American Journal of Multidisciplinary Research & Development (AJMRD)
Volume 03, Issue 10 (October - 2021), PP 54-62
ISSN: 2360-821X
www.ajmrd.com

Research Paper **Open Access**

MODELS REGULATION OF ROOM MICROCLIMATE PARAMETERS STORAGE OF AGRICULTURAL PRODUCTS

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Abstract: The paper presents an algorithm for the developed mathematical model of heat and mass transfer processes in storage rooms for various perishable agricultural products, which allows you to determine the operating modes of the heat pump regulator.

Key words: mathematical models, algorithm, regulation, microclimate, vegetable storage, heat and mass transfer processes, heating, ventilation, optimization.

I. Introduction

Modern technologies used in countries with developed agricultural production are characterized by a high profitability of obtaining vegetable products due to low storage losses, and, consequently, reducing the cost of reproducing low-quality products. It is believed that special attention should be paid to the development of the storage industry, since the costs of reproducing low-quality products are several times higher than capital investments [5].

The problem of providing a microclimate in storage rooms for various perishable products is complex. At the present stage of its development, the problem of protecting the storage microclimate from the external climate has been practically solved. However, the main problem that must be solved by ventilation in the design mode of cooling the embankment or stack of containers remains unexplored - determining the expected provision of microclimatic storage parameters and the required combination of interrelated design parameters of the "ventilation - embankment (stack)" system. This will allow not only to reasonably choose a ventilation solution from among those used, but also to start developing highly efficient technologies [7].

The main reason for the lag in this area is ventilation of moisture transfer during cooling of products by streams, which are widely used in modern storage ventilation systems. The accumulation of research materials related to the ventilation of BAP embankments over the past decades has occurred in the study of the lack of study of the flow of multidimensional non-stationary processes of heat and such processes in one-dimensional flows [9].

The regularities of the formation of storage parameters in a complex structure of a stack of lattice containers during cooling of BAP in them have not yet been studied [10].

In modern potato and vegetable storages, using progressive constructive and space-planning solutions, ventilation is the main means of regulating heat and moisture exchange processes in an embankment or a stack of containers at all stages of storage of vegetable products. At the same time, in the cooling period calculated for ventilation, the main problem is solved - a gradual decrease in the temperature of the entire mass of products to the technologically specified one, in compliance with the time indicators and the cooling rate AZ deg / h.

To study multidimensional unsteady heat transfer processes in potato and vegetable storage facilities, an optimized thermophysical model for average mounds was adopted. It takes into account the property of BAP to self-regulation of relative humidity in the massif, biological heat release, the effect of moisture exchange on the temperature field. The heat transfer coefficient is determined experimentally, which makes it possible to separately solve the problems of heat transfer and moisture exchange. The correctness of the adoption of such a model is confirmed in the work by special studies [2].

In the literature on industrial and chemical technology, only one-dimensional problems of heating (cooling) physical bodies have been considered. The massif of biologically active products has special structural and thermophysical properties, it constantly emits not only heat, but also moisture. In addition, the embankment is cooled by multidimensional flows [3].

Statement of tasks. The mathematical model of the studied phenomenon for calculating three-layer heat-shielding coatings and wall panels that are often used in practice (Fig. 1) includes a set of thermal conductivity equations for a three-layer fence (1), a layer of agricultural products (2), an energy equation for air (3) supplied into the product embankment, the differential equation of mass transfer in the air inside the embankment (4), as well as the initial (5) and boundary (6) ... (10) conditions:

