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Combined electric accumulation unit for air heating

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ABSTRACT

The article describes the functional-technological scheme of an energy-efficient combined electric accumulation heating system for agricultural production facilities. A distinctive feature and novelty is the use of direct and accumulative heating. They presented the main provisions of the heat power and aerodynamic calculation for the standard size range of such plants. Based on the results of experimental studies, a number of functional dependencies were obtained that characterize the operating modes of the plant. They determined the value of the heat transfer coefficient from the heat accumulating core to the circulating air depending on the coolant flow rate $\alpha = f(G)$. They presented the technical characteristic of the combined electrical unit current sample.

KEYWORDS: heat accumulation, convector, combined electrical unit, heat transfer, electrical heating, agriculture

Unidad combinada de acumulación eléctrica para calentamiento de aire

RESUMEN

El artículo describe el esquema funcional-tecnológico de un sistema de calefacción de acumulación eléctrica combinada de eficiencia energética para instalaciones de producción agrícola. Una característica distintiva y novedosa es el uso de calentamiento directo y acumulativo. Presentamos las principales previsiones de la potencia térmica y el cálculo aerodinámico para el rango de tamaño estándar de tales plantas. Con base en los resultados de los estudios experimentales, se obtuvieron una serie de dependencias funcionales que caracterizan los modos de funcionamiento de la planta. Determinamos el valor del coeficiente de transferencia de calor desde el núcleo acumulador de calor hasta el aire circulante dependiente del caudal de refrigerante $\alpha = f(G)$. Presentamos la característica técnica de la muestra de corriente de la unidad eléctrica combinada.

PALABRAS CLAVE: acumulación de calor, convector, unidad eléctrica combinada, transferencia de calor, calefacción eléctrica, agricultura.

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Introduction

Agriculture is a large consumer of thermal energy (Tikhomirov et al., 2017). With the introduction of heat accumulators and electricity metering differentiated during the day, the efficiency of its use in technological processes of heat supply to various agricultural production facilities can be increased significantly (Tikhomirov et al., 2019). This ensures the alignment of daily schedules of electrical loads in networks; the reduction of costs to maintain the indicators of electricity quality among consumers; the reduction of the need for the energy system to reserve additional capacity; the possibility of a tangible increase in the power supply of the economy without a significant increase of electric network and distribution substation load; the reduction of electrical equipment set capacity by 40 ... 60%; the reduction of current annual electricity costs for consumers (Kagan et al., 1980).

The results of patent document analysis on the subject under consideration in the World Intellectual Property Organization (WIPO) database confirmed the research relevance. Over the past five years, they received more than 182 patent documents in the field of thermal energy accumulation. The leaders are the UK, China, Russia, the USA and Germany. Various materials and methods of thermal energy accumulation are considered and studied: by solids through their internal energy increase, by phase transition heat use, in saturated liquids, by the means of compressed gas, etc. (Dincer and Ezan, 2018).

The review of latent thermal energy storage devices for material stability increase and efficient load control is presented in (Rathod and Banerjee, 2013; Pielichowska and Pielichowski, 2014). The energy efficiency of heat storage device use with single-phase heat-accumulating materials was substantiated in (Kukolev and Kukelev, 2004).

The aim of the study is to develop an energy-efficient heat accumulating heating system and the methods for its operation calculation in agricultural production.

1. Design diagram of combined electrical unit

Two types of heating devices with heating elements — direct (convector) and accumulative heating (Fig. 1), are located in one thermally insulated casing of our electric unit for air heating (Patent 2638696), in contrast to the known designs.

Depending on the operating mode, these devices can operate simultaneously or separately. The mixing chamber 5 (Fig. 1) is located in the upper part of the electrical unit general case to combine air flows from the heat accumulator 2 and the convector 7, which significantly increases the heater efficiency (Trunov et al., 2016).

