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DESIGNING THE STRUCTURE AND DETERMINING THE MODE CHARACTERISTICS OF THE GRAIN DRYER BASED ON THERMOSIPHONS

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Energy consumption, environmental issues, product quality are actual problems related to grain drying processes. It is necessary to pay attention to designing new structures of energy-efficient grain dryers.

A structure of an energy-efficient grain dryer based on thermosiphons has been designed; its energy consumption is 3.5...6.8 MJ/kg depending on surface temperature and air flow rate. The dryer includes a layer heater, a drying chamber, a heat generator, a heater, a noria for loading the product, and fans. The structural features of the dryer allow the drying process to be carried out without direct contact between the combustion gases and the product.

The efficiency of the designed structure was evaluated for such indicators as heat transfer coefficients to the grain flow, specific energy costs, moisture content, the relative humidity of the air leaving the dryer.

The values of coefficients of the heat transfer to the grain flow vary within 36...58 W/m²K at speeds 2.5...8 mm/s. An increase in the flow rate by 3.2 times leads to an increase in the heat transfer coefficient by 1.6 times.

The moisture content of the air at the outlet of the dryer reaches 60 g/kg, while the relative humidity is 90 %, which is several times higher than the parameters for convective mine grain dryers.

Energy consumption for drying at the surface temperature of thermosiphons T_s=142.9 °C for various grain flow rates is close to a minimum. The energy consumption is lower than in existing convective dryers.

21 % is spent on heating grain in the dryer; 54 % – on moisture evaporation; and 23.6 % are losses. If we consider the energy spent on moisture evaporation usable, the efficiency of convective dryers is only 40 % while that of dryers based on thermosiphons is 54.1 %.

It is expected that the designed structure could be a solution for small farmers in the post-harvest drying process

Keywords: thermosiphons, grain drying, specific energy consumption, air parameters, heat transfer coefficients, environmental friendliness

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1. Introduction

As a result of population growth, the global demand for energy and quality food continues to grow. Energy consumption and quality in food production processes are important parameters that need to be carefully studied and analyzed.

Grain drying accounts for up to 25 % of industrial energy consumption in developed countries [1]. Freshly harvested crops usually possess high moisture content, which makes it impossible to store the grain immediately. Approximately 30 % of the grain must be dried after harvest [2].

The most popular dryers are mine convective dryers with the mixed flow. Convective dryers are typically associated with low thermal efficiency [3]. Convective mine grain dryers often use fuel combustion gases and directly guide them into the product. There is a risk of carcinogens getting into the grain.

Research on drying is a relevant task. To minimize grain losses, obtain high-quality products, reduce the burden on the environment, new designs of efficient grain dryers are needed.

2. Literature review and problem statement

Paper [4] considers techniques of intensification of processes when working with solids. In modern drying technologies, vertical thin-film dryers, dryers with a screw conveyor, microwave dryers operate effectively. The use of these techniques helps reduce the drying time, contributes to continuous processing, the uniform temperature distribution in the product. However, the considered dryers are not applicable for drying crops due to structural features.

In dryers, solar collectors, recirculation, and heat carrier drying are used to save energy [5]. For use in grain drying processes, such dryers have low productivity.

To obtain a pure drying agent, to solve the problems of penetration of carcinogens into the product, tubular heat exchangers are used. Exhaust gases are used as heat carriers [6]. In the tubes of the heat exchanger, the heat carrier moves, outside is the air flow going to drying. A dryer of the tray type of periodic action is considered in [6]. The disadvantage is that the use of such a dryer is possible only for small batches of grain. In work [7], the combustion gases of rice husks are used as heat carriers. The device shows high thermal efficiency. The disadvantage is the need for additional fans to pump the heat carrier.

Practically not considered are dryers that use water vapor as a heat carrier, applied for highly moist dispersed products, crops. The dryers provide high coefficients of heat transfer to the grain of 30...70 W/m^{2}K [8]. Disadvantages: complicated structure, additional devices are needed for steam supply, condensate drainage. Tubular steam dryers can be considered as analogs of dryers with thermosiphons (TSs), heat pipes (HPs).

It is expected that the use of TS in drying processes would help reduce energy consumption since TSs work effectively in such areas as cooling electronics and spacecraft, heat recovery [9].

TS eliminates the need for auxiliary devices or moving parts, in particular for pumping the heat carrier.

The use of TS, HP in drying processes is limited. Perhaps this is due to the high productivity of grain dryers (60...120 t/h). The use of heat exchangers of recuperators on TS in grain dryers is known [10]. Heat exchangers are used to recover heat and heat the air before drying. The use of TS for heating the grain flow is not considered.

HP is used in solar dryers. In work [11], condensation sections of HP were built into the solar collector of the dryer, which heated the air going to drying. The maximum air temperature at the outlet of the solar collector during the study was 118 °C. The average energy and exergetic efficiency of the solar collector ranged from 10 % to 30 %, and 1.9 % to 5.6 %, respectively. However, solar dryers provide low productivity if they are used when drying crops.

