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ENERGY SAVING THERMAL SYSTEMS ON THE MOBILE PLATFORM OF THE MINI-BAKERY

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Abstract: Evaluates energy resources in the production of bakery products on a moving platform of a mini-bakery, with the possibility of recuperating heat flows based on their redistribution with the creation of local heating zones. The final acceptable conditions were established and a heat and mass transfer model was presented for the possible control of the energy potential of flue gas emissions and the temperature of water at the outlet of the heat exchanger in constant time.

KEYWORDS: heat flow, ecological and economic efficiency, flue gases, energy resource, heat recovery system.

1 Introduction

Today, the efficiency of using thermal energy is a very pressing issue and it is becoming more relevant every day. Efforts are being made all over the world to find new technological and technical constructive solutions to ensure a highly economical way of using energy resources. Their direction is to reduce the consumption of fuel and energy resources. Therefore, solving the problems of the use of energy resources is an extremely important task for Ukraine. Today, for the industry of Ukraine to survive and work effectively, it is necessary to direct development to the path of energy efficiency. Undoubtedly, optimization of the heat of combustion of products remains a rather important task of industry, especially in the bakery industry. Considering the fact that more than half of the total heat during baking is spent on the evaporation of moisture from dough products, therefore, when saving fuel, there must be ways to ensure optimal baking thermal regimes without changing the characteristics of the kinetics of heat distribution in the products.

Heat loss largely depends on the fuel burning in the ovens of the bakery ovens, on the rhythm of loading the furnace, and the range of products. According to various data [1 - 3], heat losses make up 9-20% of the produced heat. Therefore, today the main direction of reducing the heat loss of the initial combustion products is possible due to lowering their temperature. The way of recovery of thermal energy, i.e. reuse of the heat of smoke emissions, is quite well known. At many enterprises of the food industry, especially bread factories, thermal energy recovery is carried out by reusing the heat of smoke emissions with the help of a heat exchanger. They are mainly used to heat the air of the fuel mixture, which allows to return part of the heat to the heat generator. The heat of flue gases can be used to heat water for technological needs, heat air for

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combustion, pre-heat the water return network in the hot water supply system, etc. Contact and surface types can be used as heat exchangers. The use of a surface heat exchanger allows the flue gases to cool down to a lower temperature (less than 40 °C). There are various technologies for reducing the temperature of exhaust gases, which can be reduced to 65-70 °C due to the use of latent heat.

In many bakery ovens, the actual flue gas temperature is 210-230 °C and there is no provision for further lowering of the flue gas temperature. Taking into account the main direction of the development of the energy industry to increase the efficiency of the use of natural fuel, it is possible to reduce the temperature of the emission of flue gases at the bakery oven. Therefore, lowering the flue gas temperature becomes very important.

2 Literature review

Ways of heat exchange intensification to obtain positive effects have been known for a long time. But there is little information on the implementation of such technologies in bakery ovens, and there are practically no data on mini-bakeries. Scientific literature [4, 5] practically does not reflect this problem, and if there is, it is mostly not systematized regarding the methods of intensification.

Modern mini-bakeries mainly use rotary and dead-end ovens. The main direction of their possible energy saving is the allocation of waste gas heat utilization. This direction of reducing losses is effective at low capital investment costs and economical for the use of thermal energy in the technology of production of bakery products. Especially, in our opinion, this applies to the proposed new design of a mobile mini-bakery on wheels [6].

The production of bakery products in a mini-bakery, according to the technological process, consumes a significant amount of potential thermal energy, while the technology itself is a source of synthesis of potential energy flows. The heat potential generated by the activity of the mini-bakery depends significantly on its temperature state and, unequivocally, determines heat loss by scattering into the environment.

In the previous work of the authors [6, 7], the problems of ensuring the efficiency of the mini-bakery were considered, the low energy efficiency of obtaining thermal energy and its high cost due to the lack of connection to the energy source in field conditions were determined. Such heat carriers include the state of the heated environment near the bakery oven, water heating tanks, and smoke pipe with a temperature of 210-245 ^{0}C.

With this in mind, today it is possible to use secondary heat resources in various combinations to create an energy-efficient heat system. For most cases, there are schemes for obtaining heat during heat supply, which provide for the solution of a certain sequence in the technology. In addition to the sequence, there is compliance with the interaction of energy and material flows and their use due to transformation among themselves. This complex of interactions should provide for the minimization of energy costs.

Solid fuel is the energy potential of incoming raw material flows, in the process of transformation it becomes the potential of flue gases. Recovery of secondary energy resources should be solved with parallel synchronized flows of the heat system of the bakery oven. When implementing phase transitions, it is effective to use heat exchange processes on the surface of the smoke pipe of the bakery oven.

The surface of the smoke pipe of flue gas emissions is a source of flow, which must be considered as a secondary source when setting operating parameters. The use of the potential differences of the flue gas flow is the completion of the work cycle and means the inevitable loss of the energy potential of the system to the environment. The only solution to limit the loss of

such potential is the use of heat exchangers. After all, the properties of heat exchangers provide contact with transformations of energy potentials in a wide range of temperatures. This allows you to use the energy potential of the stove, which is ultimately spent as heat.

However, in cases of multi-stages associated with a change in the assortment of bakery products, there is a clear dependence of the effects of various factors, as they take place in a thermal and temporal combination or in several components. When analyzing the information posted on the Internet sites of heat exchanger manufacturers, when they were examined, it was shown that to intensify the heat exchange in the gas-tube heat exchanger, the mode movement of the heat carrier, geometric parameters are often used. Here we note that not all of the specified information and literary sources have calculated dependencies of the evaluation of the heat exchange efficiency in the heat exchanger. Therefore, the theory of experimental planning is often used [4, 5] in combination with calculations of the obtained dependence. Thus, methodological approaches to the development of the rational use of heat flows of flue gases in the baking of bakery products are based on a systematic analysis of the technology and design of the bakery oven [7, 8].

The analysis of literary sources showed that the authors focus their research in a significant amount on the topic of possible transformations of energy potentials. There are scientific publications aimed at the effective operation of devices when applying technological innovations for heat recovery of flue gases with the use of high-quality regulation systems. Based on the analysis of the heater, the authors [8] established the main input, disturbing factors and output parameters of control and automatic regulation on the developed information structure, as a cybernetic system. The author [9] in his studies of regeneration and recovery of spent heat gave an assessment of the potential of high-temperature devices, and the author [10] added the prospects of recovery of low-temperature thermal energy emissions.

The utilization of the heat of the flue gases of the bakery oven can be used in contact with the water heater. The recovery of thermal energy in the reuse of the heat of smoke emissions, the heat of exhaust air, the use of heat pipes at geothermal sources is reflected in works [11]. More detailed information is provided in [12].

