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THEORETICAL AND EXPERIMENTAL ASSESSMENT OF STATIONARY AND DYNAMIC RUNNING OF ONE FLAT PLATE SOLAR COLLECTOR

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Această lucrare îşi propune să evalueze performanțele unui colector solar plar în condiții atmosferice de funcționare. În acest scop s-a construit o instalație pilot care permite construirea caracteristicilor panourilor solare. Spre deosebire de modelele folosite în standardele curente, modelul matematic prezentat în aceasră lucrare foloseşte pe lângă ecuațiile de bilanț energetic şi ecuații de transfer de căldură între componentele colectorului. Rezultatele experimental obținute pentru regimurile staționare au permis ajustarea coeficienților de transfer de căldură şi folosirea lor în regimuri dinamice. Comparația rezultatelor experimintale cu cele ale integrării numerice a ecuatiilor arată o bună corelare.

This paper aims to evaluate the performance of a solar flat plate collector in real outdoor conditions. For this purpose, it has been built a pilot installation that allows the construction of solar panels characteristics. Unlike the models in the current standards, the mathematical model presented in this paper go beyond the energy balance equations and add heat transfer equations of collectors components. Obtained experimental results for stationary regimes allowed the adjustment of the heat transfer coefficients and their use in dynamic regimes. Comparison of experimental and numerical integration results shows a good correlation.

Keywords: solar flat plate collector, outdoor experiment, mathematical model

1. Introduction

Performance characteristics of solar thermal collectors are an important issue in the present economic conditions. At this time we use more than one standards for solar thermal collectors, both in stationary (ISO9806-1 ISO9806-3, ASHRAE93-77), and “cvasidynamic conditions” (EN12975). Methods for standard testing are presented by [1, 2]. Literature shows operating characteristics of solar collectors in real working conditions different from those indicated by standards [3]. Standard equations have hardly detectable coefficients in the collector physical model. Therefore, this work aims to build the physical model and evaluate the solar collectors, for real conditions of operation in Bucharest.

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2. Mathematical modeling of flat solar collector

Mathematical modeling of solar flat plate collectors is completed both for stationary operation regimes and dynamic operating regimes. For the modeling of these types of regimes we have used heat balance equations of panel components. We generally consider that the parts of the solar collectors are (Fig. 1): 1) outer housing, 2) insulation, 3) working fluid channel, 4) solar radiation absorbent surface (called absorbent), 5) glass layer and 6) air layer between absorber and glass. In the same fig. are shown the important temperature used: T_a is the ambient temperature, T_g is the glass layer temperature, T_c is the absorber layer temperature, and T_f is the average temperature of the working fluid.

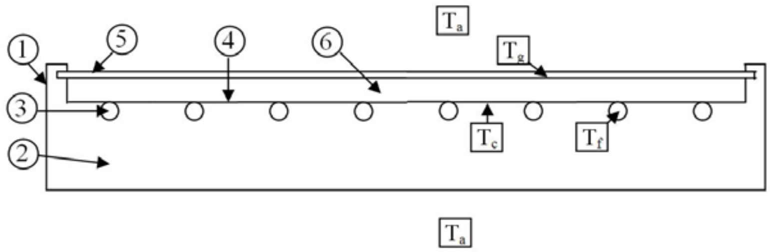

Fig. 1. Solar flat plate collector parts

2.1. Solar flat plate collectors as stationary

Given the geometry of the solar collector, the equivalent electric circuit of the thermal solar panels was built (Fig. 2). It goes from the idea that, during normal operation, the highest temperature is the absorber one. Heat flows have two directions: 1) to the working fluid (which is a useful flow) and 2) loss to the outside (on the top and bottom of the solar collector).

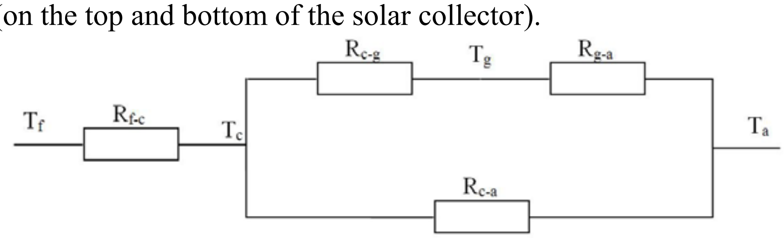

Fig. 2. Electrical equivalent circuit of the associated solar flat plate collector

Important thermal resistances for solar collectors are: $R_{f\text{-}c}$ - between fluid and absorbent, $R_{c\text{-}g}$ - between the absorber and glass, R_g - between the glass and the environment, and R_c - between the absorber and the environment on the bottom plate of the solar collector.

