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THERMODYNAMIC ANALYSIS OF THE HEAT PUMP-ADSORPTION AIR DRYING SYSTEM USING A RECOVERY

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This paper describes the development of a heat pump and adsorption system that incorpo-rates a recuperator. A theoretical model is developed to analyze the system's operation numeri-cally. The study includes a numerical analysis of the thermodynamic efficiency, an examination of the changes in air parameters at different points in the system, an investigation of the impact of variations in temperature and relative humidity of the surrounding air, and also an analysis of the effect of the regeneration air temperature on the system's performance. Graphical representations of the system's efficiency are generated by modifying the parameters of the outside air and the re-generation air temperature following the heat pump condenser. The study also explores the in-fluence of the recuperator's efficiency on the overall energy efficiency of the system. The results demonstrate that this advanced system significantly reduces the specific electrical energy con-sumption for air drying compared to systems lacking a recuperator or a recuperator and a heat pump, regardless of the outdoor air parameters.

Keywords: protection of metal equipment from corrosion, conservation of energy equipment, air drying, adsorption dryer, heat pump, heat recovery.

1. Introduction

The prevention of corrosion in metal structures is a pressing concern within the contemporary energy industry. The detrimental effects of corrosion contribute significantly to emergency incidents and financial expenses using energy equioment which is exploited at the high temperatures and pressures. It is crucial to address corrosion in order to mitigate the risk of equipment failure, as any component affected by corrosion has the potential to jeopardize the entire system's integrity.

In today's operating conditions, power equipment operates intermittently, experiencing extended periods of inactivity. During these downtime periods, it is crucial to protect the equipment from the adverse effects of corrosion, which poses the greatest risk. Without proper preservation measures, the equipment's lifespan is shortened, resulting in increased financial expenses for repairs and depreciation charges [1].

The presence of moisture in the air acts as the primary catalyst for the corrosion process, with higher relative humidity accelerating the deterioration of metal components in equipment [2]. However, when the relative air humidity is low (below 40 %), corrosion processes nearly come to a halt, preserving the integrity of the metal. Therefore, to effectively preserve power equipment, it is essential to maintain the relative humidity of the air in contact with the metal below 40 %. [3].

There are many ways of dehumidifying the air, which have different physics of the process. The main methods of air drying include: condensation, adsorption, absorption and membrane methods [4].

In work [5] the development and analysis of the operation of adsorption air drying systems with an electric regeneration air heater and an already improved system of heat pump adsorption air drying, where the heat pump unit uses the heat of the exhaust regeneration air in the evaporator of the heat pump to heat fresh regeneration air in the condenser.

The purpose of the work is to increase the energy efficiency of the air-drying scheme by combining a heat pump-adsorption air drying unit with a recuperator for the preliminary utilization of the heat of the exhaust regeneration air after the adsorption rotor. These actions are caused by high costs of electricity for heating the regeneration air in ordinary adsorption dryers, which increases the financial costs of withdrawing energy equipment into reserve.

2. Description of the operation of the scheme

The adsorption dehumidifier operates based on the characteristic of silica gel to absorb moisture from the air within a moderate temperature range (-40°C to +40°C) and release the adsorbed moisture when exposed to hot regeneration air (+60°C to +150°C). To ensure a continuous process of drying the working air and regenerating the adsorbent, a specially designed rotor is employed (see Figure 1). This rotor features a honeycomb structure, which maximizes the contact surface area and facilitates the airflow through the silica gel volume. The rotor is divided into two zones, with a ratio of 1 to 3. The larger zone is dedicated to the passage of the working air, where drying and heating take place, while the preheated regeneration air flows through the smaller zone [6]. In this smaller zone, moisture is evaporated from the rotor, resulting in humidification and cooling of the regeneration air. Fig. 2 depicts a heat pump-adsorption system with a recuperator for energy conservation in equipment.