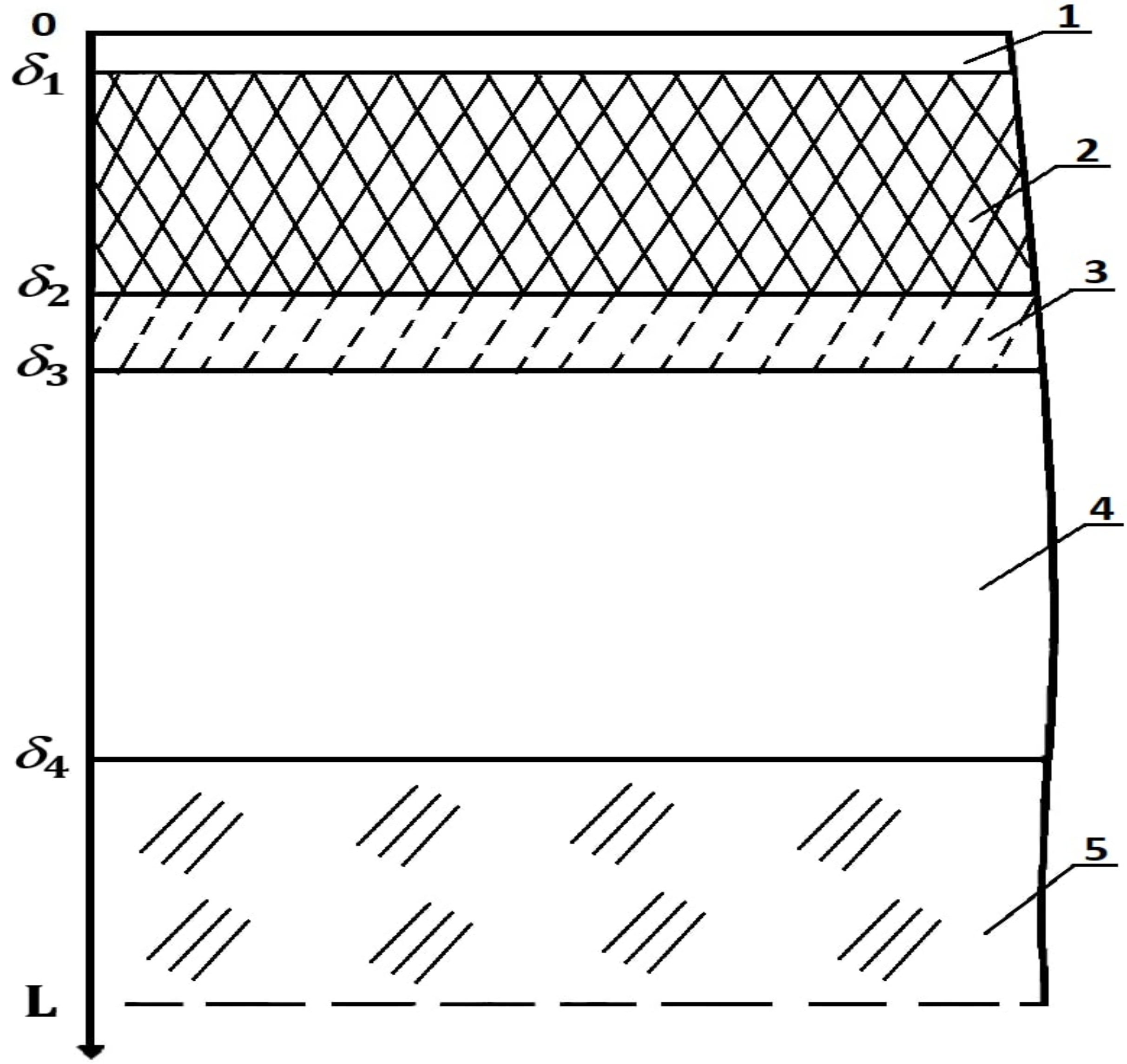

Fig. 1. Calculation diagram of the heat resistance of the storage cover:
1 - roofing layer; 2 - insulation; 3 - bearing layer; 4 - storage air layer; 5 - layer of the product embankment

The use of ventilated walls makes it possible to improve the temperature, humidity and air conditions of the enclosure, increase its durability and reliability, and ensure the maintainability of the structure during the operation of the storage facility.

The thermophysical model adopted in the research corresponds to the mathematical model of reproducing the non-stationary process of heat transfer in the average statistical bulk of the product, which takes into account: the change in the characteristics of the layer by years and types, internal heat release, the interaction of convective and evaporative cooling, the ability of the BAP to form its own relative air humidity in the layer. The coefficient α_V is determined experimentally.

The initial equations of convective heat transfer for air and a layer during cooling of an average embankment by two-dimensional flows, presented in dimensionless parameters, have the form [4]:

$$\frac{\partial \bar{t}}{\partial \bar{\tau}} + \frac{\bar{\omega}_x}{\varepsilon} \cdot \frac{\partial \bar{t}}{\partial \bar{x}} + \frac{\bar{\omega}_y}{\varepsilon} \cdot \frac{\partial \bar{t}}{\partial \bar{y}} = \frac{\alpha_V \cdot K_1}{\varepsilon}(\bar{\theta} - \bar{t}), \qquad (1)$$

$$\frac{\partial \bar{\theta}}{\partial \bar{\tau}} = \alpha_V \cdot K_2(\bar{t} - \bar{\theta}) + K_2 \cdot \frac{q_V}{(\theta_o - t_o)}, \qquad (2)$$

with initial and boundary conditions

$\bar{\theta}(0,\bar{x},\bar{y}) = 1,\ \bar{t}\,(\bar{\tau},|\bar{x}| \le \overline{b_0}\,, 0)$=0, (рис. 1, а),

