THEORETICAL AND EXPERIMENTAL ANALYSIS OF TWO PASSIVE SOLAR HEATING SYSTEMS

Dan CONSTANTINESCU[*]

ABSTRACT

This paper presents the analysis on mathematical models of two passive solar heating systems:

– INCERC system with air natural flow in the collecting greenhouse;

– Solar space system without air flow.

The results supplied by the analysis and processing of the experimental data obtained in CS 3 Bucharest solar house (INCERC) are also presented; these results prove the theoretical ones valid. The paper also includes a description of the transient mathematical model specific to the analysis of solar space system without air flow in various constructive variants.

REZUMAT

Lucrarea prezintă modelarea matematică a proceselor termice specifice pentru două sisteme pasive de utilizare a energiei solare:

– sistem INCERC cu vehiculare naturală a aerului în sera captatoare;

– sistem spațiu solar neventilat.

Se prezintă validarea experimentală pe suportul casei solare CS 3 București.

I. Introduction

Passive solar heating is presently a major concern of Romania groups working on building research and design. The first theoretical and experimental research studies were worked out in INCERC and used the following experimental bases: the INCERC 1 solar cabin (1974), the Campina solar houses (1976, 1977), the solar apartment buildings in Medgidia (1981) and CS 3 Bucharest solar house at INCERC (1982). Three passive solar heating systems were analysed:

– the INCERC system, similar to the Trombe system with air flow through the greenhouse space collecting solar radiation;

– the Trombe system without air flow;

– the direct gain system with triple glazing.

This paper introduces the theoretical and experimental analysis of the first two systems which are presently used in Romania typified projects for dwelling and industrial buildings.

The INCERC passive system differs from the traditional Trombe system both by the collecting wall material and by the position of air flow slots. The collecting wall used in the INCERC system is of double-layer type. The outside layer is covered with black paint is made of 0.15 m thick ALC[1] and the inside wall adjoining the dwelling space is made of 0.25 m thick hollow brick. Considering the fact that ALC is a heat insulating material, the wall is characterized by a quick thermal response, namely by high temperature values on the collecting wall which is a considerable thermal protection for sunless hours. The air natural circulation slots allow the quick damping of air movement under conditions of turbulent flow and thus the convective component of the useful heat flow is considerably increased as compared to the Trombe solution. The paper presents the basic mathematical modelling of heat transfer process specific to this type of wall. The transient analysis of the thermal response is performed analytically and is based on a few hypotheses proved valid experimentally.

[1] Autoclaved lightweight concrete

[*]National Building Research Institute – INCERC, Bucharest, Romania

Air natural flow in the vertical collecting greenhouse is described in a reduced number of studies that have already been published. The aspects concerning the development of the velocity and temperature field at laminar flow are studied in [1] and [2]. Turbulent natural flow is studied in [3]. The above mentioned papers, mainly [1] and [3] use a very accurate mathematical model and the results that are obtained are probably the closest to the convective heat transfer theory. Nevertheless, the transient character generated both by the thermal capacity of the collecting wall and by variation of the climatic factors is not sufficiently put into evidence as the calculation system is not included in a technical solution. Paper [4] suggests a transient calculation method without performing a detailed analysis of the natural convection phenomenon in the collecting greenhouse. In papers [5] and [6] the collecting wall is integrated in a structure without thermal capacity and space heating is performed only by convection. Our study suggests a method of analytical calculation and the analysis of convective and conductive heat transfer phenomenon includes the variation of the air flow-rate circulated in the collecting greenhouse. A simplified calculation method is generated [7]. The theoretical results are presented together with the ones experimentally obtained at the CS 3 Bucharest solar house.

The second passive solar heating system was necessary as frequent practical imperfections occurred in the INCERC system, mainly at the collecting greenhouse. The greenhouse built on sites proved leaky which has a negative influence on the energy performance of the system. In order to ensure the possibility of the beneficiaries to tighten the greenhouses, they are placed at 0.60 m off the black surface of the collecting wall. The wall is no longer supplied with air flow slots and the heat flow-rate is purely conductive. The analysis presented here is the result of modelling the transitory condition for two structures (concrete and ALC) of glazed wall with and without thermal protection in the sunless hours. The results of this solution (0.30 m thick concrete with thermal protection) are compared to those obtained experimentally in the INCERC solar cabin [8].