Paper [12] reports the structure of a screw dryer based on a ring HP. It was used in the drying of ceramic raw materials. The overall heat transfer coefficient was 35 % higher than that of a conventional steam jacket dryer, due to the thermal conductivity of the ring heat pipe. The designed dryer is compact and saves energy in drying processes. Its disadvantage is the use of screw devices in drying processes that could lead to injury to the grain.

The efficiency of the drying process is assessed by the higher relative humidity of the exhaust air φ_2. If the relative humidity is low, this indicates that there is unused thermal energy left in the air. In experiments [13], the relative humidity φ_2=45...75 %. Mine convective dryers have low rates in this regard.

The moisture content of the exhaust air d_2 is the most appropriate quantity to describe the performance of the dryer, although it says little about energy efficiency. In experiments [13], for a mine grain dryer, d_2=21...23 kg/kg; temperature, t_2=70...85 °C. This indicates a low driving force of the drying process and is due to the fact that in convective dryers, the air is used simultaneously as a heat- and moisture carrier. An option to overcome such problems may be the transition from convective heat transfer to conductive and the use of air only as a moisture carrier.

An indicator of the energy efficiency of dryers is the specific energy consumption q, MJ/kg. For adiabatic drying, paper [14] gives values from 3.5 to 4 MJ/kg. For the most common mine dryers, this figure is several times higher, which is a drawback.

It is necessary to pay attention to designing new energy-efficient structures of grain dryers, which would ensure the production of a quality product (without carcinogens). The use of TSs in drying processes is limited, although they show high efficiency in electronics and the chemical industry. It seems promising to design a grain dryer based on TS. One must evaluate the operation of the designed structures for such indicators as heat transfer coefficients to the grain flow, specific energy costs, as well as indirect parameters, such as moisture content and the humidity of the air leaving the dryer.

3. The aim and objectives of the study

The purpose of this work is to design a pilot industrial sample of an energy-efficient grain dryer based on TS, to test its performance. This would make it possible to reduce the specific energy consumption during drying through the use of highly efficient heat pipes – thermosiphons; to avoid contact of flue gases with grain; to improve the environment during drying.

To accomplish the aim, the following tasks have been set:

– to determine the influence of the grain flow rate on the heat transfer coefficient to the grain flow;

– to determine the effect of air flow rate on the temperature of the grain flow and the drying rate;

– to define the characteristics of the humid air at the outlet of the dryer;

– to determine the specific energy costs and distribution of thermal energy in the grain dryer.

4. The study materials and methods

4. 1. Design of a setup

A diagram of the grain dryer is shown in Fig. 1. The dryer includes a layer heater 1, a drying chamber 2, a heat generator 3, heater 4, noria 5 for loading the product, fans 6 and 7 (Fig. 1, 2).

The dryer works as follows. During the combustion of fuel, combustion gases are formed in heat generator 3. Fan 6 induces suction. Due to the created vacuum, the combustion gases enter the first layer heater 1, then heater 4. At the same time, more complete use of the energy of combustion gases is achieved. In the layer heater, about 40 % of the energy of the heat carrier is used, in the heater – 60 %. The air in the heater is heated from the ambient temperature to 60 °C. The grain fed by noria 5 enters layer heater 1 and is heated to the drying temperature. The grain in layer heater 1 and the drying chamber 2 moves in a dense gravitational layer. Fan 7 operates to pump. It supplies air from the environment to heater 4. The heat from the combustion gases is transferred to the air from the environment. The heated air is supplied to the drying chamber 2, saturated with moisture from the product, and discharges through channels on the chamber body in the form of a steam-air mixture.

The flows of grain and heat carrier are separated by a partition consisting of metal lattices, the space between which is filled with thermal insulation.

The layer heater includes a gas duct 1, a grain heating chamber 2, smooth thermosiphons 3 consisting of a finned evaporation section 4, and a condensation section 5. The thermosiphons are arranged in a staggered manner (Fig. 3, *a*) to ensure compact installation and efficient heat exchange. The flows of the grain and heat carrier are separated by wall 6, inside of which there is thermal insulation 7.

Based on the devised procedures and structural solutions, a prototype of a layer heater was fabricated. Its photographs are shown in Fig. 3, *b*.

The drying chamber 2 includes two parallel mines, between which pressure distribution chambers are located. In the mine, there are boxes 8 with a length of 1000 mm, a height of 125 mm, a width of 100 mm. The step of the location of the boxes: horizontally, 200 mm; vertically, 200 mm. In each row, there are 9 boxes, and the supply and outlet boxes alternate. The thickness of the blown layer of grain is 220 mm. The minimum passage for grain is 100 mm. Each mine includes three modules with overall dimensions of 2010×955×955 mm.