Since the stationary regime does not allow accumulation of heat in the collector, equations defining the stationary were adapted after [4] and will not take account of heat flows to the working fluid. Only heat flows to the outside are considered. The following are the heat balance equations for the glass surface and absorbent collector. Balance equation for the glass surface is:

$$G(1-\tau_g)+(h_{wind}+h_{r,g-a})(T_a-T_g)+(h_{c-g}+h_{r,c-g})(T_c-T_g)=0 \tag{1}$$

, where G is the incident global solar radiation on the solar panel, τ_g is the transmission of the glass factor, h_{wind} is the convection heat transfer coefficient to outside, $h_{r,g-a}$ is the radiation heat transfer coefficient between glass and ambient, h_{c-g} is the convection heat transfer coefficient on the inside surface of glass, and $h_{r,c-g}$ is the radiation heat transfer coefficient between the glass and absorbent surface.

Heat balance equation for absorbent surface is:

$$G\tau_g\alpha_c+(h_{c-g}+h_{r,c-g})(T_g-T_c)+h_{c-f}A_f(T_f-T_c)+\frac{T_a-T_c}{R_{in}}=0 \tag{2}$$

, where α_c is the radiation absorption coefficient of the absorbent layer, h_{c-f} is the heat transfer coefficient between the fluid and absorbent, A_f is the area inside the working fluid channel report to the area of active solar panel, and R_{in} is the resistance between the heat absorbent and atmosphere on the bottom side of the panel.

If neglecting the thermal radiation, the thermal resistance at the bottom of the collector becomes:

$$R_{in}=\frac{\delta_{in}}{\lambda_{in}}+\frac{1}{h_{wind}} \tag{3}$$

, where λ_{in} is conduction coefficient of the insulation material, and δ_{in} is the insulating material thickness.

The heat transfer coefficient of in the outer wind is given by the relation 4 [5, 6, 7], the radiation heat transfer coefficient between glass and environment is given by relation 5, while the thermal radiation coefficient between the absorbent material layer and the glass is given by the relation 6:

$$h_{wind}=2{,}8+3{,}0u_{wind} \tag{4}$$

$$h_{r,g-a}=\varepsilon_g\sigma(T_g^2+T_a^2)(T_g+T_a) \tag{5}$$

$$h_{r,c-g}=\frac{\sigma(T_g^2+T_a^2)(T_g+T_a)}{\frac{1}{\varepsilon_g}+\frac{1}{\varepsilon_c}-1} \tag{6}$$

, where the wind speed is u_{wind}, ε_g is radiation emission factor of the glass, σ is the

Stefan-Boltzmann constant, and ε_c is the emission radiation factor of absorbent layer.

Coefficients for thermal convection between the glass and the absorbent material have been studied by [8, 9]. For this work we used heat transfer coefficients calculated for 35 ° inclination of the closed parallelepiped air cavities, for which Nu = 2.7.

2.1. Solar flat plate collectors as dynamic

Compared to the case of stationary modeling, in the dynamic heat accumulation in the solar collectors appear. Therefore, the equations of heat balance components changes. Heat balance equation for the glass becomes:

$$\rho_g \delta_g C_g \frac{dT_g}{dt} = G\alpha_g + (h_{wind} + h_{r,g-a})(T_a - T_g) + (h_{c-g} + h_{r,c-g})(T_c - T_g) \quad (7)$$

, where ρ_g is the glass density; glass layer thickness is δ_g; and C_g is glass specific heat.

Balance equation for surface heat absorbent and copper pipes is:

$$\rho_c \delta_c C_c \frac{dT_c}{dt} = G\alpha_g \alpha_c + (h_{c-g} + h_{r,c-g})(T_g - T_c) + h_{c-f} A_f (T_f - T_c) + \frac{T_a - T_c}{R_{in}} \quad (8)$$

, where ρ_c is the absorbing surface density; δ_c is the absorption layer thickness; Cg is the absorption layer specific heat.

3. Description of experimental facility

The experimental facility is located in the Polytechnic University of Bucharest (in Bucharest), geographical coordinates of them being 44°26'17"N and 26°02'52"E. Facility consists of a solar thermal flat collector with selective surface absorption, a circulation pump, a boiler with two serpentine, measuring equipment and control. The experimental scheme is shown in Fig. 3.

