MATHEMATICAL MODEL OF THE DYNAMIC MODE OF HEAT EXCHANGE IN A SOLAR AIR HEATER

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***Abstract**: The mathematical model of dynamic mode thermoexchange in a flat solar air heater (SAH) is given. The reliability of the mathematical model is confirmed by the experience data. A mathematical model of the dynamic mode of heat exchange in a solar air heater is obtained. A comparison of experimental and calculated data shows their good convergence. The average statistical deviations of the calculated and experimental data do not exceed ±11%. The reliability of the mathematical model is confirmed by experimental data on the temperature regimes of SAH.*

***Keywords:** solar air heater, mathematical model, temperature regimes, energy balance, heat-technological process.*

QUYOSH HAVO ISITGICHIDAGI ISSIQLIK ALMASHINISHINING DINAMIK MODELI

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***Annotatsiya:** Yassi quyosh havo isitgichida (QHI) issiqlik almashinuv jarayonining matematik modeli berilgan. Matematik modelning ishonchliligi tajriba ma’lumotlari bilan tasdiqlangan. Quyosh havo isitgichidagi issiqlik almashinuvining dinamik rejimining matematik modeli olingan. Eksperimental va hisoblangan ma’lumotlarni taqqoslash ularning yaxshi yaqinlashishini ko‘rsatadi. Hisoblangan va eksperimental ma’lumotlarning o‘rtacha statistik og‘ishlari ±11% dan oshmasligini ko‘rsatdi. Matematik modelning ishonchliligi QHI ning harorat rejimlari bo‘yicha eksperimental ma’lumotlar bilan tasdiqlangan.*

***Kalit so‘zlar:** quyosh havo isitgichi, matematik model, harorat rejimlari, energiya balansi, issiqlik-texnologik jarayon.*

МАТЕМАТИЧЕСКАЯ МОДЕЛЬ ДИНАМИЧЕСКОГО РЕЖИМА ТЕПЛООБМЕНА В СОЛНЕЧНОМ ВОЗДУХАНОГРЕВАТЕЛЕ

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***Аннотация:** Приведена математическая модель теплоообмена в динамическом режиме в плоском солнечном воздухонагревателе (СВ). Адекватность математической модели подтверждается данными опыта. Получена математическая модель динамического режима теплообмена в солнечном воздухонагревателе. Сравнение экспериментальных и расчетных данных показывает их хорошую сходимость. Средние статистические отклонения расчетных и экспериментальных данных не превышают ±11%. Надежность математической модели подтверждается экспериментальными данными по температурным режимам СВ.*

***Ключевые слова:** солнечный воздухонагреватель, математическая модель, температурные режимы, энергетический баланс, теплотехнологический процесс.*

Introduction.

The aggravation of the tension of the energy balance associated with the limited reserves of fuel and energy resources, the increase in the cost of energy carriers, the deterioration of the environmental situation cause the need for energy-saving environmentally-friendly measures in all countries of the world, including in the Republic of Uzbekistan [1-5].

Thermal drying is a heat-consuming heat-technological process, as a result of which commercial or intermediate products are obtained. The issues of energy saving during thermal drying are part of the overall task of increasing its efficiency and should be considered, taking into account a set of factors that determine economic and environmental economic efficiency [2,3,4,6,7].