The authors of [13] logically and consistently supplemented the capabilities of heat exchangers with the help of heat recovery in drying units of the food processing industry. In the work of the authors [14], it was noted that the systematic approach to the recovery of industrial thermal waste can be evaluated by the level of reduction of the impact on the ecology of the environment. The authors [2, 15] reveal in detail ways to ensure a reduction in specific fuel consumption, which is possible by 20-25% when burning gas or fuel oil, 35% when using low-calorie coal, wood and bioresources of local origin.

The basis of the method of calculating recuperative heat exchangers is the determination of the necessary heat exchange surface to ensure the given heating temperature. In their research [16], the authors proposed a verification method for calculating a metallic tubular recuperator of given structural dimensions, based on the method of successive approximations. In practice, a circumstance often arises that allows the geometrical parameters of a recuperative heat exchanger to be determined by the place of its installation. The task of its calculation is reduced to determining the conditions of heat exchange, the final temperatures of heat carriers under optimal designs of the heat exchanger. At the same time, the authors [14 - 18] did not explain the method of finding the value of the temperature pressure, without which the determination of the heat transfer coefficient of the heat exchanger is impossible.

A study of the use of energy saving at enterprises in the bakery industry showed a low potential for savings [1], which is 30-45% on average. A significant share of the loss of energy

resources is due to the release of flue gases (heat) from industrial bakery ovens. Flue gases create an additional negative impact on the environment.

Therefore, the purpose of the work is to establish the design parameters of the heating system depending on the temperature of the exhaust gases and the productivity of the furnace during its optimization of technical and economic performance indicators.

3 Materials and method

Characteristics of heat recovery. The developed bakery is a complex of necessary equipment for baking 1200 kg of products. The complex is placed in a container on a moving platform of a car. The main investigated unit of technological equipment is the developed wood-fired bread oven with a heat exchanger mounted on the chimney. The laboratory installation and general scheme of heat recovery for heating water from low-temperature flue gas emitted from the furnace is shown in Fig. 1.

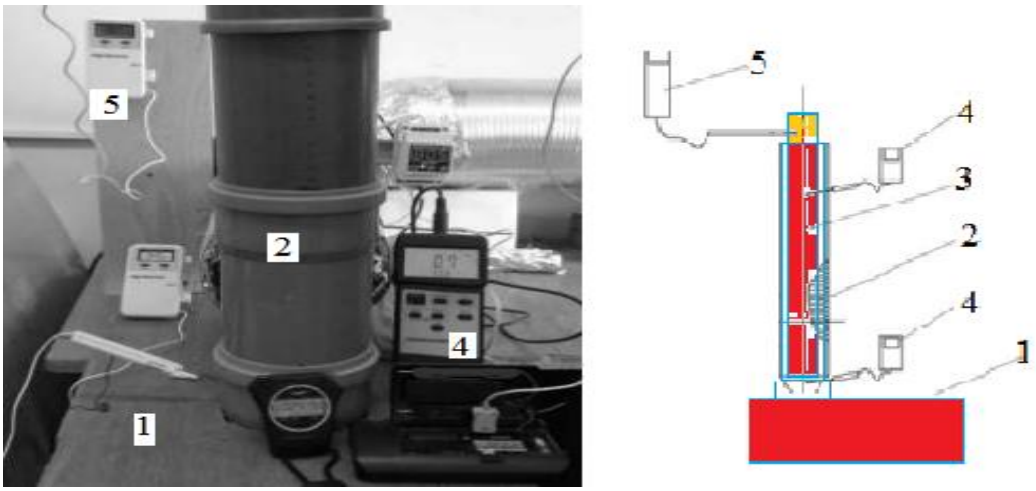

Fig. 1. Installation for experimental research of heat exchange of flue gas emissions and water heating: a) - general view: b) - installation diagram: 1 - bread oven; 2 – heat exchanger; 3 – smoke pipe; 4 – thermometer; 5 - anemometer

The proposed recuperative heat exchanger (Fig. 2. 3-D model) works in periodic or stationary thermal mode. It is a container of small capacity, where after certain intervals of time, water is filled and flue gases heat it up, going out into the atmosphere. The whole process of baking bread with the construction of the geometry of the heat exchanger allows you to trace the temperature distribution of the smoke chambers, the speed and heating of the water in the heat exchanger by the surface of the smoke pipe. Figure 2 shows the scheme of the experimental study of heat transfer and the 3-D model of the heat exchanger.

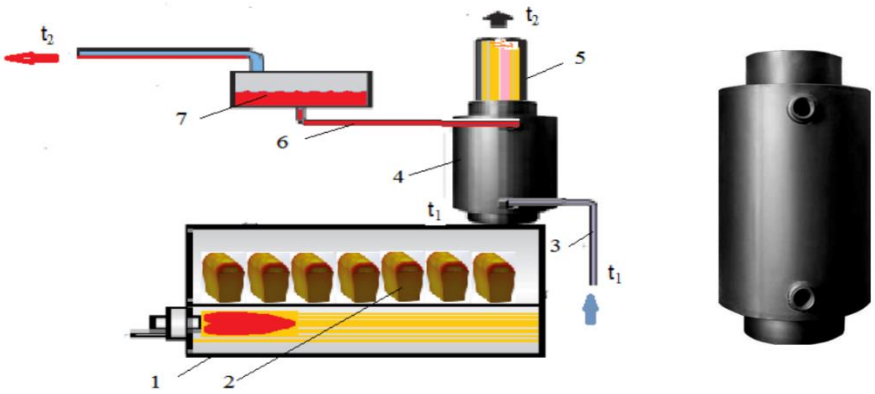

Fig. 2. Scheme of recuperation of the researched bread baking process and 3-D model of

the heat exchanger: 1- bread baking oven; 2 – bread; 3 – cold water pipeline; 4 – heat jacket; 5 – smoke pipe; 6 – heated water pipeline; 7 - water collector

For the study of the process, the maximum permissible conditions were set, namely: the temperature of the coolant at the entrance to the heat exchanger, which was equal to 220 ^{0}C, the pressure of the external environment 101325 Pa, the temperature of the water at the entrance - 20 ^{0}C and the temperature of the wall of the surrounding medium on both sides (20 ^{0}C).

All research, both theoretical and experimental, was carried out in the laboratories of the departments of automotive technology and food technology equipment in the city of Ternopil, the Ivan Pulyu State Technical University (Ternopil National Technical University) and Mykulinetsky Brewery LLC.

