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ECOLOGICALLY CLEAN ENERGY FROM THE SOLAR AIR COLLECTORS

Heat efficiency of the air collectors' system with an area of 3×2 m^2 was assessed as a function of solar energy radiation at different time intervals of their application. Variants of the mathematical model of air collector were used in the analysis of the effect of air stream distribution between both collector channels on the outlet air temperature. The calculation results proved that application of the absorber with selective surface coating caused an increase in solar air collector efficiency by 8%.

1. INTRODUCTON

The sun is the greatest but still not sufficiently utilised potential of energy for our globe. Converting the solar radiation energy into useful energy does not interfere with the ecological equilibrium but is difficult because of the periodicity, changing intensity and scattering of radiation. The exposure conditions change considerably both in annual and daily cycles – the highest exposure takes place in early afternoon from April to September. The average annual exposure on the horizontal surface is 650 W/m^2.

Solar energy is converted into heat energy in solar liquid collectors, the devices being widely used nowadays. These devices are characterized by rather high efficiencies (about 60–80%) and low operating costs. Solar air collectors have several advantages in comparison with liquid collectors. First of all they are less expensive and besides there are no problems connected with freezing or boiling the liquid and with corrosion of the metallic parts. Air collectors must satisfy different conditions. Efficiency of conversion of the solar radiation energy into usable heat in these devices should be the highest. In many cases, however, high efficiency of the collector is less important than its price.

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Solar air collectors are used more and more frequently to heat the interior space. For heating of the buildings where heat is required only in the daytime (e.g., schools or offices) the simplest heating installations consisting of air collectors and ventilators are used. Usually the collectors forming a unit are connected in parallel and placed at the south elevation of a building. Air is sucked into the collector outside or inside the building. In the dwelling houses where heat is necessary also in night-time, a rock bed of outstanding heat storage capacity is added [1].

The solar radiation is absorbed in the collectors by an absorber. There occurs the heat exchange between the absorber plate and the air flowing along it and the temperature of the latter rises (figure 1). Usually air is heated in the space over and under the absorber. This space is separated from the surroundings by the transparent cover from the side of solar radiation and by the insulation from the rear side of the collector.

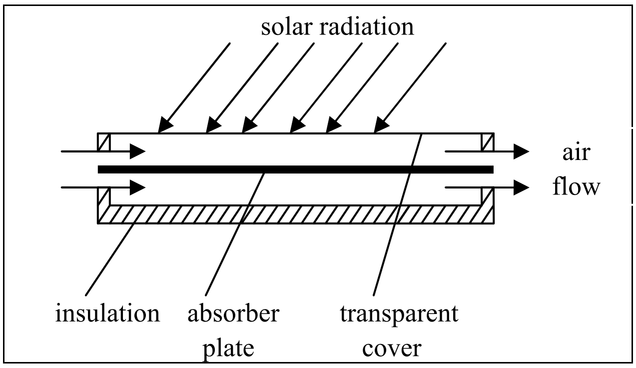

Fig. 1. The cross-section of an air collector

The efficiency of the solar energy absorption heavily depends on the optical characteristics of the absorber and the selective surface coating. The effective transmittance-absorption coefficient $(\tau\alpha)_e$, which is the product of the absorption of the absorber plate and the transmittance of the cover, is the parameter of these characteristics. In proper constructions, $(\tau\alpha)_e$ approaches 0.90.

2. EFFICIENCY OF THE SYSTEM OF SOLAR COLLECTORS

The efficiency of the system of solar air collectors was estimated based on the analysis of the collector heat efficiency as a function of solar radiation energy in the period from September 10 to October 21, 2005. The daily changes of weather were taken into consideration. Collector heat efficiency is assessed based on the parameters of heated air, i.e., inlet and outlet temperatures and mass stream.

Heat effect was calculated as follows:

$$W_h = \int_{\tau_1}^{\tau_2} Q_e d\tau = \int_{\tau_1}^{\tau_2} \frac{G}{1+Y_{\text{in}}}(i_{\text{out}} - i_{\text{in}}) d\tau\,, \tag{1}$$

where: Q_e – the effective power [W], τ – the time [s], G – the air mass flow [kg/s], Y_{in} – an absolute humidity of the air at the inlet of the collector [kg/kgps], i_{in}, i_{out} – the specific air enthalpy at the inlet and outlet of the collector, respectively [J/kgps].

Daily amounts of the solar radiation on the frontal plane of the collector was calculated from the equation:

$$I_c = \int_{\tau_1}^{\tau_2} I_\beta d\tau\,, \tag{2}$$

where I_β is the stream of the solar radiation reaching the frontal surface of the collector [W/m^2].