The urgency of solving the issues of energy saving during thermal drying is due to the fact that the latter is characterized by high-energy intensity and, as a result, is associated with environmental issues (emissions of fuel combustion products into the atmosphere). Solar energy has been widely used to solve energy-saving problems in heat-technological drying processes. As the experience of many countries shows, the main direction of using solar energy in agriculture is its conversion into low - and medium-temperature heat for the purposes of heat-technological processes. In regions at low latitudes (less than 45^0), the use of solar energy heat is most effective for drying and heat treatment of agricultural raw materials. In particular, in the conditions of the republic (45^0-37^0 north latitude), this is mainly drying and heat treatment of raw cotton, cereals, vegetables, fruits, etc. the use of solar energy in the drying processes of agricultural raw materials can provide savings of 20-33 % of the fuel consumed in agriculture in the drying processes of raw materials [4,5,6]. To calculate the thermal efficiency of the solar air heater (SAH), the initial data are the following thermal parameters: ambient temperature, the value of the total solar radiation incident on the beam-receiving surface of the SAH, the operating time of the SAH, the flow rate of the coolant (air and waste oil), the initial temperature of the coolant, materials and geometric dimensions of the SAH. The heat engineering calculation of the heat balance of the SAH was performed according to a well-known method [7].

Methods and materials

Theoretical research is based on the methods of technical thermodynamics, solar engineering, the theory of heat and mass transfer, hydrodynamics, and aerodynamics. Field studies were carried out on an experimental installation of a solar air heater with a heat accumulator and a computational experiment. The main structural element of the proposed system is the SAH. To determine the parameters of the SAH, a heat engineering calculation was performed according to the following method [8-22].

The solar air heater (Fig.1) with a width of b=1 m, a length of L=5 m is covered from above with glass 1, the bottom of the chamber is covered with a blackened heat receiver 2 (Fig.2,3). The height of the channel h=0.1 m. The heat receiver absorbs the past solar radiation, heats up and connects heat to the air flow in the channel.

Figure 1-3 shows diagrams reflecting the processes of heat exchange in a solar air heater

When developing a mathematical model of the heat exchange regime in the SAH, the following assumptions were made:

1) values G, t_{out} - permanent;

2) air temperature t_{air} the length of the channel varies linearly: $t_{air} = \frac{t_k + t_{out}}{2}$;

3) heat loss through the side walls is not taken into account;

The system of equations of the mathematical model for the dynamic mode of heat transfer in the SAH has the following form:

$$\frac{dQ_b}{d\tau} = a_2 F(t_l - t_в) + a_3 F(t_2 - t_{air}); \tag{1a}$$

$$\frac{dQ_b}{d\tau} = GC_{air}(t_к - t_{out}); \tag{1b}$$

$$I_{ab} = \frac{dQ_{в}}{d\tau} + \frac{dQ_{hr}}{d\tau}. \quad (1c)$$

Equation (1 a) describes the process of heat exchange on the internal surfaces of the SAH. Equations (1b), (1c) express the heat balance of the SAH.

The boundary conditions for equations (1a)-(1c) are the equations of thermal balance on the surfaces of the glazing 1, the heat receiver 2 and the bottom 3 (Fig.3):

$$a_{12}(t_1 - t_l) + I_{пг} / F = a_1(t_1 - t_H) + a_{11}(t_1 - t_o) + a_2(t_1 - t_{air}); \quad (2)$$

$$I_{ab} 2 / F = a_3(t_2 - t_{в}) + a_{12}(t_2 - t_1) + \sum_j \frac{\lambda_j}{\delta_j}(t_2 - t_3); \quad (2a)$$

$$\sum_j \frac{\lambda_j}{\delta_j}(t_2 - t_3) = a_1(t_3 - t_{out}) + a_{13}(t_3 - t_n) \quad (2b)$$

The mass air flow rate is calculated according to the formula

$$G = wpF_o \quad (3)$$

The solar radiation absorbed by the glazing I_{ab1} and the heat receiver I_{ab2} is determined by the relations.

$$I_{ab} = I_{ab1} + I_{ab2};\ I_{ab1} = l_i \cdot k_1;\ I_{ab2} = l_i \cdot a_1 \cdot k_2 \cdot F;\ I_i = I \cdot \cos i \quad (4)$$

where

$$\cos i = \cos a_k \cdot \sinh_0 + \sin a_k \cdot \cosh_0 \cdot \cos\psi;\ \psi = \psi_0 - \psi_k \quad (4a)$$

$$cos\psi_o = \frac{\sinh_o \sin\varphi - \sin\delta_o}{\cosh_o \cosh_\varphi};\quad \sin\psi_к \frac{cos\delta_o \sin\tau_o}{\cosh_o};$$

$$\sinh_0 = \sin\varphi + \cos\varphi \cdot \cos\delta_0 \cdot \cos\tau_0 \quad (4c)$$

