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Applications Oriented Research on Solar Collectors at the "Politehnica" University of Timişoara

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1. Introduction

Solar energy, as a free and clean energy source, is successfully used in the present in the European Union countries for individual and social buildings climatization, water heating and cleaning, electricity and motor force production, heating in swimming pools etc. Examples are numerous so that we briefly present a selection.

The Self-Sufficient Solar House (SSSH) project has been started within the Fraunhofer Institute for Solar Energy Systems in Freiburg, Germany in 1988. The construction of the building started in 1991 and has been accomplished in October 1991. The aim of the project has been to demonstrate that the needs of thermal and electric energy of a residence may be satisfied exclusively by solar means. Measurements and calculation showed that, for a primary equivalent energy of 600 MWh invested in the construction of the residence, the system returns that energy in a period of 10-20 years and its life time is about 80 years, meaning that SSSH provides an economy of primary energy for about 60 years and diminishes the pollution of the ambient (Sthal et al., 1994).

The Northern Solar Heating Demonstration Project (NSHDP) has been started in Denmark in 1985 with the aim of domestic hot water production for a block with 150 flats. The system works with an efficiency of 25-33% and provides an annual thermal energy of 485-585 kWh for every square meter of collecting surface (Pedersen, 1993).

Through the Thermosyphoning Air Panels project (TAP), an elementary school situated 26 km north of London, Great Britain, with a living surface of 1631 m^2, has been architecturally adapted to the use of solar energy in 1989 (Lo et al., 1994). The school is used from Monday to Friday by 181 pupils and 11 teachers between 8 am and 3.30 pm.

The solar collector is the active part of the solar to thermal energy conversion chain. The importance of the role played by the solar collector in solar applications explains the high amount of research performed for increasing its efficiency and for decreasing its price and impact on the ambient. Such research goals are fulfilled by devising new materials and innovating geometries. Simple, efficient, cheap and ambient-friendly solar installations are the attractive ones (Haberl et al., 2008). Some examples are cited below.

The energy autonomy of mountain meteorological stations and huts in the Italian Alpes is achieved by means of thermal, photovoltaic and biogas systems. The accumulation boiler is placed 3 m below the collector. The liquid agent circulates by gravity and thermosyphoning (De Beni et al., 1994).

The use of a small photovoltaic (PV) module for the generation of the electricity supply of the fan attached to a thermal collector increases the overall efficiency by 3-5% and improves energy autonomy (Eicker & Steeberger, 1996).
Insulating transparent materials reduce thermal losses and allow reaching temperatures by 80-120 K above ambient (Folkerts et al., 1996).
Polymer based materials, immune to the action of hot, salted water and to freezing enter the structures of sheets used for covering swimming pools (Rommel et al., 1996).
The research in the field of Solar Energy performed within the Physics Department from the "Politehnica" University of Timişoara, Romania has had the goal to evaluate and exploit the solar potential of the western part of the country. The results are relevant for the Danube-Kris-Mures-Tisza region. Procedures, equipment and installations for hot air production, drying of ceramic products, bitumen melting, home climatization, stock raising, hot water production, solar radiation measurement and environment protection have been devised. An air solar collector, a Trombe wall and solar collectors built from recyclable materials are treated in this chapter. Next, some thermal applications of Solar Energy that rely on the solar collectors are presented.

2. Air solar collector

The plane solar collector represented in Fig. 1 has been devised, constructed and studied. The elements in Fig. 1 are: transparent plate - 1, absorber - 2, pipe for air circulation - 3, insulating sheet - 4, support - 5, chassis - 6. The elevation angle is denoted by s and the overheight by h. The pipe for air circulation is represented in Fig. 2.

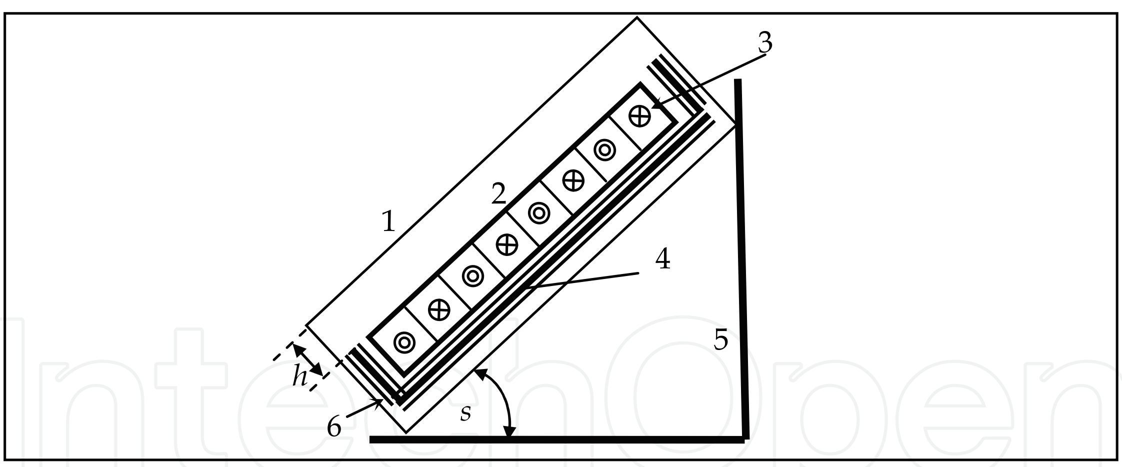

Fig. 1. Air solar collector

The absorbing plate plays also the role of a wall for the air circulation pipe. Two neighbouring corridors share a common wall, so that the transport factor is unity, $F_t = 1$. The air flow rate may be in the interval 25 – 135 m³/h and the length of the pipe is $L_{cd} = 12.4\ \text{m}$. The air admission and evacuation holes are square shaped, with a side $a = 0.25\ \text{m}$. The collector surface is $A = aL_{cd} = 3.1\ \text{m}^2$. The surface of the collector is tilted by an angle $s = 45\ \text{deg}$. The air is circulated by a low power fan, having a power of 30 W for an air flow rate of 135 m³/h, while the power of the installation is about 800 W. A photovoltaic panel may be used for powering the fan. The absorption – transmission equivalent product is $(\tau\alpha)_{eff} = 0.844$ and the thermal insulation is characterized by $U = 4.73\ \text{Wm}^{-2}\text{K}^{-1}$.

Some regions of the absorbing surface are shadowed by the window supports and by the walls of the chassis. The first effect has a daily variation, while the second one may be considered to have a hourly variation. A cross section through the lateral window support is represented in Fig. 3. The fluid crosses n times the shadows created by the central and lateral supports , with n = 5 the number of pipes. The total length of the shadow may be expressed as

$$L_1 = n\left[h\tan(\theta) + b - d\right], \tag{1}$$

where h is the overheight (Fig. 1), b is the width of the central support (Fig. 2), d is width of the insulation (Fig. 3) and θ is the angle between the incoming sun ray and the normal to the absorber. For example, at equinox, $\theta = \omega\Delta\tau$, with ω – the apparent angular speed of the Sun and $\Delta\tau$ - time from noon.

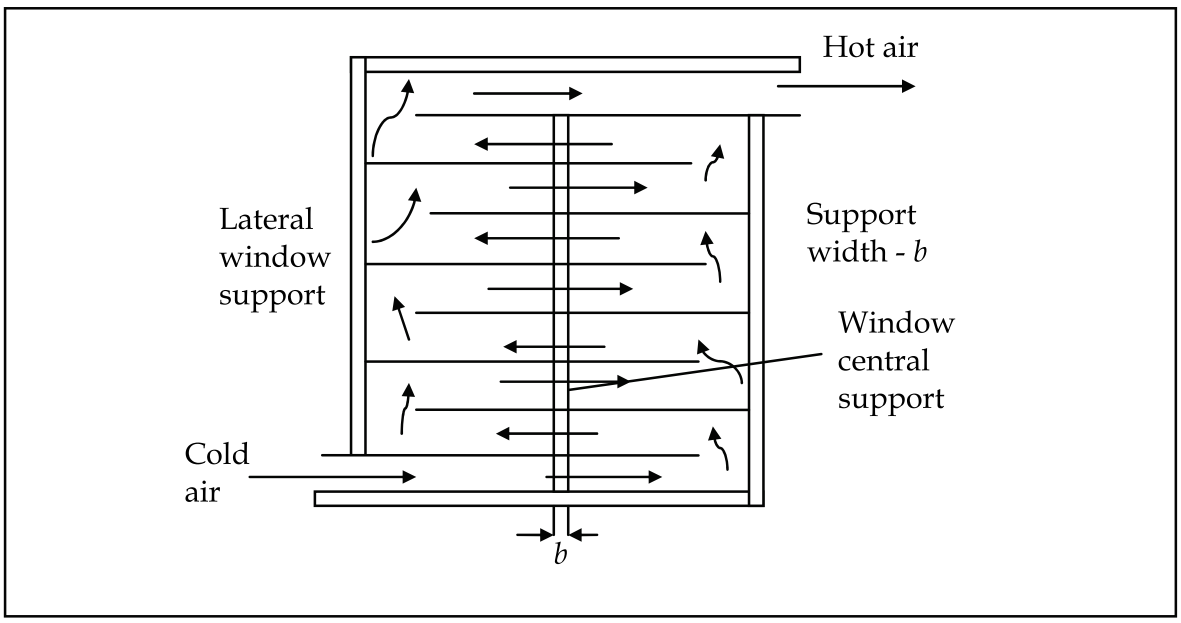

Fig. 2. Pipe for air circulation.

The length of the pipe that is irradiated allowing for the heat to be absorbed is

$$L = L_{cd} - L_1 \,. \tag{2}$$

The surface of the fluid that is irradiated, $A_c = aL$, results:

$$A_c = a\left[L_{cd} - n\left(h\tan(\theta) - d + b\right)\right] \tag{3}$$

so that the fraction of surface that is effectively used is

$$f = 1 - \frac{n\left[\tan(\theta) + b - d\right]}{L_{cd}} \,. \tag{4}$$

The variation of the fraction f with the hour is represented in Fig. 4. It may be seen that $f \approx 0.85$ for a time interval of 4 – 6 hours centred at noon.