The thermosiphon heater consists of several rows of finned thermosiphons 1 enclosed in the body and divided by a partition into 2 zones. Combustion gases pass through thermosiphon evaporators, and air from the environment passes through condensers (Fig. 4).

The use of modern heat transfer means (two-phase thermosiphons) in the heater makes it possible to organize an effective countercurrent movement of gas flows with the transverse flow around the finned pipes.

Technical characteristics of some elements of the dryer are given in Table 1.

The principle of operation of the MDGG gas burner, which is used in the heat generator, is based on the jet mixing of natural gas with air and the formation of a combustible mixture on the section of the pylons stabilizing the process. The micro diffusion process of gas combustion proceeds in a short flare, providing a uniform temperature field in the combustion chamber of the thermal unit and low environmental performance.

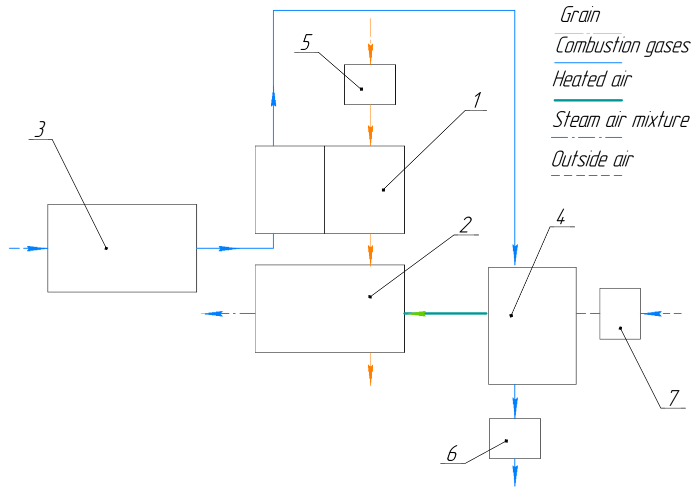

Fig. 1. Diagram of a grain dryer based on thermosiphons: 1 — layer heater; 2 — drying chamber; 3 — heat generator; 4 — heater; 5 — noria; 6 — fan; 7 — fan

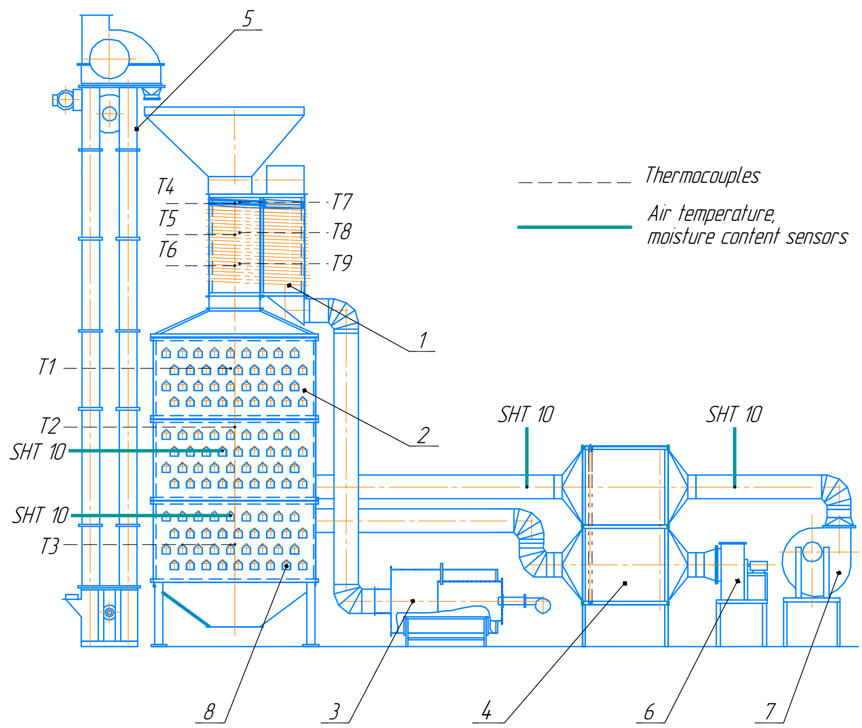

Fig. 2. Structure of the grain dryer based on thermosiphons: 1 — layer heater; 2 — drying chamber; 3 — heat generator; 4 — heater; 5 — noria; 6 — fan; 7 — fan, 8 — box

Table 1

Technical characteristics of dryer elements

No. of entry	Equipment type	Specifications	
		G, m^3/h	N, kW
1	Fan VC-14-46 No. 4	8000	4
2	Fan VC 14-46-5	20 000	15
3	Gas block heat generator MDGG-25 BA-20R-SUSH-01	–	250