Following points of measurement for working fluid were arranged (in summer water is preferred): income and outlet of solar thermal collectors for temperature and a flow rate measurement. Also, points of measurement of atmospheric parameters for temperature, wind speed and global solar radiation intensity have been arranged. All measuring instruments are compatible in terms of accuracy of measurement with standard EN12975. Sampling was done every 30 s, resulting in a number of sets of size 2880 for each day.

Solar collectors have absorbent surface of copper, coated with selective absorption paint. Used glass has anti reflexive properties. The technical characteristics of solar thermal collector are presented in Table 1.

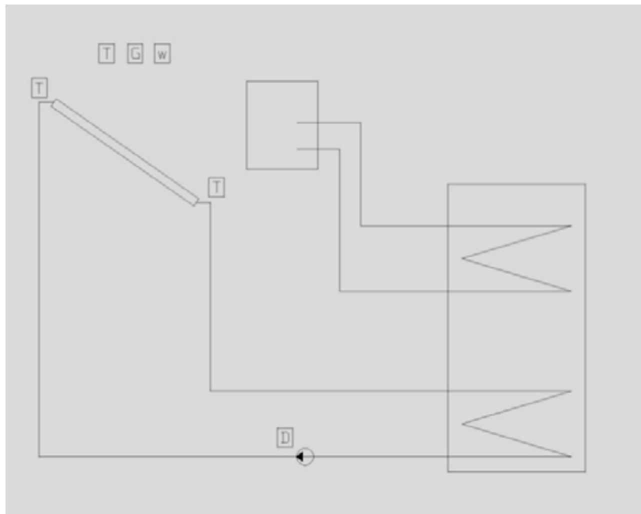

Fig. 3. Principle scheme of the experimental facility

Table 1

The technical characteristics of the solar collector

Glass	
Thickness, m	0.004
Conduction coefficient, W/mK	0.84
Specific heat, J/kg/K	900
Density, kg/m³	2500
Absorption factor	0.88
Emission factor	0.85
Transmission factor	0.75
Inner air layer thickness, m	0.03
Copper	
Density, kg/m³	8920
Conduction coefficient, W/m/K	401
Specific heat, J/kg/K	385
Absorption factor absorbent surface	0.9
Emission factor absorbent surface	0.15
Mineral wool	
Density, kg/m³	60
Conduction coefficient, W/m/K	0.04
Specific heat, J/kg/K	840
Insulation thickness, m	0.04

4. Validation of the mathematical model

4.1. Solar flat plate collectors as stationary

To calculate the stationary characteristic of the collector data have been acquired on 3, 13, 15, 16, 17, 20, 21 and 23 August, with conditions in accordance with EN12975 tests. Calculation of stationary regimes of operation was done with the following of the algorithm:

- Heat flux on the outer glass surface has been imposed;
- Using imposed heat flow, ambient temperature (T_a) and wind speed (u_{wind}), glass temperature (Tg) was calculated;
- Using imposed flow and glass temperature (Tg), absorbent material temperature (Tc) was calculated;
- Using absorbent material temperature (Tc), ambient temperature (T_a) and wind speed (u_{wind}), heat flow on the outside bottom side of the solar collector was calculated.