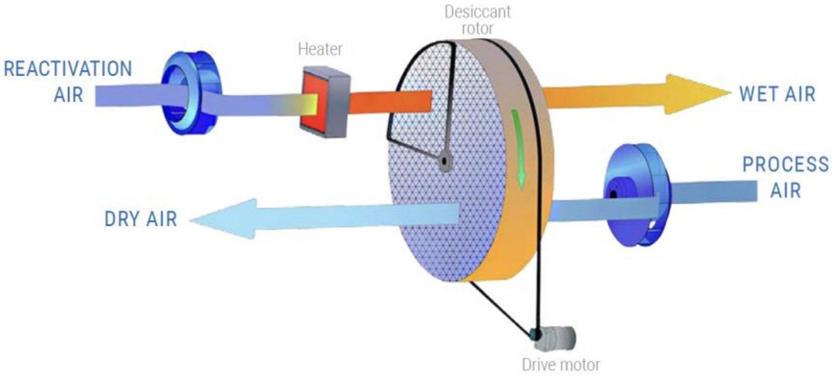

Fig.1. The principle of operation of the adsorption dehumidifier.

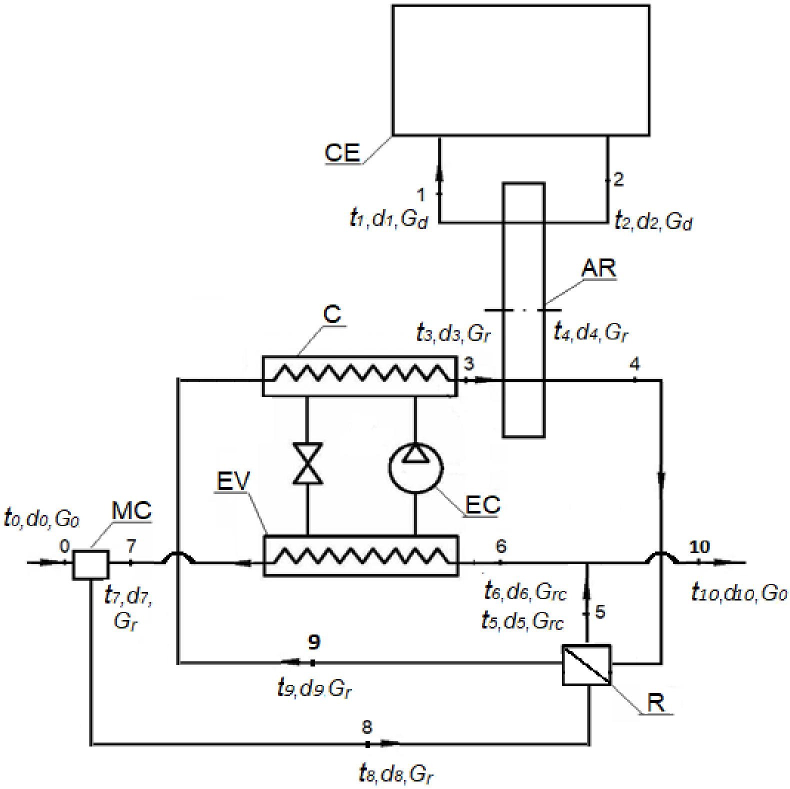

Fig.2. Schematic diagram of a heat pump-adsorption air drying scheme using a recuperator:
C – HP condenser; EV – HP evaporator; EC – compressor; AR – adsorption rotor; MC - mixing chamber; R – recuperator; CE is a canning facility.

This system incorporates a silica gel adsorption rotor to achieve thorough air drying for preservation purposes. The system also utilizes a heat pump and recuperator to efficiently heat and regenerate the air, with a portion of the regeneration air being recirculated. This recirculation process results in substantial energy savings compared to electric heating methods. The operational principle of the system can be described as follows: The upper section of the diagram illustrates the canning object (CE), from which moist air (working air) is directed into the adsorption rotor. Inside the rotor, the air is heated, dried, and then returns to the canning object with a temperature of t_1 and moisture content of d_1.

In the lower part of the diagram, the scheme illustrates the preparation, supply, and removal of regeneration air from the rotor. Simultaneously, external air (0) with a temperature of t_0, moisture content of d_0, and mass flow rate of G_0 is drawn from the surrounding environment and directed into the mixing chamber. In the mixing chamber, it combines with recirculated regeneration air (7), which emerges from the heat pump evaporator with a temperature of t_7, moisture content of d_7, and a mass flow rate of G_r.

Once mixed, the air mixture (8) passes through the recuperator, where it is heated to a temperature of t_9 (9), before entering the heat pump condenser. Inside the condenser, the air mixture is further heated to a temperature of t_3 (3) and then directed into the rotor for regeneration, allowing the removal of adsorbed moisture. While passing through the rotor, the regeneration air undergoes cooling and humidification.

