In conclusion, we will note that the features of unsteady and steady state vapor formation of microdepressions in a heater surface described herein are in satisfactory agreement with the theoretical model developed in [4].

LITERATURE CITED

1. E. I. Nesis, Liquid Boiling [in Russian], Moscow (1973).
2. V. I. Tolubinskii, Heat Exchange in Boiling [in Russian], Kiev (1980).
3. E. I. Nesis and N. B. Chigarev, Inzh.-Fiz. Zh., 48, No. 6, 962-965 (1985).
4. E. I. Nesis, Inzh.-Fiz. Zh., 23, No. 1, 51-58 (1972).

HEAT EXCHANGE IN A FLAT SOLAR COLLECTOR WITH HEAT PIPES AND HONEYCOMBED FILL

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A flat solar collector with honeycombed fill and a system of heat pipes has been developed.

Existing systems for collecting and converting solar energy consist of the following basic elements: the solar collector, a heat storage system, a standard heat source, and a unit for controlling the operation of the system. The solar collector absorbs and converts solar radiation into thermal energy, which is then transferred to the user or is stored. The heat-transfer agents can be a liquid or air, depending on the problems addressed by the system.

A design of a solar collector with high efficiency operating in both liquid and air systems without significant design modifications has extensive practical applications. Such a universal construction is possible with the use of heat pipes in a solar collector; the high efficiency is achieved also by using a special honeycombed fill between the absorber and the collector coating. A diagram of such a collector is shown in Fig. 1.

A collector of this type has the following advantages.

1. The total thermal resistance between the surface of the channel wall and the heat-transfer agent is reduced. In a traditional collector the fluid flow in the channels is laminar, as a result of which the heat-transfer coefficient is very low — of the order of several tens of $W/(m^2 \cdot K)$. The inner surface is also limited. As a result the temperature differential between the absorber and the heat-transfer agent increases, which increases the losses of heat from the collector. They can be lowered by increasing the fluid flow rate or by replacing the parallel system of channels with a serial system. This increases the hydraulic resistance and increases the energy consumed by the pump circulating the heat-transfer agent. The use of heat pipes removes this problem. The thermal resistance of a heat pipe is low, and the efficiency of heat tranfer from the condenser to the heat-transfer agent can be increased by finning its outer surface.

2. The versatility of the design lies in the fact that air or water can be pumped through the heat exchanger. Different variants of the design can be obtained by varying the number of fins placed on the condensers of the heat pipes as well as by varying the ratio of the lengths of the evaporator and condenser.

3. In a traditional solar collector the distribution of liquid from the collecting channel to separate channels is nonuniform. This makes it difficult to regulate the head of the heat-transfer agent and to determine the corresponding temperature at the collection inlet. The efficiency of the collector drops at the same time.

A collector with heat pipes consists of two parts: the absorber with the heat-pipe evaporators and a system of heat-pipe condensers, which constitute the heat exchanger. The heat-

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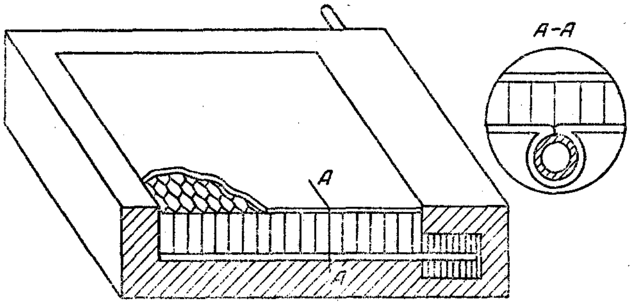

Fig. 1. Flat solar collector with heat pipes and honeycombed fill.

transfer agent in the heat exchanger flows serially over the heat pipes and is heated. The flow velocity is determined by the final temperature which it must achieve.

Operation of a Heat Pipe in a Solar Collector. In calculating and designing heat pipes the conditions which they must meet are checked first. These conditions are described in quite great detail by the dependences presented in [1, 2].

Among the standard heat-transfer agents water is characterized by the highest quantity of heat transferred by the heat pipe, but on the other hand its high freezing temperature limits its use in Central European conditions, where the ambient air temperature if often lower. Glycols can be used as laternative heat-transfer agents (see Table 1) [3].

The total coefficient of heat transfer between the surfaces of the evaporator and the heated heat-transfer agent of the solar setup is expressed by the dependence

$$U = (R_{p,e} + R_{w,e} + R_v + R_{w,c} + R_{p,c} + R_{z,c})^{-1}. \tag{1}$$

The thermal resistance $R_{z,c}$ is a function of the physical properties of the heat-transfer agent, the heat-transfer coefficient, and the degree of extension of the outer surface of the heat-pipe condenser.

