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DEVELOPMENT OF THE METHOD OF THERMOTECHNICAL CALCULATION OF THE ENERGY-SAVING EFFECT IN HELIUM-DRYING EQUIPMENT

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Abstract. *In the article the developed method of heat engineering calculation of energy-saving effect of helio drying equipment is considered. The dependence of thermal efficiency on the amount of aspirated air in the air layer of the helio dryer.*

Key words: *helium-drying equipment, chilled and drying chamber, solar air collector.*

INTRODUCTION

The heat engineering calculation in the helio drying equipment of this type consists in determining the amount of additional energy that is generated in the channel (a ventilated air gap) formed by the outer blackened wall surface of the drying chamber that simultaneously functions as the heat receiver of the helio air collector and the translucent pellicle sheath [1]. The supply of heat to the heat exchange surface of the heat receiver (radiation-absorbing panel) of this type of solar air heaters occurs from two sides: as a result of absorption of solar radiation passing through the translucent shell and heat transfer through the walls of the drying chamber. A schematic diagram of the components of the heat balance of a helio air collector of the type considered is shown in fig.1.

In accordance with fig.1, we compile balance equations for determining the useful flow of the surface of the heat receiver and the wall of translucent insulation.

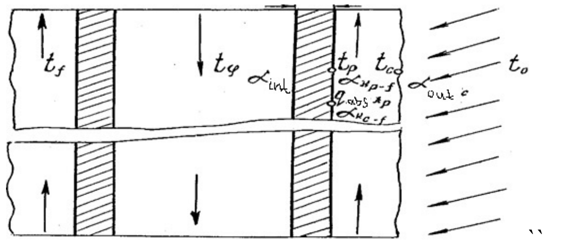
In a cross section in the flow direction in the layer and the solar radiation absorbed by the heat transferred from the heater (drying agent) is heated within the chamber outer wall surface of the drying chamber (i.e., thermoreceiver) to a temperature t_p. From the heat collector, the heat is transferred by convection to the heat carrier (spent drying agent) with temperature t_f and by radiation to the inner surface of the translucent shell with temperature t_c. The heat carrier can receive some amount of heat (by convection) from the inner surface of the light of the transparent shell, ша $t_c>t_f$. Otherwise, heat is transferred from the coolant to the transparent transparent shell by convection.

MATHEMATICAL DESCRIPTION

In this case, the expression for the flow of useful energy has the form [1].

$$q_{useful} = \alpha_{\kappa_{p-f}}(t_p - t_f) + \alpha_{\kappa_{c-f}}(t_c - t_f) \ , \qquad (1)$$

$$t_{\varphi_{entry}} = 100^oC \qquad \delta_W \qquad t_{f_{exit}} = 100^oC$$



$$t_{\varphi_{exit}} = 46^oC \qquad t_{f\,entr} = 46^oC$$

Fig.1. *Schematic diagram of the components of the heat balance of the solar air collector*

In the figure 1, $\alpha_{\kappa_{p-f}}$, $\alpha_{\kappa_{c-f}}$ - respectively, the coefficients of convective heat transfer of the surfaces of the heat receiver and the light of the transparent shell forming the air layer.

The amount of radiant energy absorbed by the outer surface of the drying chamber, i.e. heat receiver (q_{absop_p}) and heat, Resulting from the heat loss of the drying chamber (q_{mn_κ}), equal $K_{p-f}(t_p-t_f)$, Is transmitted to the heat carrier (by convection) and to the inner surface of the transparent transparent shell (radiated by the receiver) (by radiation), i.e.

$$q_{absop_p} + K_{p-f}(t_p - t_f) =$$
$$= \alpha_{\kappa_{p-f}}(t_p - t_f) + \alpha_{r_{p-c}}(t_p - t_c) \qquad (2)$$

where K_{p-f} - Coefficient of heat transfer from the main coolant inside the drying chamber with temperature - t_f to the outer surface of the drying chamber, $\alpha_{r_{p-c}}$ coefficient of radiant heat exchange between the outer surface of the heat receiver and the inner surface of the transparent shell. Value K_{p-f} in (2) is determined from the condition that heat flows on conjugate surfaces are equal, i.e.

$$K_{p-f} = \alpha_{p-f}\left(1 + \alpha_{p-f}\frac{\lambda_w}{\delta_w}\right)^{-1}, \qquad (3)$$

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where λ_w and δ_w respectively, the thermal conductivity and the thickness of the wall material of the drying chamber.