$\bar{\theta}(0,\bar{x},\bar{y}) = 1,\ \bar{t}\,(\bar{\tau},0,\ 0 \le \bar{y} \le \overline{b}_0\,, 0)$=0, (рис. 1, б),

где

$\bar{t} = \frac{t-t_0}{\theta_0-t_0}$, $\quad \bar{\theta} = \frac{\theta-t_0}{\theta_0-t_0}$ $,\bar{\tau} = \tau \cdot \frac{\omega_0}{L},\ \ K_1 = \frac{L}{C} \cdot p \cdot \omega_0\ , K_2 = \frac{L}{C_3} \cdot p_3(1-\varepsilon) \cdot \omega_0\ ,\ \ \bar{x} = \frac{x}{L}, \bar{y} = y/L$.

For the circuit in Fig. 1, and instead of L, 0.5L is taken, $\bar{b}_0 = b_0 / L$.

According to the literature and the results of special studies carried out by us in the course of the study [2, 3, 4, 10], it was concluded that the movement of air in dense layers of vegetable products can be considered irrotational (potential). It was also found that the presence of initial turbulence in the embankment zone near the source does not affect the quality indicators of the processes under study. For this case, the most convenient and simpler solution is obtained for the one-dimensional problem of heat transfer in a layer without internal sources of heat. At a constant flow rate, the dimensionless layer height and its cooling time are as follows:

$$\eta = \frac{\alpha_v y}{C \cdot p \cdot \omega}, \qquad (3)$$

$$\xi = \frac{\alpha_v}{C_3 p_3 (1-\varepsilon)}\left(\tau - \eta = \frac{\varepsilon \cdot y}{\omega}\right) \qquad (4)$$

To describe η and ξ in the general case of a two-dimensional flow, the system of equations (1), (2) was transformed to new independent variables - the potential $\bar{\varphi} = \bar{\varphi}(\bar{x},\bar{y})$ and the flow function $\bar{\Psi} = \bar{\Psi}(\bar{x},\bar{y})$, and then to the dimensionless parameters $\eta = \eta(\bar{\tau},\bar{\varphi}),\ \ \xi = \xi(\bar{\tau},\bar{\varphi})$.

The equations describing η and ξ for a two-dimensional flow in the BAP embankment are obtained:

$$\eta - K_i \int_{\bar{\varphi}_0}^{\bar{\varphi}} \frac{\alpha_V(\bar{\varphi})}{\bar{\omega}^2(\bar{\varphi})} d\bar{\varphi}, \qquad (5)$$

$$\xi = \alpha_V(\bar{\varphi}) K_2 \cdot \bar{\tau} - K_2 \cdot \varepsilon \int_{\bar{\varphi}_0}^{\bar{\varphi}} \frac{\alpha_V(\bar{\varphi})}{\bar{\omega}^2(\bar{\varphi})} \cdot d\bar{\varphi}. \qquad (6)$$

In the particular case, when $\omega = const(\alpha_V = const), \varphi = \omega \cdot y,\ \bar{\varphi}{=}\bar{\omega} \cdot \bar{y})$
(5), (6) take the form (3), (4).
(1) includes three differential equations. Here t_j (x, τ), a_j, λ_j, (j = 1,2,3) are the temperature, the coefficients of temperature and thermal conductivity at j = 1 for the outer layer of the coating (wall), at j = 2 for the insulation , for j = 3 for the inner coating layer (wall).
The right-hand side of equations (2) and (3) includes terms that take into account the contact heat transfer between the porous body (stored product) and the air moving inside the embankment with a velocity v. Here $\beta_1 = \alpha F_n / (p_b\ c_b \varepsilon)$, $\beta_2 = \alpha F_n / (p_n\ c_4)$, $p_n = p_n\ (1\text{-}\varepsilon)$, α is the heat transfer coefficient between the product and the air inside the embankment, which is determined by the shape, size of inclusions (elements of a porous body), physical properties, air velocity and is taken in the form of a formula obtained experimentally by V.Z. Zhadan: a = 1.5 + 43v.
The right-hand side of equations (2) and (4) includes the last terms that take into account heat and moisture transfer taking into account the linearization f (t_4) - equilibrium. The design mode of ventilated air layers is determined by the winter temperatures of the outside air, the minimum permissible temperature of the bin wall in contact with the product, the minimum and permissible storage temperatures and conditions that exclude overcooling of the product; falling out of condensate in the air gap; overheating of products.
Taking the theoretical model of the processes under study according to A. Antselius and B. I. Kitaev, introducing additional terms into the equations that take into account internal heat release, independent of the layer height and time $(\bar{q}_V = q_V/\alpha_V(\theta_0 - t_0) = const)$, the distribution of dimensionless air temperatures $\bar{t}_{at}$ and products $\bar{\theta}$ in the embankment are described by the expressions:

$$\bar{t} = \bar{t}' + \bar{q}_V \cdot \int_0^{\xi} e^{-\lambda} \cdot \left(\int_0^{\eta} e^{-s} \cdot I_0\left(2\sqrt{\lambda \cdot s}\right) ds\right) d\lambda; \qquad (7)$$

$$\bar{\theta} = \bar{\theta}' + \bar{q}_V \cdot \left[\int_0^{\xi} e^{\lambda-\xi} \cdot \left(\int_0^{\eta} e^{-s} \cdot I_0\left(2\sqrt{s(\xi-\lambda)}\right) ds\right) d\lambda + e^{-\eta} \cdot \int_0^{\xi} e^{\lambda-\xi} \cdot I_0\left(2\sqrt{\eta(\xi-\lambda)}\right) d\lambda\right], \qquad (8)$$

где

$$\bar{t}' = \Gamma(\eta,\xi) = 1 - e^{-\xi} \int_0^{\eta} e^{-s} \cdot I_0\left(2\sqrt{\xi \cdot s}\right) ds; \qquad (9)$$

$$\bar{\theta}' = \Gamma_1(\eta,\xi) = 1 - (\xi,\eta) = e^{-\eta} \int_0^{\xi} e^{-s} \cdot I_0\left(2\sqrt{\eta \cdot s}\right) ds; \qquad (10)$$

In equations (7) - (10), the values of η and ξ are determined by the obtained formulas (5) and (6).

When solving the problem of the analytical description of the velocity field in the bulk structure of vegetable products, the following assumptions are justified: the structure of the layer is isotropic, with the same density; movement is seen as internal in the pore channels; the air density is constant; the dependence of pressure on speed is linear and is observed throughout the volume of the embankment.

Results and Discussion. It has been established that in the embankment of vegetable products the boundary of the area with a linear law of resistance expands to values Re_e = 400 ... 625 and more. It is also known that local losses prevail in dense layers, which excludes the formation of microvorticity, and the use of two-dimensional flows in the area of motion excludes macrovorticity.