After leaving the rotor, the regenerated air (4) with a temperature of t_4 and moisture content of d_4 goes through the recuperator, where it cools down to a temperature of t_5 and moisture content of d_5. Subsequently, it splits into two streams: one is discharged into the environment (10), while the other is routed through the heat pump evaporator (6), where it undergoes cooling and partial drying. Finally, it returns to the mixing chamber, where it mixes with fresh air.

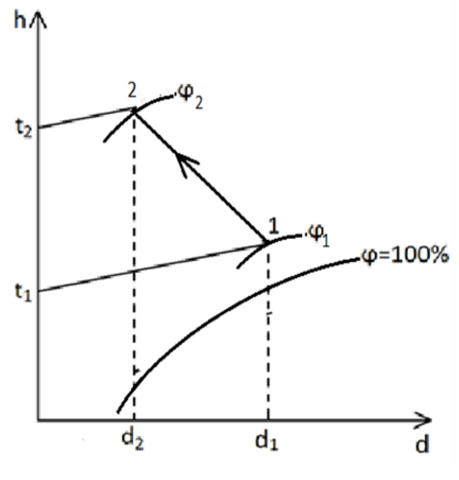

Fig.3. The process of drying the working air when passing through the adsorption rotor.

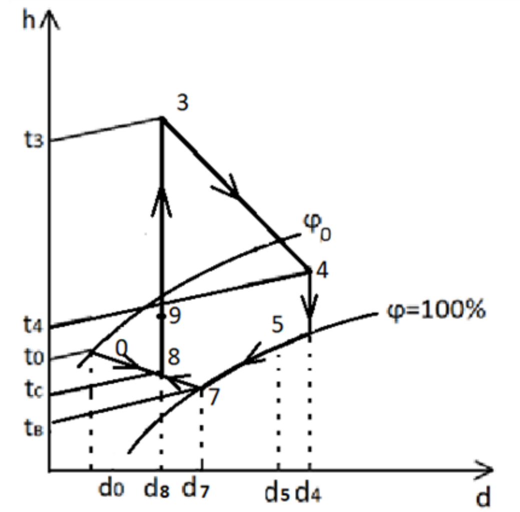

Fig. 4. The process of preparing and changing of regeneration air when passing through the adsorption rotor, HP and recuperator.

Figures 3 and 4 present the working processes of drying the working air and changing the state of the regeneration air in the elements of the heat pump-adsorption installation using a recuperator in the hd diagram of moist air.

3. Thermodynamic analysis of scheme efficiency

The efficiency analysis of the system can be conducted by examining the thermodynamic state of the adsorption rotor's regeneration scheme. This analysis relies on determining the parameters at key points within the scheme. These parameters include the temperature and humidity of the regeneration air after the evaporator, both at the entrance and exit of the mixing chamber, as well as at the exit of the adsorption rotor or the entrance of the heat pump evaporator. Additionally, the airflow of ambient air and the recirculation air entering the evaporator are also essential unknowns. The relationships governing these parameters can be derived from heat and material balance equations for individual elements and the entire system.

Considering the moisture evaporation process in the rotor as isoenthalpic, where the enthalpies h3 and h4 are equal, the overall heat balance equation for the heat pump-adsorption air drying system (with the enthalpy of the condensate after the heat pump evaporator neglected) can be written as

$$Q_0 + L_{ec} = Q_{ex}, \tag{1}$$

where Q_o is the heat flow supplied to the system with fresh air, kW;

Lec- heat pump compressor drive power, kW;

Qex - heat flow discharged into the environment with exhaust air, kW,

or in more detail in the form

$$G_0 h_0 + \frac{Q_{ev}}{\varphi - 1} = G_0 h_4, \tag{2}$$

where Q_{ev} is the heat flow removed from the recirculation air in the evaporator, kW;

G_0 – consumption of fresh air, kg d.a./h;

h_4- enthalpy of regeneration air after the adsorption rotor, kJ/kg d.a.;

h_0- enthalpy of fresh air, kJ/kg d.a.;

φ is the HP transformation coefficient.

The effective coefficient of heat transformation of the heat pump is defined as:

$$\varphi = \varphi_T \eta_{HP}, \tag{3}$$

where η_{HP} is the loss coefficient, which takes into account the real processes carried out by the working body in HP, which, according to a number of sources, can vary in the range of 0.6...0.8 (we assume $\eta_{HP} = 0.6$);

φ_T is the theoretical coefficient of transformation of HP.

