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THERMAL PERFORMANCE OF TRAPEZOIDAL-SHAPED SOLAR COLLECTOR/ENERGY STORE

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Abstract. *A simple, low-cost solar water heater has been developed for operation in Mediterranean Europe or regions of similar latitude (40º–45º north). It takes the form of a trapezoidal-shaped water store in direct contact with an inclined flat-plate solar collector assembly. This cross-section induces thermal stratification in the water store, and provides sufficient energy storage to meet typical daily hot water demand. Its thermal performance is critically dependent on the waterside convective heat transfer coefficient on the backward-reclining collector plate; previously evaluated by Cruz et al, (1999). In the current design, the absorber plate inclination to the horizontal was fixed at 45º (close to the local latitude) in order to yield maximum solar gain over a typical year. The energy saving provided by the solar collector/thermal store demonstrator largely depends on the amount of thermal stratification within the trapezoidal storage cavity. This was evaluated via both computations and measurements of the temperature field. A thermal network analysis model was then used to assess the energy saving potential of the composite system. It indicated that a 30 to 70% reduction could be obtained in contrast to direct heating; where the smaller saving occurred at times of greatest use or hot-water take-off.*

Keywords. *Solar Energy, Solar Collector/thermal Store, Buoyancy-driven Convection*

1. Introduction

In the aftermath of the 1992 Earth Summit in Rio de Janeiro (the UN Conference on Environment and Development) there has been a greater awareness of the need to devise strategies aimed at sustainable development (Hammond, 1997 and 2000): balancing economic and social development with environmental protection. The World Commission on Environment and Development (1987) in the influential Brundtland Report defined sustainable development as meeting "the needs of the present without compromising the ability of future generations to meet their own needs." Hence, governments have been encouraged to conserve depleting fuel resources, and to make greater use of renewable energy sources. Solar energy, directly or indirectly, is the major source of renewable energy available to humankind. However, the utilisation of solar thermal energy is inhibited by a number of barriers to market penetration. These include the high initial (or capital) cost of stand-alone systems, compared with grid-connected ones, and the need for an integrated means of thermal energy storage to even out diurnal and seasonal variations in solar gain.

The main requirement for a compact solar water heating system is that the absorbing surface is in direct contact with a large mass of water. The 'black' collector surface receives heat via the absorption of the solar energy, and consequently the mass of fluid (water) in contact with it is heated. The most important features of such systems (Abhat, 1980) include having a large storage capacity per unit volume (their compactness), a suitable operating temperature range, high storage efficiency, long life, and inexpensiveness. Several theoretical and experimental investigations have been reported on different designs for compact solar water heater (see, for example, Garg and Rani, 1982; Goetzberger and Rommel, 1987; Sodha et al., 1981; Sokolov and Vaxman, 1983; Tanishita, 1970; and Vaxman and Sokolov, 1985).

1.1 The Problem Considered

A simple, low-cost solar water heater has been proposed for operation in the Portuguese climate (Cruz et al, 1999). It takes the form of a trapezoidal-shaped water store in direct contact with an inclined flat-plate solar collector assembly (illustrated in Fig. 1). This geometry was suggested by earlier research on buoyancy-driven convection in complex-shaped, air-filled cavities by the last author (see Reis and Smith, 1987 and 1993). The trapezoidal cross-section induces thermal stratification in the water store, and provides sufficient energy storage to meet typical daily hot water demand. This design has similarities to that of Sokolov and Vaxman (1983), except that they employed a full-height collector and a water-side baffle plate adjacent to the absorber to direct the natural water circulation. Such simple geometries yield 'passive' solar devices with a low capital cost when compared to conventional 'active' systems. In the current design, the inclination angle of the absorber plate to the horizontal was fixed at 45^o, close to the local latitude (~ 41.5^o) as recommended by Duffie and Beckman (1974) and Sibson (1985). This inclination can be shown to yield the maximum solar gain averaged over a typical year in Famalicão (Portugal), and indeed for much of Mediterranean Europe.

2. Experimental Assembly

2.1 Design Requirements

The objective of the experimental programme was to design and build a versatile installation for simulating the proposed simple, low-cost and compact solar water heating system. This system would be less efficient than a conventional one, but should benefit from being self-operating and have a much lower running cost and initial investment. The present experimental assembly was therefore devised with aim of testing the trapezoidal-shaped solar/thermal store (Fig. 1) when located in Northern Portugal; principally around the towns of Oporto and Famalicão, where the latitude is 41.5^o north. Solar energy engineering practice [Duffie and Beckman (1974) and Sibson (1985)] suggests that the best inclination for a fixed flat-plate solar collector over the whole year is that at the latitude where the solar system will be installed. Consequently the tilt angle for the test rig heater plate (which represents the flat-plate solar collector) was fixed at 45^o; a value that could be practicably obtained within normal manufacturing tolerances. It can readily be shown (Moita, 1988) that the average annual incident solar radiation on the absorber plate is essentially constant over the range of tilt angles $25^o<\phi<50^o$ at this location.