The coefficients of convective heat transfer on the inner surfaces of the SAH channel a_2, a_3 are determined by the modified Nussell equation [12-19].

$$Nu_{kj} = 0{,}024\left[I + (d_э / L)^{2/3}\right]\mathrm{Re}_{j_b}^{0{,}786}\, \mathrm{Pr}\, j^{;0{,}45}; \quad (5)$$

$$\mathrm{Re}_j = w_j d_э / v_j;\ Nu_j = a_j d_э / \lambda_j;\ \mathrm{Pr}_j = v_j / a_j \quad (5a)$$

The coefficients of convective heat transfer on the outer surfaces a_1, a_4 are calculated by the empirical expression:

$\alpha_j = (I - \beta A/90)\alpha_j(w)$: by $w \le 5$ m/s $\alpha_j(w) = 6.43 + 3.57w$;

by $w \le 5$m/s $\alpha_j(w) = 7.7_w{}^{0.71}$ (6)

where $\beta = 90 - \alpha_k$; A=0,25.

The coefficient of heat transfer by radiation on the inner surface α_{12} is determined by the formula [4].

$$a_{12} = \sigma \frac{(T_1 + T_2)(T_1^2 + T_2^2)}{1/\varepsilon_1 + 1/\varepsilon_2 - 1} \quad (7)$$

The coefficients of heat transfer by radiation on the external surfaces α_{11} and α_{13} are determined relative to the outdoor air temperature [4] as follows:

$$\alpha_{11} = \sigma\varepsilon(T_1 + T_0)(T_1^2 + T_0^2);\quad \alpha_{13} = \sigma\varepsilon(T_3 + T_n)(T_3^2 + T_n^2); \quad (8)$$

where $T_0 = 0.0552\, T_{out}^{1.5}$; $T_n = T_{out}$

Heat losses in the SAH are determined by the formula

$$\frac{dQ_{hi}}{d\tau} = a_1(t_1 - t_{out}) + a_{11}(t_1 - t_0) + a_4(t_2 - t_{out}) + a_{12}(t_2 - t_n) \quad (9)$$

Results and discussion.

As a result of solving the control system (1a)-(1c) under boundary conditions (2)-(2b), omitting cumbersome transformations, we obtain

$$t_{air} = \frac{1}{A}(expE - Bt_{out} - D); \quad (10)$$

$$t_k = \frac{2}{A}(expE - Bt_{out} - D) - t_{out} \quad (10a)$$

where

$$D = (\frac{A_4 A_1}{A_2} - \frac{a_3}{a_{12}})\frac{I_{n\partial 1}}{F} + \frac{A_4}{A_2}\frac{I_{ab1}}{F}; \quad (10b)$$

$$A = \frac{A_4}{A_2}(A_1 a_2 + a_3) - A_5; \qquad B = \frac{A_4 A_3}{A_2} - A_6;$$

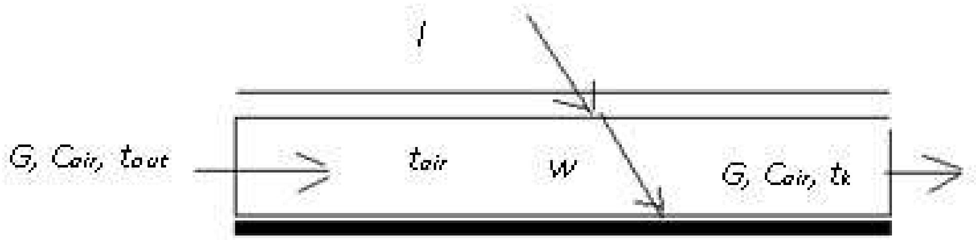

Figure 1. Schematic diagram of heat exchange in the SAH.