Calculations of a heat pipe with a capillary structure made of a copper netting (with the following geometry (m): evaporator length 1.22, adiabatic zone length 0.1, condenser length 0.15, outer diameter 0.019, and inner diameter 0.017) show that the thermal resistances $R_{p,e}$, $R_{w,e}$, R_v, $R_{w,c}$, $R_{p,c}$ fall into the range 10^{-9}–10^{-4} $m^2 \cdot K/W$. The values of these resistances are so low that the expression (1) assumes the form

$$U = 1/R_{z,c}. \tag{2}$$

The evaporators of the heat pipes are placed on the flat surface of the absorber. The amount of energy received by the heat pipe is described by the expression [4]

$$Q_u = EL_r\eta_z[(\tau\alpha)I - U_L(T_c - T_a)] \tag{3}$$

under the condition that $T_c(E) = \text{const}$.

In the case when the surface of the heat pipe is not finned, the transferred heat is determined by the dependence

$$Q_u = \frac{1}{R_{z,c}}(T_c - T_f) = k_c\pi r_{p,0}^2 C(T_c - T_f). \tag{4}$$

Substituting (4) into Eq. (3) we obtain

$$Q_u = \frac{1/U_L}{\dfrac{1}{EL_r\eta_z U_L} + \dfrac{1}{Cr_{p,0}\pi k_c}}[(\tau\alpha)I - U_L(T_f - T_a)]. \tag{5}$$

The distance between the channels in the absorber is usually approximately 0.1 m; Eq. (5) then implies that the maximum power per heat pipe does not exceed 100 W.

Reduction of Heat Losses Owing to the Use of Honeycombed Fill. The losses of heat from the absorber, determined by the coefficient U_L, affect the amount of thermal energy received in the solar collector. Different types of thermally insulating fill are employed to reduce the thermal losses. In Poland the aircraft industry produces aluminum so-called honeycombed

TABLE 1. Physical Properties of Glycols (with a concentration of 80% by weight)

Indicators	Ethylene glycol	Diethylene glycol	Triethyl-ene	Propylene glycol
Heat-transfer coeff. W/(m·K)	0,2766	0,2595	0,2422	0,2249
Specific heat, J/(kg·K)	3224	3140	3120	3475
Freezing temp., °C	—45	—37	—38,5	Unknown
Specific weight, kg/m³	1022	1032	1037	962

fill, which looks like a bee honeycomb, which can be used successfully in solar collectors. It is placed between the absorber and the glass cover and partitions the air space into small parts, limiting the convective motion of the air. In addition, the honeycombed fill suppresses radiative heat losses. At the same time, an important condition must be met: the walls of the fill must have a coating whose reflection coefficient is close to unity, while the reflection coefficient for the thermal radiation must approach zero. Thus the solar radiation flux incident at different angles on the wall of the honeycombed fill is transferred, with small losses, to the surface of the absorber, while the radiative heat-loss flux is absorbed by the walls of the fill and, as a result of secondary reflections, some of it returns to the absorber.

In the mathematical model proposed it is assumed that the absorber is a black surface for solar and thermal radiation, while the protective surface is black for thermal radiation, the walls of the honeycombed fill have a smooth surface for thermal radiation and a rough surface (scattering) for solar radiation, $\rho = \rho^s$, $\rho^d = 0$, $\rho_s = \rho_s^d$, $\rho_s^s \simeq 0$ [5]:

$$q(T_c) = \sigma T_c^4 + \frac{\mathrm{Nu}\, k_a}{L}(T_c - T_p) - \sigma T_p^4 F^s_{c-p,L} - 2{,}42a \int\limits_{z=0}^{L} (1-\rho^s)\, \sigma T^4(z)\, F^s_{dz-c,z} dz, \tag{6}$$

$$2\sigma T_p^4 + h_w(T_p - T_a) - 2{,}42a \int\limits_{z=0}^{L} (1-\rho^s)\,\sigma T^4(z)\, F^s_{dz-c,(L-z)} dz - \frac{\mathrm{Nu}\, k_a}{L}(T_c - T_p) - \sigma T_c^4 F^s_{c-p,L} - \sigma T_n^4 = 0, \tag{7}$$