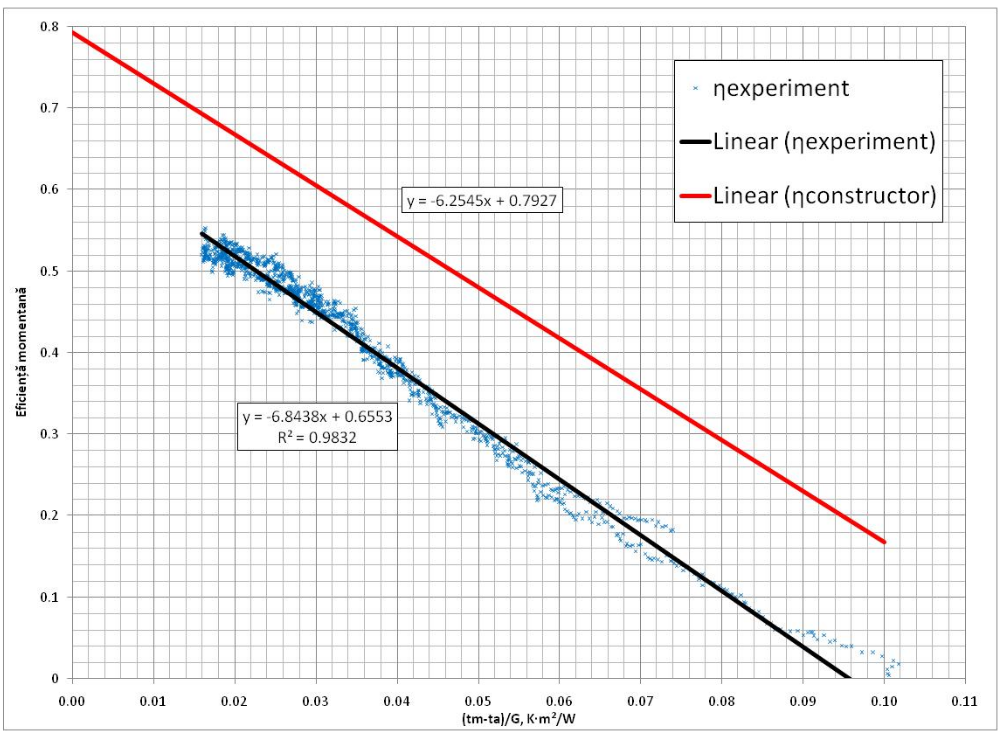

Fig. 4. Literature and experiment solar collector stationary characteristics

Taking into account the EN12975 conditions (G = 1000 W/m², Ta =

30°C), establish the characteristics in the stationary solar collectors require optical loss coefficient η_0. We have equaled η_0 with optical yield obtained experimentally. In Fig. 4, stationary running points of solar collectors from experimental study and literature [10] are presented.

From the experiment we observe an output optical eff. $\eta_{0experiment} = 0.6553$, lower than literature $\eta_{0constructor} = 0.7927$. This is explained by the deposition of dust on the surface of the collector. In the city of Bucharest lodging dust exceeds 280 g/m²/ year, leading to decreases in optical yields up to 25% [11]. Under these conditions, typical yield obtained by applying the mathematical model is presented in Fig. 5.

