Performance Evaluation of Copper-Water Thermosyphons in Glazed Box Solar Water Heater

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Abstract

A thermosyphon is a vacuumed metal tube or pipe, partially filled with a liquid as working fluid and sealed and exhibits a high effective conductance. When used in a solar collector, its performance may be expressed in terms of the heat it transfers to the tank water at its condenser with the heat it absorbs from the sun at its evaporator.

In this study a set of fifty-two thermosyphons, made from 19.05-mm-diameter copper tubes with distilled water as the working fluid, were tested for performance as to how well these thermosyphons effectively transfer the heat absorbed from the sun to the tank water. The insolation, wind speed and the temperatures of the various key components of the solar water heater were recorded by a data logger every minute for the whole twenty-four hour period. Heat gains and losses by the components were calculated in terms of the increase or decrease of temperatures.

The results show that the thermosyphons are effective in transferring the heat it absorbs to the tank water. However, the results also show that when the condenser section of the thermosyphons were still at a higher temperature to that of the evaporator section, heat flowed back thereby losing the heat stored by the tank water.

Keywords: thermosyphons, solar collector, water heater, solar energy

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Introduction

With the increasing demand for energy, researches on alternative sources of energy are encouraged to address the growing demand for energy. Heat from the sun is now considered in many heating requirements, such as comfort heating of houses in some cold countries, or heating water for hot beverages, for laundry, or in showers. With the sun giving the earth as much as 1353 Watts of energy per square meter [NASA (1971) as cited in Duffie and Beckman (1991)], one can assess how abundant solar energy is.

One problem with solar energy, however, is the efficiency in harvesting it. The energy from the sun, before reaching the collector, passes through the atmosphere enclosing the earth. The atmosphere may have clouds, depending on the weather, that can greatly decrease the amount of solar rays reaching the collector. Further, energy from the solar collector may not be transferred to the medium efficiently, so solar water heater developers are considering utilizing technologies that enhance the transfer of heat.

In this research, a solar water heater was fabricated utilizing copper thermosyphons as heat transfer mechanism and its performance was evaluated. It made use of water in copper thermosyphons as solar energy collecting elements in a flat plate solar collector utilizing clear glass sheets as glazing. So far there are no performance data available regarding the use of copper-water thermosyphons in solar collectors with double sheet glass glazing. Therefore, the performance data produced in this study will be directly or indirectly beneficial to engineers and designers should they make use of the same technology in other applications, or perhaps to further develop a more efficient solar water heater.

This investigation focused on the evaluation of the performance of copper-water thermosyphons which were arranged close enough to each other that they approximated a flat plate collector and were inclined at 45 degrees in a solar water heater. The investigation involved computation of solar energy, heat gains and heat losses, and efficiency. Because of the complexity in computing wind heat transfer coefficient, the wind direction was not taken as a factor in the convective heat transfer calculation.

Further, the expected corrosion of the metal tank was assumed to have no effect on the heat transfer coefficients.

Objectives of the Study

This study aimed to fabricate a set of copper-water type thermosyphons and a glazed box solar water heater where the thermosyphons were later installed as heat transfer mechanism and to evaluate their performance based on the local weather conditions.

Specifically, it tried to answer the following:

1. How efficient in transferring heat will the copper-water thermosyphons be when installed in solar application?
2. How do the heat transfer properties of thermosyphons behave where there is no sun or at night time?

Review Of Literature

Flat-Plate Collectors

According to Athey (1976) the simplest method of utilizing the energy of the sun to generate electric power is to use a flat-plate collector system. He designed his evaluation of a flat-plate collector system to determine the number of flat-plate collectors required to generate a given amount of electricity with optimum efficiency. Variable parameters considered were the temperature of the heat transport fluid, both to and from the collector field. In the analysis, the efficiency of the flat-plate collectors was coupled with the efficiency of the thermal cycle to calculate optimal overall system efficiencies. Overall system efficiencies for the system were on the order of 35 per cent or less. Over two million 4 ft by 4 ft collectors would be required to produce 100,000 kW(e).