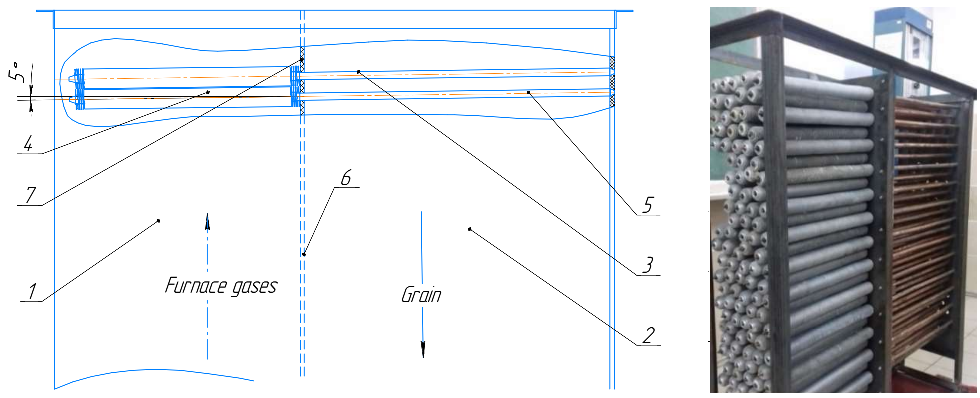

Fig. 3. Structure of the layer heater: *a* — diagram; *b* — photograph; 1 — gas duct; 2 — grain heating chamber; 3 — thermosiphon; 4 — evaporation area; 5 — condensing section; 6 — dividing wall; 7 — thermal insulation

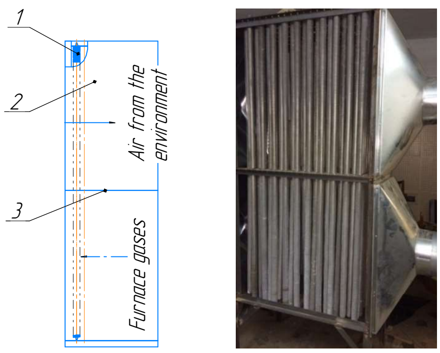

Fig. 4. Thermosiphon heater structure: *a* — diagram; *b* — photograph; 1 — finned thermosiphon; 2 — body; 3 — dividing wall

Fresh wheat from the farm was used as a drying material. The harvesting humidity of wheat was 22...23 %, it was dried to a conditioned humidity of 14 %. The mass flow rate of the grain ranged from 75 kg/h to 550 kg/h. The grain flow temperatures used were 60 °C and 70 °C. The volumetric flow rate of drying air ranged from 45 m^3/h to 185 m^3/h.

4. 3. Research methods

4. 3. 1. Methods of heat transfer research

To determine the coefficients of heat transfer from the surface of TS to the grain flow in the layer heater, the following experimental technique was used.

The product was loaded into the dryer, the test time was measured using an electronic stopwatch (Flotti F108, China).

Adjustment of grain consumption was made using a gate. Using a folding tray, a sample of the product was taken over time τ. Grain consumption was kept constant. The sample was taken three times throughout the experiment. The sample was weighed on a laboratory electronic scale (ER-Plus-06, Technovagi, Ukraine). The weight of the product *m* was in kg. Mass flow rate G_g (kg/s) was determined from the following formula:

$$G_g = \frac{m}{\tau}. \tag{1}$$

In the tests, the temperature of the grain flow was measured in two ways.

In the first case, the temperature of the grain flow at the inlet (t_i) and outlet (t_o) of the layer heater was determined, after which the temperature value was averaged. Samples were taken from the grain flow at the inlet and outlet to average the temperature. Determining the sample temperature during the drying process was carried out with thin chromel-copel thermocouples with a diameter of 0.18 mm. Samples for determining the temperature of the grain were contained in the Dewar's vessel in order to take into consideration the inertia of the thermocouples. The thermocouples were introduced into the center of the tank. The average temperature of the grain was determined from the formula:

$$t_{av} = \frac{t_i + t_0}{2}. \tag{2}$$

In the second case, the thermocouples were placed according to the height of the layer heater (Fig. 2). Several thermocouples (T7–T9) were used to measure the temperature of the grain flow. The data from the primary temperature converters entered the ADC, were converted into a digital signal, and entered into the PC. The data recording interval is 10 s. Measurement was carried out with chromel-copel thermocouples, the signal was transmitted to the Arduino ADC module (Arduino Software, Italy) and the computer.

The surface temperature of thermosiphons T_s was measured at intervals of 100 s. The surface temperature of the thermosiphons was determined by thermocouples (T4–T6) stamped into the surface.

The heat exchange process was studied at grain humidity close to the equilibrium humidity of the surrounding air at a given temperature.

The heat flux transferred to the grain was determined from the formula:

$$Q_h = G_g \cdot C \cdot (t_o - t_i), \tag{3}$$

where *C* is the heat capacity of wheat grain, J/kgK [15].