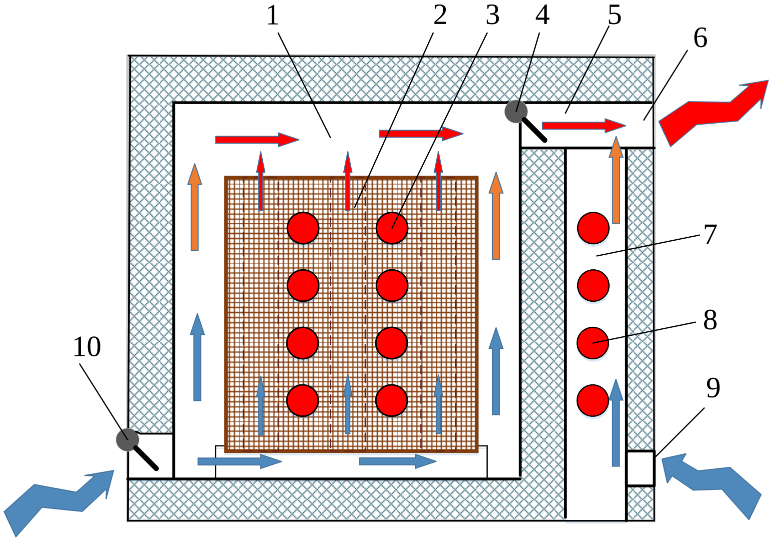

Figure 1 - The accumulation unit scheme:

1– heat accumulator chamber; 2 – heat accumulator core; 3 – accumulator core electric heaters; 4 – adjustable air damper; 5 – mixing chamber; 6 – heated air outlet; 7 – convector; 8 – convector electric heaters; 9 – air inlet; 10 – air damper.

The single-phase heat-accumulating core is enclosed in a protective heat-insulating casing, which has two openings: the inlet bottom and the outlet top with an adjustable air damper 4, which automatically or manually controls the flow of heated air and, accordingly, the heat transfer of the unit (Trunov et al., 2016).

2. Calculation method of electrical accumulation device

The methodology for electric storage device calculation to heat air includes thermal, structural, electrical and aerodynamic calculations. The technique is intended to justify the heat, power and structural parameters at a given power of the heating device for energy-efficient provision of the established operating modes.

2.1. Heat calculation

In general, the process of charging and discharging heat storage devices was solved in (Kukolev and Kukelev, 2003); however, particular solutions are required to calculate specific structures. One of the main parameters of a heater is the volume of the heat-accumulating core V_c, which is determined from the condition of the heat balance, and the following temperature values (average over the thickness of the accumulating core) are taken: when the "charging" process is completed, t_{max} = 500 - 600 ºC, when the "discharge" process is completed t_{min} = 100 - 120 ºC.

$$V_c = P_u \tau_d / \rho_c c_c (t_{max} - t_{min}) \qquad (1)$$

Where P_u is the useful (given) power of the storage heater, W; τ_d — core discharge time, h; ρ_c is the density of the core material, kg/m^3; c_c - specific heat of the core material, J/kg°C.

Based on the existing experience in the design of storage electric heating plants, the volume of the device V is determined from the relation (Trunov and Rastimesin, 2015):

$$V = (1{,}8 - 2{,}0)\ V_c \qquad (2)$$

Using the obtained value of V, they choose the relationship between the heater length, width and height for structural reasons. The height of the heater is determined by the results of aerodynamic calculation. The selected structural parameters of the unit make it possible to determine the outer surface area F_{ext} of the device.

The maximum temperature of the outer surface at the completion of the "charging" process is t_{ext} max = 40 ° C; upon completion of the "discharge" process - t_{ext} min = 20 ºC.