Based on the results of this analysis, it can be shown that the limiting factor in the use of the flat-plate collector system for electric power generation is the efficiency of the collectors. An increase in the overall system efficiency can occur only if the collector efficiency can be increased at the higher surface temperatures (Athey, 1976).

Thermosyphon in Solar Collectors

A simplified simulation method for solar thermosyphon collectors was developed by Huang and Hsieh (1985). They utilized the Hottel-Bliss-Whillier theory as absorber, Close's model for the thermally stratified tank, and a newly defined loop resistance relation. They found that the time step required in numerical computation can be expanded to 15 minutes with good accuracy compared to other methods for which the time step is between 1 and 40s. Verifying experiments in single-day and long-term scales conducted outdoors for different operating conditions, the results showed very good agreement with the simulation.

Jouhara et al. (2008) experimentally investigated the thermosyphon thermal performance with water as well as the dielectric heat transfer liquids FC-84, FC-77 ,and FC-3283 as the working fluids. Their study, however, is only applicable for miniature application. The copper thermosyphon was 200 mm long with an inner diameter of 6 mm. Each thermosyphon was charged with 1.8 ml of working fluid and was tested with an evaporator length of 40 mm and a condenser length of 60 mm. Results showed that the thermal performance of the water charged thermosyphon outperformed the other three working fluids in both the effective thermal resistance as well as maximum heat transport.

The study of Chotivisarut and Kiatsiriroat (2006) involved 19.05-mm-diameter thermosyphons. They investigated the nocturnal cooling potential in a well-insulated room by using thermosyphon as a thermal radiator. The radiator consisting of forty-eight (48) thermosyphon heat pipe tubes each 19.05 mm in diameter rejected heat to the sky for producing cool water in a 1.0 m^3 insulated rectangular tank during nighttime. The cooled water was filled in a set of six heat exchangers each 0.87 m^2 in surface area installed at the room ceiling which absorbed heat and reduced the temperature inside the tested room during the daytime. The experimental site was located in Chiang Mai, Thailand. The thermal load inside the room was performed by an electrical heater. The initial water temperature in the storage tank at 27°C gradually cooled down to 12.1°C within 4 nights. The cooled water was fed into the room when the electrical heater supplied heat at 1000 W. From the experiment, it was found that the cooled water absorbed heat and indoor temperature was decreased around 12.8°C or 21.8 % compared to that without water cooling. And accordingly, energy balance had shown only a 3.34 % error between the calculated value from thermosyphon theory and that of experimental results.

Performance measurement

A thermosyphon is a device with very high thermal conductance. Its performance is often expressed in terms of 'equivalent thermal conductivity' (Reay and Kew, 2006). That is how many times fold the device could be more conductive compared to that of the metal alone.

Conventionally, the overall performance rating of a thermosyphon solar water heater considers the thermal performance of the system during the energy-collecting phase and the system cooling loss during the cooling phase (Chang et al. 2004). Chang *et al.* (2004) suggested that the performance rating should also take into consideration the heat removal efficiency of the system during the system application phase. Their study modified the CNS 12557 B7276 test standard and employed a precise, on-line operation to derive the heat removal efficiency of a system. The thermal performance and heat removal efficiency of 12 systems with capacities in the range of 102–446 L were evaluated. An efficiency coefficient, η_0, was defined, which represented the synthesis of the characteristic thermal performance η_s, and the characteristic heat removal efficiency, η_R. The proposed modified efficiency coefficient was given by $\eta_0 = \eta_s \eta_R$ and represented the quasi overall performance of a solar heating system. The coefficient provided an effective measure of the amount of energy provided to the user from a system which collects and stores heat from solar radiation (Chang *et al.* 2004).

Methodology

Performance

Instead of determining the 'equivalent thermal conductivity' of the thermosyphon when used in a solar water heater as shown in Figure 3.1, the heat transfer to the water with the use of thermosyphons was compared with the heat absorbed by the thermosyphons from the sun.