$$h_w = 2{,}8 + 3{,}0V, \tag{8}$$

$$\frac{d^2T(z)}{dz^2} = \frac{1}{k_s t} q(z), \tag{9}$$

$$\frac{dT(z)}{dz} = 0 \rightarrow z = 0,\ z = L, \tag{10}$$

$$q(z) = (1-\rho^s)\,\sigma T^4(z) - (1-\rho^s)\,\sigma T_p^4 F^s_{dz-p,(L-z)} - (1-\rho^s)\sigma T_c^4 F^s_{dz-c,z} - \\ - \int\limits_{z=0}^{L} (1-\rho^s)^2 \sigma T^4(z)\, dF^s_{dz-dz',(z-z')} - (1-\rho_s^d)\, IE_{dz-p,(L-z)}, \tag{11}$$

$$F^s_{dz-c,z} = F_{dz-c,z} + \sum_{n=1}^{\infty} [\rho^s(z_n)]^n [F_{dz-c,z/n+1} - F_{dz-c,z/n}], \tag{12}$$

$$F_{dz-c,z/n} = \frac{0{,}5 + [z/2an]^2}{\{1 + [z/2an]^2\}^{0,5}} - \frac{z}{2an}, \tag{13}$$

$$dF^s_{dz-dz',(z-z')} = dF_{dz-dz',(z-z')} + \sum_{n=1}^{\infty} [\rho^s(z_n - z)]^n dF_{dz-dz_n,(z_n-z)}, \tag{14}$$

$$dF_{dz-dz_n,(z_n-z)} = \left\{1 - \left|\frac{z'-z}{2a(n+1)}\right| \frac{\left|\frac{z'-z}{2a(n+1)}\right|^2 + \frac{3}{2}}{\left[\left|\frac{z'-z}{2a(n+1)}\right|^2 + 1\right]^{3/2}}\right\} d\left\{\frac{z'}{2a|n+1|}\right\}, \tag{15}$$

$$F^s_{c-p,L} = 1 - (1-\rho^s)\frac{L}{a}\sum_{n=1}^{\infty}(\rho^s)^{n-1}\left\{\left[1 + \left|\frac{L}{2an}\right|^2\right]^{0,5} - \frac{L}{2an}\right\}. \tag{16}$$

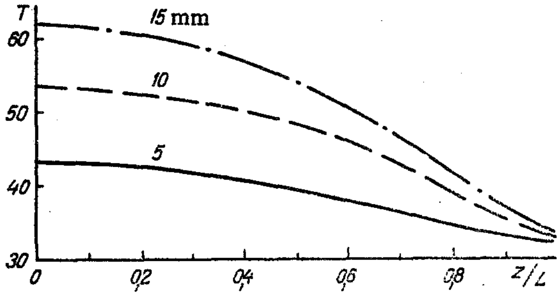

Fig. 2. Temperature distribution along the z axis of a wall of a honeycombed fill ($\rho_s^d = 0.80$; $\rho^s = 0.15$). The numbers on the curves are the height of the fill; the heat losses are identical. T is in °C.

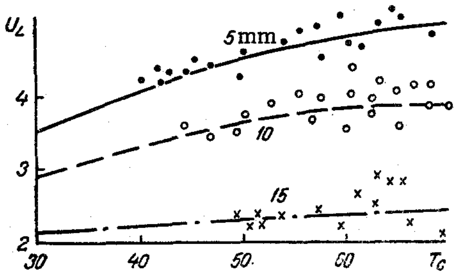

Fig. 3. Heat loss factor as a function of the temperature T_c for an aluminum honeycombed fill ($\rho_s^d = 0.80$; $\rho^s = 0.15$) with a height of 5, 10, and 15 mm. The curves show the calculation and the dots are experimental values. T_c is in °C.

Equation (6) describes the heat flux from the absorber and is the difference of the absorbed and emitted fluxes. The expression (7) expresses the balance of energy in the protective coating, while the formulas (8), (9), and (10) express the energy balance in a unit cell of the wall of the fill at a height z.

The calculations shows that the losses are all the higher the lower are the height of the fill L and the coefficient of reflection of thermal radiation for the wall of the fill. The maximum thickness of the wall of aluminum fill for which the effect of conduction on heat losses $q(T_c)$ is negligibly small and was determined (0.1 mm). This corresponds to the characteristics of the fill produced. The values of the reflection coefficients $\rho^d = 0.80$ and $\rho^s = 0.15$, for which the calculations were performed based on the mathematical model, were obtained as a result of measurements with the help of a spectroscope. They correspond to the properties of an aluminum surface coated with white paint. Figure 2 shows the temperature distribution along the wall L. The coefficient of heat losses from the absorber U_L is a function of L and the temperature, and is insignificant in magnitude (see Fig. 3).