Because the solar radiation energy changes and the resulting temperature of the heated air changes as well, in the analysis of this problem the daily amounts of heat effect and solar radiation (calculated every one minute within the 8 hour test from 8 a.m. to 4 p.m.) have been considered. On the basis of experiments the relations for September and October have been obtained (figure 2).

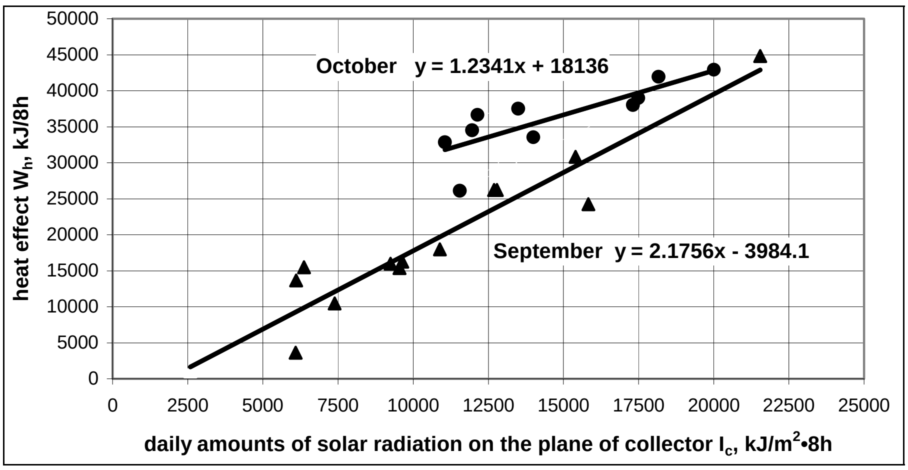

Fig. 2. Heat effect W_h of the solar collectors system versus daily (8 hours) amounts of the solar radiation I_c

Parameters of the installation of solar air collectors used in the experiments were as follows:

• the length $L = 3 \times 2$ m, the width $W = 1$ m, the height of the air channel over and under the absorber $H = 0.025$ m,

• optical characteristics of the individual collector parts: transmittance of the glass covers 1 and 2 $\tau_{g1} = \tau_{g2} = 0.8$; the absorptivity of the glass covers 1, 2 and of the absorber plate $\alpha_{g1} = \alpha_{g2} = 0.04$, $\alpha_a = 0.9$; the emissivity of the glass cover and of the absorber $\varepsilon_{g1} = 0.83$, $\varepsilon_{g2} = 0.2$, $\varepsilon_a = 0.9$. In this case, the absorber with a non-selective plate has been used ($\alpha_a = \varepsilon_a$).

3. MATHEMATICAL MODEL OF THE AIR COLLECTOR

In figure 3, the scheme of heat exchange in the solar air double-channel collector is presented. At the top, an absorber is insulated from the surroundings by two glass covers. Heat is exchanged between different parts of the collector as follows:

glass cover 1 – outer air: forced convection (h_w), radiation (h_{rot}),

glass cover 2 – glass cover 1: *conductivity* (h_p), radiation (h_{rss}),

absorber plate – glass cover 2: forced convection (h_{k1}),(h_{a1}) and radiation (h_{ras}),

absorber plate – rear wall of the collector: forced convection (h_{k2}), (h_{a2}) and radiation (h_{rab}),

rear wall of the collector – outer air: conductivity (U_t).