Heat obtained by the inner surface of the transparent transparent shell $\alpha_{r_{p-c}}(t_p - t_c)$, transferred to the heat carrier $\alpha_{\kappa_{c-f}}(t_c - t_f)$ and is lost in the environment $\alpha_{out_c}(t_c - t_o)$, i.e.

$\alpha_{l_{p-c}}(t_p - t_c) = \alpha_{\kappa_{c-f}}(t_c - t_f) + \alpha_{out_c}(t_c - t_o)$, (4)

where α_{out_c} - coefficient of total heat transfer of the outer surface of the translucent shell.

When writing the balance equation (4), the temperature distribution along the thickness of the material of the translucent shell is not taken into account because of its excessive smallness (polyethylene pellicle (film) 0.2 mm thick).

To represent the density of the useful energy flux as functions of α_{out_c}, $\alpha_{\kappa_{p-f}}$, $\alpha_{\kappa_{c-f}}$, K_{p-f}, t_p, t_f and t_o it is necessary from the system (1), (3) and (4) exclude the surface temperatures of the heat receiver (t_p) and the translucent shell (t_c).

From the joint solution of equations (2) and (4) we have

$$t_p - t_f = \frac{(q_{absor_p} + K_{p-f}t_f)(\alpha_{r_{p-c}} + \alpha_{\kappa_{c-f}} + \alpha_{out_c}) + \alpha_{r_{p-c}}\alpha_{out_c}t_o - t_f(K_{p-f}\alpha_{r_{p-c}} + K_{p-f}\alpha_{\kappa_{c-f}} + K_{p-f}\alpha_{out_c} + \alpha_{out_c}\alpha_{\kappa_{p-f}})}{K_{p-f}\alpha_{r_{p-c}} + \alpha_{\kappa_{p-f}}\alpha_{\kappa_{p-c}} + K_{p-f}\alpha_{\kappa_{c-f}} + \alpha_{\kappa_{p-f}}\alpha_{\kappa_{c-f}} + \alpha_{r_{p-c}}\alpha_{\kappa_{c-f}} + K_{p-f}\alpha_{out_c} + \alpha_{\kappa_{p-f}}\alpha_{out_c} + \alpha_{out_c}\alpha_{r_{p-c}}} \quad (5)$$

and

$$t_c - t_f = \frac{(q_{absor_p} + K_{p-f}t_f)(\alpha_{r_{p-c}} + t_o(K_{p-f}\alpha_{out_c} + \alpha_{\kappa_{p-f}}\alpha_{out_c} + \alpha_{out_c}\alpha_{r_{p-c}}) - t_f(K_{p-f}\alpha_{r_{p-c}} + K_{p-f}\alpha_{\kappa_{c-f}} + K_{p-f}\alpha_{out_c} + \alpha_{out_c}\alpha_{\kappa_{p-f}} + \alpha_{out_c}\alpha_{r_{p-c}})}{K_{p-f}\alpha_{r_{p-c}} + \alpha_{\kappa_{p-f}}\alpha_{\kappa_{p-c}} + K_{p-f}\alpha_{\kappa_{c-f}} + \alpha_{\kappa_{p-f}}\alpha_{\kappa_{c-f}} + \alpha_{r_{p-c}}\alpha_{\kappa_{c-f}} + K_{p-f}\alpha_{out_c} + \alpha_{\kappa_{p-f}}\alpha_{out_c} + \alpha_{out_c}\alpha_{r_{p-c}}} \quad (6)$$

Substituting (5) and (6) into (1) and after some algebraic transformations we have

$q_{absor} = \eta_{hl}\left[q_{absor_p} + K_{p-f}(t_f - t_p) - k_{rc}(t_f - t_o)\right]$, (7)

where η_{hl} - thermal efficiency of the heat collector of the solar air collector; k_{rc} - the reduced coefficient of heat losses through the translucent shell of the drying chamber.