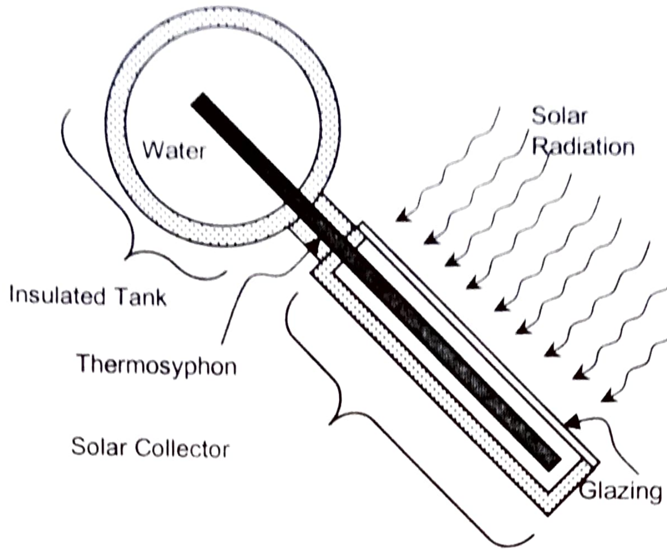


Figure 3.1 Thermosyphons in solar water heater

At a steady state, the useful output of a solar collector, Q_u is the difference between absorbed solar radiation, *S*, and thermal losses (Duffie and Beckman, 1991). That is

$$\dot{Q}_u = A_C\left[S - U_L\left(T_{P,m} - T_a\right)\right] \qquad (3.1)$$

Where A_C = Collector area = 1.0m^2, U_L = over-all loss coefficient (W/m^2K), $T_{P,m}$ = mean thermosyphon temperature (*K*) and T_a = ambient air temperature (K)

At any time *t*, the amount of solar energy reaching the earth surface, is given by (Alagao, 1994) at

$$H = \frac{(12\pi)\overline{H}}{(t_2 - t_1)}\sin\left[\frac{\pi(t - t_1)}{t_2 - t_1}\right] \qquad (3.2)$$

Here t_1 and t_2, sunrise and sunset time, may take the values from online Sun Calculator such as that in www.timeanddate.com. The term $\overline{H}$ is the average daily radiation in W/m^2. The $\overline{H}$ is the average for a 40-year study period, and is equal to 268 W/m^2 for Manila, Philippines (Alagao, 1994). The irradiance due to solar radiation is also called insolation. And, in this study the symbol *I* was used to also mean *H*. However, in this study, to monitor the irradiance due to solar radiation, or insolation, a solar pyranometer was employed.

The Absorbed Solar Energy

The absorbed solar radiation, S, is lesser than that incident to it. The glazing materials reflect a portion of the energy and the portion not reflected are not all transmitted to the black surface of the tubes. When beam radiation, the diffuse sky radiation and ground-reflected radiation are taken as one, the absorbed solar radiation, S, is given by (Duffie and Beckman, 1991) as

$$S = I_b R_b (\tau\alpha)_b + I_d (\tau\alpha)_d \left(\frac{1+\cos\beta}{2} \right) + \rho_g (I_b + I_d)(\tau\alpha)_g \left(\frac{1-\cos\beta}{2} \right) \quad (3.3)$$

Where $(1 + \cos\beta)/2$ and $(1 - \cos\beta)/2$ are view factors from collector to the sky and from collector to the ground respectively, and the subscripts *b*, *d* and *g* represent beam, diffuse and ground.