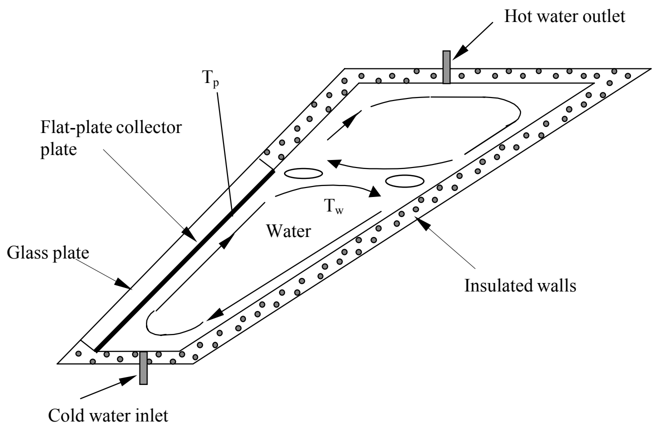

Figure 1. Cross-sectional representation of the trapezoidal-shaped solar collector/thermal store

The full-scale test rig was devised so that the inclination angle of the hot plate, as well as some of the other geometric parameters, could be varied. However, the results reported here (as indicated above) are for one geometry with an optimal tilt angle appropriate to the Portuguese climate: ϕ=45°, D=0.125 m, H'=0.5 m, H"=0.25 m, and U=0.375 m (see Fig. 2).

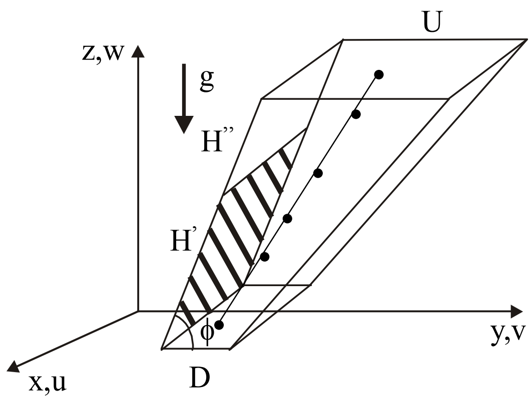

Figure 2. Geometric parameters (solid circles denote fluid temperature measurement locations)

2.2 The Experimental Test Facility

The casing of the test rig (shown schematically in Fig. 1) was fabricated from 11 mm thick steel, with the guarded hot plate made of aluminium (500x500x10 mm) in contact with the water. The side-walls were constructed from 5 mm thick steel, and external thermal insulation was provided by 40 mm thick rockwool. The water was in contact with 6 mm thick 'bakelite' plates that provided the internal lining. The volume of the cavity for a hot plate tilt angle of 45º was about 66.3 litres. An important factor in the construction of the experimental installation was the sealing of the joints, and this was achieved using a transparent silicon gel mastic.

In order to measure the wall temperature at the internal surfaces, as well as the heat flux through the aluminium plate, an array of 56 thermocouples were connected to a data acquisition system. The main heater was connected to a variable transformer used to control the power input.

3. Heat Transfer Mechanisms in the Passive Solar Heater

A partial schematic view of the cross-section of the proposed combined flat-plate collector and water store with one transparent (glass) cover is illustrated in Fig. 3. The solar radiation is incident on the cover glass, which is mostly transmitted with only a small fraction absorbed and partly reflected. About 90% of the energy transmitted through the cover glass is found to be absorbed by the absorber plate. The temperature of this plate consequently increases, and heat is then transferred to either the water store or the external environment. On the water-side of the absorber plate, the heat is transferred by natural (strictly 'buoyancy-driven') convection to the water. The conduction through the insulation on the sides of the plate is negligible. The air space between the absorber plate and the cover glass is sealed, and buoyancy-driven convection occurs between the two surfaces. The glass cover reaches a temperature between that of the absorber plate and the ambient. This cover is essentially opaque to the long-wave radiation from the absorber plate. A 'greenhouse effect' enables the temperatures achieved to be significantly above those obtained in a flat-plate collector without a cover. The external convection heat transfer depends on the wind velocity, and significant cooling effects can be experienced in high-wind conditions. In contrast, the exterior long-wave radiation is governed only by the effective 'sky temperature'. The highest temperatures that can be achieved in a flat-plate collector depend on a successful reduction of the heat loss. This (in turn) will depend on adequate side insulation of the absorber, and the suppression of buoyancy-driven convection between the absorber plate and the cover glass.

3.1 Theoretical Model of the Passive Solar Water Heater

Any solar collector is required to absorb solar radiation, and transfer the absorbed heat to the fluid with minimum heat loss. In assessing the performance of a collector/water store, it is therefore important to determine its ability to absorb solar radiation and estimate the heat losses. The collector efficiency represents the fraction of incident energy on the outer surface of the cover that is actually stored by the fluid in the store. If the heat losses increase, then the efficiency will fall. It is consequently necessary to understand the heat transfer mechanisms that arise in these solar devices (conduction, convection and radiation) in order to be able to evaluate their performance.

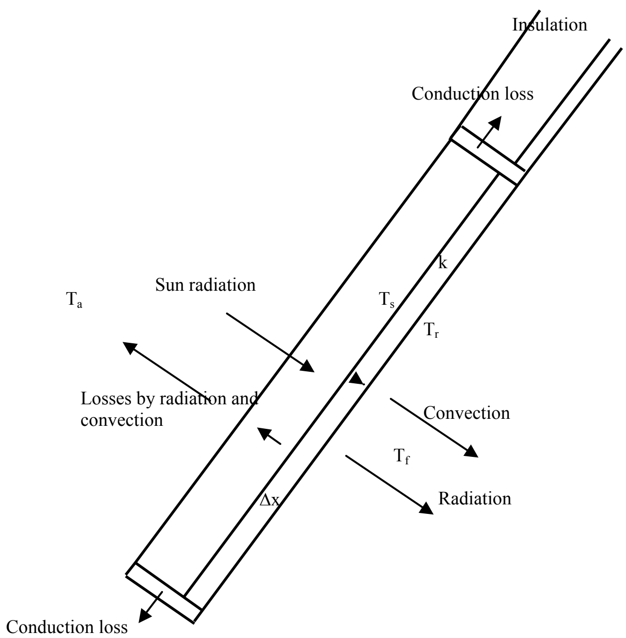

Figure 3. Absorber plate cross-diagram (adapted from Luiz, 1985)