Values η_{hl} and k_{rc} is determined from expressions

$$\eta_{hl} = \left[1 + \frac{K_{p-f}(\alpha_{r_{p-c}} + \alpha_{\kappa_{c-f}} + \alpha_{out_c}) + \alpha_{out_c}\alpha_{r_{p-c}}}{\alpha_{r_{p-c}}\alpha_{\kappa_{c-f}} + \alpha_{\kappa_{p-f}}\alpha_{r_{p-c}} + \alpha_{\kappa_{p-f}}\alpha_{\kappa_{c-f}} + \alpha_{out_c}\alpha_{\kappa_{c-f}}}\right]^{-1} \quad (8)$$

and

$$k_{hl} = \alpha_{out_c}\frac{\alpha_{r_{p-c}}\alpha_{\kappa_{c-f}} + \alpha_{\kappa_{p-f}}\alpha_{r_{p-c}} + \alpha_{\kappa_{p-f}}\alpha_{\kappa_{c-f}} + K_{p-f}\alpha_{\kappa_{c-f}}}{\alpha_{r_{p-c}}\alpha_{\kappa_{c-f}} + \alpha_{\kappa_{p-f}}\alpha_{r_{p-c}} + \alpha_{\kappa_{p-f}}\alpha_{\kappa_{c-f}} + \alpha_{out_c}\alpha_{\kappa_{c-f}}} \quad (9)$$

In the particular case, when the walls of the drying chamber are ideally heat-insulated, i.e. $K_{p-f}=0$, then we obtain the expressions known from [2-3] for q_{absor}, η_{hl} and k_{rc}

$$q_{absor} = \eta_{hl}\left[q_{absor_p} - k_{rc}(t_f - t_o)\right], \quad (10)$$

where

$$\eta_{hl} = \left[1 + \frac{\alpha_{out_c}\alpha_{r_{p-c}}}{\alpha_{r_{p-c}}\alpha_{\kappa_{c-f}} + \alpha_{\kappa_{p-f}}\alpha_{r_{p-c}} + \alpha_{\kappa_{p-f}}\alpha_{\kappa_{c-f}} + \alpha_{out_c}\alpha_{\kappa_{c-f}}}\right]^{-1} \quad (11)$$

$$k_{hl} = \alpha_{out_c}\left[1 + \frac{\alpha_{out_c}\alpha_{r_{p-c}}}{\alpha_{r_{p-c}}\alpha_{\kappa_{c-f}} + \alpha_{\kappa_{p-f}}\alpha_{r_{p-c}} + \alpha_{\kappa_{p-f}}\alpha_{\kappa_{c-f}} + \alpha_{out_c}\alpha_{\kappa_{c-f}}}\right]^{-1} \quad (12)$$

In the absence of radiant absorption on the outer surface of the wall of the drying chamber, i.e. $q_{absor_p} = 0$, Which corresponds to the operating mode of the drying unit at night, the solution (7) takes the form

$q_{absor} = \eta_{hl}\left[K_{p-f}(t_p - t_f) - k_{rc}(t_f - t_o)\right]$. (13)

As it follows from the formula (7), the economy of traditional energy sources in the use of the considered heat recovery method depends on the thermal efficiency of the solar air heater (ventilated air layer), the bottom (the radiation-absorbing panel) of which is combined with the outer blackened surface of the drying chamber, η_{hl}, determined by the formula (8).

As follows from (8), value η_{hl} with other things being equal (mean $\alpha_{r_{p-c}}, \alpha_{r_{p-c}}$ and k_{p-f}) depends on $\alpha_{\kappa_{c-f}}$ *and* $\alpha_{\kappa_{p-f}}$.