As suggested by Duffie and Beckman (1991), it is more convenient to define an average transmittance-absorptance product as the ratio of the absorbed solar radiation, *S*, to the incident solar radiation, I_T, at a particular angle of the sun. So the absorbed solar radiation is expressed in terms of the average transmittance-absorptance product, $\tau\alpha_{ave}$, and is given by (Duffie and Beckman, 1991) as

$$S = (\tau\alpha)_{ave} I_T \quad (3.4)$$

The diffuse radiation is low, since the site was elevated and have some greens around, thus taking the $(\tau\alpha)$ of beam radiation as the $(\tau\alpha)_{ave}$ as reasonable.

The transmittance is actually a function of the wavelength of the incident radiation and the angle of incidence. The solar absorptance of ordinary blackened surfaces is also a function of the angle of incidence and wavelength of the radiation. However, the sun, as the radiation source, transmits radiation of wavelengths from 0.3 to approximately 2.5μm (Duffie and Beckman, 1992), and ordinary clear glass is transparent to radiation in the range of 0.3 to approximately 2.5μm. Thus the transmittance is determined mainly based on the incidence angle for the solar time. For a specific type of glass, a range of transmittance values can be extracted from Figure 5.3.1 in the book of Duffie and Beckman (1990) or using equation 5.2.2 by Duffie and Beckman (1991).

$$\tau_a = e^{-KL/\cos\theta} \quad (3.5)$$

Where ***a*** means only the absorption losses considered.
The glazing materials were clear glass sheets, 3/16-inch-thick each. According to Duffie and Beckman (1991), K value varies from approximately 4 m^{-1} for "water white" glasses to approximately 32 m^{-1} for poor glasses. The value taken in this study was 5, not of premium quality yet water white edge. Thus the *KL* value was *KL* = 0.0238.

The evaporator sections of the thermosyphons were spray-coated with flat black lacquer. From Table 8-4 of Heat Transfer by Holman (1997) the absorptivity of flat black lacquer surface is 0.96. Thus, the equation for transmittance is:

$$\tau(\theta) = e^{-0.0238/\cos\theta} \quad (3.6)$$

Using Snell's law, and having the first medium as air, n_1 = 1, and the second as glass, n_2 =1.526, we have:

$$\theta_2 = \sin^{-1}\left(\frac{1}{1.526}\sin\theta_1\right) \quad (3.7)$$

Where the incident angle in this study was computed based on the length of day and in proportion thereof to 180°. That is:

$$\theta_1 = Abs\left(t_i - t_{solar\,noon}\right)\frac{\pi\, rad}{day\, length} \quad (3.8)$$

Where *day length* = t_{sunset} - $t_{sunrise}$, and $t_{sunrise}$ was taken as 0.

For N covers, the transmittance considering only reflection losses, the following equations of Duffie and Beckman (1991) were used:

$$\tau_{rN} = \frac{1}{2}\left[\frac{1-r_\perp}{1+(2N-1)r_\perp} + \frac{1-r_{//}}{1+(2N-1)r_{//}}\right] \quad (3.9)$$

Where

$$r_\perp = \frac{\sin^2(\theta_2-\theta_1)}{\sin^2(\theta_2+\theta_1)} \quad (3.10)$$

$$r_{//} = \frac{\tan^2(\theta_2-\theta_1)}{\tan^2(\theta_2+\theta_1)} \quad (3.11)$$

Then, $\tau \cong \tau_a \tau_r$ (3.12)

The average transmittance-absorptance product, $(\tau\alpha)$, may be reasonably approximated as (Duffie and Beckman, 1991):

$$(\tau\alpha) \approx 1.01\tau\alpha \quad (3.13)$$

Collector Overall Loss Coefficient

The collector overall loss coefficient, U_L, is the sum of the top, bottom, and edge loss coefficients, thus

$$U_L = U_t + U_b + U_e \quad (3.14)$$

The top loss coefficient is due to energy loss through the top cover. The energy loss through the top is the result of convection and radiation. As cited in Duffie and Beckman (1991), Klein (1979) developed an empirical equation based on the procedure of Hottel and Woertz (1942) and Klein (1975), which could be adapted for tubes. For N covers or glazing,

$$U_t = \left\{ \frac{N}{\frac{C}{T_{P,m}}\left[\frac{(T_{P,m} - T_a)}{(N+f)}\right]^e} + \frac{1}{h_w} \right\}^{-1} + \frac{\sigma(T_{P,m}^2 + T_a^2)}{(\varepsilon_P + 0.00591Nh_w)^{-1} + \frac{2N + f - 1 + 0.133\varepsilon_P}{\varepsilon_g} - N} \quad (3.15)$$