II. Mathematical model used in the analysis of the INCERC heating system

The working hypotheses part of mathematical modelling of thermal processes of CS 3 Bucharest solar house are the following:

– the collecting greenhouse is tight against outside;

– air flow is turbulent;

– air temperature variation on the height of the collecting greenhouse is linear;

– the air volume flow-rate circulated in the greenhouse varies linearly with the average temperature of the collecting wall black surface.

The calculation scheme is presented in fig. 1.

The heat balance equations specific to the heat transfer processes are the following:

– for the window surface (S_G):

$$\alpha_r\,(\vartheta\big|_{x=\Delta} - t_G) = \alpha_{cv}^{G}(t_g - t) + K_g\,(t_G - t_e) \tag{1}$$

– for the collecting wall surface (S_p):

$$(\alpha\tau)\cdot I = \alpha_{cv}^{P}(\vartheta\big|_{x=\Delta} - t) + \alpha_r(\vartheta\big|_{x=\Delta} - t_G) + \lambda\cdot\frac{\partial\vartheta}{\partial x}\bigg|_{x=\Delta} \tag{2}$$

– for the heat carrier:

$$Gc_{pa}\,(t\big|_{y=H} - t_i) = S_p\,[\alpha_{cv}^{P}(\bar{\vartheta}\big|_{x=\Delta} - \bar{t}) + \alpha_{cv}^{G}(\bar{t}_G - \bar{t})] \tag{3}$$

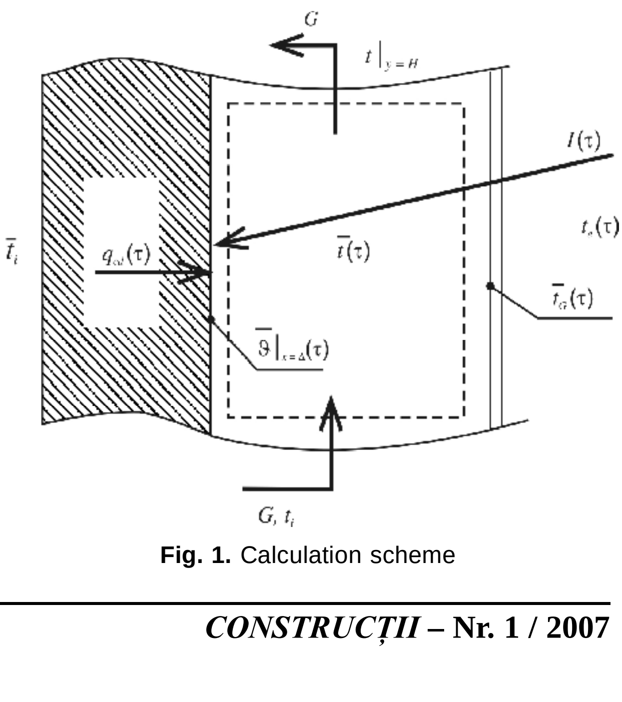

Fig. 1. Calculation scheme

$$V / S_P = A_1 \cdot \overline{\vartheta}\big|_{x=\Delta} + A_2 \qquad (4)$$

$$t = B_1(\tau) \cdot y + t_i \qquad (5)$$

– for the collecting wall inside surface:

$$\left.\frac{\partial \vartheta}{\partial x}\right|_{x=0} = 0 \qquad (6)$$

The bar-marked symbols stand for average values on surface. The equation of the conductive transfer through the collecting wall is Fourier equation:

$$\frac{\partial \vartheta}{\partial \tau} = a \frac{\partial^2 \vartheta}{\partial x^2} \qquad (7)$$

In the equation written above, parameters $(\alpha \tilde{t})$, I and t_e are time variable functions.

The convection heat exchange coefficient is determined using relation [9]:

$$\alpha_{cv} = 1.60 \cdot \Delta t^{0.33} \qquad (8)$$

A set of α_r values was numerically tested and the hypothesis of accepting a unique value α_r = = 6.96 W / m^{2}K resulted.

The temperature of the window surface is provided by equation (1) taking into account the possibility of expressing the convective thermal flow by a temperature difference linear function (error σ = 2.4 %).

$$t_G = \frac{\alpha_r \vartheta\big|_{x=\Delta} + 4.74t + K_G t_e + 17.70}{\alpha_r + 4.74 + K_G} \qquad (9)$$

Relation (2) is a IIIrd type boundary condition for equation (7). Relation (2) is written as follows in terms of the numerical values of thermal conductivity λ = 0.41 W / mK and of the collecting wall absorptivity:

$$\left.\frac{\partial \vartheta}{\partial x}\right|_{x=\Delta} = -23.19 \cdot (\vartheta\big|_{x=\Delta} - \tilde{t}) \qquad (10)$$

where:

$$\tilde{t} = 0.16 t_E(\tau) + 0.84t + 3.11 \qquad (11)$$

$t_E(\tau)$ having the function of an equivalent temperature.