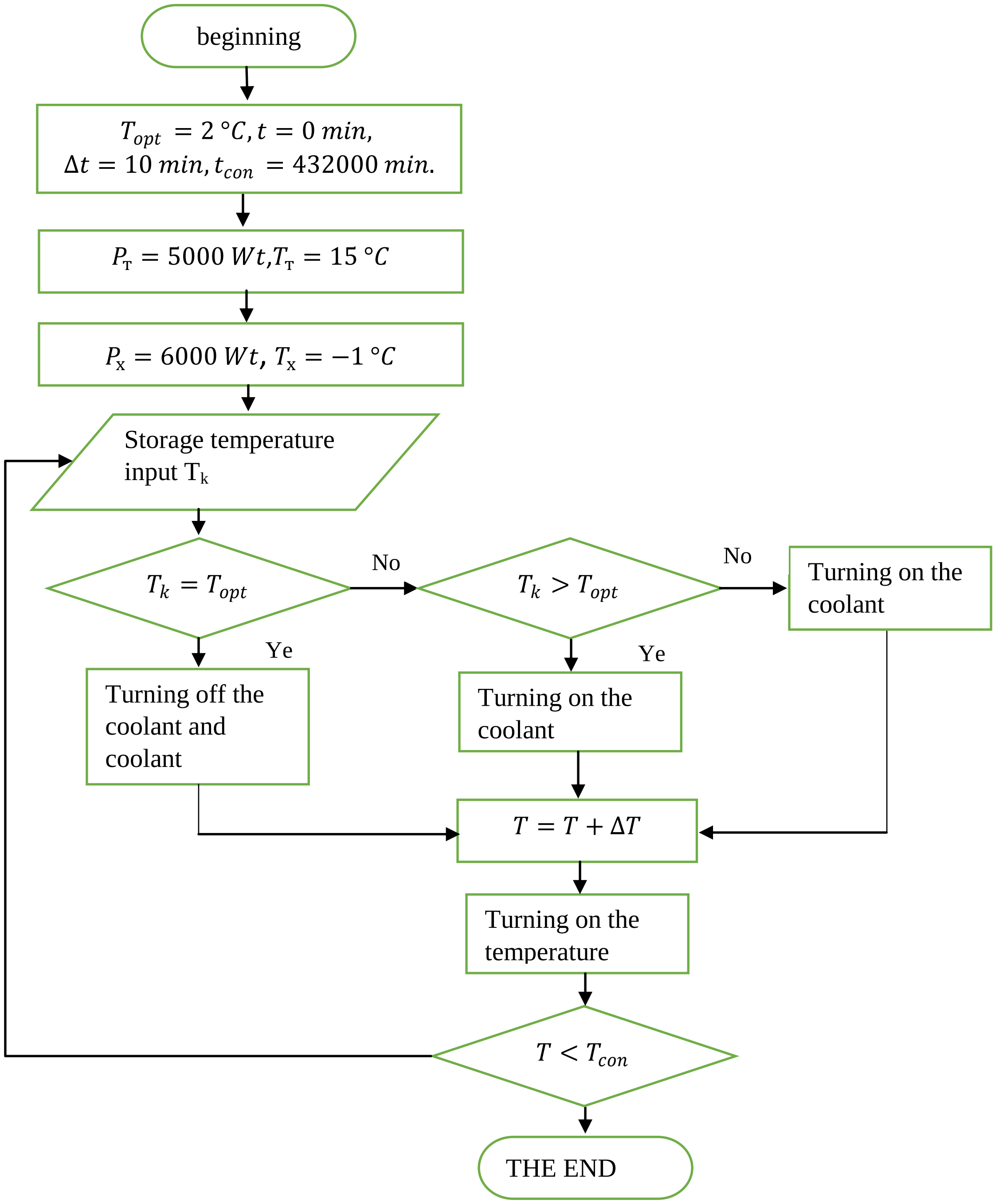

Fig. 2. Algorithm for matching the operating modes of the heat pump with the controlled and controlled parameters of the microclimate of the potato storage: T_{opt} - the optimum storage air temperature (°C); t - storage duration (min.); Δt - time increment (min.); t_{jun} - storage duration - 10 months (min.); P_T, P_x- heat and coolant

power (Wt); T_T, T_x is the temperature of the heat and coolant (°C), T_x is the temperature of the potato storage (°C).
This algorithm (Fig. 2) allows you to determine the operating modes of the heat pump regulator. If the temperature of the potato storage does not correspond to the set value, then using the electrical controllers of the heat pump in the converter, the required temperature is set, necessary for supplying air to the potato storage.
TM is a set of series-connected thermocouples made of two dissimilar elements with p- and n-type conductivity. The degree of cooling and heating will be proportional to the magnitude of the current. When the polarity of the current is reversed, the hot ("hot" junction) and cold ("cold" junction) sides of the module are reversed.
The efficiency of TM operation depends on the quality factor of the semiconductor substance, including electrical conductivity, thermal conductivity and Seebeck coefficient.
The heat output of TM is determined by the formula (Tsvetkov Yu.N.):

$$Q_r = \alpha \cdot T_r \cdot I + 0{,}5 \cdot I^2 \cdot R - k \cdot \Delta T \cdot \delta, \quad Bm, \qquad (11)$$

and the heating coefficient –

$$k_0 = \frac{Q_r}{W} = \frac{\bar{\alpha} \cdot T_r \cdot I + 0{,}5 \cdot I^2 \cdot R - k \cdot \Delta T \cdot \delta}{\bar{\alpha} \cdot \Delta T \cdot I + I^2 \cdot R}, \qquad (12)$$

where α - coefficient of thermoelectric power TM, V / °C; T_r - hot junction temperature, ° C; I- current strength, A; R - resistance, Om; k - coefficient of thermal conductivity, Wt / (m · °C); ΔT - temperature difference between junctions, ° C; W - power consumption of TM, Wt; δ is the thickness of the ceramic plate, m.
Taking into account the maximum possible hot junction temperature of 74 ° C and the empirical expression (13), the formula for determining the heat output of TM is as follows:

$$Q_r = I^2(0{,}5 \cdot R - 1{,}5 \cdot k \cdot \delta) + I(74 \cdot \alpha - 18{,}26 \cdot k \cdot \delta) + 36 \cdot k \cdot \delta. \qquad (13)$$

The heating coefficient can be expressed in terms of the cooling coefficient, which depends on the current strength ε = f (I) and the heat flux Q_0 (Wt) supplied from the external environment to the cold side of the TM:

$$k_{om} = \frac{Q_0 + W}{W} = \varepsilon + 1. \qquad (14)$$

The process of maintaining the temperature regime in the potato storage when using a regulator with TM is as follows. In heating mode, the solid filler expands due to the action of TM and the heat carrier, acting on the stem that opens the valve. In this case, the required coolant flow is supplied to the converter (Fig. 4). In this case, the balance of the heat flux in the power sensor is as follows:

$$Q = Q_{тм} + Q_{ж}, \quad \text{Вт}, \qquad (15)$$

where Q_{TM} is the heat flux controlled by an electric regulator with a solid filler and TM, Wt; Q_{zh} is the heat flow of the energy carrier circulating in the NPIE converter, Wt.