The coefficient of heat transformation of the ideal Carnot cycle φ_T taking into account thermal irreversibility in the evaporator and condenser HP is determined by the ratio

$$\varphi_T = \frac{1}{1 - \frac{T_{EV}{}^{HP}}{T_C{}^{HP}}} = \frac{1}{1 - \frac{273 + t_{EV} - \Delta t_{EV}}{273 + t_C + \Delta t_C}}, \tag{4}$$

where $T_{EV}{}^{HP}$ – the absolute temperature of evaporation of the refrigerant in the HP evaporator, K;

$T_C{}^{HP}$ – the absolute temperature of condensation of the refrigerant in the condenser HP, K;

t_7– the temperature of the exhaust air at the outlet of the HP evaporator, °C;

t_3– air temperature at the outlet of the condenser HP, °C;

Δt_{ev} – temperature difference between the flows of exhaust air and refrigerant at the outlet of the HP evaporator, °C;

Δt_c is the temperature difference between the refrigerant and regenerative air flows at the outlet of the HP condenser, °C.

Numerical values of temperature differences in the condenser and evaporator of HP are given in the literature. According to [7], $\Delta t_{ev} = \Delta t_c = 10$°C can be taken for the "air-air" HP type for the evaporator and condenser.

Taking into account the expression for the heat flow of the HP evaporator

$$Q_{EV} = G_{rc}(h_4 - h_7) \tag{5}$$

and material balance equations for the dry air of the mixing chamber

$$G_0 + G_{rc} = G_r, \tag{6}$$

where h_7 is the enthalpy of recirculation air after the evaporator, kJ/kg d.a.,

G_{rc}– consumption of air recirculation flow, kg d.a./h,

G_r– consumption of regenerative air flow, kg d.a./h, equation (2) after transformations can be written in the form

$$K\frac{h_5-h_7}{\varphi-1}=(1-K)(h_5-h_0) ,\qquad (7)$$

where *K* is the coefficient of recirculation of regenerative air K= G_{rc} / G_r.

Then, from equation (7) we get the expression for the regeneration air recirculation coefficient

$$K=\frac{1}{\frac{(h_5-h_7)}{(\varphi-1)(h_5-h_0)}+1} .\qquad (8)$$

Other parameters of the system will be determined from the material and heat balances of individual elements of the scheme. At the same time, from the equation of the thermal balance of HP

$$Q_{ev}+L_{ec}=Q_c, \qquad (9)$$

which can be represented as

$$\frac{G_{rc}(h_6-h_7)}{\varphi-1}=\frac{G_r(h_3-h_9)}{\varphi}, \qquad (10)$$

we get the formula for the air enthalpy at the outlet of the HP evaporator

$$h_7=h_5-\frac{h_3-h_9}{K}\bullet\frac{\varphi-1}{\varphi} . \qquad (11)$$

The enthalpy of the air mixture after mixing recirculation and fresh air in the mixing chamber is determined from the equation of the heat balance of the mixing chamber

$$G_o h_0+G_{rc}h_7=G_r h_8 , \qquad (12)$$

where

$$h_8=(1-K)h_0+Kh_7 . \qquad (13)$$

From the material balance of the adsorption rotor

$$G_d(d_2-d_1)=G_r(d_4-d_3) , \qquad (14)$$

where G_d is the flow of air that is dried and is a preservative agent, kg d.a./h, and d_1, d_2, d_3, d_4 are the moisture content of the air at the corresponding points of the diagram, we will get the expression for the moisture content of the discharge air

$$d_4=d_3+\frac{G_d}{G_r}(d_2-d_1) . \qquad (15)$$

From the equation of the material moisture balance for the mixing chamber

$$G_o d_0+G_{rc}d_7=G_r d_3 \qquad (16)$$

we get an expression for the moisture content of the air at the outlet of the HP evaporator

$$d_7 = \frac{d_3 - (1-K)d_0}{K} \,. \tag{17}$$

Knowing d_7, we can obtain the air temperature at the outlet of the HP evaporator from the interpolation equation on the saturation line (φ=100%) on the hd diagram of moist air [8]

$$t_7 = 14{,}752\ln(\mathrm{d}_7) - 18.929 \tag{18}$$

Since d_3 = d_8, we can determine

$$d_8 = Kd_7 + (1-K)d_0 \,. \tag{19}$$

According to (18), we determine t_8, and accordingly based on the efficiency of the recuperator η_{rec}, which lies within 0...1, we can determine the regeneration air temperature t_9

