_1)}{(h_2) - h_{ev}} \varepsilon_{HP} . \tag{25}$$

5 Numerical analysis of parameters of heat pump systems

Analysis of the presented schemes is carried out by implementing the constructed theoretical models by the method of successive approximations. The corresponding calculations were made for conditions of typical production premises of the confectionery factory in the Kiev zone (Ukraine). At the same time, the initial data for air parameters in the middle of the

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premises were agreed with the European standards for indoor temperature and humidity for some industrial products and production processes [13]. To ensure comfortable working conditions in the production premises, the following parameters of the internal air were chosen [13,14]:

- room temperature $t_r = 18\,^{\circ}\mathrm{C}$,
- air humidity in the room $\varphi_2 = 50\%$,
- air temperature at the entrance to the room $t_1 = 15\,^{\circ}\mathrm{C}$.

Under given conditions, moisture content was determined at the entrance and exit of the room, that is, at points 1 and 2 of the working process diagram in Fig. 2: $d_1 = 6.096 g_{moisture}/kg_{dryair}$ and $d_2 = 6.391 g_{moisture}/kg_{dryair}$.

Accepted conditions were the basis for further analysis of HPS air conditioning parameters. In this case, the above parameters of the schemes were determined by numerical solution of the system of Eqs. (3)–(5), (7), (12), (13), and (16) by the iteration method for HPS air conditioning with recirculation of exhaust air through HP condenser and system Eqs. (18), (4), (5), (13), (22), (24), (25) for HPS air conditioning with exhaust air recirculation via HP evaporator.

The realization of the constructed mathematical models allows us to obtain the dependence of the relative flow of fresh air on the temperature and relative humidity of the ambient air for the indicated HPS air conditioning.

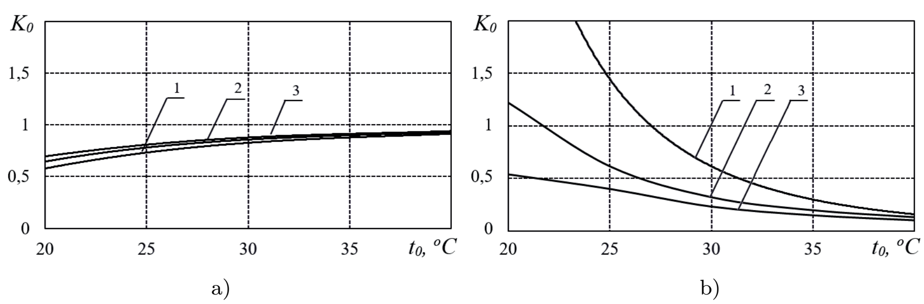

Figure 3: Dependence of the relative consumption of fresh air on the ambient temperature for HPS air conditioning with recirculation of exhaust air through the HP condenser (a) and HP evaporator (b) at different values of relative humidity of the atmospheric air: 1 – $\varphi = 40\%$, 2 – 50%, 3 – 60%.

As can be seen from Fig. 3a, when the ambient temperature increases, the relative fresh air consumption increases, which is due to an increase in the thermal load of the HP evaporator, an increase in the air flow that is removed into the atmosphere and, as a consequence, a decrease in the flow of recirculation. In this case, the value of K_0 practically does not depend on the relative humidity of the ambient air. On the contrary, in the second scheme (Fig. 3b), the value of K_0 decreases with an increase in both the temperature and relative humidity of the ambient air, due to the increase in its enthalpy and the limitation of air parameters at the exit from the mixing chamber.

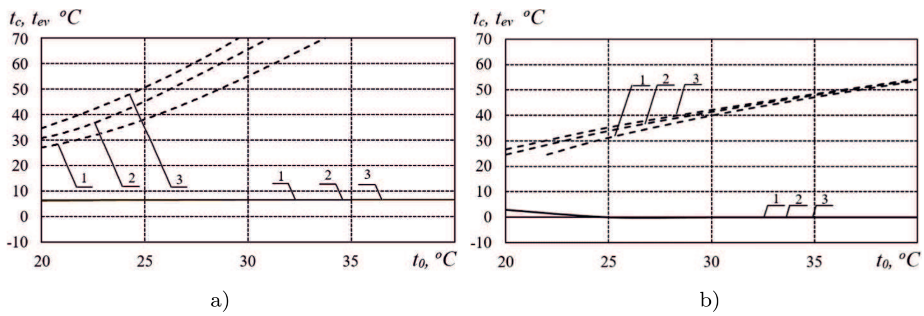

Figure 4: Dependence of the air temperature at the outlet from the HP condenser (dashed lines) and the HP evaporator (solid lines) from ambient temperature for HPS air conditioning with recirculation of exhaust air through the HP condenser (a) and HP evaporator (b) at different values of relative humidity of the atmospheric air: 1 – $\varphi = 40\%$, 2 – 50%, 3 – 60%.