Taking into account (3) and (4) balance equations, the result is:

$$B_1(\tau) = \frac{3.41\vartheta\big|_{x=\Delta} - 3.93t_i + 0.517t_E - 14.64}{0.223\overline{\vartheta}\big|_{x=\Delta} + 3.19} \qquad (12)$$

By replacing in relations (11) and (10) the following relation is obtained:

$$\left.\frac{\partial \vartheta}{\partial x}\right|_{x=\Delta} = -23.19(\vartheta\big|_{x=\Delta} - \phi(y, \tau)) \qquad (13)$$

where:

$$\phi(y, \tau) \cong 0.16t_E(\tau) + 0.84t_i + 3.11 + \\ + y \cdot (0.192\overline{\vartheta}\big|_{x=\Delta} - 4.34) \qquad (14)$$

which is obtained by linearizing as against variable $\overline{\vartheta}\big|_{x=\Delta}(\tau)$ (error σ = 5.71 %). Using the following as against dimensionless notations

$$\text{Fo} = \frac{a\tau}{\Delta^2} \text{ and } \tilde{x} = \frac{x}{\Delta}, \; \tilde{y} = \frac{y}{H}$$

the problem is stated as follows:

$$\frac{\partial \phi(\tilde{y}, \text{Fo})}{\partial \text{Fo}} + \frac{\partial \Theta}{\partial \text{Fo}} = \frac{\partial^2 \Theta}{\partial \tilde{x}^2}$$

$$\left.\frac{\partial \Theta}{\partial \tilde{x}}\right|_{\tilde{x}=0} = 0$$

$$\left.\frac{\partial \Theta}{\partial \tilde{x}}\right|_{\tilde{x}=1} + 23.19 \cdot \Theta\big|_{\tilde{x}=1} = 0 \qquad (15)$$

$$\Theta\big|_{\text{Fo}=0} = f(\tilde{x}) - \phi(\tilde{y}, 0)$$

where

$$\Theta = \vartheta - \phi(\tilde{y}, \text{Fo})$$

and $f(\tilde{x})$ is the function of temperature field distribution in the collecting wall thickness at the initial moment. Considering that this linear distribution has the form

$$f(\xi) = D_1 \xi + D_2$$

the following solution being obtained by integrating system (15):

$$\vartheta(\xi, \eta, \mathrm{Fo}) = \phi(\eta, \mathrm{Fo}) +$$

$$+\sum_j \frac{[(D_1 + D_2)\cdot m_j \cdot \sin m_j - D_1 \cdot (1-\cos m_j)]\cdot \cos(m_j \xi)}{m_j (m_j + \sin m_j \cos m_j)} -$$

$$-2\cdot \sum_j \frac{\sin m_j \cos(m_j \xi)}{m_j \sin m_j \cos m_j}\cdot \left\{\phi(\eta, 0)\cdot \exp(-m_j^2 \cdot \mathrm{Fo}) + \right.$$

$$\left. + \int_0^{\mathrm{Fo}} \frac{\partial \phi(\eta, S)}{\partial S}\cdot \exp[-m_j^2 \cdot (\mathrm{Fo} - S)]\,\mathrm{d}S \right\} \tag{16}$$

Function $\bar{\vartheta}\big|_{\xi=1}(\mathrm{Fo})$ is determined in terms of expression $\phi(\eta, \mathrm{Fo})$ (relation 14). The integral equation is obtained:

$$\bar{\vartheta}\big|_{\xi=1}(\mathrm{Fo}) = 1.238\cdot \left\{E(\mathrm{Fo}) + \right.$$

$$\left. +\sum_j \frac{[(D_1 + D_2)\cdot m_j \cdot \sin m_j - D_1 \cdot (1-\cos m_j)]\cdot \cos m_j}{m_j \cdot (m_j + \sin m_j \cdot \cos m_j)} \right\} -$$