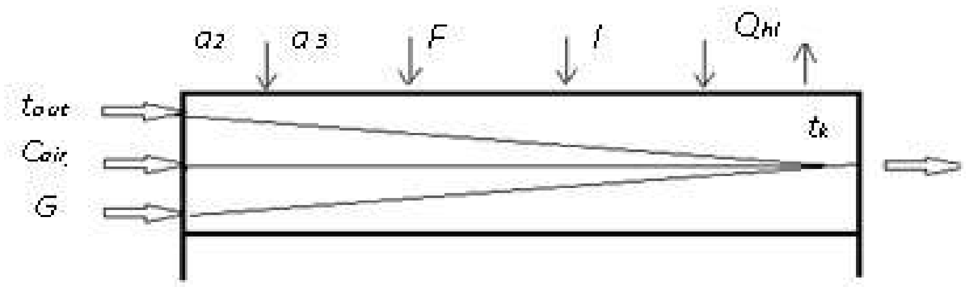

Figure 2. Information structure of the heat exchange process in the SAH:

G, C_{air}, t_n-input variables; t_k-output variable; α_2, α_3, F, I, Q_{hl} -parameters that determine the conditions of the process.

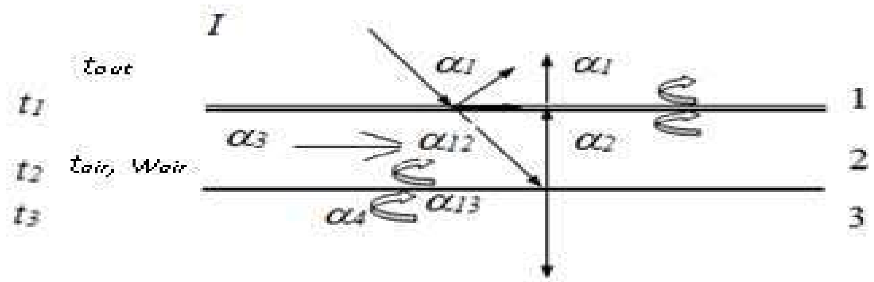

Figure 3. Diagram of the components of the SAH heat balance:

1-glass, 2-heat receiver, 3-bottom.

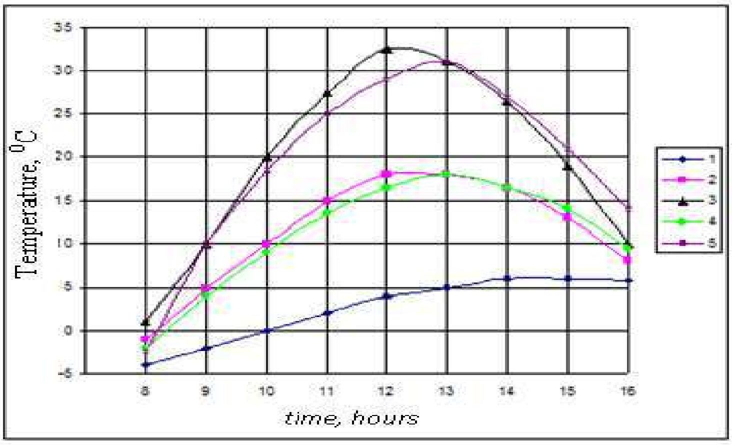

Fig. 4. Change in the air temperature: external t_n (1), average mass t_{air} (2,4) and at the outlet t_k (3,5) SAH for 17.01.2020; 2,3-experiment; 4,5-calculation.