The coefficient of heat transfer from the condensation sections of the thermosiphons of the layer heater to the grain flow was determined from the formula:

$$\alpha = \frac{Q_h}{F \cdot \Delta t_{av}}, \tag{4}$$

where *F* is the surface area of the condensing sections of TS, m^2.

The parameters of the ambient air were determined using the psychrometric hygrometer VIT-2 (Steklopribor, Ukraine).

The temperature of the grain flow in the mine was determined by thermocouples T1–T3 located at the height of the mine (Fig. 2).

4. 3. 2. Determining grain moisture

The moisture content of the grain ω_g (%) was determined by the digital moisture meter Wile 65 (Farmcomp, Finland; accuracy, 0.5...1.0 %).

Grain samples of the same volume were taken, the samples were hermetically packed. Humidity was measured by the moisture meter when the temperature of the samples became the same. Thus, the effect of temperature error in measuring humidity was reduced.

The test was carried out by a weight method [16].

4. 3. 3. Determining the parameters of the air leaving the dryer

To determine the parameters of the air coming from the dryer, the digital humidity and temperature sensor SHT 10 (Sensirion, 4.5 % RH) was used.

The range of experimental studies is given in Table 2.

Table 2

Range of experimental studies

Air flow rate, L, m^3/h	25...75
Grain flow rate, G_g, kg/h	550
TS surface temperature, °C	120

The interface of the ADC converter program was designed in such a way that the change in air parameters over time was reflected in the form of graphical dependences under an online mode. The use of such a scheme, together with the use of frequency adjustment of the fan drive, made it possible to quickly intervene in the course of the test and simplified its plan.

4. 3. 4. Specific energy costs for drying dispersed products

Determining the energy efficiency of the drying process was carried out according to the following methodology.

Specific energy costs for the drying process q, J/kg, were determined from the formula:

$$q=\frac{Q_t}{W}, \tag{5}$$

where W is the amount of evaporated moisture, kg.

The total consumption of thermal energy for drying Q_t, J was determined from the formula:

$$Q_t=G_f\cdot Q_l^c, \tag{6}$$

where G_f is the mass of the fuel, kg; Q_l^c is the lowest heat of combustion of fuel, J/kg.

Gas block heat generator (Fig. 2) runs on natural gas. The amount of gas was determined using the gas metering system "FLOWTEK-TM-1-3" (UKRGAZTECH, Ukraine).

To determine the distribution of heat in the drying process in a dryer based on thermosiphons, an equation of heat balance was built:

$$Q_t=Q_h+Q_a+Q_e+Q_l, \tag{7}$$

where Q_h is the heat consumption for heating the grain, J;

Q_a is the heat consumption for air heating, J;

Q_e is the heat consumption for water evaporation, J;

Q_l is the loss of heat to the environment, J.

The heat consumption for heating the air Q_a was determined from the formula:

$$Q_a=L\cdot(I_1-I_0)\cdot\tau_d, \tag{8}$$

where I_0, I_1 is the enthalpy of air before and after the heater, J/kg, it was determined from the I-x Ramzin diagram based on data from SHT-10 sensors (Fig. 2);

$L=\upsilon\cdot S\cdot\rho$ is the air consumption for the drying process, kg/s;

υ is the air velocity, determined using the thermoanemometer TTM-2-04-02 with analog and digital output signal (ZAO Prompribor, Ukraine);

S is the cross-sectional area of the channel, m^2;

τ_d is the drying time, s;

ρ is air density, kg/m^3.

The loss of thermal energy to the environment from the surface of the drying chamber Q_l was determined from the formula:

$$Q_l=\sum Q_{li}, \tag{9}$$

where thermal energy loss from the i-th surface:

$$Q_{li}=\alpha\cdot F_{ch}\cdot(t_{si}-t_0)\cdot\tau_d, \tag{10}$$

where F_{ch} is the area of the drying chamber, m^2;

t_0 – ambient air temperature, °C;

t_{si} is the temperature of the i-th surface of the drying chamber, °C (determined by the infrared thermometer Fluke 64 MAX (USA);

α is the heat transfer coefficient from the heated surface to the environment, W/m^{2}K:

$$\alpha=9.74+0.07\cdot(t_{si}-t_0). \tag{11}$$

The energy consumption for evaporation of moisture from the material Q_e was determined from the formula:

$$Q_e=W\cdot r, \tag{12}$$

where r is the heat of vaporization at the temperature of the material, J/kg.

5. Results of studying the dryer based on thermosiphons

5. 1. Results of studying the heat transfer from the condensation areas of layer heater thermosiphons to grain flow

The coefficient of heat transfer α, W/m^{2}K, from the condensation sections of the thermosiphons of the layer heater to the grain flow was determined. The dependence of the heat transfer coefficient on the average grain flow rate υ_g, mm/s, at the TS surface temperature T_s=142.9 °C was established (Fig. 5). With an increase in the average grain flow rate, the value of the average heat transfer coefficient on the TS surface increases. This is due to the flow conditions of the pipe surface flow.