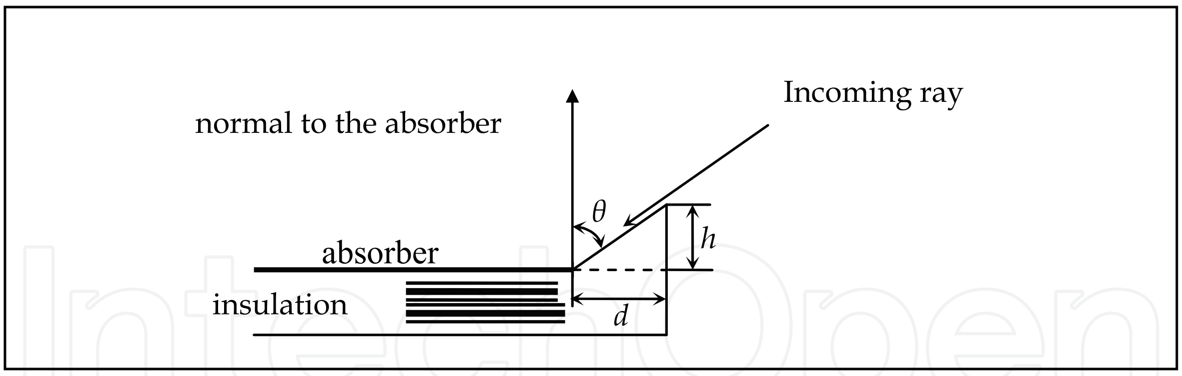

Fig. 3. Shadowing of the surface.

In order to find the equations that characterize the system, we note that the heat obtained by thermal conversion is transferred to the working agent. The fluid enters the collector at a temperature T_{fi} and exits at a temperature T_{fe}. The energy balance for the fluid that flows through a small segment of pipe, of length Δy, is

$$\dot{m}C_pT_f\Big|_y - \dot{m}C_pT_f\Big|_{y+\Delta y} + q_u'\Delta y = 0\,, \tag{5}$$

where $\dot{m}$ is the mass flow rate, C_p is the isobar specific heat of the fluid, q_u' is the heat flux absorbed by the unit length of a current tube and T_f is the temperature of the fluid.

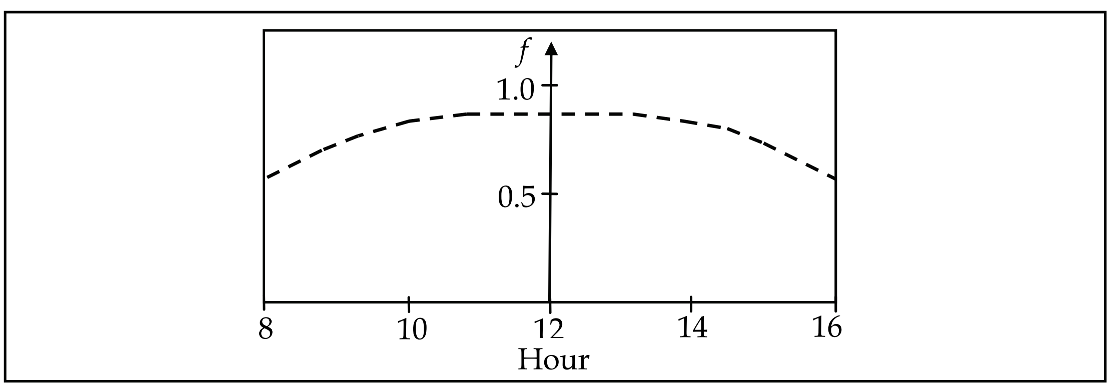

Fig. 4. Irradiated fraction of surface versus hour.

The flux absorbed per unit length may be expressed as

$$q_u' = aF'\left[S - U\left(T_f - T_a\right)\right] \tag{6}$$

where $S = (\tau\alpha)_{eff}\,G_c$ is the total flux density absorbed by the black plate, G_c is the solar flux density in the plane of the collector, F' is an efficiency factor and U is the coefficient of heat loss in the ambient.

By manipulating (5) and (6), the equation of the temperature may be obtained:

$$T_f = T_a + \frac{S}{U} + \left(T_{fi} - T_a - \frac{S}{U}\right)\exp\left(-\frac{F'AU}{\dot{m}C_p}y\right). \tag{7}$$

By setting $y = L$, the temperature at the collector output T_{fe} may be obtained.

If the collector is functioning in an open regime, the input temperature is equal to the ambient one $T_{fi} = T_a$, which, substituted into (7) yields (Luminosu, 1983)

$$T_f = T_a + \frac{S}{U}\left[1 - \exp\left(-\frac{F'aU}{\dot{m}C_p}\right)y\right]. \tag{8}$$

For $y = L$, and by using $A_c = aL$, the temperature at the output of the collector results:

$$T_{fe} = T_a + \frac{S}{U}\left[1 - \exp\left(-\frac{F'U}{\dot{m}C_p}A_c\right)\right]. \tag{9}$$

The temperature rise $\Delta T = T_{fe} - T_a$ versus the radiant power density absorbed by the black plate S is represented in Fig. 5. The curves are linear and start from the origin. Temperature rises as high as 50°C may be achieved.

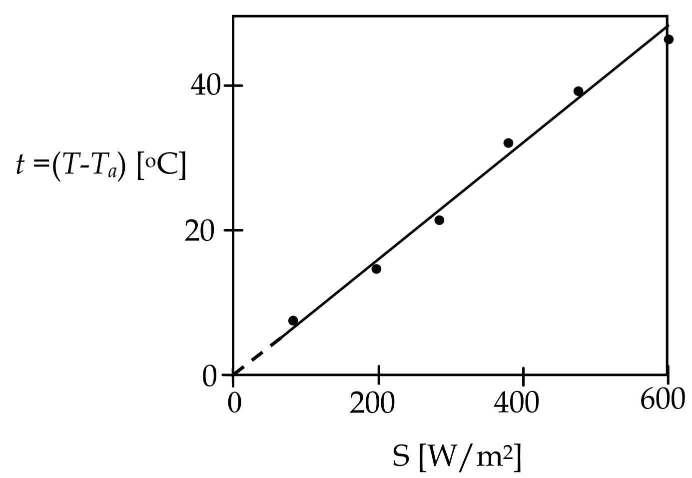

Fig. 5. Temperature rise versus absorbed power density.

The energy flow for the air collector in open state (heat per time unit or power), $\dot{Q}_u = \dot{m}C_p\left(T_{fe} - T_a\right)$ is (De Sabata & al. 1983):

$$\dot{Q}_u = \dot{m}C_p\frac{S}{U}\left[1 - \exp\left(-\frac{F'UA_c}{\dot{m}C_p}\right)\right]. \tag{10}$$

The collector power versus the density of the flux absorbed by the black plate is represented in Fig. 6, at various mass flow rates of the fluid. The power increases with the incoming radiation and the flow rate. At large flow rates, at noon, the power may increase up to 800 W.

The specific power is the ratio of the energy flow to the collecting surface

$$\dot{q}_u = \frac{\dot{Q}_u}{A_c}. \tag{11}$$

The values of the specific power are listed in Table 1. Measurements have shown that this quantity reaches larger values in the afternoon than before noon for similar values of the incident flux. This result may be explained by the fact that the carcass of the device provides additional heat to the fluid when the radiation intensity decreases.

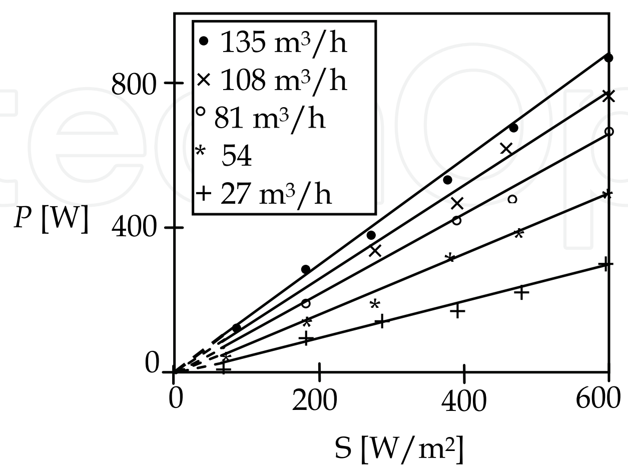

Fig. 6. Collector power versus absorbed radiation, parameterized by the flow rate

S[W/m²]	100	200	300	400	500	600
$\dot{q}_u$ [W/m²]	33	74	124	152	200	218

Table 1. Absorbed flux density and specific power.

The instantaneous efficiency of the collector is

$$\eta_i = \frac{\dot{Q}_u}{A_c G_c} . \tag{12}$$

Equations (10) and (12) imply

$$\eta_i = \frac{\dot{m} C_p S}{U A_c G_c}\left[1 - \exp\left(-\frac{F' U A_c}{\dot{m} C_p}\right)\right]. \tag{13}$$

The variation of the efficiency with the absorbed flux, for various values of the flow rate is represented in Fig. 7.
The long term performance of the collector is given by the average efficiency in the considered time interval

$$\tilde{\eta} = \frac{\dot{Q}_{u,average}}{A_c \tilde{G}_c} \tag{14}$$

where $\dot{Q}_{u,average}$ is the average value of the power provided by the collector and $\tilde{G}_c$ is the average value of the incident radiant power density in the considered time interval.
The hourly variation of the average efficiency is represented in Fig. 8, parameterized by the flow rate.

The curves presented in Fig. 8 show that efficiencies are high around noon, when the incidence angles are small and the absorption – transmission products are high. The time variation of the incidence angle determines changes of the absorption-transmission product which, at its turn, determines the variation of the efficiency. The curves present maxima at noon, but they are asymmetric with respect to noon: the slopes of the curves are smaller in the afternoon when the fluid is additionally heated by the metallic support. At high flow rates (135 m³/h), the efficiency of the collector approaches 40%. This reasonably high efficiency and the unsophisticated design recommend this solar collector for home climatization and for drying applications in industry.

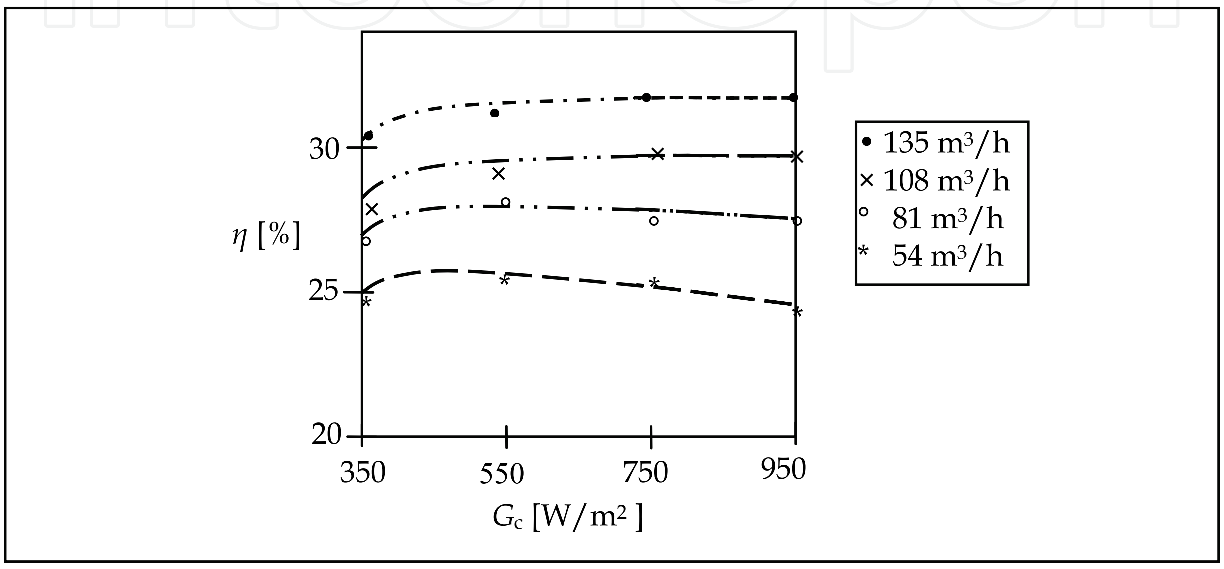

Fig. 7. Efficiency versus irradiation.