At the beginning of the heating cycle, there are practically no heat losses and the average value of the heat power loss P_{los} from the heating device during the charging process into the surrounding space is calculated for the outdoor temperature of the device surface:

$$t_{ext} = (t_{ext\ max} + t_{ext\ min})/2, \qquad (3)$$

where $t_{ext\ max}$ and $t_{ext\ min}$ – the maximum and the minimum temperature of the heater outer surface, °C.

$$P_{los}=k_r\, \alpha_{ext}\, F_{ext}\, (t_{ext} - t_0), \qquad (4)$$

where α_{ext} is the heat transfer coefficient from the outer surfaces of the heater to the surrounding space (room), W/m^2 °C; t_0 is the temperature of the surrounding space, °C; k_r - the safety factor.

Heat losses associated with the radiation of the heat-transfer surface are not taken into account. For our calculations we can take the following according to the source (Trunov and Rastimesin, 2015) with sufficient accuracy:

$$\alpha_{ext} = 8\ \mathrm{W/(m^2\,°C)};\ k_r = 1{,}2;\ t_0 = 15–20\ °\mathrm{C}$$

The main structural parameters of the storage heater include the thickness of the insulation layer δ_{ti}.

To determine δ_{ti}, one should proceed from the general principles of thermal conductivity theory, considering the lining as a single-layer flat wall. The required wall thickness of the heat-insulating layer (lining), at which the heat loss of the device will correspond to Plos, is determined from the following equation:

$$\delta_{ti} = [\lambda_l F_{av}(t_1 - t_{ext\,max})] / P_{los} \qquad (5)$$

where: λ_l – the coefficient of the lining material thermal conductivity, W/m°C; t_1 - the temperature of the lining inner layer makes 450-500 °C by the end of the charging process.

$$F_{av} = (F_{int}+F_{ext})/2, \qquad (6)$$

where F_{int} is the internal surface area of the lining.

At the first stage of the calculation, F_{av} value is taken equal to the F_{ext} area (Kagan et al., 1980). The accepted values are specified after the calculation the heat accumulating core heating.

After the loss power and thermal insulation thickness are determined, they determine installed (connected) power P_{inst} of the heat accumulator electric heating elements.

$$P_{inst} = \kappa_r P_u\ \tau_d/\tau_{ch} + P_{los}, \qquad (7)$$

where: κ_r = 1,1–1,2 – power reserve factor taking into account the aging of heating elements, the possibility of supply voltage reduction, etc.; τ_{ch} - the duration of the heat accumulator charging period, h.

The power spent on heat accumulating core heating is determined by the following formula:

$$P_c = P_{inst} / \kappa_r - P_{los}. \quad (8)$$

The calculation of the core heating process is reduced to the calculation of plate one-sided heating in constant heat flow mode. The temperature on the surface of the core t_c (from the lining side) is determined by the following expression (Sweenchanski, 1975).

$$t_c = t_{c1} + \frac{P_c \delta_c}{f_c \lambda_c}(2\mathrm{Fo} + 2/3), \quad (9)$$

where: t_{c1} – the initial temperature (average) of the core by the end of the discharge process (taken at 100 °C and specified at the stage of discharge process calculation); λ_c - the coefficient of the core material thermal conductivity, W/m°C; δ_w - the wall thickness between the air channel and the lining, m; f_c - the surface area of the air channels of the core, m^2; $F_o = a_c \tau_{ch} / \delta_c^2$ – Fourier number; a_c – the thermal conductivity ratio of the core material, m^2/s.

The number of channels, the geometric dimensions of their cross section are selected for structural reasons (Fig. 1). They take the wall thickness δ_w roughly between the air channel and the lining equal to 0.065-0.1 m.

Based on the obtained value of the volume of the heat accumulator core V_c and thermal insulation ($\Delta V = V - V_c$), as well as the selected material from which the named structural elements are made, it is possible to determine the amount of accumulated heat by the following expression:

$$Q_{ak} = c_m \gamma_m V_c \Delta t\,; \quad (10)$$

where: c_m – the heat capacity of the core material, J/kg°C; γ_m – the specific gravity of the core material, N/m^3; Δt is the temperature difference, °C.