Devices and tools. The heat exchanger is equipped with measuring devices for the experimental study of the heat exchange of flue gas emissions. When measuring the volumetric flow of flue gas, a metal tubular rotary flowmeter is used, which is located at the outlet of the flue. Pt100 thermocouples were used to measure flue gas temperature at the inlet and outlet. Similarly, Pt100 thermocouples were used to measure the water temperature at the inlet and outlet of the heat exchanger. The Benetech GM533A device was used to determine the temperature regimes in the bread making process. A Pitot tube [19] and a differential manometer HT-1890 were used to determine the heat flow. Some of the above-mentioned measuring instruments were provided by the confectionery factory “Tera". The measurement range and error of the devices are shown in Table 1.

Table 1. Device parameters

Device	Model	Range	Errors
Metal rotary flowmeter	CGYL-LZ-25	0–30 l/min	0,3 l/min
Glass rotary flowmeter	LZB-6WB	0–20 l/min	0,3 l/min
Thermocouple pressure sensor	CGYL-202 –50	–50 до 50 kPa	0,25 kPa
Pt100	SWB-B	0-100 ^{0}C	0,25^{0}C

Description of the experiment and model of heat and mass transfer. Before the technological operation of proofing the dough blanks, the oven was ignited with firewood. During the period of heating and baking of bread in the oven, the temperature of the outgoing gases and the velocity were measured. The speed of gas movement and water temperature changed over time in the ring channel. The same measurements were carried out in the heat exchanger, where there was water with an initial temperature of 20 ^{0}C and a final temperature of 68 ^{0}C at the end of baking bread.

In the process of baking products, latent heat of the oven surface and flue gases are released. Waste heat of gases (hidden heat) is realized in the heat exchanger as a synchronous process of recuperation of water heating by flue gases. The process of recovery of waste heat is shown in Fig. 3.

The physical process of energy transfer takes place under the influence of flue gas flow and temperature difference. It is transferred in the form of a certain amount of heat to water with a lower temperature with the onset of thermodynamic equilibrium. The heat transfer of

the flue gases to the water through the outer surface of the flue at different temperatures can be slowed down or increased by adjusting the gas emissions and water supply for heating. Heat transfer resistance mainly depends on the gas boundary layer and the formed liquid film of water in the wall layer.

The recovery of thermal energy (heat) transferred from the combustion of fuel through the heat carrier of flue gases is transferred to the water that is in the heating stage. The material flows of the heat carrier (flue gases) move in the direct flow of the chimney. To establish the efficiency of recuperation, the area of water heating and coolant flow is conditionally divided into several zones. This allows you to set the level of change in the temperature of the heat carrier and water heating. After all, along the length of the heat exchanger, each zone has a heat output with its own value, both as a heat carrier and as a receiver.

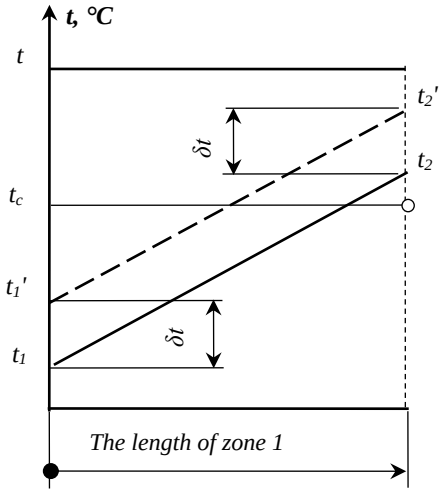

Fig. 3. Graphs of heat carrier and water temperatures in initial zone 1

We denote the water temperature at the entrance to the initial zone 1 as t_1, and at the exit from the heat exchanger (from the initial zone 1') as t(k). It is this difference that most significantly determines the level of heat loss. The flue gas (heat carrier) passing through the initial zone 1 has a temperature of t. Accordingly, $t > t_2 > t_1$, where t_2 is the water temperature at the exit from zone 1.

The interaction of the coolant through the wall of the smoke pipe with water affects the decrease in temperature. At the beginning of the zone, it is $t + \delta t$ ($t_1 + \delta t$) (see Fig. 3), and at the end - $t_{2r} + \delta t$ (where δt is the final temperature difference between the coolant and water). They depend on the thermophysical parameters of the coolant and water, the heat transfer surface, the material of the chimney wall, etc.

Therefore, in the first approximation, we believe that these indicators are the same at the beginning and at the end of zone 1.

Then for the coolant we get:

$$t_1' = t + \delta t \quad \text{and} \quad t_2' = t_{2r} + \delta t,$$

Then the average temperature of the coolant for heating water in zone 1' will be:

$$t_c = \frac{t_1' + t_2'}{2} = \frac{t_1 + t_2}{2} + \delta t.$$

The final flue gas temperature t(k) will be higher than the water temperature t_W. Let this excess

also be the value of δt. So,

$$t_{(к)} = t_c + \delta t = \frac{t_1 + t_2}{2} + 2\delta t ,$$

The temperature difference of the smoke pipe at the exit from the furnace and at the entrance to the heat exchanger is:

$$\Delta t = t_{(к)} - t_1 = \frac{t_1 + t_2}{2} - t_1 + 2\delta t = \frac{t_2 - t_1}{2} + 2\delta t$$

As we can see from the last expression, an increase in heat transfer can be achieved both by reducing the temperature difference $t_2 - t_1$ and by reducing the value of δt. Approximation of the display t_2 to the temperature t1 is possible if the length of the initial zone l is reduced. This means an increase in the heating length (number of zones). In the proposed heat exchanger, the conditions of the length of the initial zone l are positively solved.

The transient process of flue gas emissions is associated with the violation of stationary regimes. Violation of the regime leads to a change in the energy losses of the flue gas flow to a gradual increase in the temperature of the water in the hydraulic circuit of the heat exchanger. Under the condition of limiting the growth of the water temperature during the period of its exit from the heat exchanger, there are ways of possible regulation of the change in the supply rate. At the same time, there is a redistribution of heat with a decrease in water temperature. Table 2 shows the range of experimental parameters, including flue gas temperature, flue gas flow. All temperatures are in Kelvin.

Table 2. Range of experimental parameters

Initial water temperature	293 K (20 ^{0}C),
Initial smoke temperature	493 K (220 ^{0}C)
Smoke speed	4 m/sec
Inlet water velocity	0.25-1 m/sec
Inner pipe diameter	90 mm, wall thickness 1 mm
Outer pipe diameter	170 mm and 1200 mm long
Diameter of water inlet/outlet pipes	50 mm
Material steel, thermal conductivity	51.9 W/(m*K), density 7870 kg/m^3

Construction of the model form and setting of boundary conditions. The main goals of modeling include determining the design parameters of the proposed heat exchanger, which will ensure the cooling of flue gases to a temperature of 85-95 degrees Celsius. Accordingly, the parameters of the heat exchanger, which will ensure the final result, are determined.