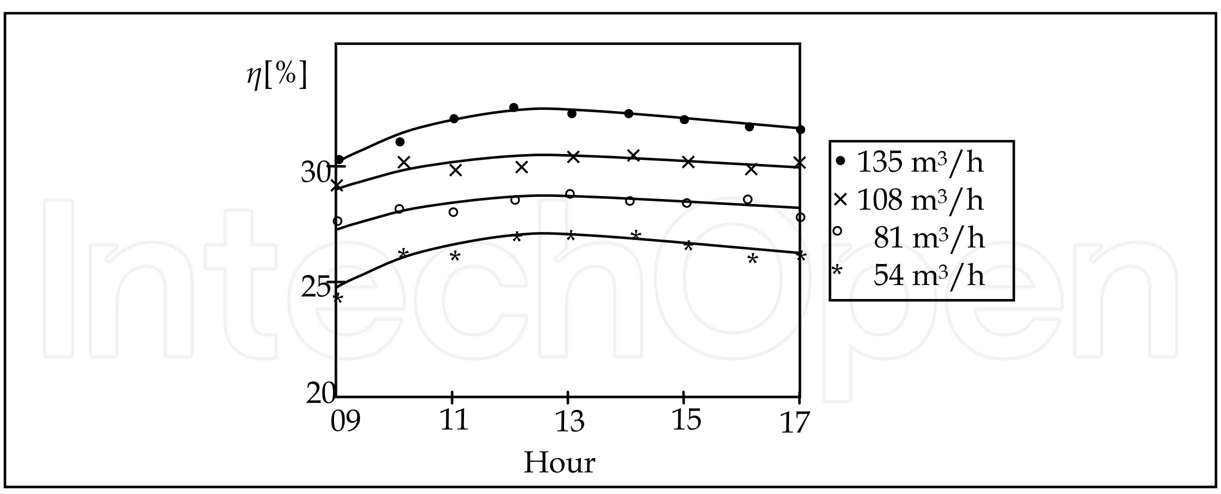

Fig. 8. Hourly variation of the average efficiency.

3. Trombe wall

The Trombe wall is the main element of heating systems for buildings based on passive solar gain. For an outside temperature t_{ext}=0°C and an inside temperature t_{int}=20°C, a simple wall (without solar installations) transfers heat towards the interior if the normal solar irradiation is greater than 465.2 W/m² (Athanasouli & Massouporos, 1999). Such conditions are met in Timişoara, Romania during transition months, between 11 am and 1 pm. In order to increase the contribution of the wall to the energy required for heating the room and in order to decrease energy losses during night time, the wall may be covered with a glass plate during daytime and additionally with a curtain at night fall (Ohanesion & Charteres, 1978). The solar panels mounted on the eastern and southern walls of a school supplied each year a thermal energy of 2469 kWh during classes (Lo et al., 1994).

An experimental setup with Trombe wall has been built at the "Politehnica" University of Timişoara in order to evaluate the opportunity of implementing passive solar installations in the region. The installation has been used for heating a living room, complementary to electric power, during transition months (March, April, September and October). The Trombe wall has been placed on the southern wall of an ordinary building with four rooms at the ground floor, otherwise heated by classical means. The three rooms that were not heated by solar means have been maintained at a temperature of 21 ± 1 °C, so that the heat lost through the door of the target room could be neglected (De Sabata et al., 1986a, 1986b).

The dimensions of the solar heated room were $2.80 \times 4.75 \times 1.75$ m and the dimensions of the window on the southern wall were 1.0×0.75 m. The walls made with bricks were 0.39 m thick and were plastered with lime and mortar. The concrete foundation was $h = 1$ m deep and 0.49 m thick. The underground water layer is situated at a depth smaller than four meters and it has a temperature $t_f = 10$ °C. The surface of the Trombe wall was $A_T = 8.8$ m² (Fig. 9). The curtain from *I* covered the wall during night time. The air dampers $L_{1,2,3}$ controlled the direction of the air flow. A water container *C* was attached to the passive wall in order to raise the inside air humidity. The small power fan *F* ($P = 10$ W) contributed to the uniformity of the thermal field.

The heating of the room has been provided by a radiator with thermostat *R* and the Trombe wall. The heat supplied by the two devices balanced the thermal losses of the room through the eastern wall, the floor and the window (Luminosu, 2003a). Temperatures at points 1..12 have been measured with the thermometer *V*, having an error of ± 0.1 °C. The global radiation intensity *G* has been measured with an error of $\pm 5\%$ by means of an instrument built in our laboratory (Luminosu et al., 2010), the electric power with an aem1CM4a instrument (*N* on Fig. 9), with an error of ± 5 Wh and the air humidity has been measured with the hygrometer *H*, having an error of $\pm 5\%$. Additionally, the velocity of the air current has been measured with the anemometer FEET (*A*, Fig. 9), error $\pm 10\%$ and the illumination at the centre of the room has been measured with a Lux PU150 light meter.

The average air velocity has been found to be $\overline{v} = 0.15$ m/s, which corresponds to the upper comfort limit and, due to the additional water container, the humidity has been kept in between the limits 35..70%, a range well inside the comfort limits. The lighting at the centre of the room has been in the range 50..70 lx in the horizontal plane; these values have been achieved by operating the blinds and by turning on the 12 W ECOTONE light bulbs for about 4 hours a day.

The measured values of the solar radiation (1), temperature at the upper air damper (2), temperature at the centre of the room (3) and ambient temperature versus hour are presented in Fig. 10. Measurements have been performed in autumn (October and November) and spring (mid February and March). Temperature ranges of 14..17.5 °C at the centre of the room, 21..31°C at the upper air damper and 18..22 °C near a wall shared with an adjacent room have been obtained.

The daily average radiant energy has been $\bar{H}_d = 99.1\ \text{MJ}$. Adding up the hourly measured heats resulted in the following average daily heats: the heat lost by the room $\bar{Q}_{dL} = 22.4\ \text{MJ}$, the heat supplied by the passive wall $\bar{Q}_{dT} = 10.26\ \text{MJ}$ and the electric energy for heating $\bar{Q}_{del} = 12.31\ \text{MJ}$

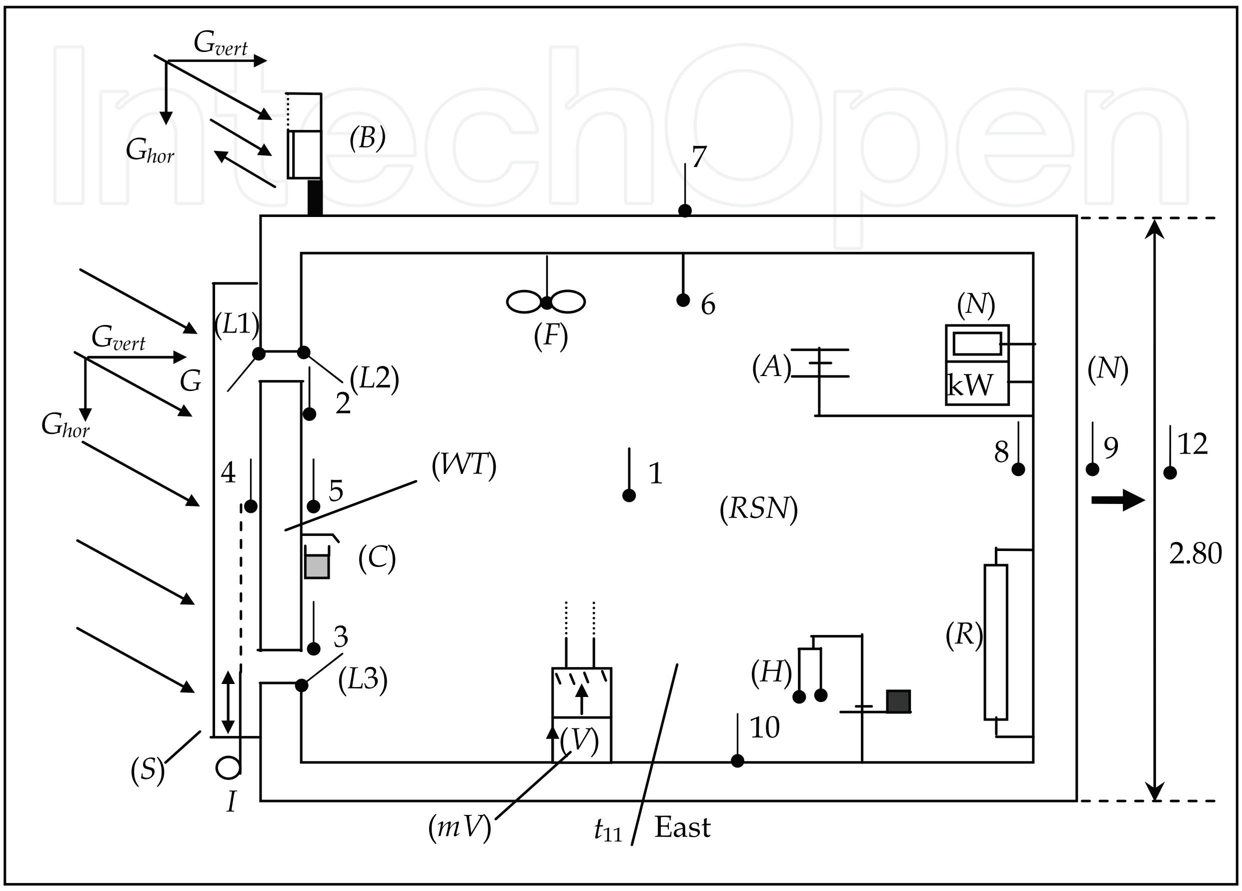

Fig. 9. Room with Trombe wall and measuring points.

The power of the Trombe wall has been $\bar{P}_T = 237.5\ \text{W}$. As the average number of days with clear sky during the transition months is $N = 46$, the annual average heat supplied by the wall is $Q_{yT} = NQ_{dT} = 131\ \text{kWh}$. The daily efficiency of the passive wall is $\bar{\eta}_T = 100 \times \frac{\bar{Q}_{dT}}{\bar{H}_d}$. Depending on the season, the efficiency of the considered wall varied between 7.8 and 10.4%. The specific annul heat of the wall is $\bar{q}_{yT} = \frac{\bar{Q}_{yT}}{A_T} = 14.9\ \text{kWh/m}^2$.