Fig. 3 shows the proposed absorber plate of the solar water heater, where Δx is the thickness of the absorber plate, T_s and T_r are the temperatures on both sides of the absorber plate, T_f is the temperature of the fluid, and T_a is the air ambient temperature. The equivalent heat flow, or thermal network analysis, circuit for the collector, insulation and losses is shown in the Fig. 4. It is possible to evaluate the steady-state loss from the sides and the cover glass assuming fully realistic heat transfer processes. However, to simplify the idealised (theoretical) representation of the solar water heater, the walls of the store were considered adiabatic. In addition, it is usual with solar collectors is to assume that T_s and T_r are the same, and noted as T_p. The heat transfer to the water as mainly a convective process.

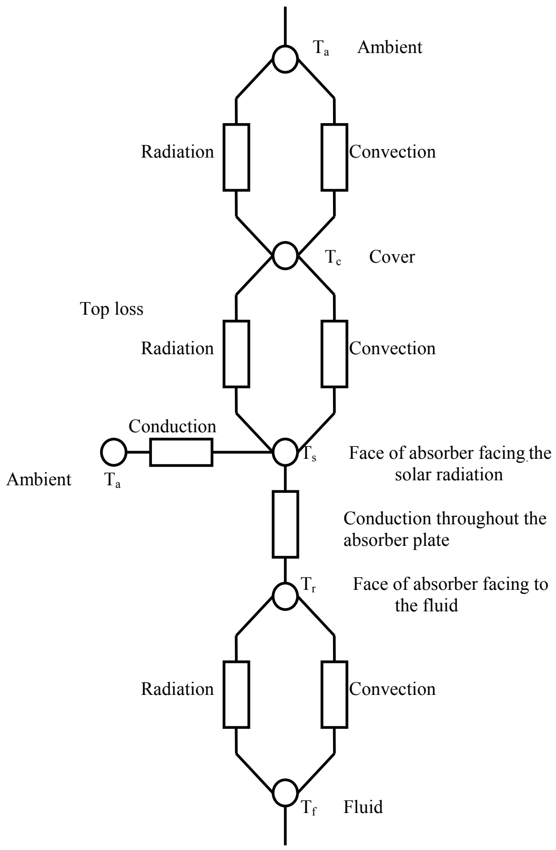

Figure 4. Equivalent heat flow circuit for the collector, insulation and losses (Adapted from Luiz, 1985)

The energy balance on the absorber plate at a given instant, assuming steady-state conditions, is given by:

$$q_{useful} = q_{absorbed} - q_{lost\,plate\,to\,cover} = q_{absorbed} - q_{lost\,cover\,to\,ambient} \tag{1}$$

where q_{useful} is the useful energy, $q_{absorbed}$ is the energy collected by the absorber, $q_{lost\ plate\ to\ cover}$ is the energy lost from the absorber plate to the cover, and $q_{lost\ cover\ to\ ambient}$ is the energy lost from the cover to the ambient. These can be expressed by:

$$\begin{cases} q_{loss\,plate} = h_{pc}\left(T_p - T_c\right) + \dfrac{1}{\dfrac{1}{\varepsilon_p} + \dfrac{1}{\varepsilon_c} - 1}\sigma\left(T_p^4 - T_c^4\right) \\ q_{loss\,cover} = h_{ca}\left(T_c - T_a\right) + \varepsilon_c\sigma\left(T_c^4 - T_{sky}^4\right) \end{cases} \tag{2}$$

where ε_p is the emissivity of the absorber plate, ε_c is the emissivity of the cover glass, T_p is the temperature of the absorber plate, T_c is the temperature of the cover (glass), and T_a is the ambient air temperature. The second term on the right hand side is the heat loss by long-wave radiation between the absorber plate and the cover, while the long-wave radiation loss from the cover glass to the surrounding sky is represented by the fourth term. Duffie and Beckman (1980) suggest that the following convection correlations are appropriate for solar collectors:

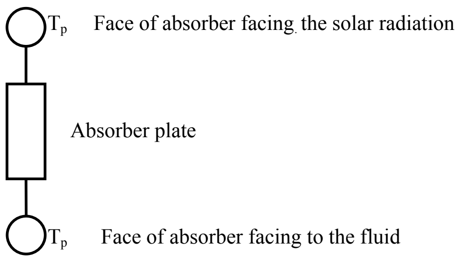

Figure 5. Equivalent heat flow circuit in absorber plate. The temperature T_p is the same in both sides (Adapted from Luiz, 1985)

$$h_{pc} = \frac{\left[\left(0.06 - 0.0017\frac{\phi}{90}\right)Gr^{1/3}\right]k}{L} \tag{3}$$

where ϕ is defined in degrees, and

$$h_{ca} = \frac{\left[0.825\left(Gr\cos\phi\, Pr\right)^{\frac{1}{4}}\left(1+\left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right)^{-\frac{8}{27}}\right]^{2}k}{L} \tag{4}$$

The convection coefficient h_{pc}, applies to buoyancy-driven convection between inclined parallel plates with a slope ϕ, while h_{ca} relates to external convection from the cover glass to the ambient. Here L is the characteristic length (height of the absorber plate) and k is the thermal conductivity of the air at the mean temperature. For windy conditions (forced convection) the convection coefficient is given by h_{ca}=5.7+3.8V, where V is the wind velocity (m s^{-1}) (Duffie et al., 1980, Incropera et al., 1985).