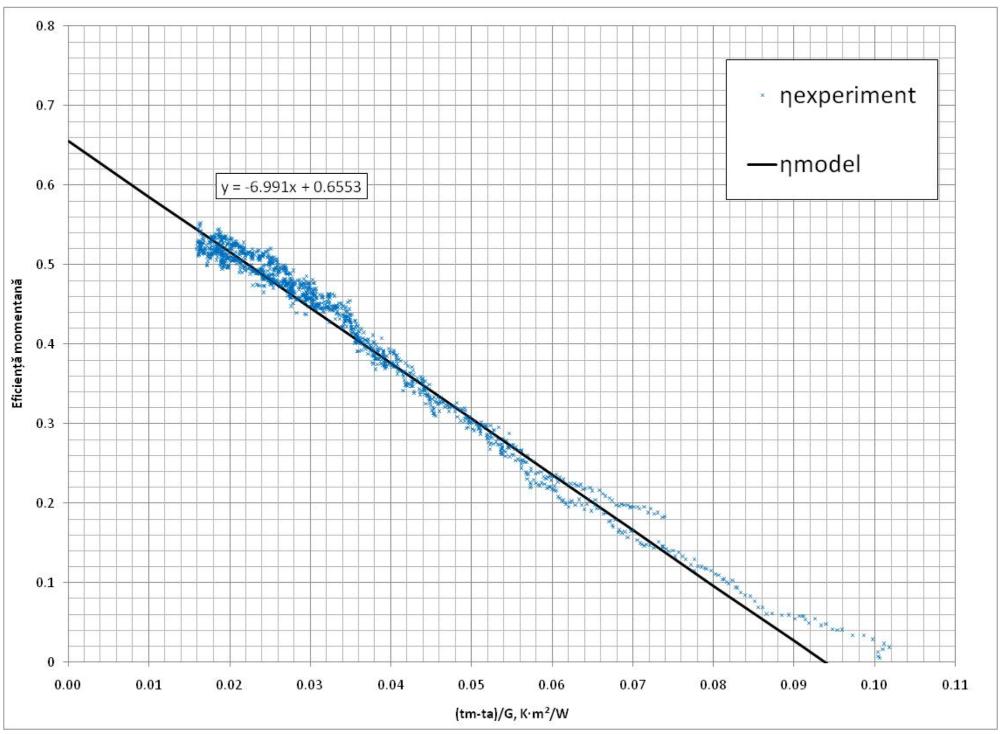

Fig. 5. Model results comparison with experimental data

4.2. Solar flat plate collectors as dynamic

Integration schemes for dynamic equations have been done for the day of 08.21.08. In Fig. 6, the most important atmospheric factors are presented: G is solar global radiation intensity, TP2 is water outlet temperature, TSR is inlet water temperature and TE is ambient temperature. Around 19H a sharp decrease in solar radiation intensity is observed. This is caused by pyranometer shadowing; the solar collector was not subject to this phenomenon.

For the dynamic regime calculation we used equivalent average thermal resistance between fluid and absorbent R_{c-f} = 0.75 m²K/W, obtained in experimental procedures without the presence of solar radiation.

Given that in dynamic regimes direct solar radiation is not normal to the solar collector the reflection phenomenon was considered. We proposed the shape shown in Fig. 8 for this correction factor. It is noted that the rated coefficient of reflection in glass is not symmetrical to the noonday sun. This is because solar panels were oriented SV, with a deviation of 12.5 ° compared to S. Also, the coefficient of total reflections in glass take into account the proportion of indirect solar radiation in the global solar radiation measured.

Numerical integration was done using the modified Euler method. The integration step is equal to the distance between sampling (30s). Initial conditions were considered stationary, stationary model presented above was used for initial parameter calculus. After numerical integration specific thermal power collected by solar panels were calculated. Fig. 8 shows the comparison of the thermal specific power obtained in the experiment with those obtained by applying the described mathematical model.