$$-1.238\cdot \left\{E(0)\cdot \Psi(\mathrm{Fo}) + \int_0^{\mathrm{Fo}} \frac{\partial E(S)}{\partial S}\cdot \Psi(\mathrm{Fo}-S)\,\mathrm{d}S \right\} -$$

$$-0.238 \left\{\bar{\vartheta}\big|_{\xi=1}(0)\cdot \Psi(\mathrm{Fo}) + \int_0^{\mathrm{Fo}} \frac{\partial \bar{\vartheta}\big|_{\xi=1}(S)}{\partial S}\cdot \Psi(\mathrm{Fo}-S)\,\mathrm{d}S \right\} \tag{17}$$

with the following solution:

$$\bar{\vartheta}\big|_{\xi=1}(\mathrm{Fo}) = \frac{1}{b_0}\cdot$$

$$\cdot \sum_j \frac{[(D_1 + D_2) m_j \sin m_j - D_1 (1-\cos m_j)] \cos m_j}{m_j (m_j + \sin m_j \cos m_j)}\cdot$$

$$\cdot \prod_j m_j^2 + \sum_j \frac{B(p_j)}{R'(p_j)}\cdot \frac{1}{p_j} \exp(p_j \cdot \mathrm{Fo}) +$$

$$+ 5.208\cdot \{E(0)\cdot P(\mathrm{Fo}) +$$

$$\left. + \int_0^{\mathrm{Fo}} \frac{\partial E(s)}{\partial s}\cdot P(\mathrm{Fo} - s)\,\mathrm{d}s \right\} \tag{18}$$

where

$$P(\mathrm{Fo}) = \frac{a_0}{b_0} + \sum_j \frac{A(p_j)}{R'(p_j)}\cdot \frac{1}{p_j} \exp(p_j \cdot \mathrm{Fo})$$

$$A(p) = \sum_j a_j p^j$$

$$R(p) = \sum_j b_j p^j$$

and values p_j are the roots of the characteristic equation.

$$R(p) = 0$$

Solution (18) is valid for the case of a 2.00 m high greenhouse in the CS 3 Bucharest solar house.

If expression $\bar{\vartheta}\big|_{\xi=1}(\mathrm{Fo})$ is know, function B_1 (Fo) and further the function of heat carrier temperature distribution on the greenhouse height may be determined using relation (12).

If temperature distribution $\vartheta(\xi, \eta, \mathrm{Fo})$ are know, the convective heat flow may be calculated (relation (3)) as well as the conductive heat flow:

$$Q_{COND} = -\lambda \cdot \frac{\partial \vartheta}{\partial \xi}\bigg|_{\xi=1} \cdot S_p \tag{19}$$

where Q_{COND} is the average value as against η. Sum ($Q_{CONV} + Q_{COND}$) represents the useful flow of the collecting greenhouse in sunny hours. For the remaining hours, the calculation method which is used in the case of unvented wall is used. The mathematical model was used in processing the data supplied by the experimental which is presently carried out in CS 3 Bucharest solar house.

The processing was performed for long periods (≈ 30 days) and a more efficient simplified mathematical model was obtained; it is based on the theory of steady-state heat transfer. This model allows to estimate the energy performance of the house but not to determine the inside temperature levels generated by the solar passive heating system.

III. Experiments performed in CS 3 Bucharest solar house

CS 3 Bucharest solar house has groundfloor and one storey with 4 apartments each of 3 rooms. Two bedrooms of each apartment are on the South facade and supplied with solar passive heating. Two rooms were arranged as measurement rooms in order to determine the collecting wall energy performance. The slots for air flow in the collecting greenhouse are differently arranged. In the storey standard room (SSR), the bottom slot admits cold air (at a temperature approximately equal to t_i) on a line parallel to the collecting wall and the upper slot is normal in the wall surface allowing hot air admission in the room. In the ground floor standard room (GSR) both slots are placed similarly to the bottom slot in solution (SSR).

The tests on visualizing air flow in the greenhouse clearly put into evidence the advantages of the (SSR) system. At the same time, visualizing tests were performed in a room supplied with normal slots on the collecting wall surface. In all cases, laminar flow was noticed in this type of greenhouse. In the collecting greenhouse working in the storey standard room (SSR) flow usually was turbulent the air bathing the whole surface of the collecting wall (fig. 2) unlike the (GSR) case where only 60 % of the collecting surface is active (fig. 3).