When the set temperature in the potato storage is reached, the heat pump with electric controls is switched off and the valve is closed. To speed up the process of closing the valve, the TM switches to the cooling mode, due to which the filler is cooled. In the cooling mode, the solid filler under the influence of TM expands and the regulator valve opens in the same way. In this case, the energy carrier with a temperature below the set value will prevent the valve from opening. Then the balance of the heat flow in the power sensor will be:

$$Q = Q_{тм} + Q_{ж}, \quad \text{Вт}, \qquad (16)$$

In this case, the energy consumption is higher than in heating mode.
The output coordinates of the parameters of the industrial premises microclimate as an object of regulation are: temperature - t_B, moisture content - d_B, partial pressure - P_B, relative humidity - φ_B, and heat content (enthalpy) - I_B. Any combination of these parameters determines the thermal and humidity state of the air in the room, and when regulating technological processes, it is necessary to take into account their relationship to exclude self-oscillating modes of mutual influences through various control channels.
The gain along the channel for changing the air velocity and coolant temperature is determined by:

$$K_{VT} = \frac{F_y \cdot (t_{вт} - t_{нв}}{C_в \cdot G_в} \cdot \left(\frac{1}{K_m} + \frac{A}{V_B} + \frac{C}{V_m}\right)^{-1}, \ \frac{°C}{м/с}, \qquad (17)$$

where F_y is the heat transfer surface, m^2; t_{wt} - initial temperature of the coolant,°C; t_{nv} - initial air temperature, ℃; $C_в$ - specific heat capacity of air, kJ / (kg • ° C); G_B is the mass of the air flow rate, kg / h; K_T - heat transfer coefficient, W / (m • ℃); V_B — speed of air movement, m / s; V_T is the speed (flow rate) of the coolant, m / s; A, C - technical characteristics of the air heater.
The amplification factor hyperbolically decreases with increasing V_T and V_B (Fig. 3), since the irrigation chamber operates in polytropic and adiabatic modes. The sprinkler chamber in the adiabatic mode is an intermediate link, and the air treatment process takes place at I_B = *const*. In the polytropic regime, the process of air processing occurs with a change in its enthalpy.

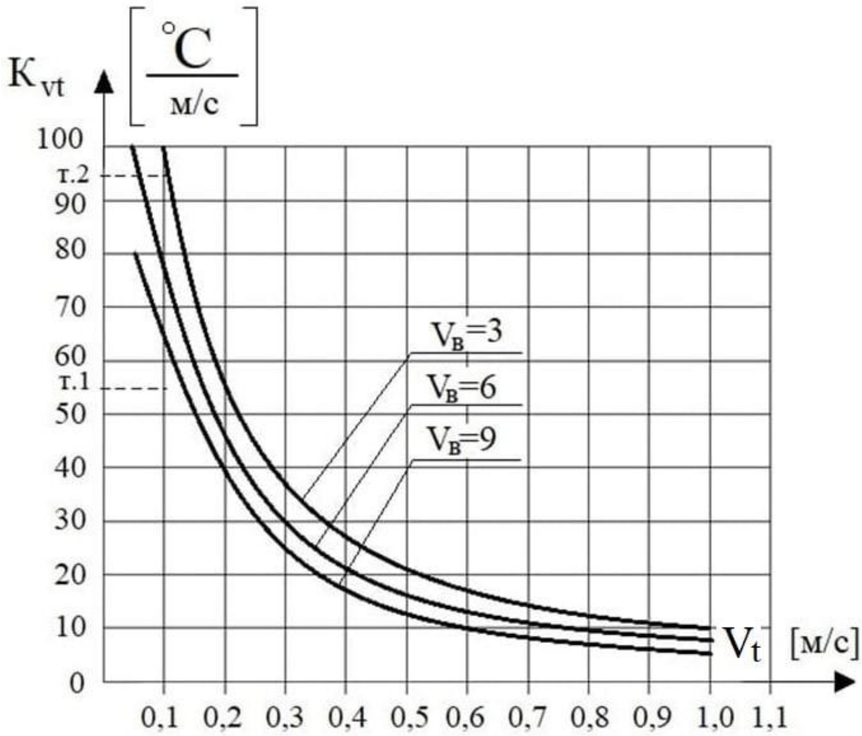

Fig. 3. Dependence of the transfer coefficient of the air heater on the flow rate of the coolant at different air speeds

By changing the flow rate of the coolant using a plunger pump with a variable speed of rotation of the output shaft, a real possibility of regulating K_VT and temperature graphs in a wide range (point 1 and point 2) was obtained, setting the optimal values according to the given algorithm (Fig. 4) in accordance with the technological requirements.