The amount of heat accumulated by the heater allows you to determine the time of its heating τ_h:

$$\tau_h = Q_{ak} / (0{,}8P_{inst} - 0{,}5P_{los}) \quad (11)$$

where: P_{los} – heat loss by the device, W.

The resulting heating device heating time must not exceed the time of the reduced tariff in the power system.

2.2. Aerodynamic calculation

The purpose of aerodynamic calculation is to determine the height of the instrument housing H_c, the interaxial distance H and the area of the inlet F_1 and outlet F_2 hole in the instrument housing (Fig. 2).

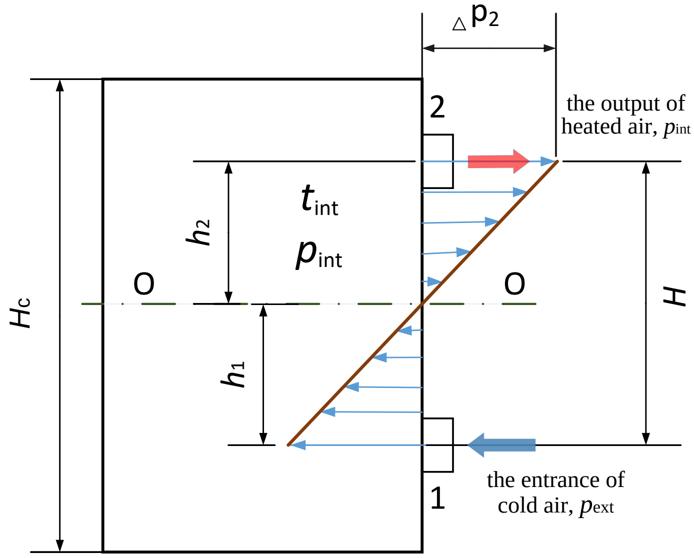

Figure 2 - The scheme of the accumulation unit aerodynamic calculation

The natural air flow in the heater is created due to the difference in air temperatures and, accordingly, the densities of air heated in the heat accumulating core of the heater and the air in the heated room (Rastimesin and Trunov, 2016).

Within the height H, which mainly affects the height of the heater unit H_c, there is a neutral plane 0-0, in which the pressure p inside the device will be the same as outside, i.e. the pressure difference is zero (Basta and Rudnev, 2010). The pressure of the external and internal air in the plane of the hole 1:

$$p_{1ext} = p + h_1 \rho_{ext} g, \tag{12}$$

$$p_{1int} = p + h_1 \rho_{int} g, \tag{13}$$

where p is the air pressure in the plane O-O, Pa; h1 is the distance from the center from hole 1 to the neutral plane, m; ρext is the density of outdoor air, kg/m^3; ρint is the density of internal air at t_{cmin}, kg/m^3; t_{cmin} is the minimum core temperature, i.e. at the end of discharge, °C; g is the acceleration of gravity, m/s^2.

According to the thermophysical properties, $\rho_{ext} > \rho_{int}$; therefore, $p_{1ext} > p_{1int}$. It follows that the outside air enters hole 1.

The pressure difference in the plane of the hole 1 is equal to:

$$\Delta p_1 = p_{1ext} - p_{1int} = (p + h_1 \rho_{ext} g) - (p + h_1 \rho_{int} g) = h_1(\rho_{ext} - \rho_{int})g, \tag{14}$$

Similarly, the pressure of the external and internal air in the plane of the hole 2 make the following: $p_{2ext} = p - h_2 \rho_{ext} g$, $p_{2int} = p - h_2 \rho_{int} g$.