1) The surface area of the outer round non-ribbed pipe:

$$F = \pi \cdot d_{out} \cdot l = 3{,}14 \cdot 0{,}017 \cdot 1{,}2 = 0{,}064 m^2,$$

where d_{out} - the outer diameter of the pipe is 0,017 m^2; l −the length of the pipe is 1,2 m.

2) Water consumption volume in the heat exchanger

$$G_V = F_{ep} \cdot V_w \cdot n,$$

where F_{ep} −cross-sectional area of the heat exchanger pipe, m^2; G_V- volume flow of water m^3/sec; V_w −speed of water movement is 0,5 m/sec; η - number of heat exchangers, 1 units.

$$\mathrm{F_{ep}} = \frac{\pi \cdot (\mathrm{d_z} - \mathrm{d_b})}{4} = \frac{3{,}14 \cdot 0{,}008^2}{4} = 0{,}00005(\mathrm{m^2}),$$

$$\mathrm{G_V} = 0{,}5 \cdot 0{,}00005 \cdot 1 = 0{,}0000251(\frac{\mathrm{m^3}}{\mathrm{c}}).$$

The feature of the design of the developed heat exchanger, which is mounted on the outer surface of the smoke pipe at a distance of 50 mm above the surface of the stove, allows selection of large values of the heat flow.

Heat exchange in the heat exchanger takes place in two stages: from the hot coolant to the smoke pipe through which it passes (flue gas), and from it to the water (cold coolant). In the static mode, it can be described in the form of heat flow balances in the pipe and water of the heat exchanger.

The heat flow that is transferred from the chimney to the cold coolant (water), W:

$$\mathrm{Q_{cc}} = \alpha_{sr} \cdot F_{tr} \cdot (t_{tr} - t_w) \quad (1)$$

where α_{sr} – coefficient of heat transfer from the chimney wall to water, W/(m^{2}degrees); F_{tr} - total pipe surface, m^2; t_{tr} – pipe wall temperature, ^{o}C; t_w – average water temperature, ^{o}C.

Heat flow at the inlet and outlet of the heat exchanger with hot water, W:

$$\mathrm{Q_{in}} = \mathrm{G_p C_{in}\, t_{in}}, \quad (2)$$

$$\mathrm{Q_{out}} = \mathrm{G_p C_{out}\, t_{out}}, \quad (3)$$

where, $\mathrm{G_p}$ – liquid mass flow rates, kg/sec; $\mathrm{C_{in}}$, $\mathrm{C_{out}}$ – heat capacity of liquid at the inlet and outlet of the heat exchanger, J/(kg degrees); $\mathrm{t_{in}}$, $\mathrm{t_{out}}$ – fluid temperature at the inlet and outlet of the heat exchanger, ^{o}C.

The heat flow that is lost from the surface of the heat exchanger to the environment, W:

$$\mathrm{Q_{ex}} = \alpha_{mn}\, \mathrm{F_{tz}} \cdot (t_{tr} - t_w). \quad (4)$$

where α_{ex} - coefficient of heat transfer from the surface of the heat exchanger to the surrounding air, W/(m^2 degrees); $\mathrm{F_{tz}}$ - the outer surface of the heat exchanger, m^2; $\mathrm{t_a}$ -outside air temperature, ^{o}C.

The heat accumulated in the metal of the chimney and in the water, J:

$$\mathrm{P_{tr}} = \mathrm{G_{tr} C_{tr} t_{tr}} \quad (5)$$

$$\mathrm{P_w} = \mathrm{V_w}\, \rho \mathrm{C_w}\, \mathrm{t_w}, \quad (6)$$

where, $\mathrm{G_{tr}}$ – mass of the heat exchanger pipe, kg; $\mathrm{C_{tr}}$ –heat capacity of the heat exchanger pipe, J/(kg degrees); $\mathrm{V_w}$ – volume of liquid (water) in the heat exchanger, m^3; p– density of water, kg/m^3; $\mathrm{C_w}$ - heat capacity of water, J/(kg deg).

The average water temperature $\mathrm{t_a}$ is taken as the arithmetic mean between the temperature at the inlet and outlet of the heat exchanger. Based on this, the equation of the dynamics of the

heat change in the smoke pipe and the temperature in the water in the heat exchanger will be described by the following equations:

$$\frac{dP_{tr}}{d\tau} = G_{\text{tr}} \cdot r - \alpha_{sr} \cdot F_{tr} \cdot (t_{tr} - t_w) \quad (7)$$

$$\frac{dP_w}{d\tau} = \alpha_{sr} \cdot F_{tr} \cdot (t_{tr} - t_w) + G_w \cdot C_w \cdot t_{pw} - G_{kw} \cdot C_{kw} \cdot t_{kw} - \alpha_{sr} \cdot F_{tr} \cdot (t_w - t_z) \quad (8)$$

Substituting equations (4, 5, 6) into equations (7, 8), we get:

$$\frac{dt_{tr}}{d\tau} = \frac{G_{\text{tr}} \cdot r - \alpha_{sr} \cdot F_{tr} \cdot (t_{tr} - \frac{t_{wn}+t_{wk}}{2})}{G_{mw} C_{mw}} \quad (9)$$

$$\frac{dP_{kw}}{d\tau} = \frac{\alpha_{sr} \cdot F_{tr} \cdot \left(t_{tr} - \frac{t_{nw}-t_{kw}}{2}\right) + G_w \cdot (C_{nw} \cdot t_{nw} - C_{kw} \cdot t_{kw}) - \alpha_{sr} \cdot F_{tr} \cdot (\frac{t_{nw}+t_{kw}}{2} - t_z)}{0.5 V_w \rho_w C_w} \quad (10)$$

The value of the heat capacity and density of water depends on the temperature. From equation (10), we find the time constant for controlling the water temperature at the outlet of the heat exchanger:

$$\text{T} = \frac{V_w \rho_w C_w}{\alpha_{mn} F_{tz} + 2\rho_w C_{wk}}$$

Knowing from modeling the optimal design of the heat exchanger, various modes of operation were also modeled.

Techniques for constructing graphs. The Flow Vision software complex, based on the finite-volume method of solving hydrodynamic equations based on a rectangular adaptive grid with local grinding, was used to determine design parameters with operating modes and process modeling. This made it possible to import geometry from ADS systems (automated design system) with the exchange of information with the finite element analysis system.

Processing of the experimental array of data was carried out using the "Statistica-12" application program package. The coefficients of the regression equation or approximating function were determined by the standard method [8, 16]. Determination of the optimal modes was carried out by substituting vectors of limited values into the objective function under the conditions of the technology of baking bread in a mini-bakery. The data of the experimental studies were processed with the help of the Microsoft Office package.