$$t_9 = t_7 + \eta_{rec}(t_4 - t_8) \,. \tag{20}$$

Having determined t_9 and calculated the enthalpy h_9, we can determine the enthalpy of regeneration air after cooling in the recuperator h_5

$$h_5 = h_4 - (h_9 - h_8) \,. \tag{21}$$

The numerical implementation of the above system of equations for air parameters at the nodal points of the system allows to determine the energy efficiency of the use of the heat pump in the rotor regeneration system by the usual ratio of the useful effect of the circuit to the energy spent on the HP compressor drive

$$\eta = \frac{Q_c}{L_{ec}} \,, \tag{22}$$

where η - coefficient of energy efficiency of the scheme.

Defining the useful effect as the heat flux used to evaporate moisture from the adsorption rotor, according to Eq.

$$Q_{us} = G_r(d_4 - d_3)r(t) \,, \tag{23}$$

where $r(t)$ - the latent heat of water vaporization, and the power of the HP compressor drive according to the equation

$$L_{ec} = \frac{G_r(h_4 - h_в)}{(\varphi - 1)} \,, \tag{24}$$

we will get the final expression for the coefficient of energy efficiency of the scheme in the form

$$\eta = \frac{(d_4 - d_3)r(\mathrm{t})(\varphi_\mathrm{T} - 1)}{K(h_5 - h_7)} \,. \tag{25}$$

4. Computational analysis of the system

By employing the method of successive approximations and utilizing equations (3), (4), (8), (11), (13), (15), (17), (18), (20), (21), and (25), a numerical analysis of the system can be conducted. This analysis allows for the assessment of how specified variables (such as the temperature and relative humidity of the outside air, temperature, and moisture content of the regeneration air before the adsorption rotor, and the efficiency of the recuperator) impact the performance characteristics of individual elements (such as the exhaust air recirculation coefficient and the heat pump transformation coefficient), as well as the overall energy efficiency of the heat pump-adsorption air drying system with a recuperator.

The parameters and operating ranges were chosen with the same constraints as in previous publication [5]. Figure 5 illustrates the graphical relationship between the recirculation coefficient and the outside air temperature, considering the efficiency of the recuperator.

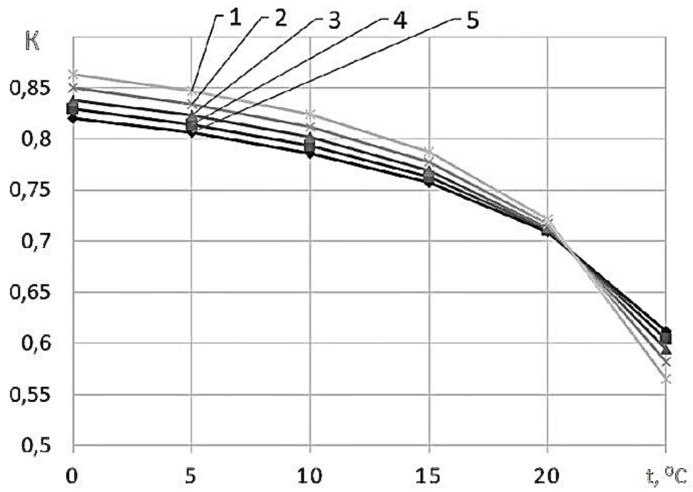

Fig. 5. Dependence of the recirculation coefficient on the fresh air temperature of the recuperator efficiency coefficient at a regeneration air temperature of 60 ^{0}C: 1- η_{rec} =0.8; 2- η_{rec} =0.6; 3- η_{rec} =0.4; 4- η_{rec} =0.2; 5- a system with a heat pump but without a recuperator.

The results demonstrate that at fresh air temperatures ranging from 0°C to 25°C, the recirculation coefficient in a system with a recuperator surpasses that of a system without a recuperator.

However, at higher temperatures, the recirculation coefficient declines more rapidly in a system with a recuperator compared to a system without one. Figure 6 shows graphical dependences of the overall energy efficiency coefficient of a system with a heat pump and recuperator, a system with only a heat pump, without a recuperator, and a system with an electric heater at different temperatures of regenerative air.