The incident solar radiation transmitted through the cover glass to the absorber plate is denoted by $G_n\left(\overline{\tau\alpha}\right)$, W m^{-2}. Convection and radiation occur from the exposed plate to the water (on the back surface of the absorber plate). If steady-state conditions are assumed, then the heat transfer through the absorber plate can be written:

$$\begin{cases} q_{useful} = G_n\left(\overline{\tau\alpha}\right) - h_{pc}\left(T_p - T_c\right) - \dfrac{1}{\dfrac{1}{\varepsilon_p} + \dfrac{1}{\varepsilon_c} - 1}\sigma\left(T_p^4 - T_c^4\right) \\ q_{useful} = G_n\left(\overline{\tau\alpha}\right) - h_{ca}\left(T_c - T_a\right) - \varepsilon_c\sigma\left(T_c^4 - T_{sky}^4\right) \\ \\ h_{pc}\left(T_p - T_c\right) - \dfrac{1}{\dfrac{1}{\varepsilon_p} + \dfrac{1}{\varepsilon_c} - 1}\sigma\left(T_p^4 - T_c^4\right) = h_{ca}\left(T_c - T_a\right) - \varepsilon_c\sigma\left(T_c^4 - T_{sky}^4\right) \end{cases} \tag{5}$$

where the useful (stored) energy is evaluated from the expression:

$$q_{useful} = h_{exp}\left(T_p - T_{film}\right) \tag{6}$$

Here h_{exp} is the buoyancy-driven convection heat transfer coefficient evaluated experimentally and numerically (Cruz, 1997; Cruz et al., 1999). Both equations [5] and [6] can be solved simultaneously to yield the temperature of the plate T_p, and ultimately the useful energy q_u.

Finally, the thermal energy storage efficiency over a period of time (t) is given by

$$\bar{\eta} = \frac{\bar{q}_{useful}}{G_n} \tag{7}$$

where $\bar{q}_{useful}$ is defined by

$$\bar{q}_{useful} = \sum_{i=0}^{t} mc_p \left(T_{final} - T_{initial}\right)_{water} \tag{8}$$

The full-scale experimental tests (Cruz, 1997) were carried out with a heat flux of 600 Wm^{-2}. Consequently the heat input to the collector in the network analysis was specified as $G_n\left(\overline{\tau\alpha}\right) = 600\ W\ m^{-2}$, so that both the analytical and experimental situations could be compared on an equivalent basis. The absorber convection coefficients were then estimated (using equations [3] and [4]) to be h_{pc}=1.29 W m^{-2} K^{-1} and h_{ca}= 5.74 W m^{-2} K^{-1}. The tilt angle of the plate to the horizontal was fixed at 45º, and the computations were then performed with the sky temperature given by $T_{sky}=T_{ambient}-6$ (Duffie et al., 1980). Table 1 and 2 summarises some of the results obtained. After 8 hours, the theoretical heat loss by convection from the plate was 86,660 J and by radiation 313,328 J. The corresponding values for the actual heat loss by convection and by radiation were 85,896 J and 313,135 J respectively. The temperature obtained for the plate was T_p=32.7 ºC, and that for the cover was found to be T_c=23.1 ºC. The buoyancy-driven convection coefficient was experimentally evaluated and found to vary from 101.2 to 147.1 W m^{-2} K^{-1} (Cruz, 1997). In Fig. 3 the collector parameters obtained using the network model are depicted, and in Fig. 7 the variation of the useful energy stored over 8 hours is plotted. The experimental average thermal efficiency η_{exp} can be obtained via:

$$\bar{\eta}_{exp} = \frac{m_1 c_p\left(T_1 - T_a\right) + m_2 c_p\left(T_2 - T_a\right) + m_3 c_p\left(T_3 - T_a\right)}{G_n} \tag{9}$$

which is based on the stratification model illustrated schematically in Fig. 8. For the above conditions, the measured efficiency was found to be 48%. In contrast, the theoretical average thermal efficiency $\eta_{theoretical}$, given by equations [7] and [8], was 51%; a difference of only 3%. This is clearly an acceptable error in engineering terms, given that in the experimental situation the losses from the walls were found to be around 5% (Cruz, 1997). Thus, the thermal network analysis model is shown to be quite adequate for describing the thermal behaviour of the solar water heater developed here. It provides an accurate representation of the solar collar/thermal store under realistic solar gain and weather conditions appropriate to the climate in Portugal.