The sensation of thermal comfort is determined by the inside temperature and the temperatures of the walls and objects the human body establishes a radiant energy exchange with. According to hygienists (Săvulescu, 1984), the radiant temperature (°C) is given by

$$t_{rad} = \sum_{j=1}^{n} f_j t_j \tag{15}$$

and the room temperature by

$$t_{room} = \frac{t_{int} + t_{rad}}{2}, \tag{16}$$

where t_{int} is the inside room temperature, n is the number of elements the body exchanges radiant energy with and f_j are the shape factors $f_j = \frac{A_j}{A}$ (A_j – area of the j'th element, A – exchange area).

The level of comfort is optimal when the room temperature is equal to the comfort temperature prescribed by hygienists. According to Bradke (in Săvulescu, 1984), an inside air temperature $t_{int} = 21^\circ C$ must have a radiant temperature correspondent $t_{rad,adm} = 16.3^\circ C$ and a comfort temperature one of $t_{comf} = 18.7^\circ C$.

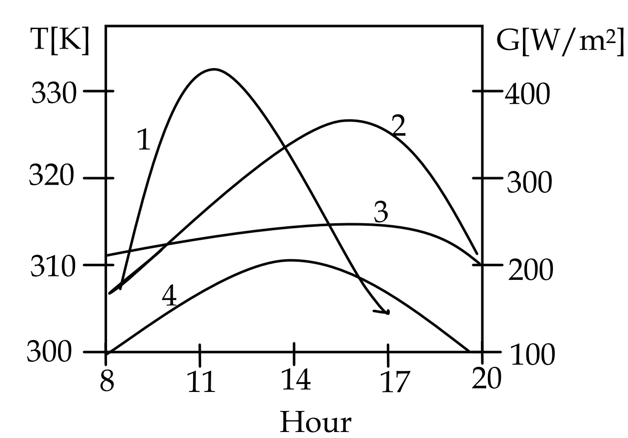

Fig. 10. Temperature of the passive wall and global solar radiation versus hour.

The shape factors f_j and the average temperatures $\overline{t}_j$ of the walls of the room heated by the passive wall, the average radiant temperature $\overline{t}_{rad}$ and the room temperature $\overline{t}_{room}$ are given in Table 2.

Radiant element	f_j	$\overline{t}_j$ (°C)	$\overline{t}_{rad}$ (°C)	$\overline{t}_{room}$ (°C)
Eastern wall	0.09	16	17.9	19.5
Southern wall	0.24	26		
Western wall	0.09	18		
Northern wall	0.24	18		
Ceiling	0.16	14		
Floor	0.16	13		

Table 2. Thermal comfort inside the room.

The Trombe wall produces a room temperature by 0.8°C higher than the comfort temperature prescribed by hygienists.

The thermal comfort factor, according to Van Zuilen (in (Săvulescu, 1984)), is given by

$$B = C + 0.25\left(\overline{t}_{\text{int}} - \overline{t}_{rad}\right) + 0.1\overline{x} - 0.1\left(37.8 - \overline{t}_{\text{int}}\right)v^{1/2}, \tag{17}$$

with x – absolute humidity inside, $\overline{x} = 12\ \text{g/kg}$; C – constant depending on the season, $C = -10.6$ in this case; v – velocity of the air.
Depending on B, the thermal sensation of comfort may be optimal $B = 0$, satisfactory $B = \pm 1$, or discomforting $B = \pm 3$. In our case we have $B = -0.325$, meaning that comfort reaches an optimal state.

4. Solar collectors from recyclable materials

Applications of Solar Energy in urban areas are facilitated by the existing infrastructure. However, in isolated locations, additional preparations that raise the costs of installations are necessary. Therefore, the possibility of using waste materials, resulted from demolishment of old buildings and from old appliances for devising low cost, small size solar collectors has been studied in our laboratory (Luminosu, 2007a). Transforming waste into raw material for a useful application has both a favorable impact on prices and on ambient. The main mechanisms of this impact are: decrease in the quantity of polluting waste; decrease in the demand for metal and glass from industry; decrease of energy consumption from classical sources; raise in the quality of life by the availability of low cost and ambient friendly energy in isolated locations; economy in transportation costs, as discarded materials are often available at the place were the collectors are built (e.g. following demolishments of old buildings); and economy in fabrication costs, as materials are often preprocessed and already cut into usable shapes, so that the collectors may be realized in modest mechanical workshops.

4.1 Solar collector from old glass plates

A first solar collector has been realized from glass plates, Fig. 11. The represented elements are: metallic frame – 1; vertical glass plates oriented towards south – 2; heated water – 3; cold water tank – 4; taps – 5, 6; mechanical support – 7; expansion bowl – 8; solarimeter – 9. Water is stored between the glass plates. One plate is transparent, while the other plate is painted in black, in order to absorb the solar radiation. The hot water is removed through

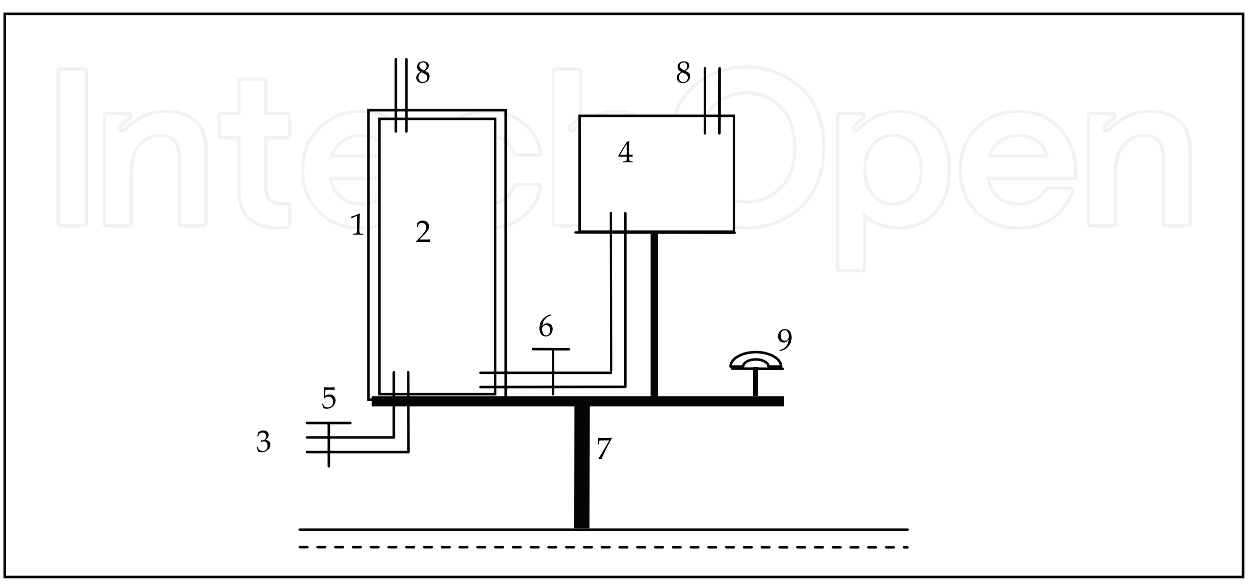

Fig. 11. Collector with glass plates.

the tap 5. The collector is filled with water contained in the tank 4, by the principle of communicating vessels, through the tap 6. The collector is positioned vertically in order to avoid breaking of the glass plates. The dimensions of the plates are 40 × 70 cm. The dimensions of the collector and the quantity of water stored between the glass plates must be kept reasonably low, by mechanical reasons related to the resistance to bending of the glass. The thickness of the water layer is 1.5 cm and the mass of water is m=4.2 kg.
The collector has been experimentally tested. Solar radiations has been measured with a solar wattmeter built in our laboratory (Luminosu et al., 2010). The water temperature T_w and the ambient temperature T_a have been monitorized. The water has been heated in time intervals comprised between 0.5 and 5.5 hours, symmetrically placed around noon. Measurements have been taken every 0.5 hours. It has been found that, under clear sky conditions, the water temperature raised by approximately 32°C with respect to the ambient temperature so that the water could be used for domestic purposes. The obtained average efficiency of the collector has been $\overline{\eta} = 33.3\%$.

4.2 Solar collector based on the heat exchanger of an old refrigerator

A second design consisted of a solar collector built around some parts of an old refrigerator. These parts are frequently available following the current replacement of old, heavy energy consuming refrigerators with modern, ecological ones. The disclosed heat exchangers and polystyrene sheets from the old refrigerators may be used for building small sized solar collectors, with favourable effects on the ambient.
The design of a collector that uses parts from an old "Arctic" refrigerator is presented in Fig. 12.

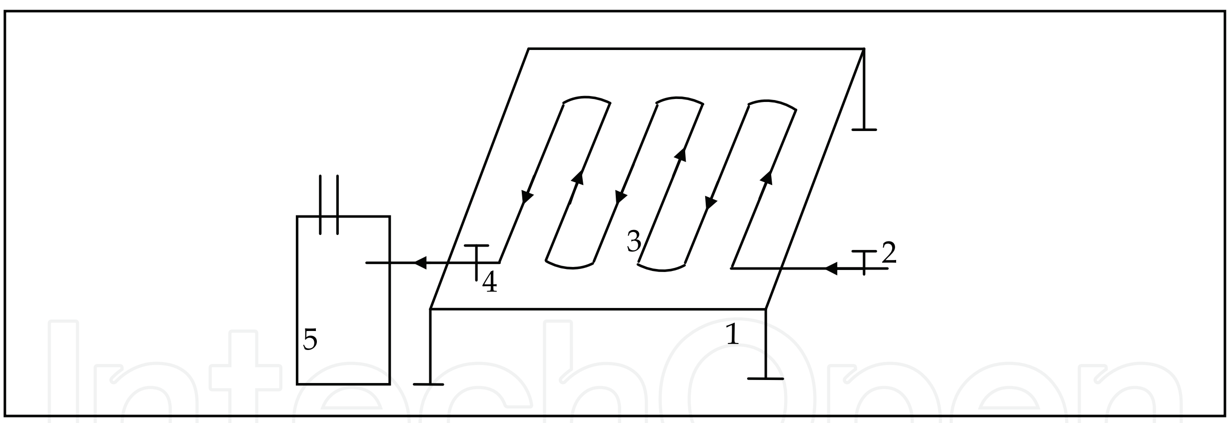

Fig. 12. Collector with pipes from an old refrigerator.