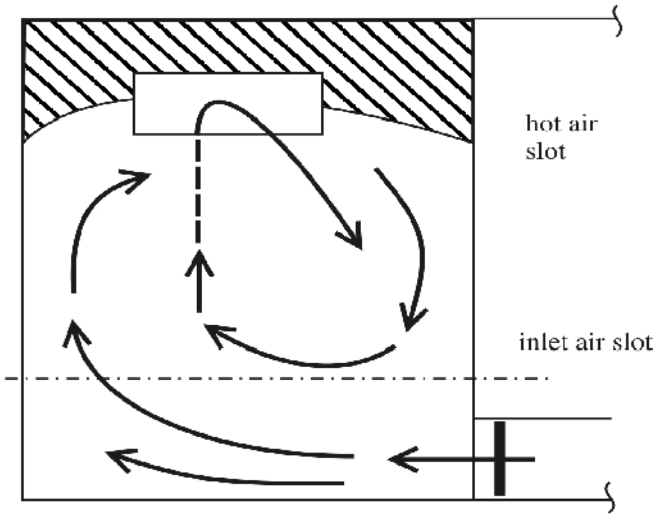

Fig. 2. SSR greenhouse air flow

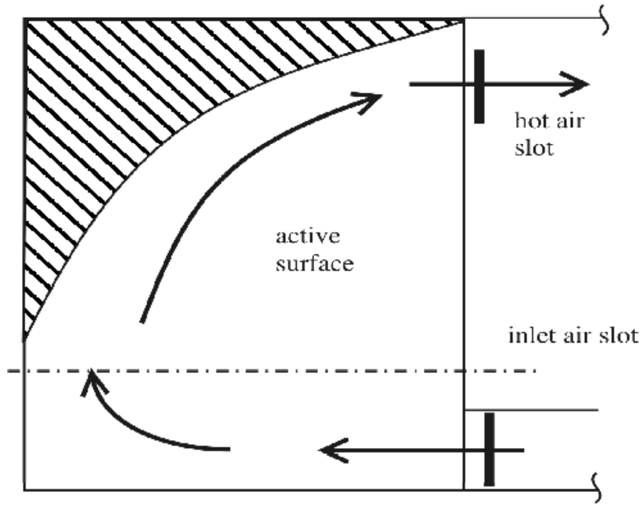

Fig. 3. GSR greenhouse air flow

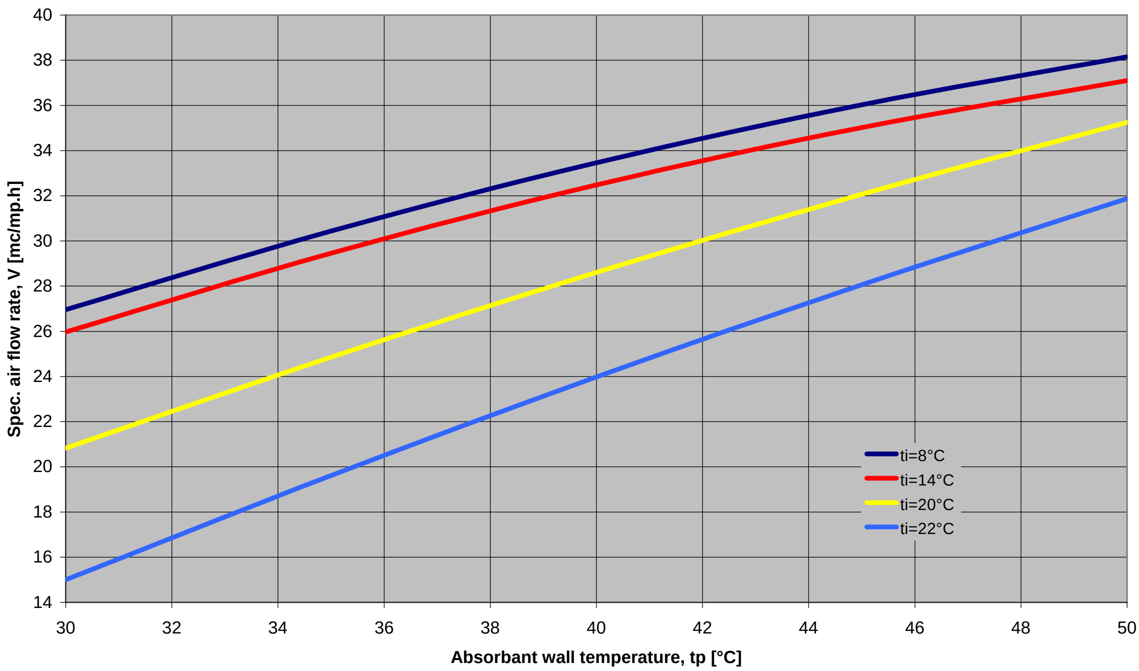

Fig. 4. The SSR system air flow rate

Air volume flow-rate values were measured by hot wire anemometer. After data processing a curve family $V = f(\overline{\vartheta}\big|_{\dot{x}=1}, t_i)$ resulted which is plotted in fig. 4. May we mention that the influence of outside temperature is not significant. The experimental correlation is accurately checked using the data supplied by the preliminary model. The energy equation is solved under the compulsory condition of equality between pressure drops and available pressure.

The solar house energy performance was determined using the mathematical model and proved valid by the rooms energy balance equation for $Q_{AUX} = 