Since $h_2 \rho_{ext} g > h_2 \rho_{int} g$, then $p_{2ext} > p_{2int}$. Internal warm air will go out of the hole 2 of the heater to the outside and the pressure difference in the plane of the hole 2 will be the following:

$$\Delta p_2 = p_{2int} - p_{2ext} = (p - h_2 \rho_{int} g) - (p - h_2 \rho_{ext} g) = h_2(\rho_{ext} - \rho_{int})g, \tag{15}$$

The total pressure required for air to move through the opening 1 and 2:

$$\Delta p = \Delta p_1 + \Delta p_2 = h_1(\rho_{ext} - \rho_{int})g + h_2(\rho_{ext} - \rho_{int})g = H(\rho_{ext} - \rho_{int})g. \quad (16)$$

According to a well-known expression in hydraulics:

$$\Delta p = U^2\rho/2g \quad (17)$$

and after its conversion relative to the air velocity U in the hole 1 and 2 respectively we get the following:

$$U_1 = \sqrt{2gh_1(p_{ext} - p_{int})/p_{ext}};\ U_2 = \sqrt{2gh_2(p_{ext} - p_{int})/p_{int}}. \quad (18)$$

The values of the air velocity at the inlet and outlet are taken for structural reasons at the level of 4–7 m/s [13]. Then

$$h_1 = \frac{U_1^2 p_{ext}}{2g(p_{ext} - p_{int})},\ h_2 = \frac{U_2^2 p_{int}}{2g(p_{ext} - p_{int})}, \quad (19)$$

Using the found value $H = h_1 + h_2$ and taking into account the design features of the heater with heat accumulation, the height of the heater body H_c is determined. The found value of the interaxial distance H determines the gravitational air pressure in the device Δp. At this pressure, air flows through the device with the speed U_1 and U_2 set at the inlet and outlet.

Inlet and outlet areas F_1 and F_2:

$$F_1 = G_1/(3600\mu U_1) = G_1/(3600\mu\sqrt{2gh_1(\rho_{ext} - \rho_{int})/\rho_{ext}}; \quad (20)$$

$$F_2 = G_2/(3600\mu U_2) = G_2/(3600\mu\sqrt{2gh_2(\rho_{ext} - \rho_{int})/\rho_{int}}; \quad (21)$$

where $G_{1,2}$ - the air flow through the holes of the device, m^3/h; μ is the flow coefficient that takes into account the pressure loss during the jet compression when it passes through the hole. Thus, the flow occupies the cross section smaller than the hole dimension (μ = 0.65).

Assuming that $G_1 = G_2$, the air flow is determined from the following expression:

$$G_{1,2} = 3600P_u/c\rho_{out}\Delta t_{ov}, \quad (22)$$

where P_u is the useful (given) device power, W; , °C; Δt_{ov} - the temperature of air overheating in the heating device, °C; ρout is the density of the air leaving the device at overheating temperature, kg/m^3.

2.3. Main provisions of electrical calculation

The tubular heating elements (THE) or spirals in ceramic tubes are used, as a rule, as the heating elements in such heating devices.

The calculation of electric heating elements includes the selection of the specific surface power of the heater at which the temperature on the heater surface does not exceed the permissible value. At that, the power released in the heating elements is transferred to the core during charging with thermal conductivity.

Tubular heating elements (THE) are used as the heating elements in the developed heating device, that are tightly installed in the heat accumulator core and located symmetrically. Heat transfer through a cylindrical wall should be taken as an ideal design scheme for the specific surface power determination of a heating element.

The calculation includes the determination of the specific surface power of the heater at which the temperature on the heater surface does not exceed the allowable value.

For tubular spiral heaters, heat transfer through a uniform cylindrical wall, the thermal field of which will be one-dimensional, and isothermal surfaces are concentric cylinders whose axis coincides with the axis of the heater can be an ideal design scheme.

The actual specific surface power of the heating element working surface has the following form (Sweenchanski, 1975):

$$W_2 = k\frac{2{,}32\pi 10^{-2}\lambda}{H\ln D_2/D_1}\left(t_1 - t_2\right), \qquad (23)$$

where k is the coefficient of the heating element configuration; λ is the coefficient of the heat accumulator material thermal conductivity; H is the step of the heaters (the distance between the heaters; D_1 is the outer diameter of the tubular heating element; the equivalent diameter $D_2 = 2/\pi(A + B)$; the values of A and B are the geometric dimensions of the heat storage core (A is the height and B is the thickness).