The latter in turn depend on the velocity (υ_f) and temperature (t_f) of the coolant (spent drying agent) in the layer under consideration

It is of scientific and practical interest to establish the optimal value of the waste drying agent flow in the channel, where it is partly regenerated.

The linear velocity of the drying agent in the ventilated interlayer is related to its volumetric flow rate (G_{sda}) by formula

$$v_f = \frac{G_{sda}}{n_{ab}}, \quad (14)$$

where n - number of spiral channels in the air layer, a, b - respectively, the channel width and height.

The problem under consideration is reduced to the establishment of a graphical dependence η_{hl} from G_{sda}.

Addiction $\alpha_{\kappa_{c-f}}$ *and* $\alpha_{\kappa_{p-f}}$ from ν_f under turbulent motion of the coolant at $(l/d_{equ})>50$ is determined from the criterial formula [4].

$Nu{,}21Re^{0,8}Pr^{0,43}$. (15)

The equivalent diameter of the channel (d_{equ}) interlayer, a certain ratio

$$d_{equ} = \frac{4F}{P} = \frac{4ab}{2(a+b)} \quad (16)$$

and with a=0,1 и b=0,2 as already noted above, is equal to 0,133 m.

Taking into account the dependence λ_f and ν_f (and correspondingly Pr_f) from the temperature of the coolant, also the values d_{equ} (0,133 m), The criterion equation (15) can be rewritten in the form

$\alpha_k = A_t G_{sda}^{0.8}$, W/(m·^{0}C), (17)

where A_f - coefficient of dependence α_k from t_f.

According to the calculations carried out t_f=40°C (λ_f =2,76·10^{-2} W/(m·°C), ν_f= 16,96·10^{-6} m^2/s и Pr_f=0,701), t_f=50°C (λ_f =2,83·10^{-2} W/(m·°C), ν_f = 17,95·10^{-6} m^2/s and Pr_f=0,699), t_f=60°C (λ_f =2,90·10^{-2} W/(m·°C), ν_f= 18,97·10^{-6} m^2/s и Pr_f=0,698), the corresponding values of A_t are 0,0913, 0,0894 and 0,0876 $\frac{W}{m^2 \cdot {}^oC}\left(\frac{m^3}{h}\right)^{-0,8}$.

Graphical processing of the results of calculations of the dependence of A_t on t_f. In the interval t_f. From 40 to 60°C allowed us to establish the following approximate expression

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$A_t = 0{,}0987 - 0{,}000185\ t_f$ (17')

Value $\alpha_{\kappa_{c-f}}$ *and* $\alpha_{\kappa_{p-f}}$ while taking into account the influence of the centrifugal inertial force in curved channels with curvature (d_{eq}/R), it is determined from [5-6]

$$\alpha_{\kappa_{c-f}} = \alpha_{\kappa_{p-f}} = \alpha_{\kappa}\varepsilon_R\ , \qquad (18)$$

where

$$\varepsilon_R = 1 + 1{,}77\frac{d_{equ}}{R}\ , \qquad (19)$$

With the size of the drying chamber 0,9 m, d_{eq}= 0,133 m value ε_R in (18) is 1,2616.

Substituting (17) and (19) into (18), we obtain

$\alpha_{\kappa_{c-f}} = \alpha_{\kappa_{p-f}}$ =0,1245(1 - 0,00187 t_f.) $G^{0,8}_{sda}$, W/(m·°C) (19')

(unit of measurement G_{sda} in (19) - m^3/h)

The value of K_{p-f} in (18) is determined from the expression obtained from the condition for the equality of the specific heat fluxes on the conjugate surfaces, i.e.

$$K_{\rho-f} = \left[\frac{1}{\alpha_{int}} + \frac{\delta_w}{\lambda_w}\right]^{-1} \qquad (20)$$

where α_{int} - coefficient of heat exchange on the inner surface of the drying chamber, λ_w and δ_w - the thermal conductivity of the material and the wall thickness of the drying chamber. In its turn

$$\alpha_{int} = \alpha_{int}^{con} + \alpha_{int}^{rad}\ , \qquad (21)$$

where α_{int}^{con}, α_{int}^{rad} - convective and radiant components of the heat transfer coefficient on the inner surface of the drying chamber.