0$ (fig. 5). The inside free running temperature and outside climate values specific to the cold season are presented in Table 1.

Tabelul 1.

Month	I [W / m²]	n [h / day]	t_{e_s} [°C]	t_{e_n} [°C]	t_e [°C]	t_i [°C]
X	510	5.9	16.6	10.0	11.6	22
XI	463	2.5	7.2	3.9	4.3	15
XII	464	1.5	2.8	– 1.0	– 0.8	8
I	457	2.1	– 3.8	– 6.8	– 6.5	5
II	512	4.8	– 4.5	– 8.8	– 8.0	5
III	463	2.5	3.7	1.1	1.4	8
IV	454	7.1	18.5	12.3	14.1	20
AVE	480	3.3	6.1	– 0.2	0.67	10.3

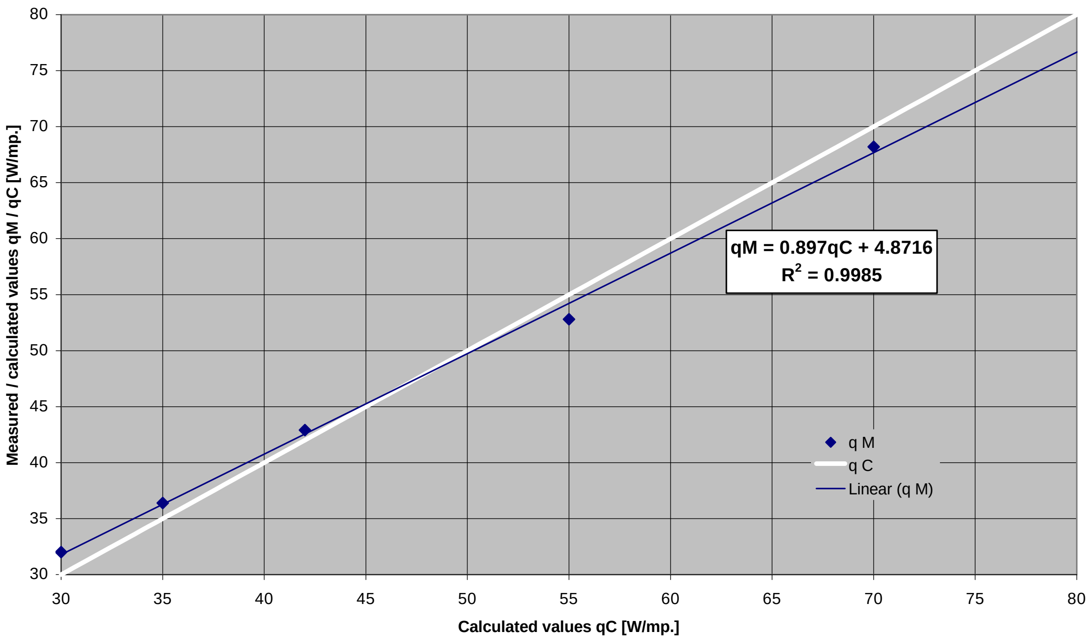

Fig. 5. Energy performance experimental validation

A reduction of about 60 % of the heat consumption is noticed thanks to the improved insulation and to the solar heating passive system. The heat collected by passive system is about 30 % in case of (SSR). As we have already mentioned, the correct service of the system depends on greenhouse tightening against air leakage.

