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HEAT BALANCE OF COMBINED HEAT EXCHANGER AERODYNAMIC HEATING DRYERS

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Abstract:
Today, the use of aerodynamic dryers for drying various types of fruit crops is very current. In them, the electric energy spent on the drive of the centrifugal fan is transformed into thermal energy due to the mutual friction of the air flows circulating in the closed chamber. In order to increase the energy efficiency of the drying process, the heat of the waste drying agent was used in the research. The presented dryer was equipped with a combined heat exchanger. In order to predict the thermal performance of the combined heat exchanger depending on external factor variables, the dependence of the temperature of the fresh drying agent at the outlet of the combined heat exchanger on the dryer operation time is theoretically determined on the basis of the heat balance equation. The air solar collector in the combined heat exchanger made it possible to increase the temperature of the drying agent at the outlet by another 10 °C without extra costs of electrical energy. A comparative analysis of the results of experimental and theoretical studies showed their high convergence.

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KEYWORDS
Heat balance equation, drying agent, combined heat exchanger, aerodynamic heating dryer

1. INTRODUCTION

Drying processes are widespread in the agricultural products processing [1-6].

The high cost of traditional energy resources makes it increasingly necessary to use alternative energy sources to reduce energy intensity and increase the efficiency of technical equipment in agricultural production [7-18]. Combined types of dryers with different sources of thermal energy are often used [19-23].

One promising direction in reducing the energy intensity of drying, for example, fruit and berry raw materials is the use of aerodynamic heating dryers that implement the principle of the isolated rotor (i.e. rotor without housing) of the centrifugal fan in the rotary heating mode. In this case, the vast majority of the electric energy supplied to the rotor is spent on overcoming aerodynamic losses in the flowing part of the impeller, in the drying chamber and is converted into heat. This is the principle of aerodynamic heating furnaces (AHF) or another name - aerodynamic loss furnaces.

In order to increase the energy efficiency of the drying process, the heat of the waste drying agent was used in the research.

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In agriculture, there are dryers that use renewable sources of energy in the drying process, such as the use of corn cobs [24], residues from vineyards and orchards [25], the use of solar collectors [26-29], etc. Also, there are dryers which work on the principle of combined heat exchanger [30,31]. In research will show the operation of such a dryer with a combined heat exchanger.

The purpose of the study is to develop the heat balance equation of the aerodynamic heating dryer with a combined heat exchanger, allowing to determine on its basis the temperature of the fresh drying agent at the outlet of the heat exchanger.

2. MATERIALS AND METHODS

The research used a batch dryer with a combined heat exchanger, Fig.1.

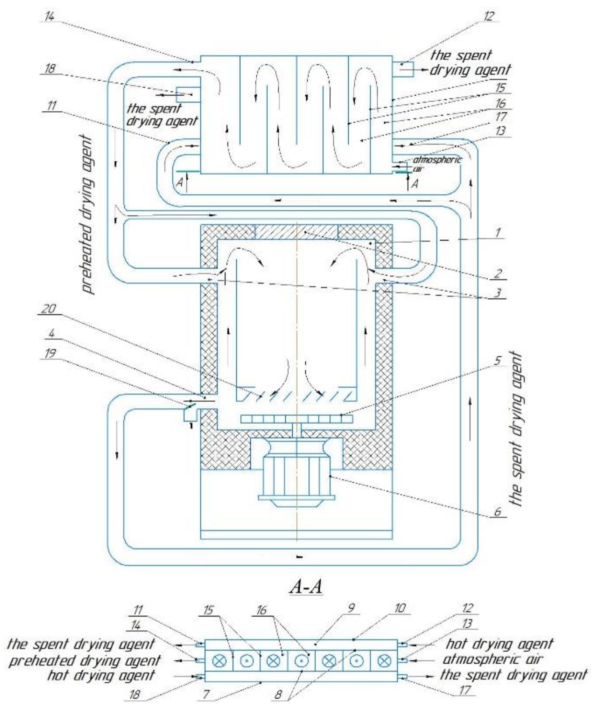

Fig. 1. Diagram of a dryer with a combined heat exchanger

The basic constituent parts of the dryer (Fig. 1) are: 1 – heat - insulated chamber; 2 – door; 3 – pipes for drying agent suction; 4 – pipes for drying agent discharge; 5 – heating rotor of a centrifugal fan; 6 – drive electric motor; 7 – combined heat exchanger; 8 – heat-receiving surfaces; 9 – air solar collector; 10 – translucent coating; 11 – pipe for entering the spent drying agent into the upper chamber; 12 – pipe for exiting the spent drying agent from the upper chamber; 13 – inlet pipe of the middle chamber; 14 – output pipe branch pipe of the middle chamber; 15 – partitions; 16 –channels for the passage of heated atmospheric air; 17 – branch pipe for entering the exhaust drying agent into the lower chamber; 18 – branch pipe for exiting the exhaust drying agent from the lower chamber; 19 – flap, (Fig.1).