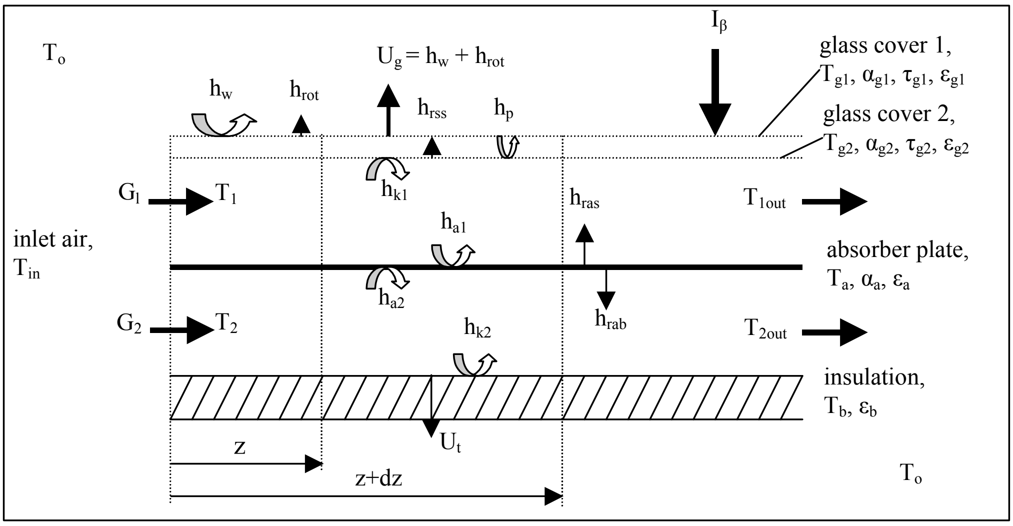

Fig. 3. Heat exchange in the double-channel collector

In the heat balances of this system, it was assumed that mass streams of the air flowing on both sides of the absorber were equal, and the values of the convective heat-transfer coefficients $h_{a1} = h_{k1}$ are equal in the channel over the absorber and also $h_{a2} = h_{k2}$ in the channel under the absorber [2], [3]. It was also assumed that the collector was in thermal equilibrium, therefore its individual parts could be described by the following heat balance equations:

- glass cover 1:

$$U_g(T_{g1} - T_o) = h_{rss}(T_{g2} - T_{g1}) + h_p(T_{g2} - T_{g1}) + \alpha_{g1} I_\beta\,, \tag{3}$$

- glass cover 2:

$$h_{\mathrm{rss}}(T_{g2} - T_{g1}) + h_p(T_{g2} - T_{g1}) + h_{k1}(T_{g2} - T_1) = h_{ras}(T_a - T_{g2}) + \alpha_{g2}\tau_{g1} I_\beta\,, \tag{4}$$

- absorber plate:

$$h_{\mathrm{ras}}(T_a - T_{g2}) + h_{\mathrm{rab}}(T_a - T_b) + h_{a1}(T_a - T_1) + h_{a2}(T_a - T_2) = \tau_{g1}\tau_{g2}\alpha_a I_\beta\,, \tag{5}$$

- rear wall of the collector:

$$h_{k2}(T_b - T_2) + U_t(T_b - T_o) = h_{\mathrm{rab}}(T_a - T_b)\,. \tag{6}$$

Air heating is described by two equations: equation (7) represents channel 1 over the absorber plate and equation (8) – channel 2 under the absorber plate:

$$\frac{G_l c_p}{W}\frac{dT_1}{dz} = h_{a1}(T_a - T_1) + h_{k1}(T_{g2} - T_1)\,, \tag{7}$$

$$\frac{G_2 c_p}{W}\frac{dT_2}{dz} = h_{a2}(T_a - T_2) + h_{k2}(T_b - T_2)\,. \tag{8}$$

Equations (3)–(6) were transformed to obtain the temperatures of: the second glass cover (T_{g2}), the absorber plate (T_a) and the rear wall of the collector (T_b) as a function of the temperatures T_1, T_2 and T_0, which were next used in equations (7) and (8). Based on an available literature the convective heat transfer coefficients, the radiative heat transfer coefficients and the coefficient U_g expressing the heat lost by the outer collector pane and gained by the surroundings have been calculated [4]. The coefficient representing the heat lost by the collector U_t through the rear wall was found from the equation of heat conduction through this wall.

Heat energy stream (effective power) taken by the air in channels 1 and 2 equals:

$$Q_e = G_l c_p(T_{1\mathrm{out}} - T_{\mathrm{in}}) + G_2 c_p(T_{2\mathrm{out}} - T_{\mathrm{in}}) \tag{9}$$

and the collector efficiency:

$$\eta = \frac{Q_e}{W \cdot L \cdot I_\beta}\,, \tag{10}$$

where W [m] is the collector width, L [m] – the collector length.

Calculations for the constructional and process parameters of the tested system of air collectors have been performed to find the outlet air temperature and the collector efficiency. It was assumed that air mass flows in both collector channels were equal, i.e., $G_1 = G_2$. The satisfactory agreement of the calculated temperature of the air flowing out of the collector with this temperature measured during the tests has been obtained, a mean temperature difference has equalled 10%.

The calculations of solar collector efficiency versus its length have been made also for the absorber with selective surface coating ($\alpha_a = 0.9$; $\varepsilon_a = 0.15$). At such values of α_a and ε_a the temperature of the air flowing in the channel over the absorber rose. This increase was caused by the reduction of heat losses through the glass covers to the surroundings. As a result, the collector efficiency increased by 8% (figure 4). From the economical point of view, however, a cheaper absorber with a non-selective surface coating is more advantageous.