IV. Mathematical model used in analysing passive systems without air flow

For practical reasons, the system has a 0.60 m wide passage between the collecting wall and the glazing. This space is not connected to the dwelling rooms. In the closed space adjoining the collecting wall natural convection currents are developed, flow being usually turbulent. The values of number Gr are higher then 2×10^7 in the range of useful temperature. Relation (20) is used in determining the convective transfer coefficients [10]:

$$\mathrm{Nu}_\delta = c \cdot (\mathrm{Gr}_\delta \cdot \mathrm{Pr})^{1/3} \qquad (20)$$

where

$$c = 0.065 \cdot (H/\delta)^{-1/9} \cdot \mathrm{Pr}^{-1/3}$$

The Unitary Thermal Response (UTR) method is used as calculation method [11]. The mathematical model includes the balance equations of the collecting walls that are considered homogeneous. The heat transfer is one-dimensional, the boundary conditions being of Dirichlet type variable in time t_{PE} (τ) and t_{PI} (τ). Heat flows are determined using the following relations:

$$q_{PI}(\tau) = \{t_{PI}(\tau)\} \cdot \{X\} + \{t_{PE}(\tau)\} \cdot \{Y\} \tag{21}$$

$$q_{PE}(\tau) = \{t_{PE}(\tau)\} \cdot \{X\} + \{t_{PI}(\tau)\} \cdot \{Y\} \tag{22}$$

where: $\{X\}$, $\{Y\}$ – matrices specific to the thermophysical characteristics of the structure under study and to the response of structures at impulsional thermal excitation (Dirac functions).

As the conductive flows are equal to the convective and radiative flows in case of $x = 0$, Δ areas of the collecting walls under study, temperature values t_{PE} (τ), t_{PI} (τ) may be expressed in terms of instantaneous values I (τ), t_e (τ), t_i (τ). In the mathematical model that was worked out, the condition of thermostatic temperature control of the dwelling space, namely t_i = 18°C is raised. Temperature t_i is actually determined by the room heat balance; situations when t_i > 18°C may also occur. Condition t_i = 18°C implies $Q\text{-}_{AUX} \leq 0$ and therefore the possibility of dwelling space air conditioning. The thermal response of the concrete and ALC walls was determined using the following possibilities of carrying out and putting into service the system:

- simple outside glazing (1G);
- double outside glazing (2G);
- night protection with blinds made of expanded polyurethane (P):
  - – with automatic control;
  - – with manual control;
- night protection with reflecting screen (RS):
  - – with automatic control;
  - – with manual control;
- without night protection (WP).

The automatic control implies the existence of devices requiting collecting wall protection at the moment when q_{PE} (τ) < 0. The manual control implies the occupants decision on protecting the outside surface of walls during afternoon and night, as in daytime hours (7^{00}-16^{00}) the surface is freely exposed no matter the direction of the heat flow q_{PE} (τ).

The climatic data that are used represent the hourly values of solar radiation intensity I (τ), of outside temperature t_e (τ) and of wind velocity w (τ) for a 5 years period (1978-1982) recording in Campina (at CS 1 and CS 2 Campina solar houses). The results obtained are presented in Tables 2 (concrete) and 3 (ALC).

The thermophysical characteristics used in calculations are as follows:

concrete: 1.62 W / mK;
c = 0.836 kJ / kgK; ρ = 2.400 kg / m^3
ALC: 0.35 W / mK;
c = 0.842 kJ / kgK; ρ = 900 kg / m^3.

Table 2.

Concrete collecting wall

Wall type	C-2GWP	C-2GP	C-2GRS	C-1GWP	C-1GP	C-1GRS
Δ [m]	0.2	0.2	0.2	0.2	0.2	0.2
Δq_{PI} [kgcf / m^2 year]	2.88	13.68	15.07	– 6.00	9.54	10.89
Δ [m]	0.3	0.3	0.3	0.3	0.3	0.3
Δq_{PI} [kgcf / m^2 year]	2.94	12.6	13.79	– 4.09	9.32	10.54

Table 3

ALC collecting wall

Wall type	ALC-2GWP	ALC-2GP	ALC-2GRS	ALC-1GWP	ALC-1GP	ALC-1GRS
Δ [m]	0.2	0.2	0.2	0.2	0.2	0.2
Δq_{PI} [kgcf / m^2 year]	5.36	10.11	11.02	1.77	6.96	7.82
Δ [m]	0.3	0.3	0.3	0.3	0.3	0.3
Δq_{PI} [kgcf / m^2 year]	6.00	9.68	10.41	3.40	7.14	7.88

V. Interpretation of results

The variant implying the highest energy gain is concrete with double glazing and reflecting screen protection, 0.20 m concrete thickness (Δq_{PI} = = 15.07 kgcf / m^2 year; 1 kgcf = 21.506 kJ).