The thermodynamic state and efficiency of each heat pump schemes are determined by the air temperatures at the outlet from the evaporator and the HP condenser. As can be seen from Fig. 4a and b, with an increase in ambient temperature, the air temperature at the outlet of the evaporator HP (solid lines) practically does not change throughout the range of change in temperature and relative humidity of the outside air φ. However, there is a significant increase in the air temperature at the output from the HP condenser (dashed lines), which is directly related to the increase in the thermal load of the HP condenser.

In Fig. 5, the dependences of the HP refrigeration ratio and the heat pump scheme as a whole for the air conditioning systems specified on the ambient air parameters are constructed.

In connection with the expansion of the temperature range of the HP cycle (Fig. 4), which leads to deterioration of its working conditions, there is a significant decrease in the HP refrigeration rate for both the first and second schemes. However, the refrigeration coefficient of the scheme, which is the ratio of the amount of cold at the entrance to the premises to the

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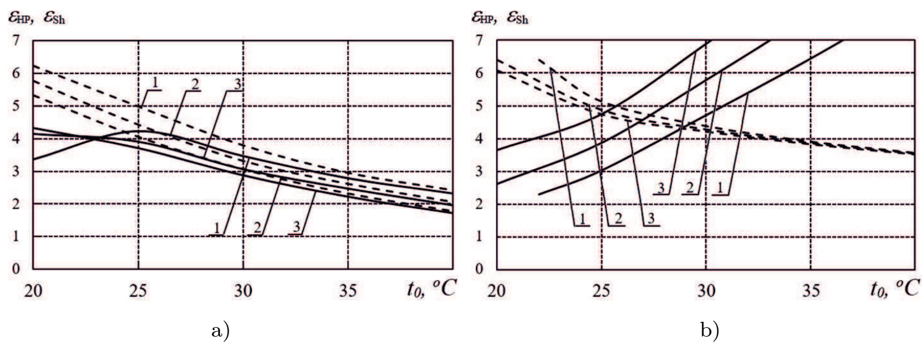

Figure 5: Dependence of the HP refrigeration factor (dashed lines) and the cooling coefficient of the heat pump circuit as a whole (continuous lines) for HPS air conditioning with recirculation of air through: a) HP condenser; b) HP evaporation from ambient temperature at different values of relative humidity of atmospheric air: 1 – φ = 40%, 2 – 50%, 3 – 60%.

spent work of the HP compressor, for the second scheme (as opposed to the first) increases with the increase of ambient air parameters. This indicates that the second scheme (with the recycling of exhaust air through the HP evaporator) ensures the cooling of the system and thus is more efficient.

6 Conclusions

The analysis showed that in order to maintain the set temperature and humidity conditions in premises with the release of excess moisture during the hot period of the year, heat pump air conditioning schemes with different versions of exhaust air recirculation (through a condenser or evaporator of HP) can be used.

The constructed theoretical models of heat pump air conditioning systems allow us to determine the required relative consumption of fresh air and air parameters at their nodal points, which provide the given conditions in the middle of the room. At the same time, as shown by the results of numerical calculations, these characteristics and air parameters to a large extent depend on the temperature and humidity of external atmospheric air and especially for the circuit with the recirculation of exhaust air through the HP condenser (Fig. 4a). In this case, with the increase in the parameters of the outside air, the air temperature at the outlet of the HP condenser goes beyond the parametric range of effective operation of conventional heat pumps.

Numerical calculations have shown that the second scheme (when air recirculation through the HP evaporator) is more able to work, since it provides the set temperature-humidity parameters of the air in the room at a lower value of the air temperature at the outlet of the HP condenser and can operate in a wider range of temperatures and humidity of the ambient air (Fig. 4b).

As a result of numerical analysis, it was found that the second scheme is more energy-efficient, because it provides a higher cooling coefficient of the circuit (or requires less specific energy consumption) in a wider range of parameters of the outside air during the hot period of the year (Fig. 5b).

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