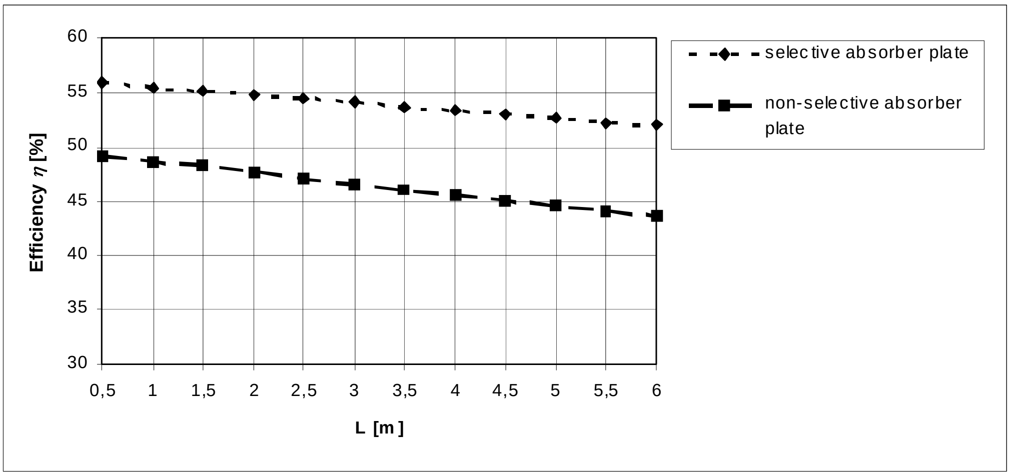

Fig. 4. Collector efficiency η versus its length L; air mass flow $G = 0.088$ kg/s, radiation intensity $I_\beta = 950$ W/m^2

Variants of the mathematical model of air collector were also used to analyze the influence of the degree of air stream distribution between collector channels $\gamma = G_1/G_2$ on the air temperature over the collector length. At the tested range of γ the temperature of air flowing out of the lower channel was always higher than the temperature of air flowing out of the upper channel. The analysis reveals (figure 5) that the best thermal conditions for the double-channel air collector were achieved when the degree of air stream distribution γ was equal to 0.9.

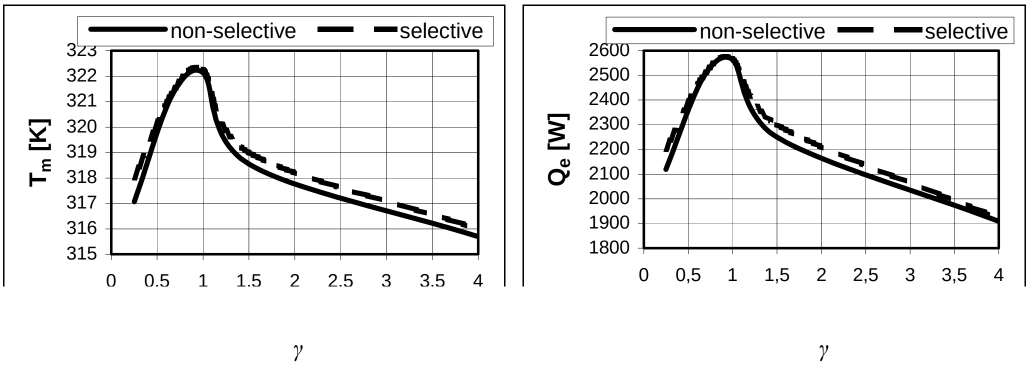

Fig. 5. The mean outlet air temperature T_m and the effective power Q_e versus the degree of distribution of air stream between the collector channels γ; air mass flow $G = 0.088$ kg/s, radiation intensity $I_\beta = 950$ W/m^2

4. RECAPITULATION

Energy of solar radiation is the greatest, ubiquitous and free thermal energy potential. Making use of the sun energy does not interfere with ecological equilibrium but is difficult because of the variable intensity of the solar radiation and its periodicity. Better and more efficient solar collectors enable us to increase utilisation of solar radiation energy and gradual elimination of conventional energy sources. For this purpose the cheap solar air collectors with a non-selective surface coating of the absorber, designed for heating the buildings, are most suitable.

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EKOLOGICZNIE CZYSTA ENERGIA Z POWIETRZNYCH KOLEKTORÓW SŁONECZNYCH

W referacie przedstawiono wyniki badań wydajności cieplnej układu kolektorów powietrznych o powierzchni 3×2 m^2 w zależności od energii promieniowania słonecznego dla różnych przedziałów czasowych ich stosowania. Wydajność cieplna kolektorów jest określona parametrami ogrzewanego powietrza: temperaturami na wlocie i wylocie z kolektora, strumieniem masowym.

Rozwiązania modelu matematycznego dwukanałowego kolektora powietrznego wykorzystywane są do analizy wpływu stopnia podziału strumienia powietrza w obu kanałach kolektora na przebieg temperatury powietrza wylotowego z kolektora. Przeanalizowano również wpływ stopnia selektywności powierzchni absorbera na sprawność kolektora.