The elements in Fig. 12 are: mechanical support – 1; tap for cold water – 2; heat exchanger – 3; tap for hot water – 4; container with warm water – 5. The heat exchanger is 0.90 m long and 0.45 m wide, the pipes circulating the working fluid are spaced by 6 cm and the collecting area is 0.405 m². The collector is oriented towards south, at a tilt angle of 45 deg. A greenhouse effect is created by means of a glass plate, 3 mm thick. The hot water is accumulated in a Dewar pot. A coefficient of thermal losses $U = 6.453\ \mathrm{Wm^{-2}K^{-1}}$ and an absorbtion – transmission equivalent product $(\tau\alpha) = 0.847$ have been determined. The collector has been studied in open circuit.
For large flow rates of the water, of up to 3.60 kg/h and for densities of the solar flux of 500..600 W/m², the raise of the water temperature may reach up to 30°C and the efficiency

may be larger than 50%. In this way, the temperature of the water in the Dewar pot reaches 50..60°C, a temperature that allows the domestic use.
In conclusion, the use of recyclable materials for devising small sized thermal solar collectors has favourable impacts both on the way of life in isolated places and on the ambient.

5. The "Politehnica" solar house

Solar houses are equipped with thermal solar systems that maintain the inside temperature at a comfortable level and produce hot water for domestic use. As maximum solar radiation and energy need are not synchronous events, several types of thermal solar installations, which complement the classical ones, have been conceived. Some examples from the literature include: a hybrid solar system with heat pump, plane collectors and storage tank with $CaCl_2 \cdot 6H_2O$ (Çomakli, 1993); thermal solar system with heat pump that relies on the heat accumulated in the roof of the building (Loveday & Craggs, 1992); and thermal solar system with plane collectors complementary to the gas installation (Pedersen, 1993). Close to our laboratory, an experimental Solar House has been built and experimented with.

5.1 The solar house and measuring devices

The building has two rooms, a lobby and an access hall. A "minimal thermal loss enclosure", situated at the first floor has been defined and provided with a double layered door and a triple layered window. The dimensions of the room are $3.5 \times 3.5 \times 2.8\ \text{m}$, giving a total volume $V_r = 35\ \text{m}^3$ and a total thermal exchange area $A_r = 63.7\ \text{m}^2$. The technical room is situated at the ground floor. A bedrock thermal accumulator, in the shape of a parallelopiped of dimensions $1.5 \times 1.5 \times 4\ \text{m}$ and filled with river stone $(C = 16.6\ \text{MJ/Kg})$ is deposited in the basement. The concrete walls are 40 cm thick and insulated with mineral wool. The main side of the building is south oriented.
The energy system shown in Fig. 13 includes the plane collectors – 1, the heat exchanger – 2, the thermal accumulator – 3, the heated room – 4 and the technical room – 5. The collecting field consists of twelve "Sadu 1" solar collectors connected in parallel. Each of the plane collectors is provided with aluminium pipes with inner diameter of 20 mm, facing south and tilted by an angle $s = 45\ \text{deg}$ from horizontal. The dimensions of the collectors are $2.0 \times 1.0 \times 0.12\ \text{m}$ and they are insulated with a 50 mm thick layer of mineral wool. The case is made of 0.8 mm steel plates. The heat-transfer fluid is water, activated by a 40 W Riello TF108 pump at a mass flow rate $m_w = 300\ \text{kg/h}$. The total collecting surface is $A_c = 24\ \text{m}^2$ and the thermal and optical parameters are $U_c = 3.7\ \text{W/m}^2$ and $(\tau\alpha)_{eff} = 0.81$.
The heat exchanger is of air-water type with copper coil and it provides a power of 60 W and a mass flow rate $m_a = 1154\ \text{kg/h}$. The heat carried by the hot water from the collectors to the coil of the heat exchanger is transferred to the air and carried to the bedrock. The direction of the air flow between the heat exchanger, tank and heated room through the nozzles *C*, *D* and *H* is determined by the slide dampers mounted at points *a*, *b*, *c* and *d* (Fig. 13). The heated room (minimum loss enclosure 4) may be heated either by solar means (the hot airflow comming fron the accumulator through nozzle *H*) or electrically from the radiator *R* equipped with a thermostat. The temperatures at points *A*, *B*, *C*, *D*, *H* (heat carrying fluid), *F* (hall), *I* (tank), *G* (exterior) and *T* (technical room) are read on the electric thermometer *V* with an error of $\pm 0.5^\circ\text{C}$. The thermometer is equipped with 1N4148 diode

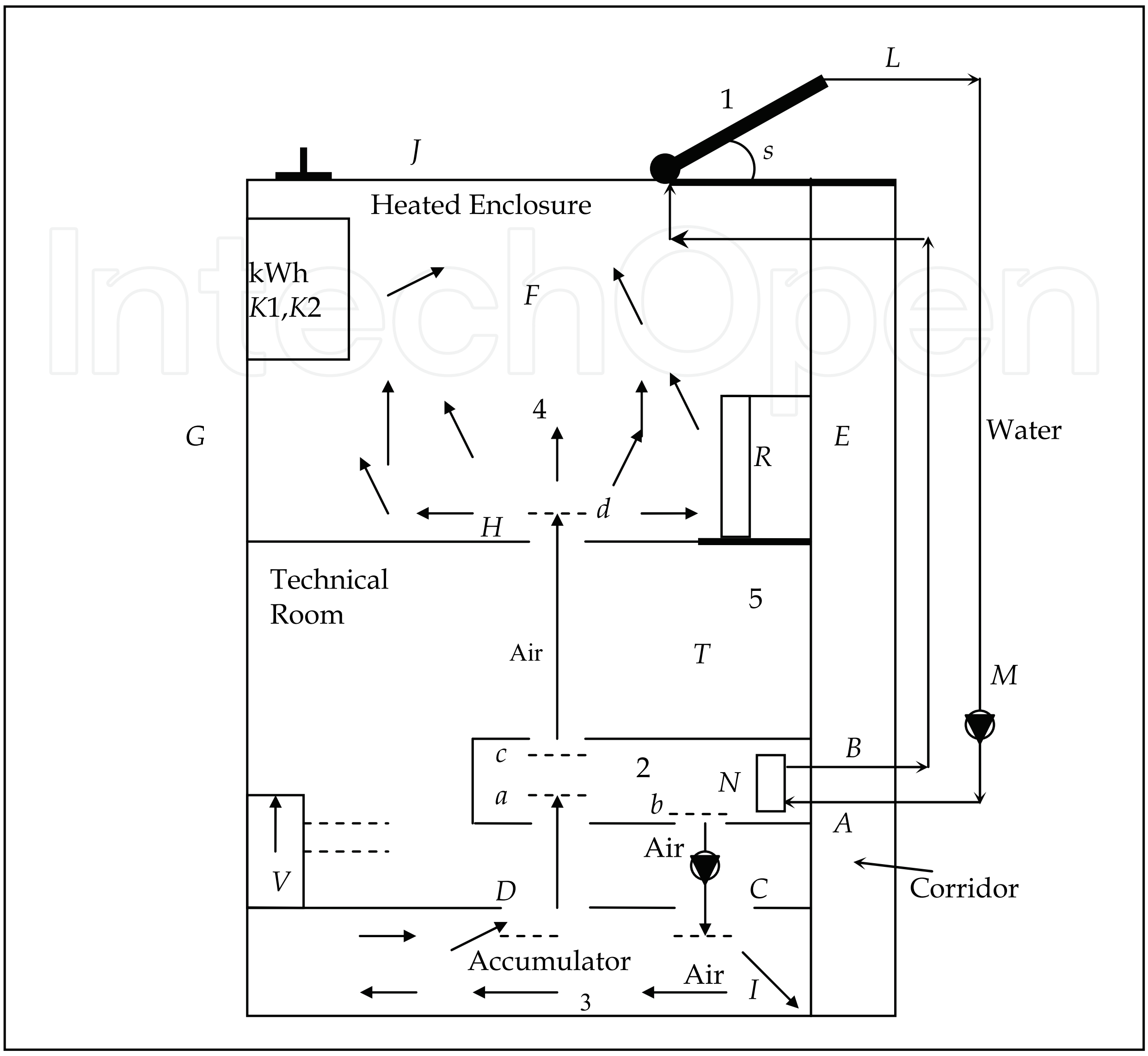

Fig. 13. Simplified chart of the energy system of the Solar House

sensors. The intensity of the solar radiation *G* is read on the pyrheliometer *J* with an error of $\pm 1\ \mathrm{W/m^2}$. The flow rate is obtained by dividing the volume recorded with the AEM BN5 water gauge, with an error of $\pm 25\ \mathrm{cm^3}$, at point *M*, by the recording period of time. The air velocity is measured with a FEET anemometer at point *N*, with an error of $\pm 0.5\ \mathrm{m/s}$, so that the air flow rate may be evaluated from $V_a = A_a v_a = 895\ \mathrm{m^3/h}$ (A_a is the area of the orifice of the nozzle). The electric energy used by the radiator *R* for heating is read on the AEM 1CM4A meter at point *K*1, with an error $\Delta Q_{el} = \pm 5 \times 10^{-3}$ kWh and the energy used by the pumps is read on a similar meter at point *K*2. A βM135 temperature detector is mounted at point *L*. The detector triggers a control circuit that starts the pumps if the water at the collectors output has a temperature over 50^{o}C.

5.2 Analytic model for the solar house

The heat loss per time unit through walls, ceiling, and through window and door openings is given by (De Sabata & Luminosu, 1993)

$$\dot{Q}_1 = \sum_{i=1}^{2} m_i A_i \frac{\Delta T_i}{R_i} \tag{18}$$

where $\Delta T_1 = T_F - T_G$ and $\Delta T_2 = T_F - T_E$; m – thermal mass coefficient, $m_1 = 0.90$ for the walls and $m_2 = 1.2$ for the window and the door; A_i - the corresponding surface areas and R_i – global thermal resistances.

The heat per time unit required to warm up the air infiltrated through the shutters of the window and the door may be expressed as

$$\dot{Q}_2 = E(iL)v^{4/3} + \dot{Q}_{door} \tag{19}$$

where E=1 (first floor), i – air infiltration coefficient i=0.081 $\mathrm{Ws^{4/3}m^{1/3}K^{-1}}$, L – lengths of the shutters, L_{door}=5.4 m, L_{window}=4.4 m; v – wind velocity, v=3.4 m/s (typical value).

The thermal resistance is given by

$$R = \frac{1}{\alpha_{int}} + \sum_{j=1}^{3} \frac{d_j}{k_j} + \frac{1}{\alpha_{ext}} \tag{20}$$

where $\alpha_{int,ext}$ - surface thermal exchange coefficients, $\alpha_{int} = 8\ \mathrm{Wm^{-2}K^{-1}}$, $\alpha_{ext} = 22.8\ \mathrm{Wm^{-2}K^{-1}}$; d_j – thicknesses of the successive layers of materials that forms the walls; k_j – heat conductivity of the layers [$\mathrm{Wm^{-1}K^{-1}}$].