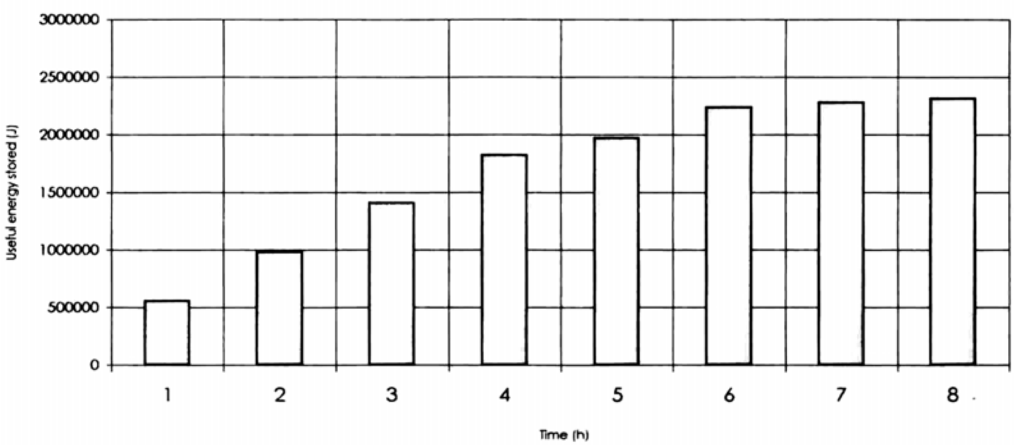

Figure 6. Theoretical useful energy stored during 8 hours

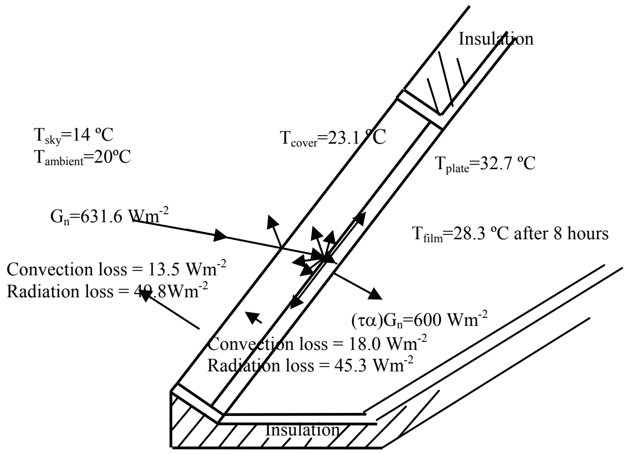

Figure 7. Values obtained with the theoretical model

	Temperature	Units
$T_{air\ ambient}$	20.0	ºC
Initial T_{water}	20.0	ºC
Final $T_{water\ (after\ 8\ hours)}$	28.3	ºC
T_{sky}	14.0	ºC
T_{plate}	32.7	ºC
T_{cover}	23.1	ºC
Average $\eta_{theoretical}$	51%	
Average η_{exp}	48%	
Error(%)	3%	

Table 1. Summary of the validation of the theoretical model (during 8 hours)

Time (h)	**T_{plate} (ºC)**	**T_{film} (ºC)**
0	25.9	20,0
1	26.5	22.0
2	27.3	23.5
3	28.2	25.1
4	29.4	26.5
5	30.6	27.1
6	31.7	28.0
7	32.5	28.2
8	32.7	28,3

Table 2. Temperatures of the plate and water given by the theoretical model (during 8 hours)

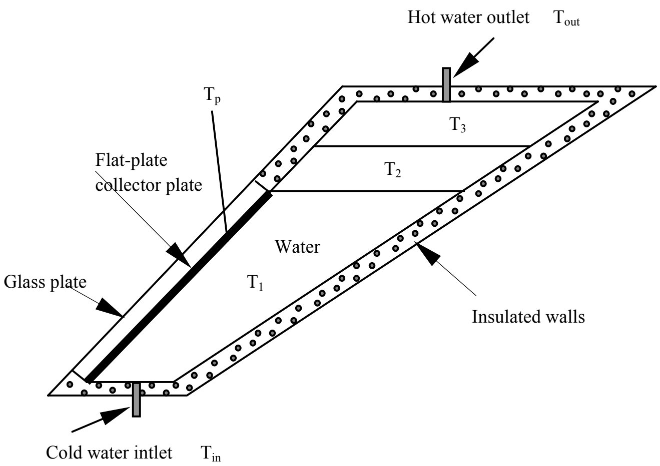

Figure 8. Schematic cross-section with notional stratification zones

4. Experimental Measurements

4.1 Temperature Variation, Energy Saving and Thermal Stratification

The stored energy in the tank was evaluated using the simple relation:

$$q_{stored} = mc_p (T_{out} - T_{in})_{water} \tag{10}$$

whilst the hot water demand (draw off) was estimated from the UK-ISES demand profile (see Table 3). The energy saving can then be evaluated using following equation

$$\eta_{saving} = \frac{T_{out} - T_{in}}{T_{set} - T_{in}} \tag{11}$$

For the situation depicted in Fig. 9, the energy saving is represented by the shaded area in Fig. 13. The corresponding percentage saving is plotted in the Fig. 14, where the maximum value of about 70% occurs in the morning. It falls to a minimum of 30% in the evening. (It is important to note that the draw-off of hot water was assumed to start after a one day heating period without any water draw-off.)