In the dryer, the values of the heat transfer coefficients vary within 36...58 W/m^{2}K at an appropriate flow rate of

2.5…8 mm/s and the diameter of the smooth part of TS of 12 mm.

In the experiments, wheat with equilibrium initial humidity was heated at given initial temperature and humidity. With an increase in the temperature of the grain flow, the equilibrium in the grain-air system was disturbed. There was the evaporation of a small amount of moisture from the surface of the grain. The humidity changed from 14 to 12 % with a change in the temperature of the grain flow from 30 to 60.5 °C. The values of the heat transfer coefficients (Fig. 5) were obtained taking into consideration the amount of heat spent on the phase transition.

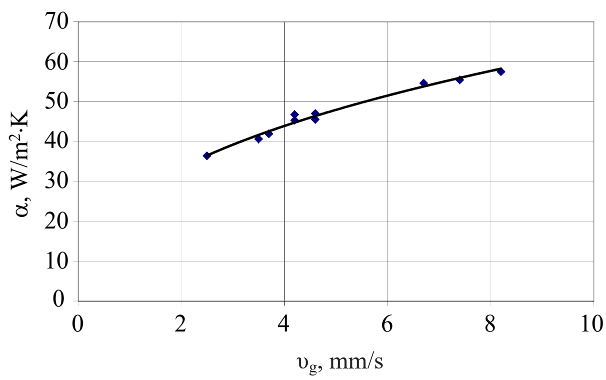

Fig. 5. Effect of average grain flow rate on heat transfer coefficient

The capabilities of the experimental installation did not make it possible to increase the grain flow rate above 8.2 mm/s. Increasing the flow rate in the grain dryer leads to a significant increase in the heat transfer coefficients. In the layer heater, TSs are set at an angle of 5°, the values of the heat transfer coefficients when flowing around the inclined pipe are higher due to more complete use of the pipe surface, mixing the flow. According to [8], TSs work most efficiently at an angle of inclination of 60°, but, due to the design features of the layer heater, it was not possible to use such an angle.

5. 2. Results of studying the effect of air flow rate on the temperature of the grain flow and the drying rate

For experiments in which the grain flow rate G_g=75 kg/h was maintained, the TS surface temperature T_s=142.9 °C, the air flow rate was changed from 45 m^3/h to L=185 m^3/h. The temperature of the grain flow in the experiment was defined as the arithmetic mean between the readings of thermocouples located according to the height of the drying mine. The following thermograms and drying curves were built (Fig. 6, 7). The temperature of the grain flow increased from 40 to 80 °C. The temperatures of the grain flow inside the mine were evenly distributed. There are no sharp differences in the readings of thermocouples.

When the flow rate increased to L=185 m^3/h, the following data were obtained (Fig. 7).

The temperature of the grain flow increased from 40 to 55 °C. The grain does not overheat to a temperature of 80 °C as in the experiment with air flow rate L=45 m^3/h. The drying rate increases (Fig. 7). Air captures more moisture, which evaporates from the surface of the grain.

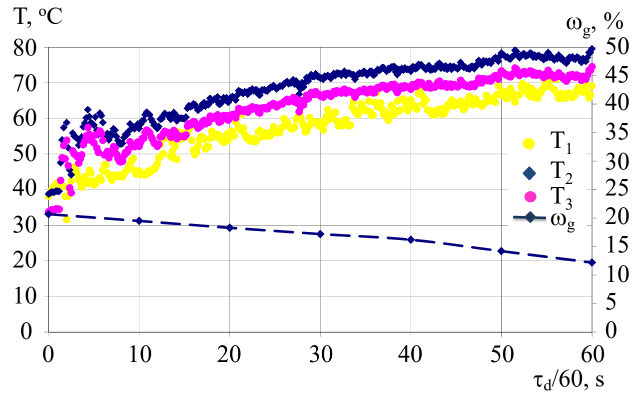

Fig. 6. Thermograms, drying curves at L=45 m^3/h
T_1, T_2, T_3, °C – grain flow temperature;
ω_g – grain moisture content

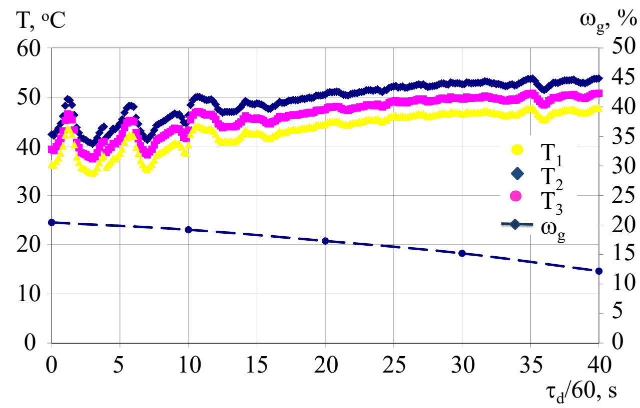

Fig. 7. Thermograms, drying curves at L=185 m^3/h
T_1, T_2, T_3, °C – grain flow temperature;
ω_g – grain moisture content

5. 3. Results of studying the characteristics of moist air at the outlet of the dryer

In the experiments, the grain flow rate G_g=550 kg/h was maintained. The moisture content of the air at the outlet of the dryer d_2 at the beginning of the drying process about 1000 s) is slightly higher than the ambient air parameters (d_0=10 g/kg). The state of the air corresponds to the period of warming up of the material without removing moisture from its surface (Fig. 8). The maximum enthalpy of air at the outlet of the dryer is I_2=213 kJ/kg.