Where

f = $(1 + 0.089h_w - 0.1166h_w\varepsilon_p)(1 + 0.07866N)$ (3.16)

C = $520(1 - 0.000051\beta^2)$ for $0° < \beta < 70°$ (3.17)

= 466.297

e = $0.43(1 - 100/T_{p,m})$ (3.18)

β = collector tilt (degrees) = 45°

ε_g = emittance or emissivity of glass = 0.94

ε_P = emittance or emissivity of plate (or tube) = 0.96

T_a = Ambient temperature (K)

$T_{P,m}$ = mean temperature of plate (or tube) (K)

h_W = wind heat transfer coefficient (W/m^2C)

The wind heat transfer coefficient was computed with the use of the Reynolds number (Re), the Nusselt number, (Nu)

$$(Nu)=\frac{hL}{k}=0.664\,\mathrm{Re}_L^{1/2}\,\mathrm{Pr}^{1/3} \tag{3.19}$$

Further, when wind speed is zero, the correlation for free convection coefficient for inclined surfaces, was suggested by Fujii and Imura (1972) as cited in Bejan and Kraus (2003)

$$(\mathrm{Nu})=0.14\left[(\mathrm{Gr}\times\mathrm{Pr})^{1/3}-(\mathrm{Gr_c}\times\mathrm{Pr})^{1/3}\right]+0.56(\mathrm{Gr_c}\times\mathrm{Pr}\cos\gamma)^{1/4} \tag{3.20}$$

The back loss coefficient is

$$U_B=\frac{1}{R_{3a}+R_{3b}+R_{3c}}=\frac{1}{\left(\frac{x}{k}\right)_{3a}+\left(\frac{x}{k}\right)_{3b}+\left(\frac{x}{k}\right)_{3c}} \tag{3.21}$$

Where, $k_{plywood}$ = 0.13 W/mK, k_{felt} = 0.04 W/mK, $x_{plywood}$ = 19.05 mm, and x_{felt} = 19.05 mm

The edge loss coefficient was taken as a fraction of the back loss in terms of the area of the edge and the area of the collector. Again according to Duffie and Beckman (1991) it is:

$$U_e=\frac{(UA)_{edge}}{A_C} \tag{3.22}$$

Heat Loss from the Tank

The tank was insulated but it was not an assurance that heat loss was zero. The temperature of the water in the tank was higher than the ambient air so heat loss was inevitable.

The heat loss from the tank was the sum of the heat loss from its horizontal cylindrical wall and that from its plane circular ends. For the heat loss from the horizontal cylinder, the length was taken as constant. The mode of heat transfer for both was a combination of inside convection, conductions and outside convection.

$$Q_L=(Q_L)_{\text{horizontal cylinder}}+(Q_L)_{\text{plane ends}} \tag{3.23}$$

The heat flow cross-sectional area at the tank ends could not be considered as constant since the outside insulation diameter was larger than the tank diameter a mean area A_m for tank ends was used. And the heat loss from the tank therefore resulted in the following:

$$Q_L = \frac{T_i - T_o}{\frac{1}{\pi D_1 h_i l} + \frac{\ln(D_2/D_1)}{2\pi k_1 l} + \frac{\ln(D_3/D_2)}{2\pi k_2 l} + \frac{1}{\pi D_3 h_0 l}} + \frac{2(T_i - T_o)}{\frac{1}{A_i h_i} + \frac{x_1}{A_{m1} k_i} + \frac{x_2}{A_{m2} k_i} + \frac{1}{A_e h_o}} \tag{3.24}$$