Raw materials are loaded into the heat-insulated chamber 1 via the door 2. At the beginning of drying in the warm-up phase of the dryer, the flaps 19 are in the position shutting off the supply of the exhaust drying agent into the chambers of the combined heat exchanger 7. The atmospheric air enters the middle chamber of the plate heat exchanger via the inlet pipe 13. Moving along the channels 16, ambient air is heated by heat exchange with the heat-receiving surface of the solar collector 9 and via the outlet pipe 14 of the middle chamber and inlet pipe 3 of insulated chamber 1 comes to the heating rotor 5, which provides further heating and circulation of the drying agent in a closed loop of insulated chamber 1. In contact with the dried raw material, the drying agent is saturated with moisture removed, and via outlet pipes 4 is partially discharged into the atmosphere. At the same time heating of atmospheric air (drying agent) in the plate heat exchanger 7 is carried out by solar radiation energy.

When the temperature of the spent drying agent discharged from the drying chamber reaches the temperature of the heated drying agent coming from the combined heat exchanger 7, the flaps 19 are put into position, providing supply of the waste drying agent through inlet pipes 11 and 17 back to the combined heat exchanger 7. This provides extra heating of atmospheric air (drying agent) by transferring the heat of the exhaust drying agent through the heat-receiving surfaces 8. The spent drying agent is removed from the combined heat exchanger 7 via pipes 12 and 18. At the same time the increase of heat regeneration coefficient and the use of solar radiation energy for heating the drying agent in the initial period of the dryer operation before it reaches the working temperature in the insulated chamber is provided.

At an average air velocity at the inlet to the heat exchanger of about 6.5 m/s, the dynamic pressure is about 25 Pa. Compared with the total pressure of centrifugal fans of similar type in the range of 1,500-2,000 Pa the extra hydraulic resistance of the heat exchanger is negligibly small relative to the hydraulic resistance in the flow part of the heating rotor and the drying chamber, which are useful in this case.

The general view of the dryer with combined heat exchanger is shown in Fig. 2.

Fig. 2. General view of the dryer with a combined heat exchanger

The roof of the drying chamber is the wall of the lower chamber of the combined heat exchanger, located on top of the dryer within its overall dimensions. The roof of the drying chamber is the wall of the lower chamber of the combined heat exchanger, located on top of the dryer within its overall dimensions. The upper enclosure of the combined heat exchanger is a transparent cover of the air solar collector. Middle chamber of the heat exchanger has 8 channels with cross section of 220x50 mm, Fig. 3.

Fig. 3. View of the middle chamber of the combined heat exchanger

To predict the thermal performance of the combined heat exchanger depending on external factor variables, it is necessary to know the dependence of the drying agent temperature at the outlet of the combined heat exchanger on the dryer operation time. Such dependence connects the parameters of external factor variables with the design and technological parameters of the combined heat exchanger, which allows simulating its output steam parameters, depending on various external conditions and the thermal mode in the drying chamber.

The effective work of the solar collector of the combined heat exchanger is provided during the day from 8 to 18 hours. For this period, we will compose the heat balance equation.

The heat balance equation of the combined heat exchanger over an infinitesimal time interval *dt*:

$$dQ_{a.\text{в}} + dQ_{c.\text{э}} + dQ_{\text{o.a}} = dQ_{\text{c.}a} + dQ_{\text{ст.}a} + dQ_{\text{в}} + dQ_{n}, \text{ J} \quad (1)$$

where:

$dQ_{a.в.}$ – the amount of heat supplied into the combined heat exchanger with atmospheric air;

$dQ_{c.э.}$ – the amount of heat supplied with solar energy and absorbed by the heat-receiving surface;

$dQ_{o.a.}$ – the amount of heat supplied to the combined heat exchanger with the spent drying agent;

$dQ_{c.a.}$ – amount of heat removed by the drying agent (heated atmospheric air) after heat exchange with heat-receiving surfaces;

$dQ_{cm.a.}$ – the amount of heat for heating the walls of the combined heat exchanger;

$dQ_{в.}$ – amount of heat removed into the atmosphere by the spent drying agent after heat exchange with heat-receiving surfaces;

dQ_n – heat losses to the environment via the enclosures of the combined heat exchanger.