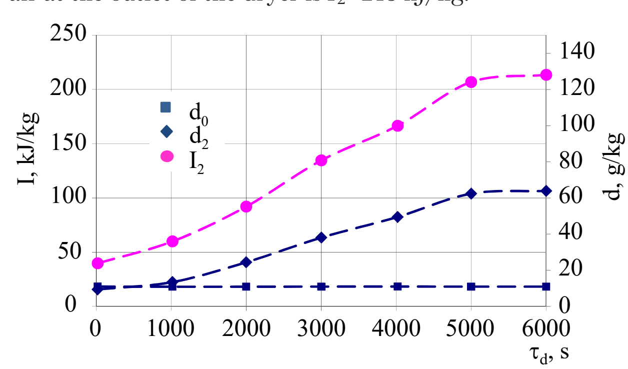

Fig. 8. Air parameters at flow rate L=25 m^3/h:
d_0, g/kg – moisture content of ambient air;
d_2, g/kg – moisture content of air at the outlet of the dryer;
I_2 – enthalpy of air at the outlet of the dryer

When analyzing the drying curves, one notices that there is a period of warming up of the material when its humidity practically does not change (about 1000 s). Further, the drying rate is constantly increasing (Fig. 9).

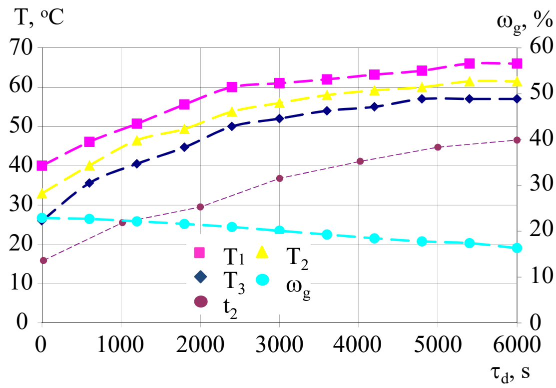

Fig. 9. Drying curves and thermograms of wheat grain G_w=550 kg/h: T_1, T_2, T_3, °C – grain flow temperature; t_2, °C – air temperature at the outlet of the dryer; ω_g – grain moisture content

The drying process took place at an average grain flow temperature of 60 °C, which meets the technological requirements. Grain moisture content is reduced from 23 % to 16 %.

5. 4. Results of studying the specific energy costs and distribution of thermal energy in the grain dryer

Specific energy consumption q, MJ/(kg of removed moisture), was defined as the ratio of the total energy expended to the amount of moisture removed. The impact on the amount of energy consumption is exerted by the surface temperature of TS, the air velocity in the intergrain space, the grain flow rate (Fig. 10).

The energy consumption for drying at the TS surface temperature T_s=142.9 °C for grain flow rate G_g=550 kg/h is approaching a minimum. In addition, an increase in the air velocity and grain flow speed also leads to an intensification of the drying process and a decrease in energy costs. In a series of experiments at an air flow rate of L=185 m^3/h, energy consumption per process is minimal.

The distribution of thermal energy in the h designed dryer was determined (Fig. 11). An analysis of the influence of each parameter on the distribution of energy in the drying process was carried out. 21 % is spent on heating grain in the dryer, 54 % – on the evaporation of moisture, and 23.6 are % losses.

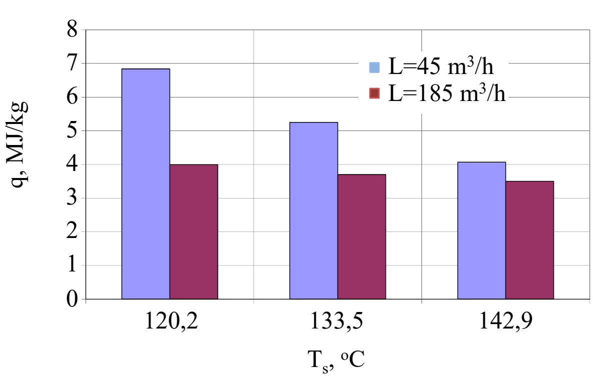

Fig. 10. Specific energy consumption of the dryer

If we consider the energy spent on moisture evaporation usable, the efficiency of convective dryers is only 40 % (Fig. 11). Compared to convective drying, in a dryer based on thermosiphons, most of the energy (54.1 %) is spent on moisture evaporation.