Value α_{int}^{con} in (21) is determined from the criterion dependence of heat (15) with appropriate consideration of the temperature and flow velocity of the primary drying agent near the wall of the drying chamber.

At the drying agent temperature in the drying chamber ($t_{f_{ent}}$)100 °C and exit from it ($t_{f_{ex}}$) 46°C [2] and consumption 1087 m^3/h, the size of the drying chamber 0,9 m, as well as the average porosity of the drying chamber 0,4, the drying rate of the drying agent in the chamber is 1,1865 m/s, value α_{int}^{con}, determined on the basis of (15) with t_f=73 °C is 3,59 W/(m^2·°C).

Value α_{int}^{con} in (21), determined from a well-known dependence

$$\alpha_{int}^{rad} = \varepsilon_{rc}\sigma(0{,}81 + 0{,}01t)\ , \qquad (22)$$

at ε_{rc}= 0,9608, t=70 °C is 8,23 W/(m^2·°C). Value α_{int}, the result obtained on the basis of (21) is 11,82 W/(m^2·°C), a K_{p-f} at λ_w=40 W/(m^2·°C) и δ_w=0,02 m is 11,75 W/(m^2·°C).

Value $\alpha_{r_{p-c}}$ at the emissivity of the outer wall of the drying chamber 0,98, the translucent sheath of the polyethylene film 0,25 [2] and the external dimensions of the drying chamber 0,9 m and the translucent sheath 1,1 m is 1,97 W/(m^2·°C). Such a low value of αl as compared to c $\alpha_{int_{p-c}}^{rad}$ (8,23 W/(m^2·°C)) due to the low emissivity of the polyethylene film than the outer wall of the drying chamber.

Value α_{out_c} analogically α_{int} also consists of convective (α_{out}^{con}) radiant (α_{out}^{rad}) components, i.e.

$$\alpha_{out_c} = \alpha_{out}^{con} + \alpha_{out}^{rad}\ . \qquad (23)$$

Due to the fact that the drying chamber developed by the authors has a rectangular shape and is located vertically [3] value α_{out_c} in (23) in accordance with [1] can be determined by the formula

Nu= 0,312 $Re^{0.8}$, (24)

In which the length of the flow is taken as the determining dimension [4] i.e. πd/2. With an average daily wind speed [1] 2 m/s and d=0,9 m and at t=30 °C value α_{out}^{con}, determined by (4.24) is 8,0 W/(m^2·°C).

Value α_{out}^{rad}, determined from the analogous formula (22), but with appropriate consideration for the temperatures of the surrounding objects, the sky, the angular coefficients of the radiant fluxes between the outer surface of the vertical cylinder and the celestial sphere, and also the surrounding objects is 1,44 W/(m^2·°C). Value α_{out_c}, determined by (23) in this case is 9,44 W/(m^2·°C).

Substituting the value K_{p-f}=11,75 W/(m^2·°C), $\alpha_{r_{p-c}}$ =1,97 W/(m^2·°C), α_{out_c} =9,44 W/(m^2·°C) the (4.28) and assuming that $\alpha_{\kappa_{c-f}} = \alpha_{\kappa_{p-f}}$, get

$$\eta_{hl} = \left[1 + \frac{152.67 + 11.75\alpha_{\kappa_{c-f}}}{(13.38 + \alpha_{\kappa_{c-f}})\alpha_{\kappa_{c-f}}}\right]^{-1} \qquad (25)$$

Taking into account the value $\alpha_{\kappa_{c-f}}$ ($\alpha_{\kappa_{p-f}}$) by (19′), the solution of (25) can be represented in the form

$$\eta_{hl} = \left\{1 + \frac{152{,}67 + 1{,}463(1 - 0{,}00187t_f)G^{0,8}}{\left[1{,}666 + 0{,}155(1 - 0{,}00187t_f)G^{0,8}\right](1 - 0{,}00187t_f)G^{0,8}}\right\}^{-1} \qquad (25')$$