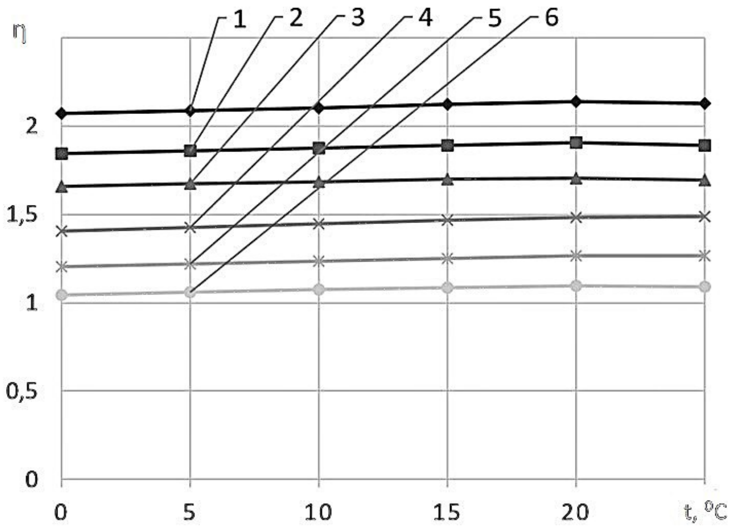

Fig.6. Dependence of the energy efficiency coefficient η on the temperature of fresh air and the temperature of regeneration air at η_{rec} =0.6: 1- t_r =60^{0}C; 2- t_r =65^{0}C; 3- t_r =70^{0}C; and without recuperator: 4 – t_r =60^{0}C; 4 – t_r =65^{0}C; 4 – t_r =70^{0}C.

In fig. 7 shows graphical dependences of the overall energy efficiency coefficient of a system with a heat pump and recuperator, a system with only a heat pump, without a recuperator, and a system with an electric heater.

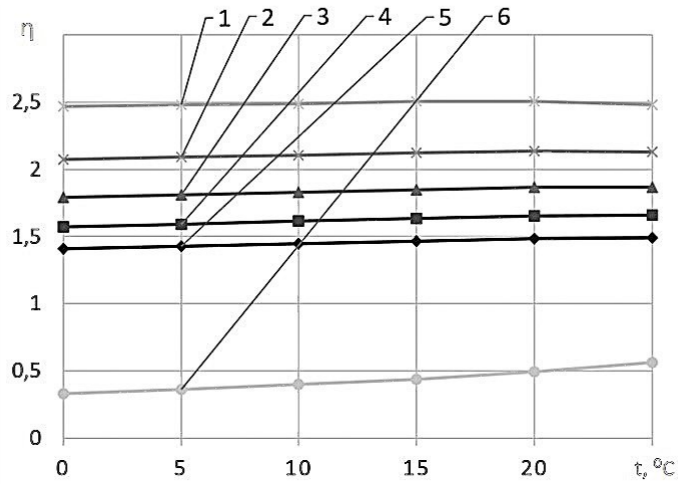

Fig.7. Dependence of the coefficient of energy efficiency η on the temperature of fresh air and the coefficient of efficiency of the recuperator at the temperature of regeneration air 60 ^{0}C: 1- η_{rec} =0.8; 2- η_{rec} =0.6; 3- η_{rec} =0.4; 4- η_{rec} =0.2; 5- a system with a heat pump but without a recuperator; 6- system without a heat pump.

It can be seen from the given graph that the energy efficiency of a system with a heat pump is 2.5-3.6 times higher than a system with an electric heater, and a system with a heat pump and recuperator can have an energy efficiency up to 6 times higher than a standard scheme with an electric heater. and 1.8 times higher energy efficiency than a scheme with a heat pump without a recuperator.

5. Conclusions

1. The combined use of an adsorption dehumidifier and a heat pump is a much more energy-efficient solution than the use of an adsorption dehumidifier with an electric heater, and the use of a heat pump with a recuperator in the system additionally increases the overall energy efficiency of the air drying process.
2. The overall energy efficiency of the considered air drying system largely depends on the coefficient of efficiency of the recuperator itself, and therefore the issue of selecting the type of recuperator requires special attention.
3. At the temperature of the regeneration air before the heat pump 60 ^{0}C and increasing the efficiency of the recuperator to the η_{rec}*=0,8,* the specific external energy consumption for air drying can be reduced by 1.8 times compared to the heat pump scheme without a recuperator and up to 6 times compared to the basic scheme without a heat pump and recuperator.

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