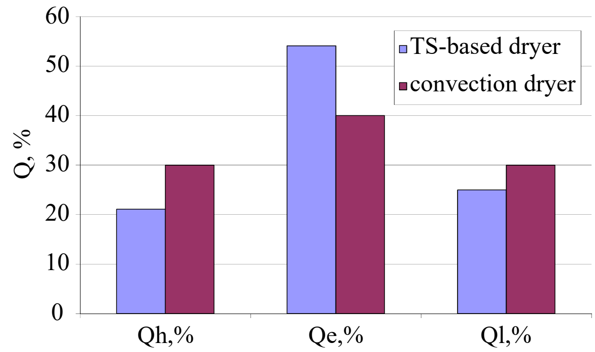

Fig. 11. Distribution of thermal energy in grain dryers

To minimize the specific energy consumption of the grain dryer, it is necessary to search for the optimal ratio of TS surface temperature, air consumption, grain flow rate.

6. Discussion of results of drying grain in the dryer based on thermosiphons

The structure of the dryer was designed, its effectiveness was confirmed by the results of our research.

The structural features of the dryer allow the drying process to be carried out without direct contact between the combustion gases and the product.

In the layer heater, TSs are set at an angle of 5° (Fig. 3), the values of the heat transfer coefficients when flowing around the inclined pipe are higher due to more efficient use of the pipe surface. Compared to the data reported in [8], the heat transfer coefficients are at the level of tubular steam dryers. The structure does not have devices for supplying steam or removal of condensate, which reduces energy costs.

Compared to mine grain dryers, the air at the outlet of the grain dryer based on TS has a high moisture content. According to [13], in experiments for convection grain dryers of mine type, the moisture content d_2=21...23 g/kg, the temperature t_2=70..85 °C, the relative humidity ϕ_2= =45...75 %. In the grain dryer based on TS, the moisture content d_2=50...60 kg/kg (Fig. 8), the temperature t_2=40...45 °C (Fig. 9), the relative humidity ϕ_2=80...95 %. That indicates a higher drying speed. This is due to the fact that the air in the dryer with TS is used as a moisture carrier, the grain flow is heated by the contact of the wet grain and the surface of TS. Thus, the driving force of the process is higher, the air can take on more moisture.

The high energy efficiency of the TS-based dryer can be explained by the efficiency of TS as a heat conductor with very low thermal resistance, as well as the absence of the need to pump the liquid compared to active heating methods. The process of transferring heat through the surface of the pipe to the grain flow in a layer heater occurs under conditions of partial condensation of water vapor in the volume of the pipe. This demonstrates serious advantages over convective heating of grain.

Our dryer is designed for a relatively small capacity (12 t/h) while the capacity of the dryers at industrial elevators reaches 200 t/h. Accordingly, it is better to use dryers of this type for small farms, for drying seed grain. The grain flow is a strong abrasive. Perhaps, with the prolonged operation, wear of the surface of the layer heater vehicle would occur. It is necessary to thoroughly clean the grain before drying because it is possible to clog the space between the tubes of the layer heater.

The current study may be advanced by optimizing the operating modes of the designed structure. To minimize the specific energy consumption of the grain dryer, it is necessary to search for the optimal ratio of the surface temperature of TS, air consumption, grain flow rate.

7. Conclusions

1. The influence of grain flow rate on heat transfer coefficients in a TS-based dryer has been determined. The values of the heat transfer coefficients vary within 36...58 W/m^{2}K at grain flow rates of 2.5...8 mm/s, the diameter of the TS pipe is 12 mm. An increase in the flow rate by 3.2 times leads to an increase in the heat transfer coefficient by 1.6 times.

2. The characteristics of the humid air at the outlet of the dryer have been determined. The moisture content of the air reaches d_2=60 g/kg while the relative humidity reaches ϕ_2=90 %, which is several times higher than the air parameters for convection mine grain dryers [16]. This indicates a higher drying rate and efficient use of energy.

3. The influence of air flow rate on the change in the temperature of the grain flow and the drying rate has been determined. An increase in the air flow rate by 4.1 times leads to a decrease in the temperature of the grain flow by 1.4 times. This is due to the intensification of heat and mass transfer by increasing the air velocity in the intergrain space. The drying speed increases by 1.5 times.

4. The influence on the amount of energy consumption in the dryer is exerted by the surface temperature of TS condenser, the air velocity in the intergrain space, as well as the grain flow rate. The energy costs for drying at the TS surface temperature T_s=142.9 °C for various grain flow rates are close to a minimum. The energy consumption is lower than in existing convection dryers [14]. 21 % is spent on heating grain in the dryer, 54 % – on moisture evaporation, and 23.6 % are losses.

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