The heat loss per time unit for the room is the sum

$$\dot{Q}_L = \dot{Q}_1 + \dot{Q}_2\ . \tag{21}$$

The hourly heat loss is $Q_{hL} = 3600\dot{Q}_L$ and the daily heat loss $Q_{dL} = \sum_1^n Q_{hL}$ (n – number of hours).

The heat lost by the room is compensated through solar and electric gains:

$$\bar{Q}_{dL} = \bar{Q}_{H\to F} + \bar{Q}_{el} \tag{22}$$

Hourly measurements have been carried out over several series of 3-4 days during spring (March, April, May) and autumn (September, October, November), 2000. In order to obtain average insolation characteristics, the experimental data have been statistically processed as described below.

The measurement period has been split into 12 h intervals, successively numbered 1, 2, ..., n; then, $n = n_1 + n_2$, n_1 – number of intervals with significant insolation, n_2 – number of intervals without solar radiation (night time and days with overcast sky). The hourly and daily average energy have been calculated with:

$$H_h = 3600 G_{hc} A_c,\ \bar{H}_h = \frac{1}{n_1}\sum_{n_1} H_h \tag{23}$$

$$\bar{H}_d = \sum_{i=1}^{p} \bar{H}_{hj} \tag{24}$$

p – number of 1 h intervals in an insolation day, $p = 1..8$.

The hourly average temperatures at points shown in Fig. 13 have been calculated using the equation

$$\overline{t}_{hq} = \frac{1}{n_{1,2}} \sum_{i=1}^{n_{1,2}} t_{hqi}, \; q = A, B, C, D, E, F, G, H, I \; . \tag{25}$$

The elements of the energy system have been labeled as follows (Fig. 13): j=0 – collecting area; j=1 – collectors, between A and B; j=2 – heat exchanger, between C and D; j=3 – accumulator, between I and H; j=4 – room, between H and F. The hourly and daily average heat have been calculated for each segment using

$$\overline{Q}_{hj} = 3600 m_x C_x \Delta\overline{t}_{hj}, \quad \overline{Q}_{dj} = \sum_{(p)} \overline{Q}_{hj} \tag{26}$$

(e.g. $\Delta t_{h1} = \overline{t}_{hA} - \overline{t}_{hB}$); the subscript x identifies the nature of the fluid: $x = a$ – air, $x = w$ – water.

The average efficiencies of the successive links have been calculated with

$$\overline{\eta}_j = \frac{\overline{Q}_{d,j+1}}{\overline{Q}_{d,j}} \; . \tag{27}$$

For example, for the collectors we have $\overline{Q}_{d,0} = \overline{H}_d$, $\overline{\eta}_1 = \dfrac{\overline{Q}_{d,1}}{\overline{H}_d} = \dfrac{\sum_1^8 3600 m_w C_w \left(\overline{t}_{hA} - \overline{t}_{hB}\right)}{\overline{H}_d}$.

The average efficiency of the system is given by:

$$\overline{\eta}_{syst} = \prod_{j=1}^{3} \overline{\eta}_j \; . \tag{28}$$

5.3 Experimental results

The hourly variation of the quantity $\overline{H}_h$ versus hour of the average day is represented in Fig. 14 (Luminosu, 2003b).

The daily average of the radiant energy has been $\overline{H}_d = 389.8\ \mathrm{MJ}$. The average hourly temperatures at points A, B, C, D and I versus hour are represented in Fig. 15.

The average temperature at A, at noon has been 83°C. The highest temperature at A, i.e. 87°C, has been reached during May and September. During March and November, the same point has reached the lowest temperature, 61°C.

The maximum average temperature of the air in the heat exchanger has been of 52°C. The temperature of the accumulator has been carefuly maintained above 30°C all throughout the measurement period $\left(t_{min,st} = 30^{\circ}\mathrm{C}\right)$. The average increase in the temperature of the tank during the daily loading period has been $\Delta t = 11^{\circ}\mathrm{C/day}$. The average decrease in temperature during the extraction of heat from the bedrock has been of 4.5°C/day. The average temperature inside the heated room has been kept at $(20 \pm 1)^{\circ}\mathrm{C}$ for an ambient (exterior) temperature variation between 4 and 15°C. The average daily heat transferred by the collectors to the heat exchanger has been $\overline{Q}_{d,A \to B} = \overline{Q}_{d1} = 291.6\ \mathrm{MJ}$. An efficiency $\overline{\eta}_1 = 0.75$ for the collecting field has been obtained.

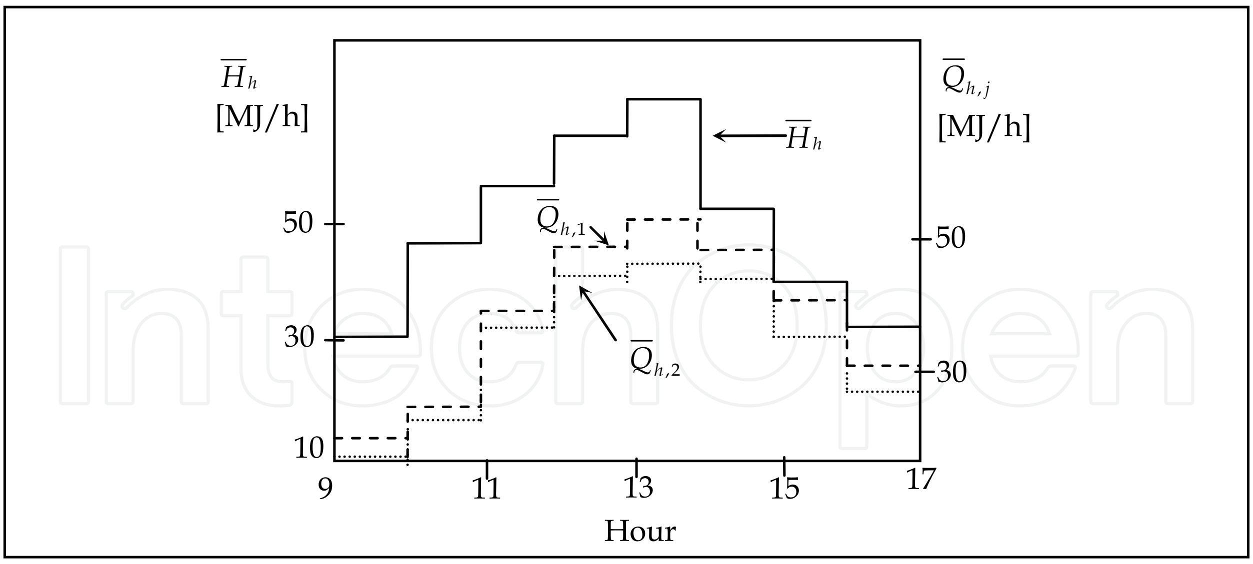

Fig. 14. Hourly averaged parameters $\overline{H}_h$, $\overline{Q}_{h,1}$ and $\overline{Q}_{h,2}$ versus hour.

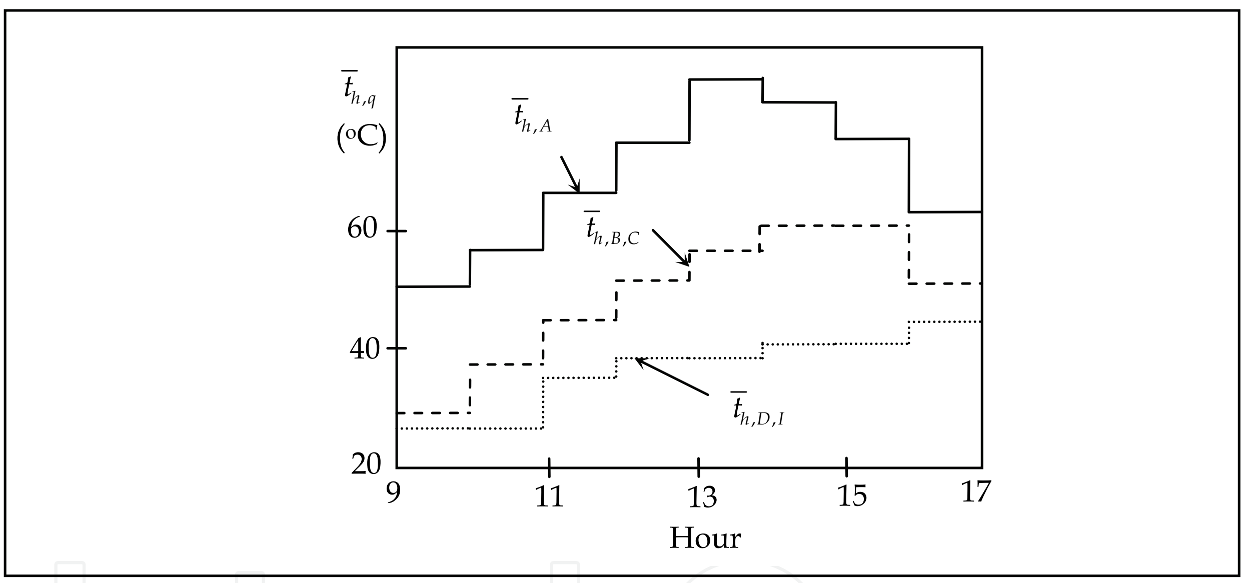

Fig. 15. Hourly averaged temperatures at points *A*, *B*, *C*, *D* and *I*.

The daily average heat swept away by the air curent from the coil conected between *A* and *B* (Fig. 13) has been $\overline{Q}_{d,D\to C} = \overline{Q}_{d2} = 239.4$ MJ, so that the average efficiency of the heat exchanger resulted as $\overline{\eta}_2 = 0.82$. The average quantity of heat transferred from the air current to the bedrock has been $\overline{Q}_{d,I\to H} = \overline{Q}_{d3} = 183.7$ MJ and the corresponding efficiency $\overline{\eta}_{3ld} = 0.77$. The room had a solar gain $\overline{Q}_{d,H\to F} = \overline{Q}_{d4} = 115.7$ MJ, so that the efficiency of the heat extraction from the storage environment resulted as $\overline{\eta}_{3ds} = 0.63$. The global efficiency of accumulation and storage of heat could then be calculated: $\overline{\eta}_3 = \overline{\eta}_{3ld}\overline{\eta}_{3ds} = 0.49$.
By using (28), one gets for the efficiency of the system $\overline{\eta}_{syst} = 0.30$.
The daily power consumption of the pumps is $\overline{Q}_{el,pumps} = 5.2$ MJ.
The average heat lost daily in the heated enclosure has been $\overline{Q}_{dl4} = 186.6$ MJ, which is compensated by solar energy $\overline{Q}_{d4}$ given above and by the energy provided by the electric radiator $\overline{Q}_{d,el,heat} = 70.9$ MJ. The solar energy ratio for room heating is

$$p = \frac{\bar{Q}_{d4} - \bar{Q}_{el,pump}}{Q_{dL4}} \times 100 = 59\% \ . \quad (29)$$

5.4 Discussion

The solar system has an efficiency of 30% with respect to the incident solar energy. The thermal energy produced by the energy chain of the residence could provide 60% of the needs of the minimum loss enclosure. As the global efficiency is the product of individual ones, a possibility to increase the efficiency is o decrese the number of elements in the series conection.