If automatic thermal protection is not under service the energy gain is reduced up to 2.88 kgcf / m^2 year. The influence of thermal protection automatic control is rendered evident by the following ration:

$$R_c^{(1)} = 0.47$$

$$R_c^{(2)} = 0.73$$

where (1), (2) represent the glazing type.

In case of ALC the same ratios are:

$$R_{ALC}^{(1)} = 0.76$$

$$R_{ALC}^{(2)} = 0.82$$

Without thermal protection the thermal performance decrease is considerable; in case of ALC it is of 50-70 % and in case of concrete it is over 80 %.

Simple glazing without thermal protection is not recommended in any case.

VI. Conclusions

a. This paper presents the analysis on mathematical models of two passive solar heating systems:

– INCERC system with air natural flow in the collecting greenhouse;

– Trombe system without air flow.

b. Starting from the calculation models that were worked out, the experimental data supplied by CS 3 Bucharest solar house (at INCERC) were analysed and processed.

c. The measured data (air volume flow-rate values, convective and conductive heat flows) are in accordance with those obtained by calculation.

d. The paper gives a description of the transient mathematical model specific to the analysis of unvented Trombe walls.

Symbols

t_G – glazing area temperature [°C]

t – air temperature in collecting greenhouse [°C]

t_e – outside temperature [°C]

$t\big|_{y=H}$ – temperature of air exhausted in room [°C]

$\bar{t}$ – air average temperature in collecting greenhouse [°C]

t_i – inside temperature [°C]

t_E – equivalent temperature [°C]

t_{PE} – temperature of unvented collecting wall outside surface [°C]

t_{PI} – temperature of unvented collecting wall inside surface [°C]

$\vartheta\big|_{x=\Delta}$ – temperature of vented collecting wall inside surface [°C]

α_r – radiation heat exchange coefficient [W / m^{2}K]

α_{cv} – convection heat exchange coefficient [W / m^{2}k]

k_G – coefficient of glazing overall heat losses [W / m^{2}K]

I – solar radiation intensity [W / m^2]

λ – collecting wall thermal conductivity [W / mK]

G – air mass flow-rate [kg / s]

c_P – specific heat of constant pressure air [kgJ / kgK]

a – thermal diffusivity of collecting wall material [m^2 / s]

Q_{AUX} – heat flow exhausted by auxiliary source [kgcf / month]

τ – time [s]

H – greenhouse high [m]

Δ – collecting wall thickness [m]

δ – greenhouse passage width [m]

m_j – eigenvalues

Fo – Fourier dimensionless number

Nu – Nusselt dimensionless number

Gr – Grashoff dimensionless number

Pr – Prandtl dimensionless number

REFERENCES

[1] Akbari H., Borgers T.R. – *Free Convective Laminar Flow Within the Trombe Wall Channel*, Solar Energy vol. 22, 1979, pp. 165-174

[2] Barra O.A., et al – *A Theoretical Study of Laminar Free Convection in 1-D Solar Induced Flows*, Solar Energy vol. 23, pp 211-215

[3] Borgers T.R., et al – *Free Convevctive Turbulent Flow Within Trombe Wall Chanel*

[4] Constantinescu D., et al – *Romanian Solar Passive Heating Systems*, INCERC, Miami, 1980

[5] Constantinescu D. – *Present and future of Solar Energy. Use of Solar Energy in Space Hating and Warm Water Producing*, Ed. Academiei, Bucuresti, 1982 (în românești)

[6] * * * – *Solar Systems in Passive and Hybrid Systems*, INCERC, Contract 32 / 1982

[7] Constantinescu D. – *Heat Balance Specific to CS 3 Bucharest Passive Solar Heating System*, Proc. Joint Seminar Romania-USA, Bucuresti, 1985

[8] *** – *Solar Energy Used in Dwelling Heating*, INCERC, Contract 192 / 1973

[9] Davidescu A. – *Heat Exchange in Industrial Plants*, Ed. Tehnica, Bucuresti, 1968

[10] Giblin R. – *Transmission de la chaleur par convection naturelle*, Eyrolles, Paris, 1974

[11] Constantinescu D. – *UTR Method and Applications in Solar Heating*, Conf. Nat. En. Sol., Iasi, 1982