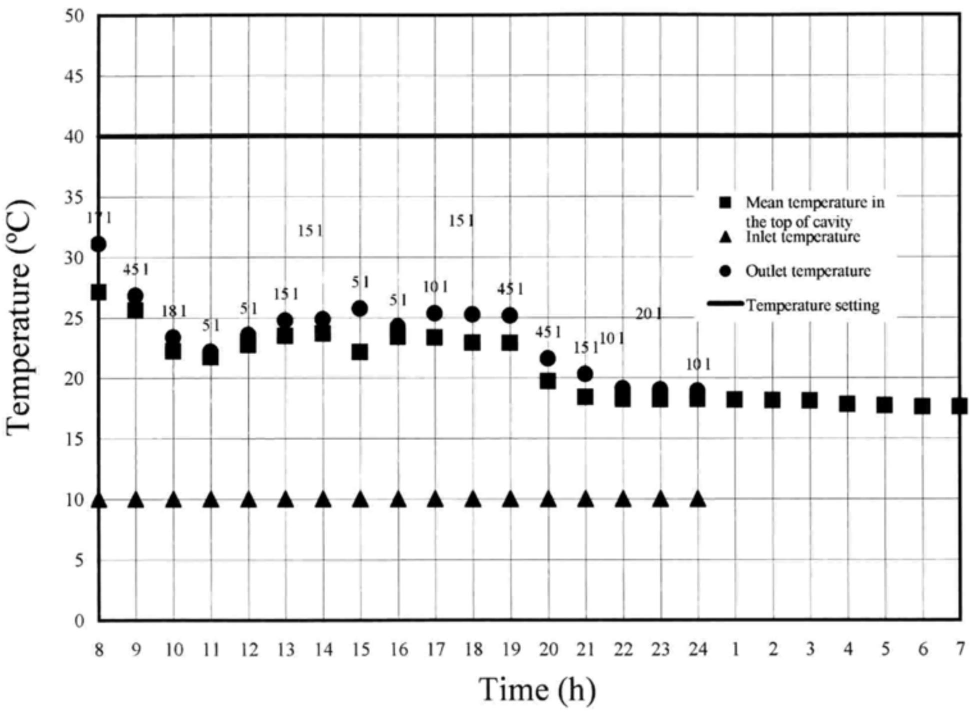

Figure 9. Hot water draw-off temperature-time profile

The temperature profiles inside the cavity of the experimental water store are shown in Fig. 9. Here the measurements were obtained for one day; starting from a temperature of 31.1 ºC. Every hour during the day some quantity of water was drawn off following the pattern suggested by Table 3. When water is drawn off, the equivalent quantity of cold water enters the store at the bottom, and consequently the temperature decreases. It obviously takes time for water temperature to recover after such a draw-off. Analysing Fig. 10 and 11, it is possible to see that the temperature decreases between 8:00 a.m. and 10:00 a.m. Later in the morning (from 10:00 a.m. to 1:00 p.m.), the temperature increases as a consequence of smaller quantities of hot water being drawn off, and eventually decreases again after 7:00 p.m.

Pattern of water usage			
hour	litres	hour	litres
7	0	16	5
8	17	17	10
9	45	18	15
10	18	19	45
11	5	20	45
12	5	21	15
13	15	22	10
14	15	23	20
15	5	24	10

Table 3. Daily hot-water usage

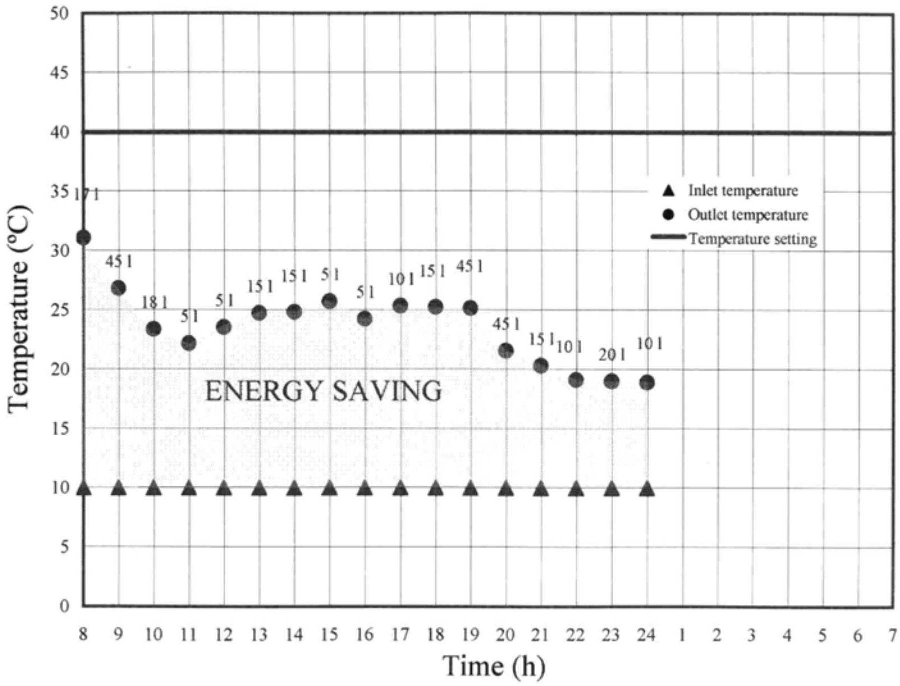

Figure 10. Energy saving The shaded area represents the percentage of the energy saved to the water at 40 ºC during a given day.