A typical value for the southern side of the roof of an average residence is $A' = 40 \text{ m}^2$. This collecting area would give each year, at the location with solar conditions similar to those considered above, a quantity of heat as high as $Q_u' = \bar{q}_{yu} A' = 3977 \text{ kWh}$.

The present study might be extrapolated to thermal systems that do not contain heat exchangers. In this case, the water collector has to be replaced with air collectors. The hot air may be directed both towards the room and towards the thermal storage tank.

As a conclusion, the development of passive and active solar architecture in the Euroregion might be beneficial for both private residences and institutional buildings.

6. Thermal system for drying ceramic blocks

Solar collectors may be used with good results as complementary sources of heat in technological processes that take place at moderate thermal levels. Such applications lead to the reduction of conventional fuel consumption and have favourable impact on the environment.

Air solar collectors are used worldwide in complex installations for the climatization of buildings and for drying industrial and agricultural products. In the case of plane solar collectors with air and bedrock between the absorbing and transparent plates, the rocks in the current tube increase the turbulence of the air flow, so that the coefficient of thermal transfer and consequently the efficiency are also increased (Choudhury & Garg, 1993). Air solar collectors with thermosyphoning and rocks in the fluid current tube are used for heating social buildings during daytime (Lo et al., 1994). Solar installations optimized through exergetic analysis are used in Mexico for drying mango fruits (Torres-Reyes et al., 2001).

At the Physics Department from the "Politehnica" University of Timişoara, a thermal system with plane collectors designed for drying ceramic blocks has been realized. The system relies on hot air from the collectors during the daytime and on heat accumulated in water tanks in the night time.

6.1 Description of the system

The thermal system has been placed on the roof of an industrial hall belonging to the Plant for Ceramics Products from Jimbolia, near Timişoara. The hall was 12 m long and had a volume $V_h = 312.5 \text{ m}^3$, Fig. 16, (De Sabata et al., 1994).

A common practice for drying ceramic blocks relies on the Johnson burner with fuel oil. At the place, the power was P=770 kW. Hot gases resulted after the burning process are blowed with a ventilator over the drying hall.

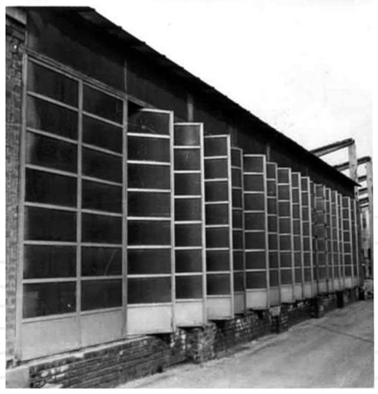

Fig. 16. Hall for drying ceramic products.

The hall was divided into 10 corridors; each corridor contained $n_1 = 8000$ (hollow) bricks posed on mobile shelves. The drying process consists of removing a quantity of water $m_2 = 0.5\ \mathrm{kg}$ from each brick, such the humidity decreases below 5% (Luminosu, 1993).
The minimum quantity of heat needed for a drying cycle is $Q_{cycle} = 25 \times 10^{10}\ \mathrm{J}$. The average quantity of water that must be evacuated in a 10 day cycle is $M = 167\ \mathrm{kg/h}$. The variation in humidity of the air is $\Delta x = 5 \times 10^{-3}$ kg of water per kg of air. The air in the hall must be renewed N=9 times per hour. As the working temperature varies between 40 and 60°C, a fraction of the heat Q_{cycle} may be obtained by solar conversion.
The longitudinal axis of the hall was oriented in the E-W direction. On the south oriented roof, a plane solar collector with air has been posed. The collectors were tilted by an angle $s = 30\ \mathrm{deg}$ and the total collecting surface was $A_C = 600\ \mathrm{m}^2$.
The path of the air current is presented in Fig. 17. The air was blown with fans placed in each corridor, having a power of 100 W.
The quality of the ceramic products is determined by the uniformity of the drying process. Consequently, the storage of thermal energy of solar origin for subsequent use during periods without sun is important.

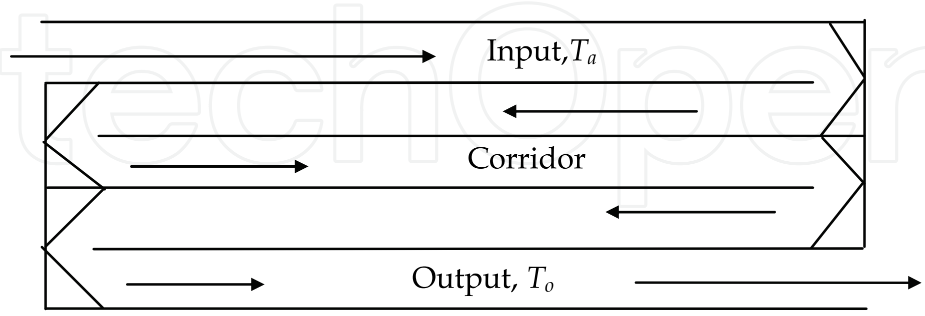

Fig. 17. Air current tube

A storage system of thermal energy as sensible heat has been designed to supply the requirements of the drying process during nighttime or for one or two days with low solar radiaton. Water collectors of type Sadu1 have been mounted on the roof of a neighboring hall, having a collecting area $A_C' = 360\ \mathrm{m}^2$. The average hourly specific power of the collectors has been $q = 2.09 \times 10^3 \text{ kJ} \cdot \text{m}^{-2} \cdot \text{h}^{-1}$. The average hourly captured thermal energy has been $Q_h{}' = 7.524 \times 10^5 \text{ kJ} \cdot \text{h}^{-1}$. The flow rate of the water through the storage installation has been $\dot{m}_w = 900 \text{ kg/h}$. The thermal energy has been stored as sensible heat in a storage tank having a volume $V' = 54 \text{ m}^3$. The hot water has been circulated with a pump at a flow rate of 800 l/h through radiators with horizontal pipes during nighttime. The air has been blown over the radiators by using fans and then the heated air was directed to the drying hall.

6.2 Experimental results

The drying procedure has been applied to ceramic blocks of dimensions $24 \times 12 \times 8$ cm. The density of the burnt material was 1300 kg/m^3.

Experiments at an industrial scale have been performed in the period April – September 1999. The drying process has been divided into cycles and it consisted of 3 cycles per month with a duration of 6..8 days/cycle. Several physical quantities have been monitorized: the solar radiation intensity G had a variation in the interval 460..920 W/m^2; the ambient temperature t_a (23..33°C); the air temperature at the hall entrance t_{in} (40..60°C); the water temperature at the output of the water collector t' (40..73°C); the flow rate of the working fluid v $(2.2..2.6) \times 10^3 \text{ m}^3/\text{h}$; the relative humidity of the air in the hall (30..35%).

The solar system has been used for heating the circulated air for 8 hours per day, in the interval 8 am – 4 pm. From the accumulation tank, heat has been extracted for time periods comprised between 8 and 16 hours per day. The Johnson burner has been used in parallel to the solar system in order to provide an air temperature at the input of the drying system of 40..60°C and the prescribed air humidity of 30..35%.

Cylindrical samples with radii of 2 cm and heights of 8 cm have been periodically extracted from the blocks. The samples have been weighted and compared with the burnt material in order to determine the mass of water from the ceramic block. The drying process was considered completed when the mass of water from the block was below 150 g.

The drying periods have been found as follows: n=6 days in June and July; n=7 days in May and August; n=8 days in April and September.

The following quantities have been determined:

a. the heat injected in the drying hall by the air collectors:

$$Q_{air} = \sum_{j=1}^{8} \dot{m}_{air} C_{air} \Delta t_j \Delta\tau \tag{30}$$

where $\Delta t_j = t_j - t_a$, t_j is the temperature of the air heated by the collectors and $\Delta\tau$ corresponds to the 8 hours interval when the air solar collectors were used;

b. the heat injected into the hall from the storage system:

$$Q_{storage} = \sum_{i=1}^{16} \dot{m}_{air} C_{air} \Delta t_i \Delta\tau \tag{31}$$

where $\Delta t_i = t_i - t_a$, t_i is the temperature of the air heated by the storage system and $\Delta\tau$ corresponds to the 16 hours interval when the storage system was used for heating;

c. the heat provided by the thermal solar system

$$Q_{syst} = Q_{air} + Q_{storage}; \tag{32}$$

d. the heat provided by the Johnson burner

$$Q_J = m_J q \tag{33}$$

where m_J is the mass and $q = 42$ MJ/kg is the calorific power of the fuel oil;

e. the total heat used for heating the hall:

$$Q_{nec} = Q_{syst} + Q_J\,; \tag{34}$$

f. the total energy cosumed for the hall

$$W = Q_{nec} + W_{electric} \tag{35}$$

where $W_{electric}$ is the electric energy that could be read on a meter;

g. the fraction of heat of solar origin from the total energy used for the hall:

$$f = \frac{Q_{syst}}{W}\,. \tag{36}$$

h. The efficiencies of the solar installations have been calculated by dividing the heat they provided by the solar energy incident on the collecting surfaces.

The monthly averages of these quantities are presented in Table 3.

Month	April	May	June	July	August	September
n (days/cycle)	8	7	6	6	7	8
G (W/m^2)	741	833	864	849	780	656
$\langle Q_{syst} \rangle$ (GJ/cycle)	80	91	84	82	81	72
$\langle Q_J \rangle$ (GJ/cycle)	173	161	166	170	173	190
$\langle Q_{nec} \rangle$ (GJ/cycle)	253	252	250	252	254	262
$\langle W_{electric} \rangle$(GJ/cycle)	0.51	0.45	0.38	0.38	0.45	0.51
$\langle f \rangle$ (%)	32	36	34	33	32	27
η_{air} (%)	53	56	60	60	57	54
$\eta_{storage}$ (%)	34	37	40	41	38	35

Table 3. Monthly averaged quantities that characterize the drying process.

The results presented in Table 3 for one year show that the solar thermal system may provide approximately one third of the thermal energy needed for the process of industrial drying of ceramic blocks. The calculated efficiencies might change from year to year following solar radiation and weather variability.

Experiments revealed that the presented system provided a uniform distribution of temperature so that a reduction by 10% of the number of blocks broken during the burning process with respect with other drying systems used within the same plant resulted.

The energy chain could be built with inexpensive and readily available materials and parts, produced by the local industry.