The heat convection correlation of the water inside the tank was a complex thing and beyond the scope of this study. The assumption that the inside surface temperature of the tank is equal to the bulk inside water temperature was reasonable, and with that assumption, the inside convection terms of the equation 3.37 may be dropped.

$$Q_L = \frac{T_i - T_o}{\frac{\ln(D_2/D_1)}{2\pi k_1 l} + \frac{\ln(D_3/D_2)}{2\pi k_2 l} + \frac{1}{\pi D_3 h_0 l}} + \frac{2(T_i - T_o)}{\frac{x_1}{A_{m1} k_i} + \frac{x_2}{A_{m2} k_i} + \frac{1}{A_e h_o}} \tag{3.25}$$

Heat Loss through the Thermosyphons

When the solar collector was not receiving solar radiation during night time, the water in the tank still lost heat through the collector. The thermosyphons which are made with copper tubes still conducted heat back to the solar collector when the collector temperature became lower than the tank water.

The thermosyphons only operated when the evaporator section (the lower section) was receiving heat, and when the upper section was heated instead, heat was only transported back to the lower section by conduction. Thus the heat loss through the thermosyphons was analyzed with heat conduction mode. The equation used for the heat loss through the thermosyphons, Q_{TL}, is:

$$Q_{TL} = k_{Cu} A_{Cu} (T_C - T_E) \tag{3.26}$$

The Experimentation

An experiment was conducted that simultaneously measured the temperatures of the thermosyphon evaporator and condenser sections of the glazing, of the back surface, the ambient air dry-bulb and wet-bulb, and of the water inside the insulated tank.

The heat delivered to the water was measured in terms of the temperature rise of the water plus the tank, which was assumed to be of the same temperature with the water.

$$Q_{Water} = \frac{\left[(mC_P)_{water} + (mC_P)_{\tan k}\right](T_o - T_i)}{t} \quad (3.27)$$

The mass of water in the tank was 100 kg and the mass of tank was 25.02 kg.

The Solar Water Heater

The solar water heater was composed of a double-glazed box enclosing the thermosyphons, the thermosyphons, an insulated tank, and support frames. Refer to Figure 3.2. The condenser sections of the thermosyphons were inserted into the tank at an inclination angle of 45 degrees and were sealed. Then the flat-black-lacquer sprayed evaporator sections of the thermosyphons were enclosed in a box made of 19.05-mm ply-board with side and back insulations and covered with two sheets of 3/16-inch clear glass.