The method of elementary heat balances is used, where the heat exchanger is conditionally divided into calculation elements. Such a breakdown of the calculation is as shown in fig. 4 and allows you to determine the rational values of the temperature field indicators within the framework of the given task. The stability of the solution is ensured by the use of an implicit versatile scheme and the control volume method with an offset grid.

Determination of the temperature field of flue gases when emitted from the furnace. The proposed system of multi-purpose management of the technological processes of the production of bakery products, aimed at increasing the productivity of the mini-bakery and reducing the specific costs of raw materials and energy resources. The problem of non-stationary thermal

conductivity in a chimney was solved with the help of the method of finite differences of initial and boundary conditions [20; 21; 25]. Optimizing the operating parameters of the furnace and the heat source requires the development of generalized equations that will make it possible to unambiguously determine the influence of the determining factors. Mathematical modeling of the temperature field of a potential energy source (smoke pipe) during operation of a bakery oven. The process of heat supply and removal in the chimney is a function of time and space. The temperature field is formed from the geometric center - the axis of the smoke pipe. Therefore, the movement of the heat flow is in the radial direction and the magnitude of the heat flow is in the axial direction. Heat flow and heat transfer from the surface is a stable factor that ensures a quasi-stationary state. This feature allows solving a three-dimensional problem using a cylindrical coordinate system. Therefore, for building the model, a cylindrical coordinate system was chosen, taking into account the background of the gas heat flow in the pipe surface [15]. The structural model is adopted by the cutout of a cylinder that has several contours (Fig. 4).

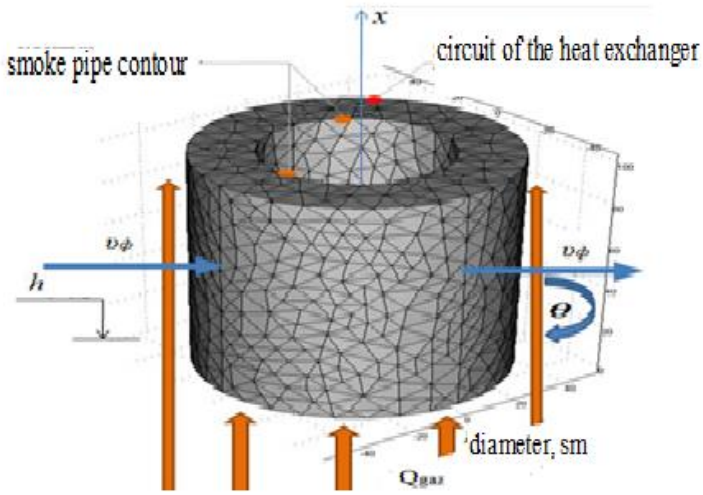

Fig. 4. Calculation model of the recuperator of the temperature field in the conditions of the movement of flue gases: υ – velocity of flue gases, m/sec; Q is tap flow, m

The temperature field of the smoke pipe of the energy source is described by a dimensionless function [15] with three dimensionless influencing parameters (11):

$$f = (Q * Fo * \Theta) \quad (11)$$

where, Fo–the Fourier criterion; Θ– dimensionless temperature; Q is the relative heat flow.

According to the dependencies and the introduction of a dimensionless criterion, the general equation will take the form (12):

$$\Theta = \sum c * Q^m * Fo^n \quad (12).$$

where, n– indicator of degree of criterion Fo; m - an indicator of the degree of the directed flow activity criterion.

The calculation results are well approximated by second-order polynomials. Applying the method of statistical processing, we obtain the criterion equation (13):

$$\theta = -5 \cdot 10^{-3} \cdot Q \cdot {F_o}^2 + 2 \cdot 10^{-4} Q \cdot F_o + 0.006 \cdot Q + 7.6 \quad (13)$$

The equation makes it possible to predict the temperature change of an energy source operating in a cyclic mode of heat flow. On the basis of data digitization, a dependence was obtained (refinement in equation 13) with an adjustment for the regeneration coefficient – kp (Fig. 5). Then the equation for the mode with correction for the regeneration coefficient will be (14):

$$\theta = -5 \cdot 10^{-3} \cdot Q \cdot {F_o}^2 + 2 \cdot 10^{-4} Q \cdot F_o + 0.006 \cdot Q + 7.6(0.0004 k_r + 1.244) \quad (14)$$

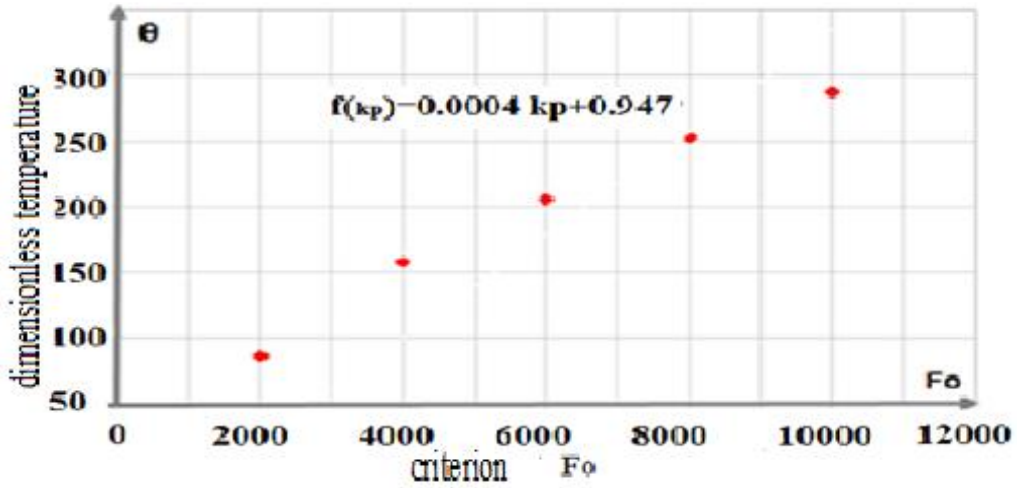

Fig. 5. The influence of the chimney operation mode on the energy efficiency of the heat exchanger

4 Results and discussion

The basis of the proposed design of a mini-bakery on a moving platform with a developed solid fuel oven is not only to ensure the quality of baking products, but also effective energy saving. Based on the obtained mathematical description of the recovery of thermal energy (heat) of flue gases and the dynamics of heat change in the flue pipe and temperature in the water of the heat exchanger, it was possible to thoroughly establish its effectiveness according to the proposed structural scheme (Fig. 2). The scheme reveals the relationship between the input parameters and the predicted controlled actions and output variables of the water heating operating mode. The given analysis of the potential heat source has limited temperature levels. Therefore, the use of this heat of gas emissions for heating water may be limited. Undoubtedly, the integration of such processes is a tool for increasing the energy efficiency of the bread oven in the mini-bakery.