7. Solar heater for bitumen melting

7.1 Experimental installation

The extension of the applications field of solar energy is possible by identifying new industrial activities for which the thermal solar conversion is appropriate, efficient and cheap. Low and medium temperature thermal solar installations (50-80°C) have the largest efficiencies (40-50%).

Bitumen has many applications in civil engineering industry and road and highway construction. In industry, bitumen is heated by classical means in a three-phase process: heating up to 50-65°C for melting; heating up to 100-125°C for the asphalt mixture; maintaining the thermal level during inactive periods.

The D80/100 bitumen used in road construction has the following physical properties: penetration at 25°C of 0.0085 m, a melting point at 47.5°C, a ductility at –25°C of 1.30 m and a density at 25°C of 1050 kg/m³. As the melting temperature is sufficiently low, it is possible to use low and medium temperature thermal solar installations in the first phase of the heating process.

At the present time, the literature on this subject is rare. At the Physics Department of the "Politehnica" University of Timişoara, an experimental setup has been devised in an outdoor laboratory in order to test the possibility of using solar energy for bitumen preheating (De Sabata & Nicoara, 1984; Mihalca & al., 1988; De Sabata, 1986c). The results have been encouraging, although the thermal conductivity of the bitumen $\lambda_{bi} = 0.174\ \mathrm{Wm^{-1}K^{-1}}$ is much smaller than the thermal conductivity of water $\lambda_{w} = 0.651\ \mathrm{Wm^{-1}K^{-1}}$ (at 60°C). Further research in this direction is still necessary in order to find the optimal solution.

The experimental installation is presented in Fig. 18. The elements are: cylinder of iron plate – 1; mechanical support for the envelope – 2; insulating support for the cylinder – 3; envelope made of glass plates – 4; thermometer – 5, indicating the temperature in the collector, T_c and the ambient temperature T_a; device for the variation of the tilt angle of the axis of the cylinder with respect to the horizontal – 6. The cylindrical tank has a length of 0.30 m, a diameter of 0.15 m a mass of 1.17 kg and it contains 6.4 kg of bitumen. The installation is facing south and the axis of the tank is tilted by an angle of 30 deg with respect to the horizontal (Luminosu & al., 2007b).

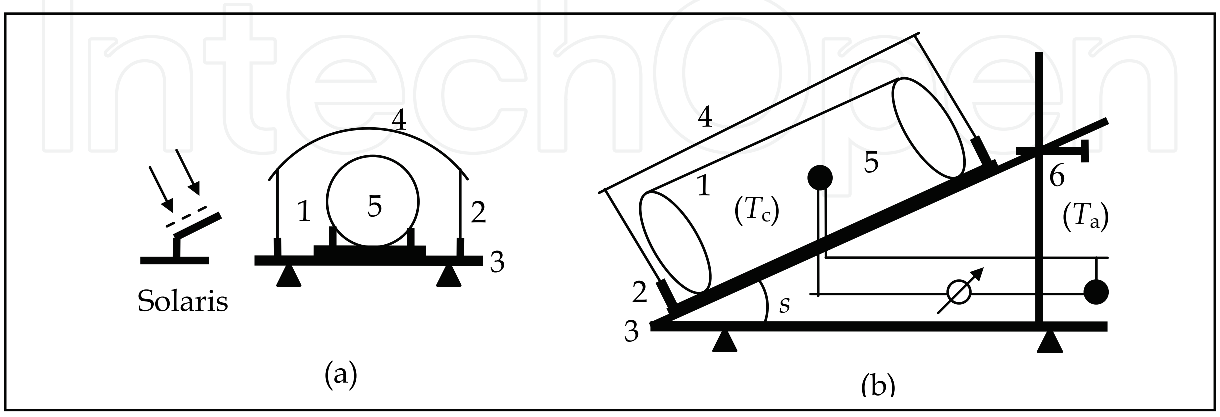

Fig. 18. Installation with semicylindrical glass envelope; (a) front view; (b) side view.

The results reported in Table 4 below have been obtained by measurements performed in 2003. The following quantities are considered: $t_{bi,k,av}$ – the hourly average temperature of the

bitumen at hour k; $t_{a,k,av}$ - the ambient hourly average temperature at hour k; $G_{k,av}$ – average hourly irradiance; $G_{d,av}$ - average daily irradiance; p – specific power; η – efficiency.
The experimental results show that the bitumen may be heated by solar means up to a temperature of 50-65°C. The thermal field in the bitumen mass is influenced by the solar radiation, the geometry of the installation and the ambient. The achieved efficiency of the laboratory installation for bitumen heating has been between 8.1 and 9.1%. The results have been favourable enough to suggest trying industrial applications.

Hourly interval $\Delta\tau$[h]	08-09	09-10	10-11	11-12	12-13	13-14	14-15	15-16	16-17	17-18	18-19
$t_{bi,k,av}$ [°C]	21,0	24,5	30,0	36,0	42,5	47,5	50,5	54,0	56,5	56,5	53,5
$t_{a,k,av}$ [°C]	18,5	19,5	21,0	24,5	26,5	28,5	30,0	32,5	32,5	31,0	28,5
$G_{k,av}$ [W/m²]	344	438	760	863	978	960	747	684	431	386	297
p [W/m²]	65,6										
$G_{d,av}$ [W/m²]	721										
η [%]	9,1										

Table 4. Hourly values of the quantities $t_{bi,k,AV}$, $t_{a,k,AV}$, $G_{k,AV}$, $I_{d,AV}$, p and η.

7.2 Industrial thermal solar system for bitumen preheating

The diagram of the solar system for bitumen preheating that has been realized at Săcălaz, near Timişoara, in cooperation with the Roads and Highways Regional Direction is presented in Fig. 19 (Luminosu et al., 2007b).
The solar installation has been placed on an existing construction. The elements in Fig. 19 are: solar collector, with a surface of 300 m²; roof made of iron plates – 2; pipes penetrating the bitumen – 3; compartment with bitumen preheated at 90-100°C - 4; heat exchanger with oil – 5; tank for bitumen heating at 100-150°C - 6; metallic meshes distanced by 0.5 m (mounted in order to homogenize the temperature in the solar trap) – 7; thermometers – 8; fire place – 9. An iron plate, having a thickness of 0.75 mm is placed between the glass plate

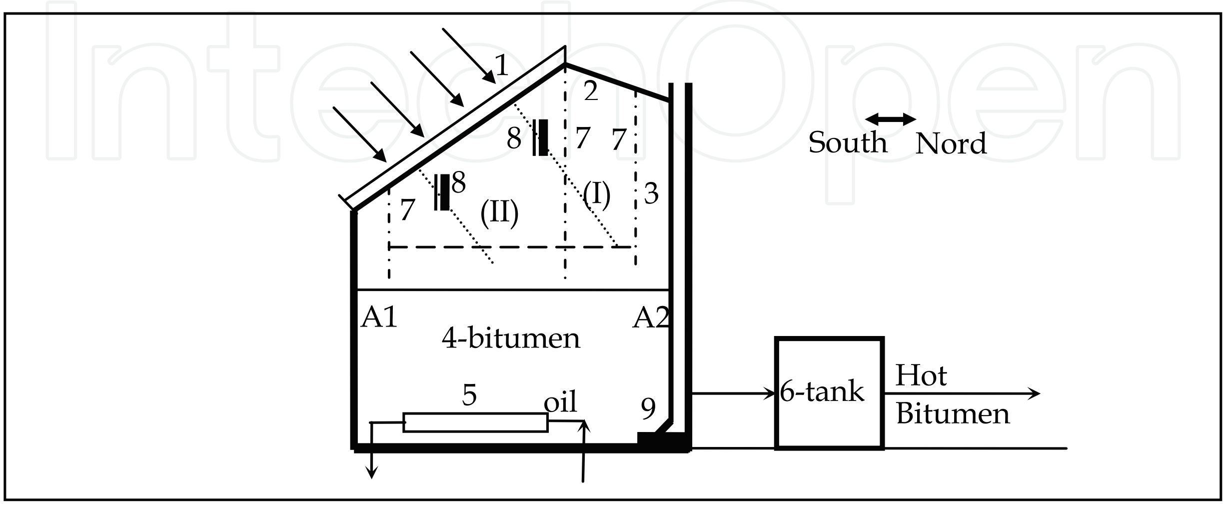

Fig. 19. Diagram of the industrial installation for bitumen preheating.

and the surface of the bitumen. The solar installation accomplishes the bitumen heating up to 50-55^{o}C, with the favourable consequence of saving conventional fuel.

Financing conditions allowed only for preliminary measurements. The temperature has been measured in the volume in between the surface of the bitumen and the roof (the solar trap). We present as examples, in Table 5, the hourly averages of the temperatures of the bitumen t_{Bi} and ambient t_a.

The maximum average temperature of the bitumen, 54-56^{o}C has been obtained around 14h30. In the daytime when measurements have been performed, the maximum average temperature in the solar trap has been larger than the ambient temperature by 27^{o}C. It has been evaluated a saving of approximately 80 kg conventional fuel for 1 m^2 collecting surface per year. A further saving of fuel is obtained if the bitumen extraction is made around 4-5 pm from the upper portion of the tank.

Hour	9h30min	10h30min	12h30min	14h30min	16h30min	18h30min
$\langle t_a \rangle$ [^{o}C]	27,5	28,5	34,0	35,0	33,5	31,0
$\langle t_{Bi} \rangle$ [^{o}C]	38.0	47,5	55,5	56,5	55,0	52,5

Table 5. Average temperatures in the solar trap.

8. Conclusion

Research in solar energy has been approached at the "Politehnica" University of Timişoara in 1976, motivated by economical and ecological problems related to classical fuels.

Solar collectors have been conceived and realized and several thermal solar installations for producing hot air and water have been devised and applied in industry. Solar energy technology has also been applied to waste water cleaning and to building climatization. A part of this experience has been presented in this chapter. The installations have been realized and tested in Timişoara, Romania. The obtained results are relevant for the south-eastern part of Europe.

The experimentally determined efficiencies of the solar installations have been comparable with efficiencies of similar installations produced in other European countries. This proves the possibility of implementing solar energy applications in the region based on the local industry and on locally devised solutions. However, a further involvement of the local industry in the field of solar energy in particular and of renewable energy in general, as well as the education of the population in this spirit are actions to be considered in the near future.

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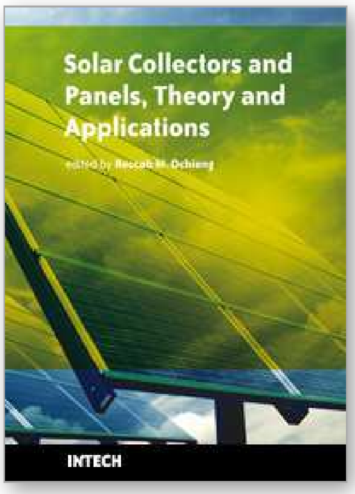

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