RESULTS AND DISCUSSION

The results of calculations for establishing the dependence $\eta_{hl}=f(G_{sda})$ on the basis of formulas (17) and (25) for t_f = 40, 50 and 60 °C are given in fig.2. As can be seen from the graphs, an increase in the average temperature of the spent drying agent in the ventilated air layer will lead to a slight decrease η_{hl} (within 1,1-1,3) and for this reason the dependence curves $\eta_{hl}=f(t_f)$ at various G_{sda} almost closely coincide with each other.

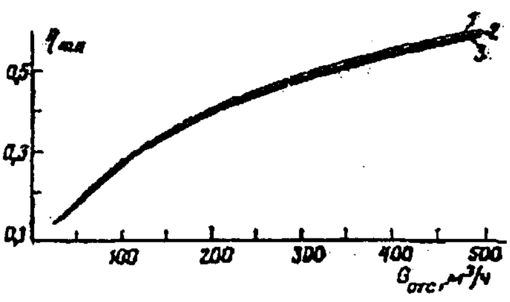

Fig.2. *Dependencies $\eta_{hl}=f(G_{sda})$ at t_f=40 (1), 50 (2) and 60°C (3)*

From figure 2 also shows that the main growth η_{hl} account for changes G_{sda} range from 25 to 250 m^3/h. So when changing G_{sda} from 25 to 250 m^3/h and t_f=40^o and 50 ^{o}C the growth of η_{hl} is 3,87 times, and at t_f = 60 ^{o}C 3,91 times.

A further increase in G_{sda} (from 25 to 250 m^3/h) does not lead to a significant increase η_{hl}. So, if you change G_{sda} from 25 to 250 m^3/h appropriate growth η_{hl} be 1,30 times at t_f=40 ^{o}C, 1,31 times at t_f=50 and 60 ^{o}C .

CONCLUSION

Thus, from the character of the dependence $\eta_{hl}=f(G_{sda})$, it can be concluded that the optimum value of G_{sda} for the developed drying plant is 250 m^3/h.

The possible changes in the value of G_{sda} according to the production need within 10-15% of this (i.e. 250 m^3/h) do not lead to significant changes η_{hl}.

It should be noted that from the heat engineering point of view, increasing the G_{sda} by more than 250 m^3/h leads to an increase in the relative humidity of the primary drying agent as a result of mixing the latter with the spent (i.e., sucked) drying agent. Reduction of G_{sda} from 250 m^3/h, as can be seen from the figure, leads to a decrease in the heat engineering efficiency of the suction.

REFERENCES

[1]. Iskandarov Z.S. Combined solar-fuel drying plants. - Tashkent: "FAN", 2005. P.120-126.

[2]. Iskandarov Z.S., Halimov A.S. Numerical Calculation of the Useful Capacity Obtained from Regenerating an Exhaust Drying Agent in a Solar Fuel Drying Installation //Applied Solar Energy,- New-York, 2014. Vol. 50, No.3, P.138-142

[3]. Halimov A.S., Agüera A.L., V. Gandara, Akhatov J.S. A study the influence of absorbing layer to the efficiency of solar drying chamber, International Symposium on Power Electronics, Electrical Drives, Automation and Motion - Sorrento (Italy) June 20-22, 2012. Pages 87-92

[4]. Avezov R.R., Kasimov F.Sh. Heat losses of large tray solar collectors with exposed evaporation surface Applied Solar Energy, 2008. Volume 44, Number 4, Pages 262-265.

[5]. Sreenivasa B.R, Hemanchandra K.R. Thermal Engineering Data Handbook, I.K. International Publishing House Pvt. Ltd, 2007. Pages 244, New Delhi, -110 016 (India)

[6]. Iskandarov Z.S. Thermal optimization of suction of spent drying agent through the air layer of the vertical chamber of a solar drying unit // Heliotechnics 2014. №5. P.49-54.

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