$$E = \frac{AF}{GCair}; \quad A_1 = \frac{(a_3 + a_{12} + a_j)a_4 + a_{13} + a_j) - a_j^2}{a_{12}(a_4 + a_{13} + a_j)}; \quad (11a)$$

$$A_3 = A_1(a_1 + a_{11}) + a_j \frac{a_4 + a_{13}}{a_4 + a_{13} + a_j)}; \quad (11b)$$

$$A_2 = A_1(\alpha_1 + \alpha_{11} + \alpha_2 + \alpha_{12}) - \alpha_{12}:$$

$$A_4 = (a_2 + a_3 \frac{a_4 + a_{11} + a_2 + a_{12}}{a_{12}}); \quad A_5 = a_3 \frac{a_2}{a_{12}} + a_2 + a_3; \quad A_6 = \frac{a_1 + a_{11}}{a_{12}}; \quad (11c)$$

A comparison of experimental and calculated data shows their good convergence (Fig. 4). The average statistical deviations of the calculated and experimental data do not exceed ±11%.

Conventional designations

α - coefficient of thermal conductivity, m^2/s;

C_{air} - specific heat capacity of air, Dj/(kgК);

F, F_o- the area of the beam-receiving surface and the cross-section of the channel, m^2;

G- mass air consumption, kg/s;

h,b,L- height, width channel length, m;

h_o- the height of the sun, deg;

I, I_i- solar radiation incident on surfaces perpendicular to the rays and inclined, W/m^2;

I_{ab} – tiny radiation absorbed by the surface, W;

i- the angle of incidence of the rays on the surface, grad;

k- the coefficient of absorption of solar radiation by the surface;

Q_{air} - the heat of heating the air in the channel, Dj;

Q_{hl} – heat losses, Dj;

Re,Nu,Pr- similarity criteria of Reynolds, Nusselt, Prandtl;

t_{out},t_k - outdoor (initial) and final air temperature, ^oC;

t_{air} - average mass air temperature in the channel, ^oC;

t_o,t_n - the temperature of the sky and the underlying surface, ^oC;

t_1,t_2, t_3,- the temperature of the glazing, the heat receiver and the bottom, ^oC;

T- absolute temperature, К;

w, $w_н$ – air and wind flow rates, m/s;

$\alpha_1 \ldots \alpha_4$- convective heat transfer coefficients, $W/(m^2 \cdot К)$;

$\alpha_{11} \ldots \alpha_{13}$- coefficients of heat transfer by radiation, $W/(m^2 \cdot К)$;

α_j- transmission coefficients of solar radiation by glazing;

α_k- the angle of inclination of the collector relative to the horizontal, grad;
δ- salt thickness, m;
δ_o- declination of the Sun, grad;
ε- surface emissivity;
λ- coefficient of thermal conductivity, W/(m·K);
ν- coefficient of kinematic viscosity, m^2/s
ρ- air density, kg/m^3;
σ- Stefan-Boltzmann constant, $W/(m^2 \cdot K^4)$;
τ, τ_o – time (h) and hour angle (grad);
φ- latitude of the area, grad;
ψ_o ψ_k – azimuths of the Sun and the inclined surface, grad.

Conclusion.

A mathematical model of the dynamic mode of heat exchange in a solar air heater is obtained.

A comparison of experimental and calculated data shows their good convergence (Fig. 4). The average statistical deviations of the calculated and experimental data do not exceed ±11%.

The reliability of the mathematical model is confirmed by experimental data of the temperature regimes of SAH.

The average statistical discrepancy between the experimental and calculated data is 6-16% .

This is due to the fact that the mathematical model does not take into account heat loss through the side walls and structural disadvantages of a real installation. The given mathematical model quite realistically reflects the dynamic mode of the heat balance of the SAH. These include the effect of mass flow and solar radiation on the output temperature and efficiency of the solar collector. The results show that a two-pass collector with a groove has the highest efficiency. Finally, as a result of a comparative study of the influence of flow and radiation, it was found that the thermal characteristics of the solar collector strongly depend on the flow rate.

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