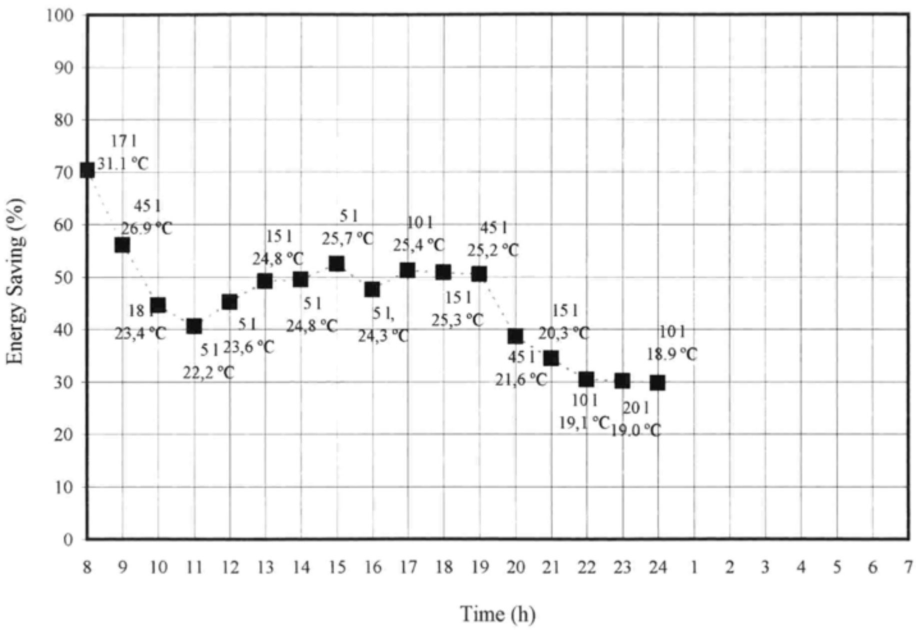

Figure 11. Energy saving during a day for a set temperature of 40º C. During the night no water was drawn off.

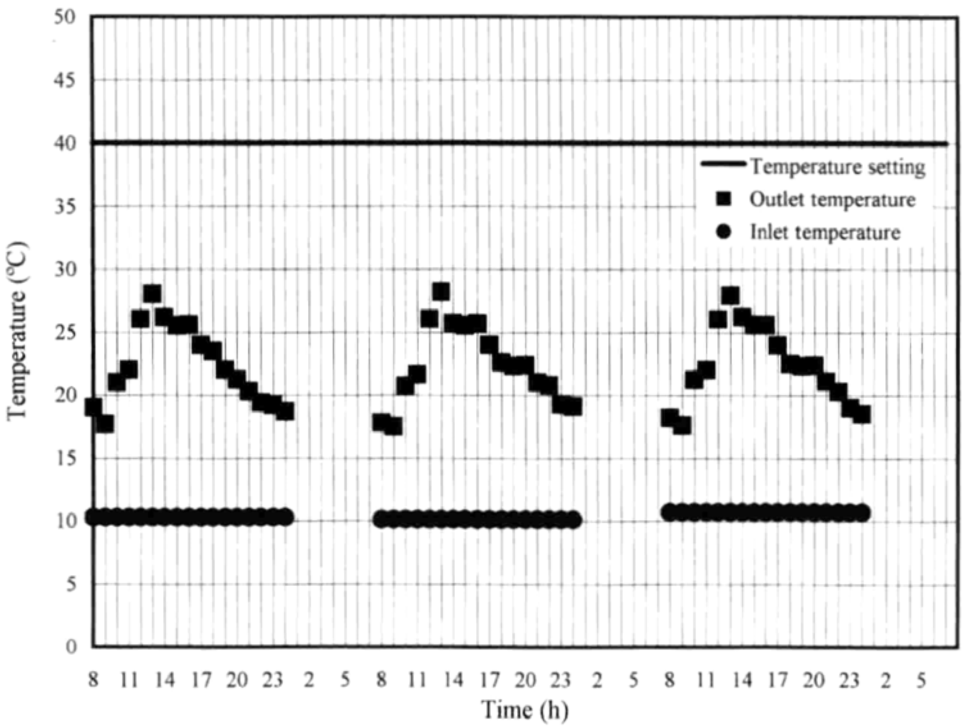

Figure 12. Three days of draw off hot water and heating

In Fig. 12 and 13 the results of 3-day simulations are presented. The temperature decreases in early morning, and then increases until 1:00 p.m.. During the rest of the day the temperature decreases as a consequence of the water demand increasing from 3:00 p.m. to 8:00 p.m. The temperature falls smoothly during the night in the absence of water draw-off. It can be seen that the energy saving is around 60%. This implies that only about 40% of energy is needed to achieve the desired water temperature; sufficient, for example, to take a shower.