The current state of integration of this methodology is described in the book [23]. Taking into account the analysis of the presented methodologies, the advantages of the processes for the development of low-potential heat recovery systems were used in the developed heat exchanger. It was also established from the published publications [24, 26] that one of the main parameters in the integration of all processes is the temperature difference in the heat recovery system. It leads to better heat recovery and increased efficiency of energy use while maintaining the limit of the minimum temperature difference. In this case, flue gas emissions between the inlet and outlet of the chimney are about 353-363K, taking into account the calculated structural and technological features of the proposed heat exchanger.

Since the shape and temperature of the coolant directly reflects the state of temperature changes, we consider this change in shape during research. The temperature distribution along the height of the chimney was determined from the entrance to the highest point of the heat exchanger. In their research [24, 25], the authors established the effect of the excess air coefficient during gas emission. This parameter characterized the gas permeability of the heating system. It also establishes that the actual water heating efficiency and flue gas coolant

efficiency is an important factor that leads to the difference between the actual water heating efficiency and its theoretical efficiency.

The process of adding and removing heat in a potential energy source is a function of time. The temperature field is formed by the design parameters of the furnace performance, and this is the formation of temperature fields of heat flow in the radial and axial directions. Despite this, due to the design, the formation of heat flow and heat removal from the chimney circuit are stabilizing factors that ensure a quasi-stationary state. The change in the shape of the distribution of the temperature of the heat carrier and the temperature of water and gas emissions with independent regulation of their supply is recorded in Tables 3 and 4.

Table 3. Dependence of the temperature of the water at the outlet of the heat exchanger on the speed.

V initial water, m/sec	0,25	0,5	0,75	1	1,5
T water average ex., K	341,59	340,38	338,6	338,22	336,83
T water average, K	341,59	340,38	338,6	338,22	336,83

To study the effect of flue gas temperature changes on the stability of water heating, a computer simulation of the heat exchanger was carried out. The computer program and the calculation procedure were verified by comparing the results with established results. The rate of heat flow, the temperature in the heat exchanger, as well as the temperature of the flue gas emission and the influence of design parameters on the efficiency of the process were evaluated. The authors [3, 27] conducted such studies of changes in the influence of the furnace function: fuel consumption, temperatures, and emissions. The database of obtained numerical values made it possible to evaluate the influence of parameters on the furnace performance. However, the influence of the temperature readings of the gas before the heat exchanger in the studies is insufficient.

Table 4. Dependence of the gas temperature at the exit from the heat exchanger on the speed.

V initial gas, m/sec	4	6	8	9	11
T gas average in., K	491,93	495,43	503,8	508,61	511,23
T gas average ex., K	411,33	421,43	432,3	435,63	445,62

According to the features of the proposed heat exchanger in fig. 5 shows the variations in the form of movement of flue gases and the distribution of the coolant in the flue pipe and the heating of water in the heat exchanger when the parameters change. From fig. 5, it can be seen that under any conditions there is a difference, which shows a decrease in the temperature of the coolant along the height of the heat exchanger. The greater the heat load (at the inlet), the more obviously this phenomenon affects the heat transfer and changes at the outlet, giving off thermal energy.

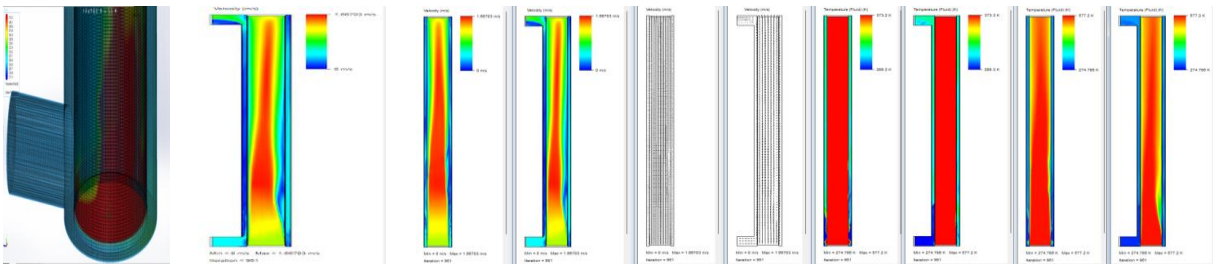

Fig. 5. Temperature distribution in the heat exchanger

The thermal distribution along the height of the heat exchanger at each point decreases with increasing water temperature. The internal heat transfer of energy along the height of the heat exchanger at each point tends to change gradually. This shows that the uniformity of the distribution of the coolant in the heat exchanger can be improved by changing the parameters (constructive and technological).

A change in the water heating temperature affects the internal energy potential of the gas, which decreases with a decrease in the heat load. The internal energy potential is determined by two factors: the temperature of smoke leaving the stove and its rapid spread in the chimney. On the one hand, when the heat load remains constant, the higher the heating of the water, the lower the energy potential of the gas flow. At a higher speed of gas exit from the chimney, water heating does not increase. On the other hand, it makes it possible to reasonably set the speed of gas propagation to the speed of water movement, which leads to possible regulation of the propagation of energy flows. So the flow rate is the dominant factor. Thanks to the total effective regulation of the thermal potential, the water heating range at the maximum height of the heat exchanger increases. Therefore, to ensure the quality of the process, at this stage, it is necessary to predict the design parameters (increase the volume of the heat exchanger) and ensure the value of the constant time of water temperature control at the outlet of the heat exchanger, according to equation (10).

According to the obtained results, in the simulation process, graphs of the dependence of the coolant temperature at the outlet on the speed were built using the Microsoft Excel software environment.