Let's determine the expressions of the heat balance components.

Amount of heat supplied to the combined heat exchanger with atmospheric air,

$$dQ_{a.\text{в}} = L_0 i_0(t) dt \quad (2)$$

where: L_0 – atmospheric air flow rate, kg/s;

$i_0(t)$ – enthalpy of atmospheric air depending on the time during the drying period, J/kg.

The amount of heat supplied with solar energy and absorbed by the heat-receiving surface,

$$dQ_{\text{с.э}} = q_{\text{с.э}}(t)F_{\text{т.п}}\varepsilon dt, \quad (3)$$

where:
$q_{с.э}(t)$ – solar energy flux density depending on the time during the drying period, W/m^2;
$F_{т.п}$ – heat-receiving surface area, m^2;
ε - surface blackness degree.

The amount of heat supplied to the combined heat exchanger with the spent drying agent, $dQ_{\text{о.а}}$

$$dQ_{\text{о.а}} = L_0 i_2(t)dt, \quad (4)$$

where $i_2(t)$ – enthalpy of the spent drying agent depending on the time during the drying period, J/kg.

Amount of heat removed by the drying agent after heat exchange with heat-receiving surfaces,

$$dQ_{\text{с}.a} = L_0 i_1(t)dt, \quad (5)$$

where $i_1(t)$ – enthalpy of the drying agent depending on the time during the drying period, J/kg.

The amount of heat supplied to heat the walls of the combined heat exchanger,

$$dQ_{\text{ст}.a} = M_{\text{ст}.a}C_{\text{ст}.a}dT_{\text{ст}.a}, \quad (6)$$

where:
$M_{cm.a.}$ – weight of the walls of the combined heat exchanger, kg;
$C_{cm.a}$ – heat capacity of the combined heat exchanger wall material, kJ/(kg K);
$dT_{cm.a}$ – temperature increment on the walls of the combined heat exchanger, K.

The amount of heat removed into the atmosphere by the spent drying agent after heat exchange with heat-receiving surfaces,

$$dQ_{\text{в}} = L_0 i_3(t)dt, \quad (7)$$

where i_3 – enthalpy of the spent drying agent after heat exchange with heat-receiving surfaces as a function of time in the drying period, J/kg.

Heat losses to the environment

$$dQ_{\text{п}} = kF(T_{\text{ст.а}}(t) - T_{\text{о.с}}(t))dt, \quad (8)$$

where:
k – heat transfer coefficient via the enclosures of the combined heat exchanger, W/(m^2 K);
F – the enclosure area of the combined heat exchanger, m^2;
$T_{cm.a}(t)$ – temperature of the walls of the combined heat exchanger depending on the time during the drying period, K;
$T_{o.c}(t)$ – ambient temperature depending on the time during the drying period, K.

Substituting these expressions (2-8) into equation (1), we obtain:

$$L_0 i_0(t)dt + q_{\text{с.э}}(t)F_{\text{т.п}}\varepsilon dt + L_0 i_2(t)dt = L_0 i_1(t)dt + M_{\text{ст}.a}C_{\text{ст}.a}dT_{\text{ст}.a} + L_0 i_3(t)dt + kF(T_{\text{ст.а}}(t) - T_{\text{о.с}}(t))dt \quad (9)$$

After transforming equation (9) we obtain:

$$\big(L_0(i_0(t) + i_2(t) - i_1(t) - i_3(t)) + q_{\text{с.э}}(t)F_{\text{т.п}}\varepsilon + kFT_{\text{о.с}}(t)\big)dt - kFT_{\text{ст.а}}(t)dt = M_{\text{ст}.a}C_{\text{ст}.a}dT_{\text{ст}.a} \quad (10)$$

Let's assume that the temperature of the walls of the combined heat exchanger is in direct dependence on the temperature of the spent drying agent, i.e.:

$$T_{\text{ст.а}} = a_{\text{а}} T_2 + b_{\text{а}}, \quad (11)$$

where:
a_a, b_a – proportional factors;
T_2 – temperature of the spent drying agent at the inlet to the combined heat exchanger, maintained by the heat regulator, K.