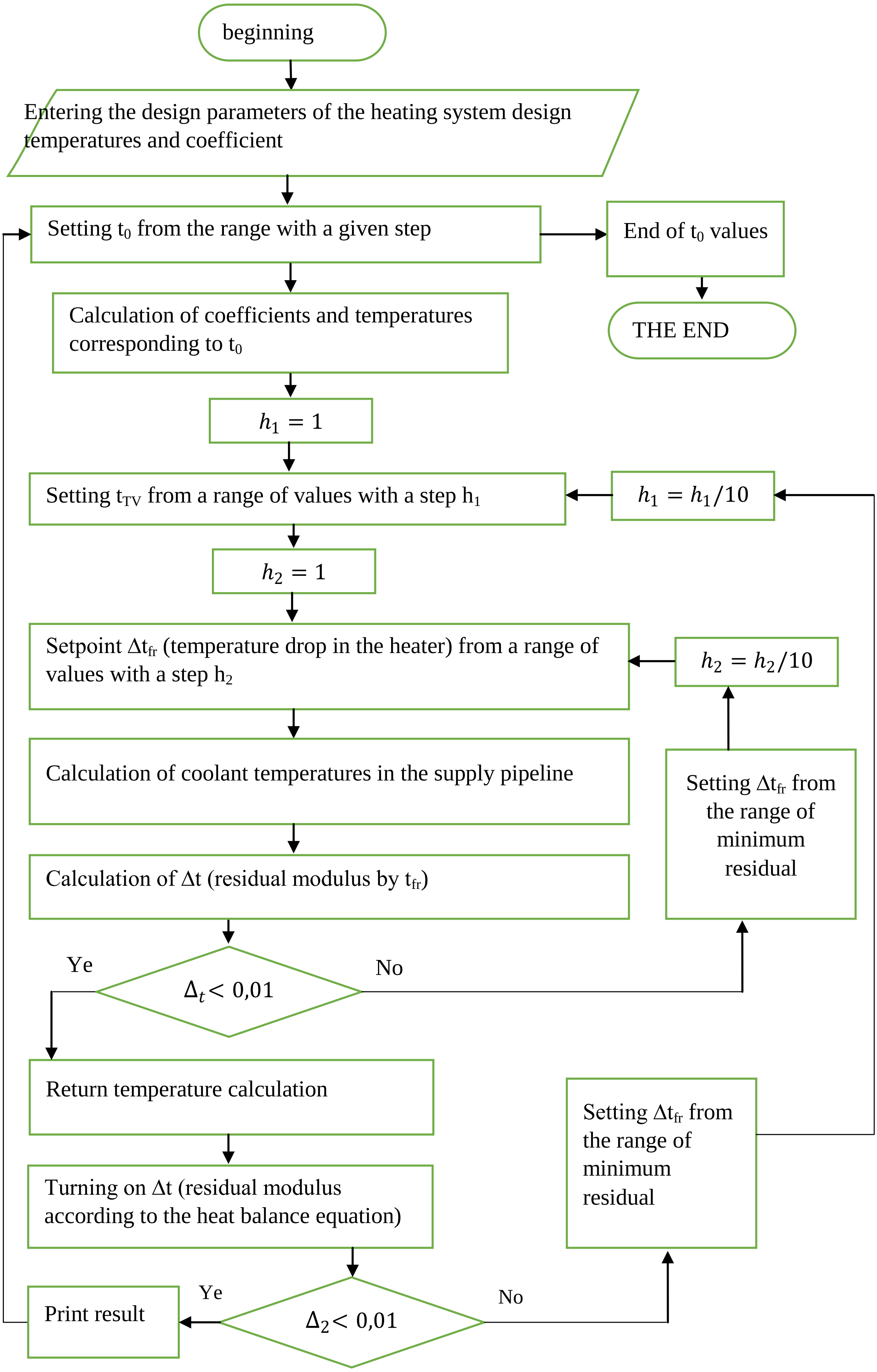

Fig.4. Algorithm for calculating temperature graphs

Energy and mathematical flows of microclimate processes in the form of differential equations make it possible to determine the temperature and moisture content of the air:

$$\begin{cases} p\cdot d_B = \frac{q_\partial \cdot n}{m_\text{в}} + \frac{G_\text{пара}}{m_\text{в}} + \frac{-d_B\cdot\left(G_\text{вент}+G_\text{инф}\right)}{m_\text{в}} + \frac{\left(G_\text{вент}+G_\text{инф}\right)\cdot d_\text{в}}{m_\text{в}} \\ p\cdot t_B = \frac{q_\partial\cdot n}{c_\text{в}\cdot m_\text{в}} + \frac{G_\text{со}\cdot c_\text{Вол}\cdot\Delta t_\text{со}}{c_\text{в}\cdot m_\text{в}} + \frac{G_\text{то}\cdot c_\text{Вол}\cdot \Delta t_\text{то}}{c_\text{в}\cdot m_\text{в}} + \\ + \frac{\left(F_\text{огражд}\cdot k_\text{огр}+c_\text{в}\cdot G_\text{инф}+c_\text{в}\cdot G_\text{вент}\right)\cdot t_\text{н}}{c_\text{вол}\cdot m_\text{в}} + t_\text{в}\cdot\left(-\frac{G_\text{вент}}{m_\text{в}} - \frac{G_\text{инф}}{m_\text{в}} - \frac{F_\text{огр}\cdot k_\text{огр}}{c_\text{в}\cdot m_\text{в}}\right)\end{cases} \qquad (18)$$

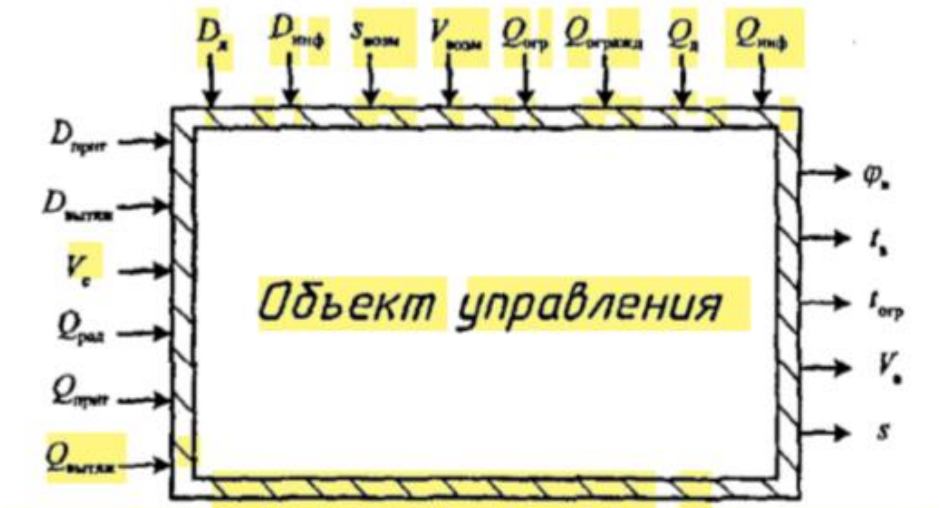

Fig.5. The structure of the simulation model for regulating the parameters of the microclimate of the industrial premises

At the stage of developing a microclimate control system, it is advisable to select an optimality criterion that allows specifying the required set of input and output variables and the form of presentation of control signals, taking into account the purpose of control.
For such a task, an adequate criterion of optimality is the quadratic criterion of optimality (the minimum flow rate of the coolant per unit of production):

$$Y = \int_0^T (\Delta t_\text{в}^2 + \Delta\varphi_\text{в}^2)d\tau \to min, \qquad (19)$$

Where Δt_B is the deviation of the air temperature in the room from the set value, °C; $\Delta\varphi_B$ is the deviation of the relative humidity in the room from the set value, %.
Therefore, the optimization problem is reduced to finding the coefficients of the PID controllers corresponding to the minimum value of the criterion. For these purposes, the Gradient descent method is used - finding the local minimum (maximum) of the function by moving along the gradient.

Conclusions. When developing the ACS of heat supply processes, it was taken into account that the dynamics of thermal processes occurring in heat exchangers is determined by a large number of parameters associated with the design of the apparatus, the conditions of heat exchange on the surfaces of products, the temperature of the media exchanging heat, and the flow rate of the coolant. Therefore, heat exchangers are objects with distributed parameters [1,6, 8, 10].

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