At that the specific surface power of the heater (TEH) will be equal to:

$$W_1 = k\frac{2{,}3210^{-2}\lambda(t_1 - t_2)H}{\pi d D_1 \ln D_2 / D_1}, \quad (24)$$

d – the inner diameter of the heating element tube, mm.

The configuration coefficient k of tubular heating elements can be determined from known curves for wire frame heaters.

The obtained value of the specific surface power of the heater W_1 must be less than or equal to the allowable specific surface power of the heater W_v.

$$W_1 \leq W_v. \quad (25)$$

They developed and tested the generalized calculation method, which allows one to determine the structural, heat-energy and electrical parameters of energy-efficient combined heat-storage electrical devices of a standard size range for air heating in agricultural premises.

3. Device main modes and experimental research

The main operating modes of the combined electric accumulating unit for air heating (Rastimesin and Trunov, 2016): the heat accumulator charging during the period of the reduced electricity tariff; the heat accumulator charging during the period of the reduced electricity tariff + operation of a direct electric heater - electric convector; the heat accumulator discharge during the period of the high tariff for electricity; the heat accumulator discharge during the period of the high electricity tariff + the operation of a direct electric heater - electric convector; independent operation of the electric convector according to a given program.

Based on the calculation results, they developed the operational experimental model of a combined electrical device with heat storage and the set power of 7.2 kW (Fig. 3, Table 1).

The graph of charging and uncontrolled discharge of the heat storage unit is shown on Figure 4. During unregulated discharge of the device, the temperature of the heat accumulator t_c and the heat flux Q (kW) are changed exponentially from the heat storage core to the circulating air:

$$t_c=545{,}3e^{-0{,}3\tau}, \tag{26}$$

$$Q=16{,}6e^{-0{,}41\tau}. \tag{27}$$

Figure 3 - Combined electrical device with heat accumulation

Table 1 – Device Specifications

Power voltage, V	380/220
Heat storage power, W	4,8
Electric convector power, W	2,4
Heat accumulator charging time, h	4
Heat storage discharge time, h	4-8
Device weight, kg	200

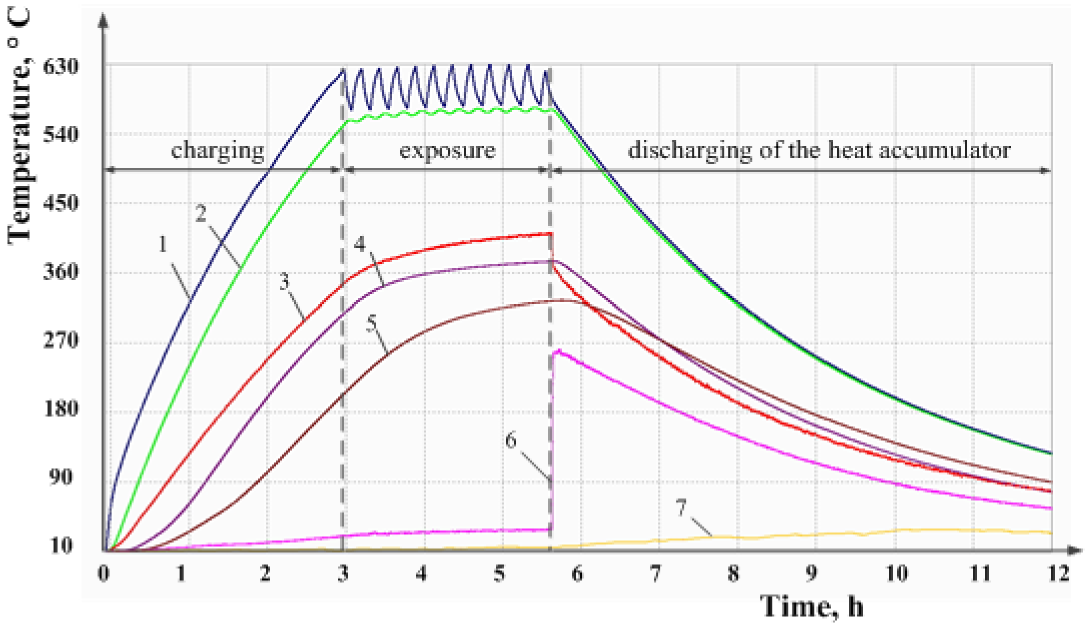

Figure 4 - Schedule of charging and unregulated discharge of the combined heat storage unit:

1 – the electric heater temperature; 2 – the heat accumulator temperature; 3, 4, 5 - the inner layer temperature of thermal insulation; 6 – the temperature of the outgoing heated air; 7 – indoor temperature.

The value of the heat transfer coefficient α (W/m^{2}K) from the heat-accumulating core to the circulating air is determined depending on the flow rate of the coolant G (m^3/h): $\alpha = 0.51G - 18.77$. This parameter is decisive for heat flux calculation during the regulated and unregulated discharge of the device heat accumulator.

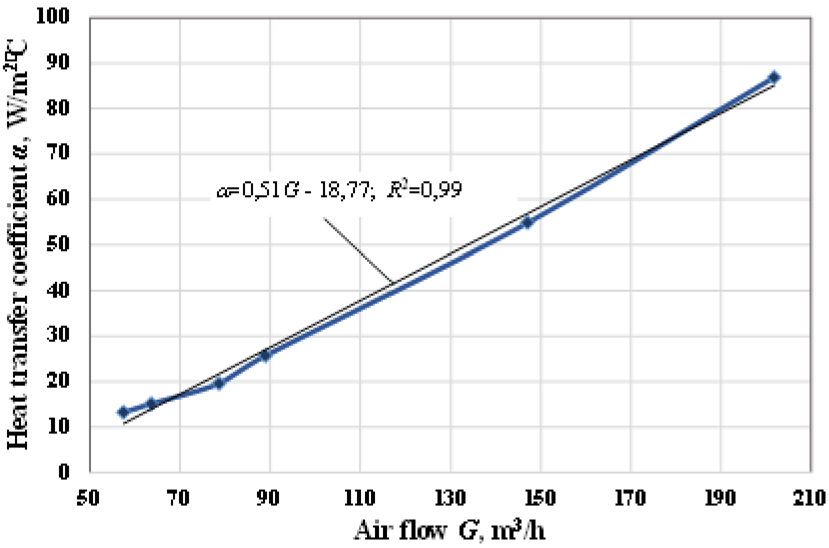

Figure 5 - The dependence of the heat transfer coefficient from the heat storage core to circulating air

4. Summary

They developed the functional diagram of an energy-efficient combined electric storage heating system for agricultural facilities. A distinctive feature is the use of direct and accumulation heating in one device. The accumulation of heat is carried out by a solid body made of chromomagnesite by its internal energy increase. The novelty of the design is protected by the Russian Federation patent (Patent 2638696).

They developed and presented the main provisions of the methodology for air heating electric storage device calculation, which includes the thermal, electrical and aerodynamic calculations necessary to justify the heat and power and structural parameters of the device at a given power and established operating modes.

Based on the calculation results, they developed the working model of an energy-efficient heating system for agricultural facilities with the power of 7.2 kW. The power of the heat accumulator makes 4.8 kW.

According to the results of experimental studies, they obtained a number of functional dependencies characterizing the device operating modes. The value of the heat transfer coefficient α (W/m^{2}K) from the heat-accumulating core to the circulating air is determined depending on the flow rate of the coolant G(m^3/h): $\alpha = 0.51G - 18.77$.

The current device sample has been introduced into production at the dairy farm. The annual economic effect amounted to 124 thousand rubles as compared to the electric heating plant without heat storage.

Conflict of interests

The author confirms that the presented data do not contain a conflict of interest.

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