Considering that $dT_{\text{ст.а}} = d(a_{\text{а}}T_2 + b_{\text{а}}) = 0$ equation (10) will be:

$$\big(L_0(i_0(t) + i_2(t) - i_1(t) - i_3(t)) + q_{\text{с.э}}(t)F_{\text{т.п}}\varepsilon + kFT_{\text{о.с}}(t)\big)dt - kF(a_{\text{а}} T_2 + b_{\text{а}})dt = 0 \quad (12)$$

For functions $i_0(t)$, $i_1(t)$, $i_2(t)$, $i_3(t)$, $q_{с.э}(t)$, $T_{o.c}(t)$ and $T_2(t)$ it is known that:

$$i_0(t) = C_{c.\text{в}}T_{\text{о.с}}(t) + 0{,}001x_{\text{н}}(r_0 + C_{\text{п}}T_{\text{о.с}}(t)), \quad (13)$$

$$i_1(t) = C_{c.\text{в}}T_1(t) + 0{,}001x_{\text{н}}(r_0 + C_{\text{п}}T_1(t)), \quad (14)$$

$$i_2(t) = C_{c.\text{в}}T_2 + 0{,}001x_{\text{к}}(r_0 + C_{\text{п}}T_2), \quad (15)$$

$$i_3(t) = C_{c.\text{в}}T_3(t) + 0{,}001x_{\text{к}}(r_0 + C_{\text{п}}T_3(t)), \quad (16)$$

where:
$C_{с.в}$ – heat capacity of dry air, kJ/(kg K);
$x_н$, $x_к$ – correspondingly, initial and final moisture content of the drying agent, g/kg;
r_0 – specific heat of vaporization at temperature 0 ^{0}C;
$C_п$ – heat capacity of vapor, kJ/(kg K);
T_1 – temperature of the drying agent at the inlet to the dryer, K;
T_3 – temperature of the spent drying agent at the outlet of the combined heat exchanger, K.

It is known that the temperature of atmospheric air is proportional to the solar energy flux density:

$$T_{o.c}(t) = a_0 q_{c.э}(t) + b_0, \tag{17}$$

where a_0, b_0 – empirical factors;

$$q_{c.э}(t) = a_{c.э}t^2 + b_{c.э}t + c_{c.э} \tag{18}$$

where $a_{c.э}$, $b_{c.э}$, $c_{c.э}$ – empirical factors.

Let's assume that the spent drying agent temperature at the outlet of the combined heat exchanger is proportional to the average temperature between the heated drying agent temperature at the outlet of the combined heat exchanger and the spent drying agent temperature at its inlet:

$$T_3(t) = a_3((T_1(t) + T_2)/2) + b_3, \tag{19}$$

where a_3, b_3 – empirical factors.

Then, taking into account expressions (13-19), equation (12) will have the form:

$$T_1(t) = K_{10}t^2 + K_{11}t + K_{12}, \tag{20}$$

where:

$$K_0 = 0{,}5a_3T_2 + b_3;\ K_1 = C_{c.в} + 0{,}001x_{н}C_{п};$$
$$K_2 = L_0T_2(c_{c.в} + 0{,}001C_{п}x_{к}) + c_{c.э}\varepsilon F_{m.n} + kF(a_0c_{c.э} + b_0 - a_aT_2 - b_a);$$
$$K_3 = a_{c.э}(\varepsilon F_{m.n} + kFa_0);\ K_4 = b_{c.э}(\varepsilon F_{m.n} + kFa_0);$$
$$K_5 = L_0(K_1b_0 - 0{,}001C_{п}x_{к}K_0 - c_{c.в}K_0) + K_2;$$
$$K_6 = L_0K_1a_0a_{c.э} + K_3;$$
$$K_7 = L_0K_1a_0b_{c.э} + K_4;\quad K_8 = L_0K_1a_0c_{c.э} + K_5;$$
$$K_9 = L_0(K_1 + 0{,}5c_{c.в}a_3 + 0{,}0005C_{п}x_{к}a_3);$$
$$K_{10} = K_6/K_9;\ K_{11} = K_7/K_9;\ K_{12} = K_8/K_9.$$

Using dependence (20) is applicable at the time of the day $t \in [8 \div 18]$ h.

3. RESULTS AND DISCUSSION

The obtained expression (20) of the dependence of the drying agent temperature at the outlet from the combined heat exchanger on the time of its operation under conditions of variable external factors allows us to model the output thermal characteristics of the aerodynamic heating dryer with a heat utilizer in the form of a combined heat exchanger with a solar collector.

Fig. 4 shows the results of experimental determination of the drying agent temperature at the outlet of the combined heat exchanger $T_1(t)$ and the theoretical dependence of this temperature obtained from the expression (20).