The experimental studies of the characteristics of the honeycombed fill were performed with the help of the method of zero efficiency [6], used only to determine U_L. In this method the thermal energy is not removed from the setup. The solar radiation absorbed by the absorber is returned only in the form of losses $U_L\ (T_c - T_a)$. The temperature on each surface is identical.

Equation (3) with $Q_u = 0$ assumes the form

$$U_L = (\tau\alpha)\frac{I}{T_c - T_a}\,. \tag{17}$$

The value of $(\tau\alpha)$ was determined with a spectrometer; the error in the experimental determination of U_L was calculated from the dependence

$$dU_L = \sqrt{\left[\frac{\partial U_L}{\partial(\tau\alpha)}\,d(\tau\alpha)\right]^2 + \left[\frac{\partial U_L}{\partial(T_c - T_a)}\,d(T_c - T_a)\right]^2 + \left[\frac{\partial U_L}{dI}\,dI\right]^2} \tag{18}$$

and equaled 12%.

The solar collector developed is characterized by high performance characteristics, in particular, low thermal resistance and low heat losses, and compactness, and it can be employed in the liquid and air setups.

NOTATION

a, hydraulic radius of a cell of the honeycombed fill, m; C, length of the condenser, m; E, length of the evaporator, m; F, angular coefficients referring to the solar radiation; $q(T_c)$, heat flux from the absorber of the collector, W/m²; h_w, coefficient of heat exchange with the surrounding medium, W/(m²·K); I, flux of solar radiation, W/m²; k_a, coefficient of thermal conductivity of air, W/(m·K); L, height of the honeycombed fill, m; L_r, distance between the heat pipes, m; r, radius, m; R, thermal resistance, (m²·K)/W; T, temperature, °K; T_n, conventional sky temperature, K; U_L, heat-loss factor W/(m²·K); V, wind velocity, m/sec; $(\tau\alpha)$, reduced absorptivity; η_z, efficiency of the collector; σ, Stefan–Boltzman constant; ρ, total coefficient of reflection of thermal radiation; ρ^s, coefficient of specular reflection of thermal radiation; ρ^d, coefficient of diffuse reflection of thermal radiation; ρ_s, total coefficient of reflection of solar radiation; $\rho_s{}^d$, coefficient of diffuse reflection of solar radiation. Indices: c, condenser; e, evaporator; p, heat pipe; w, wick; v, vapor; f, liquid.

LITERATURE CITED

1. S. W. Chi, Heat Pipe Theory and Practice (A Source Book) Hemisphere Publ. (1976).
2. P. Dunn and D. A. Reay, Heat Pipes, Pergamon Press (1975).
3. C. K. Hsieh, "Design of a system using CPC collectors to collect solar energy and to produce industrial process steam," Report ANL-79-102, University of Florida (1979).
4. J. A. Duffie and W. A. Beckmann, Solar Energy Thermal Processes, Wiley, New York (1974).
5. J. Januszewski and W. Tomczak, Inzynieria Chemiczna i Procesowa, Wroclaw (1983), Vol. 4, No. 3, pp. 467-482.
6. J. Januszewski, S. Kajl, and F. Ruszel, Cieplownictwo, Ogrzewnictowo, Wentylacjz, 18, No. 3, 35-37 (1986).

MASS OF HYDROGEN LIBERATED IN STAINLESS STEEL HEAT PIPES WITH WATER DURING LENGTHY OPERATION

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The authors describe a method of calculating the amount of hydrogen liberated, based on experimental data of long-term performance tests of heat pipes.

Heat transfer in low-temperature stainless-steel heat pipes with water has a number of special features, compared with heat transfer in heat pipes where the materials of the wall, capillary-porous structure and heat-transfer agent are compatible with steam (e.g., copper–water). This stems primarily from the influence on the thermal technology characteristics of the heat pipe of the noncondensible gas liberated from the heat-transfer agent as a result of electrochemical corrosion of the metal in contact with the agent. The hydrogen formed gradually collects in the condensation zone, reducing the effective length and increasing the thermal resistance of the heat pipe. Up till now there has been practically no detailed study of the laws of gas liberation in stainless-steel heat pipes with water, aimed at determining the mass of hydrogen formed during long-term operation. For example, the investigations of [1] did not take into account the nonuniformity of hydrogen release with time, and were conducted over a short period of continuous operation of the heat pipe (3 months). The use of a formula proposed in [2] to calculate the mass of hydrogen liberated is difficult because there are no reliable data on the parameters appearing in the formula, parameters determined only experimentally (the activation energy, the overvoltage in the metal, the number of electrons taking part in the reaction, etc.), and whose number also varies with time.

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