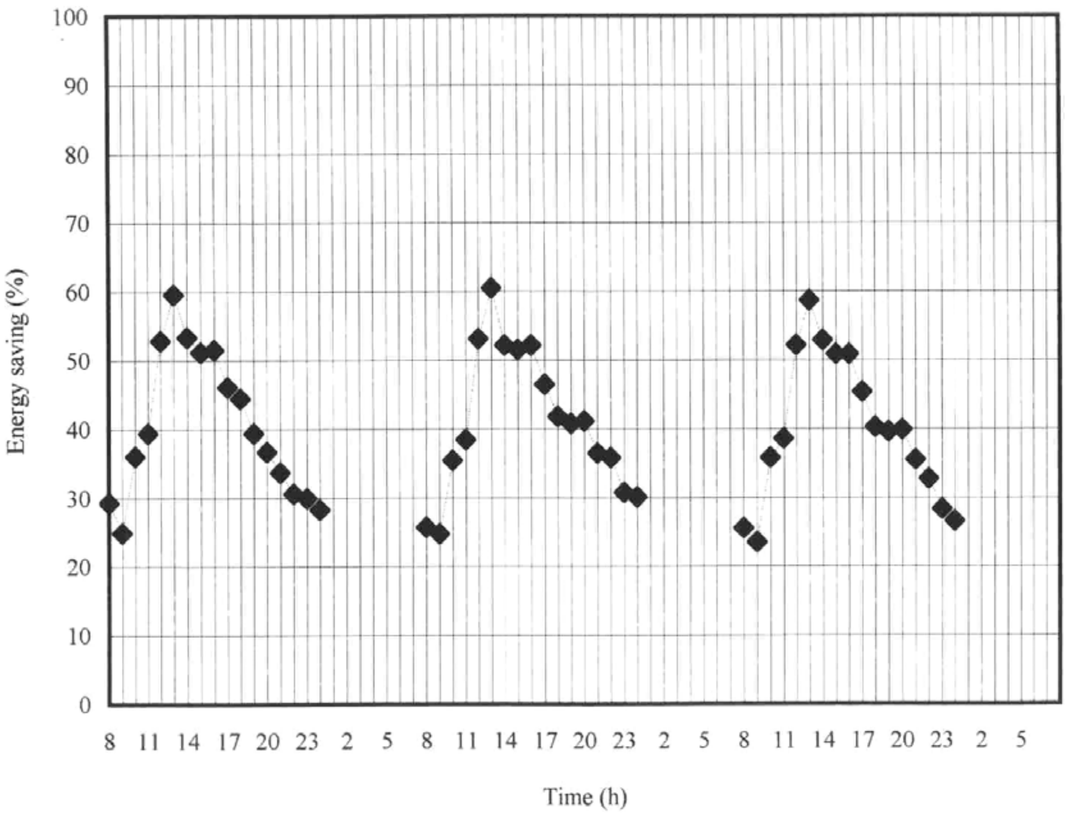

Figure 13. Energy saving for a run of three days

Finally, the thermal stratification in the store is illustrated in Fig. 14, where the top of the cavity/store is warmer during operation than the bottom. The profile represents the sort of behaviour required for an effective thermal storage device.

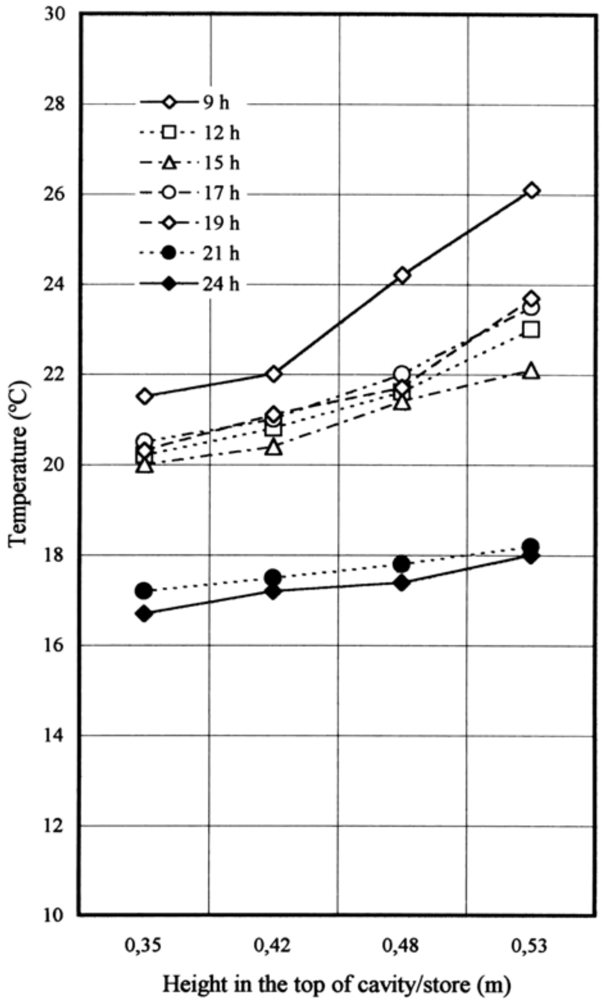

Figure 14. Stratification in the top of cavity/store for a 24 hours cycle (see figure 9)

5. Concluding Remarks

The objective of this work was the design, construction, and testing of a simple, low-cost, passive water heater; principally for the climate in Portugal or Mediterranean Europe. It took the form of a trapezoidal-shaped water store in direct contact with an inclined flat-plate solar collector assembly (Fig. 1). An important requirement for the solar collector/thermal store demonstrator was that hot water energy savings could be obtained during typical domestic use. The amount of this energy saving largely depends on thermal stratification within the storage cavity. The objective was to obtain heated water as cheaply as possible. In Portugal solar radiation is abundant throughout the year, and this makes the use of a passive solar water heaters feasible.

The performance of the novel solar collector/thermal store demonstrator was simulated using thermal network analysis. In addition, an experimental test rig was employed to validate the modelling. This consisted of a versatile facility, which had the capability to alter the tilt angle of the heated surface to the horizontal. However, a constant tilt angle of 45º was actually employed in the present study, as this is near optimum for the Portuguese climate. An elaborate array of thermocouples was connected to a data acquisition system to collect and process all the temperature information.

The experimental installation was also used to test the operation of the full-scale passive solar water heater/thermal store. The heater plate was used to simulate the heating cycle during the day, and the cut off at night. Hot water demand was simulated using the British Standard domestic heating load profile (in the absence of one for Portugal). The maximum temperature under these conditions was found to be 31ºC for a power input of 150 W. Measured energy savings varied from 70% to 30%, the latter at times of most intense use. It is important to note that the storage tank had a capacity of only around 66 litres, and that domestic hot water demand in Portugal will sometimes exceed this. Nevertheless, the feasibility of a simple, low-cost solar water heater has been demonstrated.

6. Acknowledgments

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