During the experiment, the weather was sunny and there were no cloud cover. In mathematical modeling, cloud cover can be taken into account by correcting the value of solar energy flux density by substituting the appropriate factors into the equation (18). The specific values of the factors used in the calculations were determined from the averaged data of the processing of solar flux density measurements during the three days preceding the experiment. In this case dependence (18) has the form:

$$q_{c.э}(t) = -6{,}3571t^2 + 157{,}02t - 510{,}57.$$

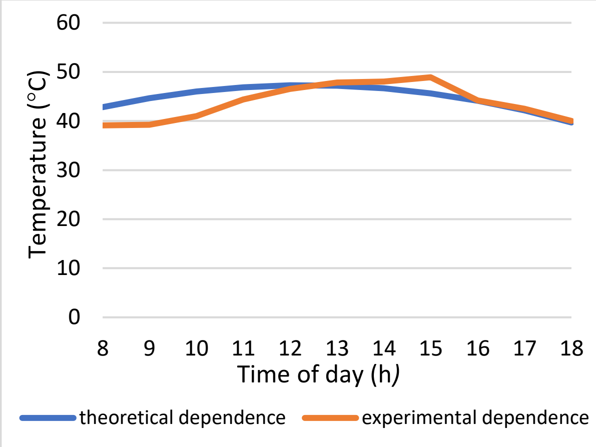

Fig. 4. Changing the temperature of the drying agent at the outlet of the combined heat exchanger

The studied dryer has the following technical characteristics: Electric motor power of the heating rotor drive – 17.5 kW, rotor frequency – 1,500 min^{-1}, outer diameter of the rotor – 700 mm; inner diameter of the rotor – 550 mm; rotor width – 150 mm; number of blades – 20; working volume of the drying chamber – 3.72 m^3, weight of loaded raw materials – 280 kg, yield of finished product – 13%; adjustable temperature range in the drying chamber in the drying process – from 40 to 80 ^{0}C; power consumption – 1 kW/h/kg of evaporated moisture. In the drying chamber, there are louvers in front of the heating rotor to adjust its performance. With the daisy-shaped circular louvers on the side wall of the dryer opposite the rotor, you can also regulate the flow rate of the spent drying agent to the heat exchanger.

The following parameter values were used in the calculation: atmospheric air flow rate L_0 – 0.3 kg/s; area of the heat receiving surface of the combined heat exchanger $F_{m.n}$ – 2.6 м²; surface blackness ε – 0.95; heat transfer coefficient via the enclosures of the combined heat exchanger k – 1.8 W/(m²·K); the enclosure area of the combined heat exchanger F – 0.9 m²; proportional factors a_a = 2.423, b_a = – 62.0976; average temperature of the spent drying agent at the inlet to the combined heat exchanger, maintained by the heat regulator, T_2 = 60 °C; heat capacity of dry air $C_{с.в}$ – 1,006 kJ/(kg K); initial and final moisture content of drying agent $x_н$, $x_к$, correspondingly – 0.009 и 0.024 g/kg; heat capacity of vapor $C_п$ – 2,040 kJ/(kg·K); empirical factors a_0=0.049, b_0=15.84; empirical factors a_3=1.4532, b_3=-19.707. The area of the drying chamber roof and the area of the combined heat exchanger top enclosure, which is a translucent cover of the air solar collector, were not taken into account in the calculation of heat loss The area of the roof of the drying chamber and the area of the combined heat exchanger top enclosure, which is a translucent cover of the air solar collector, were not taken into account in the heat loss calculation.

In accordance with the heat balance equation obtained in the experiment, the used drying agent was immediately fed into the combined heat exchanger without separating the stage of its heating to the temperature of the drying agent coming from the heat exchanger. Singling out this stage is reasonable at low atmospheric temperatures up to about 15 °C. In this case, its duration is about half an hour. At higher temperatures, the heating phase lasts about 10 minutes for a total drying time of about 13 hours.

The analysis of Fig. 4 shows high convergence of experimental and theoretical results. The presence of the air solar collector as part of the combined heat exchanger allowed in this case to increase the temperature of the drying agent at the outlet by another 10 °C without extra costs of electric energy.

4. CONCLUSION

Thus, the use of combined heat exchangers in agricultural drying plants will reduce energy intensity of drying processes not only for aerodynamic heating dryers, but also for dryers of other types using active stimuli of drying agent movement.

The dependence obtained (20) makes it possible to predict with sufficient accuracy the temperature of heated drying agent at the outlet of the combined heat exchanger of aerodynamic dryer taking into account its design parameters and environmental parameters.

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