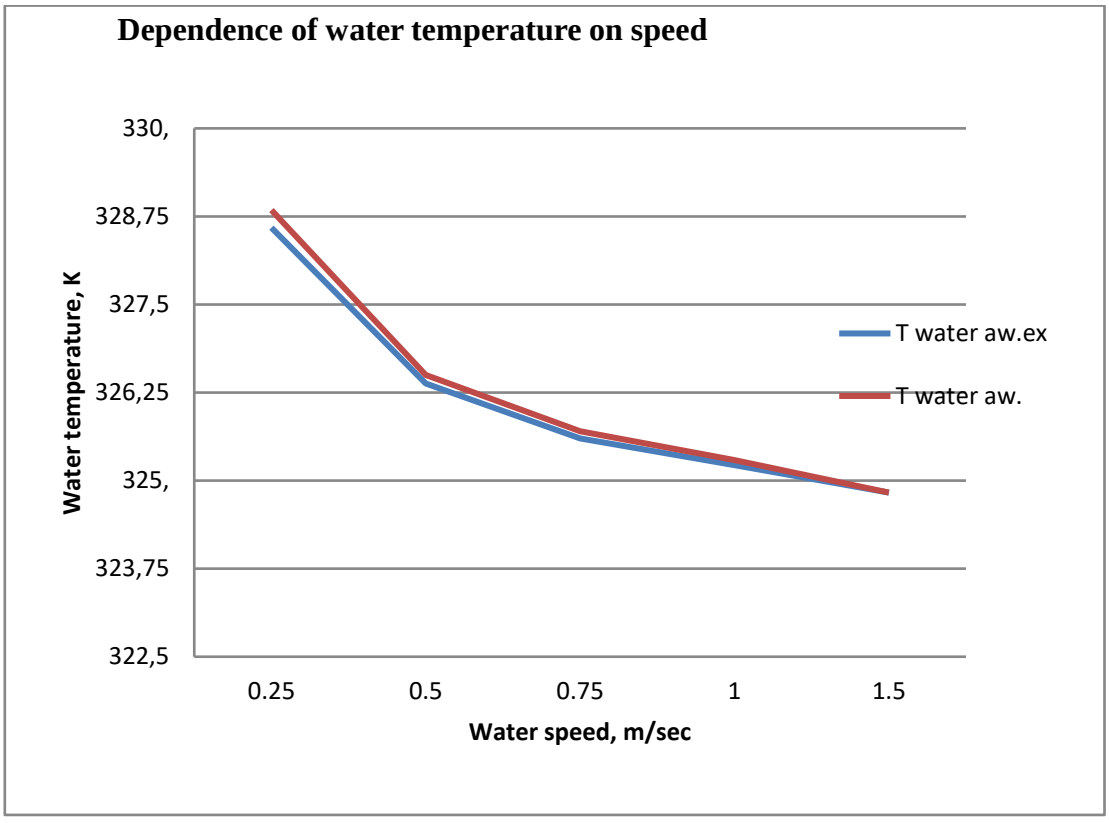

Fig. 6. Dependence of water temperature on its speed.

As can be seen from the graph (Fig. 6), the water temperature at the outlet of the heat exchanger is inversely related to its velocity at the inlet. This dependence shows that due to the increase in the speed of the water flow, it does not have time to take away all the heat and simply "flies by". This dependence is especially visible at flow velocities up to 1 m/sec, at velocities greater than 1 m/sec the dependence also persists, but it is not so sharp.

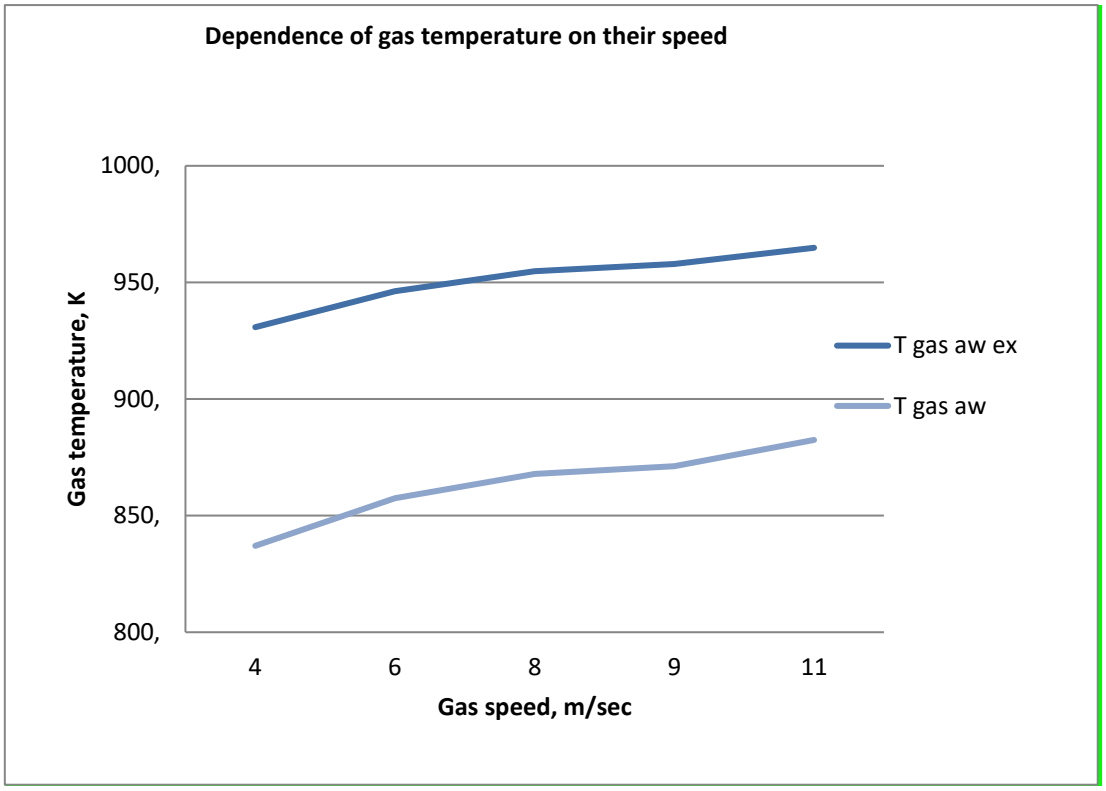

Fig. 7. Dependence of gas temperature on their speed.

From fig. 7 it can be seen that as the speed of the flue gases at the heat exchanger inlet increases, their temperature at the outlet also increases. These indicators indicate that the gases are cooled worse with an increase in speed, that is, they do not have time to give the maximum amount of heat to the water. The same data in the experiments of the authors [25, 26] show that the dependence of the temperature of gas emissions and their heat loss depending on the productivity G and they turned out to be linear. Therefore, it can be seen in Fig. 5, 6 that the amount of energy losses of the heat carrier of flue gases is increasing. The established mode of operation of the heat exchanger of the corresponding recovery mode is possible with regulatory compliance with the performance and parameters of water supply and flue gas emission in all its zones. In case of a forced shutdown of the bakery oven, in order to reduce the influence of external disturbances before and after heat exchange, it is necessary to draw water.

When the heat exchange takes place in the heat exchanger, the heated water receives all the thermal energy transferred from the smoke emissions, excluding heat losses to the environment. They are small. It is also possible to foresee their elimination with the help of heat exchanger insulation. The evaluation of these dependencies of the modes of movement of flue gases and water affects the efficiency of the heat exchanger. Therefore, in order to ensure maximum heat transfer, the selection of optimal modes of movement was carried out.

Analysis of simulation implementation systems and software. Taking into account the above and on the basis of the working and technological characteristics of the mini-bakery, a system of adaptive control of the complex of preparation of dough, dough and baking of

bakery products was considered and built. The use of smoke emissions makes it possible to establish the dependence of fuel consumption parameters, gas temperature flow in the water heating heat exchanger.