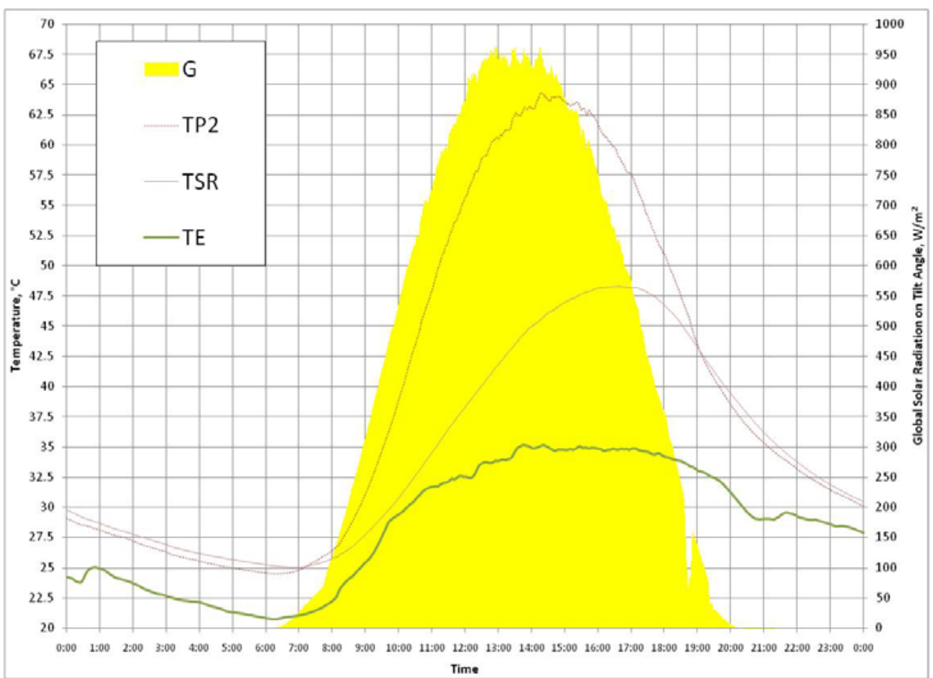

Fig. 6. Atmospheric induced parameters into the model

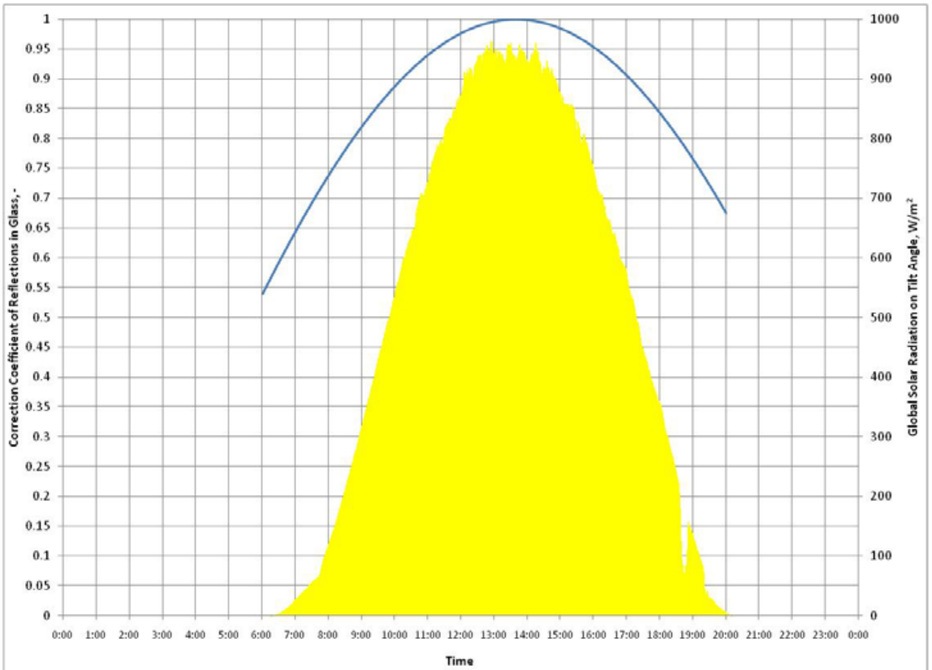

Fig. 7. Correction coefficient of solar incident radiation for total reflection in the glass

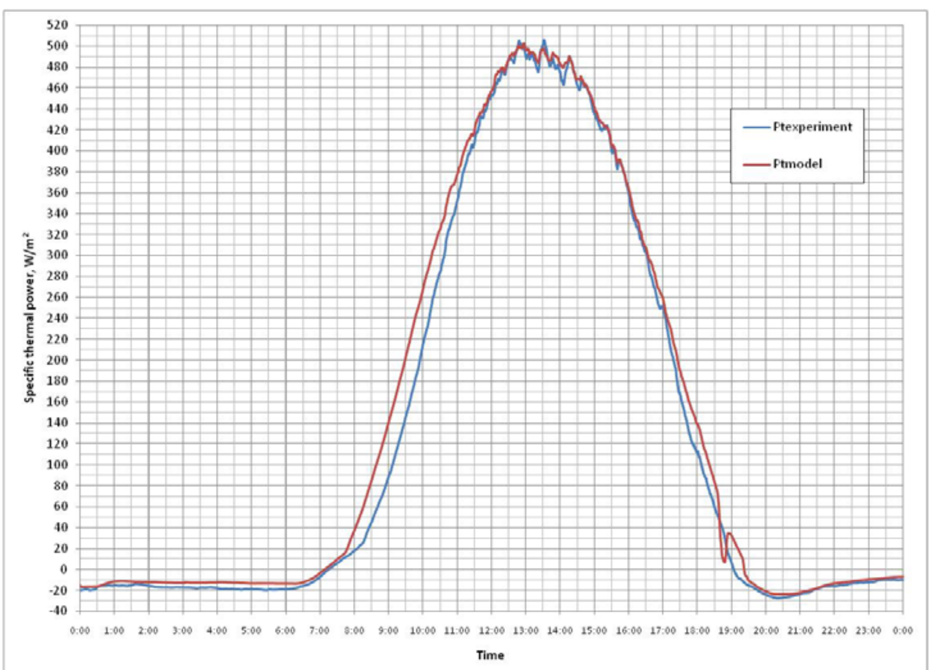

Fig. 8. Specific thermal power obtained by solar thermal collectors

5. Conclusions

Paper presented the mathematical modeling of thermal solar flat plate collectors based on their physical model. Energy balance equations were joined with heat transfer equations for the system components, achieving the models for stationary and dynamic running regimes.

Analysis of experimental data showed that in case of Bucharest, the optical efficiency of solar collectors is about 25% lower than the manufacturer, which is explained by mitadinage of surface layer by dust. Stationary characteristic regime equations of solar collector were obtained by applying the mathematical model. Coefficients for heat transfer losses were very close to those obtained in experimental study.

Modeling of dynamic collector regimes imposed the use of a correction coefficient associated with total glass reflections. The results show a good correlation of experimental data with data obtained by applying the mathematical model.

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