We established the movement and distribution of temperature flows during the emission of heat gases from the furnace chimney and the transfer of this heat flow through the wall for heating water. For a more thorough approach to technology, the following parameters should be attributed: G-parameter, which characterizes fuel consumption; baking conditions: υg is the heat flow rate of flue gases (m/sec) and T_{rk} - the temperature in the working chamber of the oven (°C).

The results of the conducted experimental studies made it possible to obtain the following dependencies (5-6):

$$Q_p = 263{,}9 * v_g + 9{,}116 * E_1 + 40{,}15 * J_{in} + 0{,}05332 * T_{rk} - 480{,}9 \rightarrow max \qquad (15)$$

$$E_1 = -2{,}21 * 10^{-4} * Q_p + 0{,}279 * G - 1{,}272 * J_{in} + 40{,}69 \rightarrow max \qquad (16)$$

where, E_1i J_{in} - strength of flour and content of ingredients in percentages, according to the recipe of bread; G – a parameter that characterizes solid fuel consumption; Q_p – oven performance; υ_g iT_{rk} – baking conditions: heat flow rate (m/sec) and oven temperature (°C).

The dependence of solid fuel consumption and furnace performance is considered in Figure 8 in the form of a multidimensional surface with consideration of the optimal point on the ordinate along the W axis.

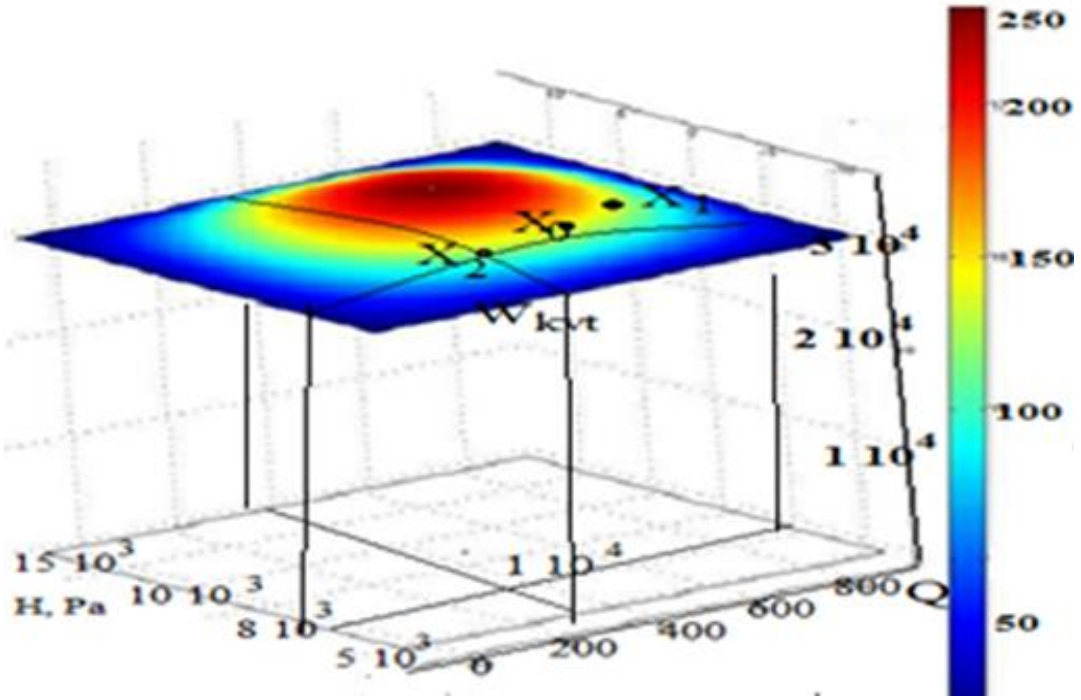

Fig. 8. Geometrical interpretation of optimization of energy consumption depending on from the productivity of the furnace and the flow in the gas exhaust tract of the chimney.

Figure 8 highlights a number of characteristic relationships. This relationship of productivity Q_p and energy consumption W with the specific costs E_1 of the designed furnace depends on the amount of fuel consumption and the rate of heat flow (υ_g). In addition, there is the heat consumption of the baking chamber, the influence of moisture from the evaporation of the bread dough in the baking chamber M, and dilution. Mode parameters depend on the

recipe composition of the products, gas penetration and the state of the bread dough (J_{in} *and* J_x) and the temperature in the chamber by zones (T_1, T_2, T_3, T_4). These parameters affect the amount of heat loss (q_2) and the temperature generated by solid fuel (*Qn/g*,*tn/g*), which should not exceed the values allowed for this type of furnace.

The regression equation for optimization of energy consumption and oven performance during baking, which describes the cause-and-effect relationship, has the form (7 - 8):

$$W = 0{,}9846G + 2{,}349Q_p + 2{,}966T_2 + 922 * v_g + 41{,}69E_1 - 634 \rightarrow min \quad (17)$$

$$Q_p = 263{,}9v_g + 9{,}116E_1 + 40{,}15J_{in} + 0{,}053T_3 + 1{,}597 \geq Q_p plan \quad (18)$$

The correspondence of expressions (17 and 18) with experimental data confirms the value of Fisher's test: for (17) F = 5.399, for (18) F = 6.129 and the root mean square error is 3.08% and 2.64%. Therefore, it is possible to use the data of the multiple regression equation when solving the fuzzy optimization of the technological process of bakery products.

The development of systems for multi-purpose management of technological processes of bakery production based on product quality control and intelligent technologies will contribute to increasing labor productivity and reducing specific losses of raw materials, energy and material and food resources.

The use of a heat exchanger in the technology of a mini-bakery allows not only to effectively use a secondary source in the heat supply of the technology of preparing products, but also an ecological direction of production.

CONCLUSION

The analysis of the results obtained from the conducted research made it possible to determine the design, properties of the heat exchanger structure, the dynamics of temperature development and heat transfer, and the integrity of the process. According to the results of the theoretical and experimental studies, taking into account the thermodynamic analysis, the temperature field of the potential energy source, the main rational parameters of the heat exchanger were established: internal diameter - 90 mm; outer diameter D=176 mm, height - 1200 mm. Conducted studies of the influence of the design parameters of the heat exchanger on the temperature of the waste gases established that it decreases by 80 °C. The use of innovative developments of equipment, technologies and the use of flue gases in the production of bakery products at a mini-bakery is mathematically substantiated, which allows reaching a new level of quality and safety of reproduction of the principles of energy and ecological and economic efficiency. Thus, the relationship between the parameters of fuel consumption, the speed of the heat flow of flue gases, and the temperature in the working chamber of the oven (°C) allows heating the water in the heat exchanger to 70 °C during one cycle of baking bread. At the same time, the change in the heat flow in the chimney is determined by the change in the temperature of the heating gases.

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