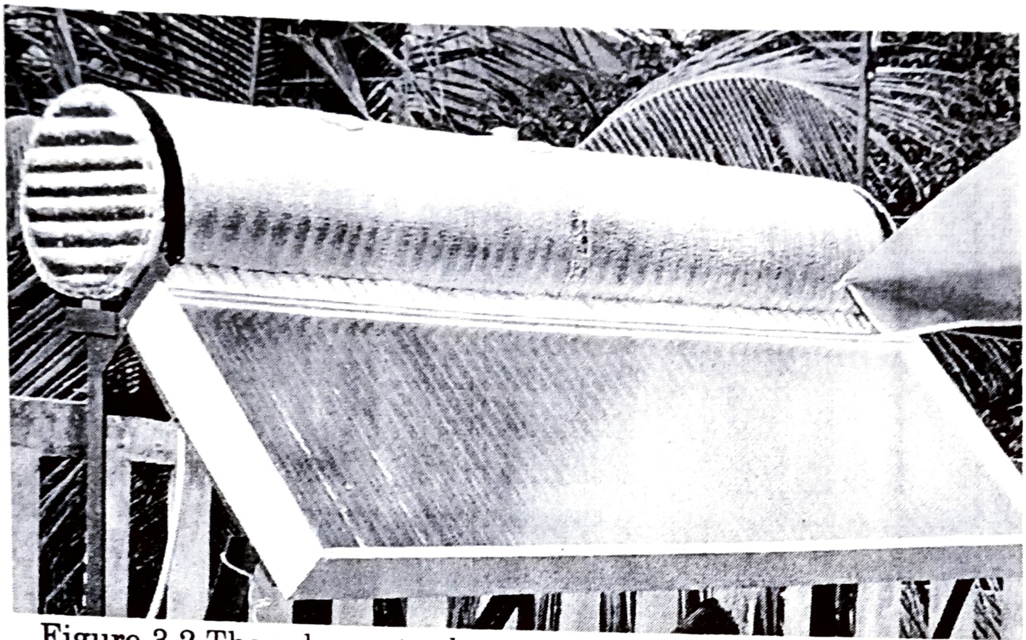


Figure 3.2 The solar water heater incorporating thermosyphons

Results and Discussion

Useful Solar Heat

The conduct of the study was a success. Fifty-two copper-water thermosyphons were fabricated, tested and functional. A glazed box solar collector and a specially made storage tank were fabricated. And the installed copper-water thermosyphons successfully heated the water in the storage tank.

The useful solar heat, which is the available heat from the sun less the collector losses, closely follows the irradiance readings indicating the response of the thermosyphons with the solar heat although there are conspicuous fluctuations from around 9 am to before noon due to clouds (Figure 4.1).

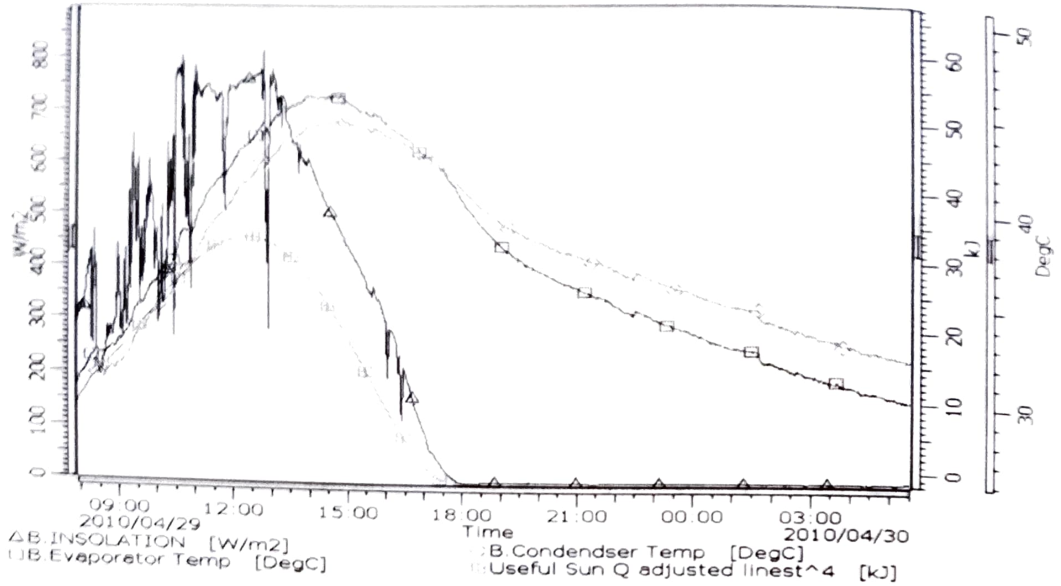


Figure 4.1 April 29 Profile of insolation, useful heat and thermosyphon temperatures

The thermosyphon evaporator temperature is shown higher than the thermosyphon condenser temperature when it is operating. Starting around 6 pm, the thermosyphon evaporator temperature dropped lower than the thermosyphon condenser. This was when the device was losing heat back to the collector.

Water Heat Gain

With the useful heat and wind speed, the water heat gain and tank heat loss is plotted in Figure 4.2. The occurrences of the wind indicate that the wind was of a transient type. And as can be seen the occurrence of the wind did not influence the trajectory of water heat gain nor the tank heat loss curve

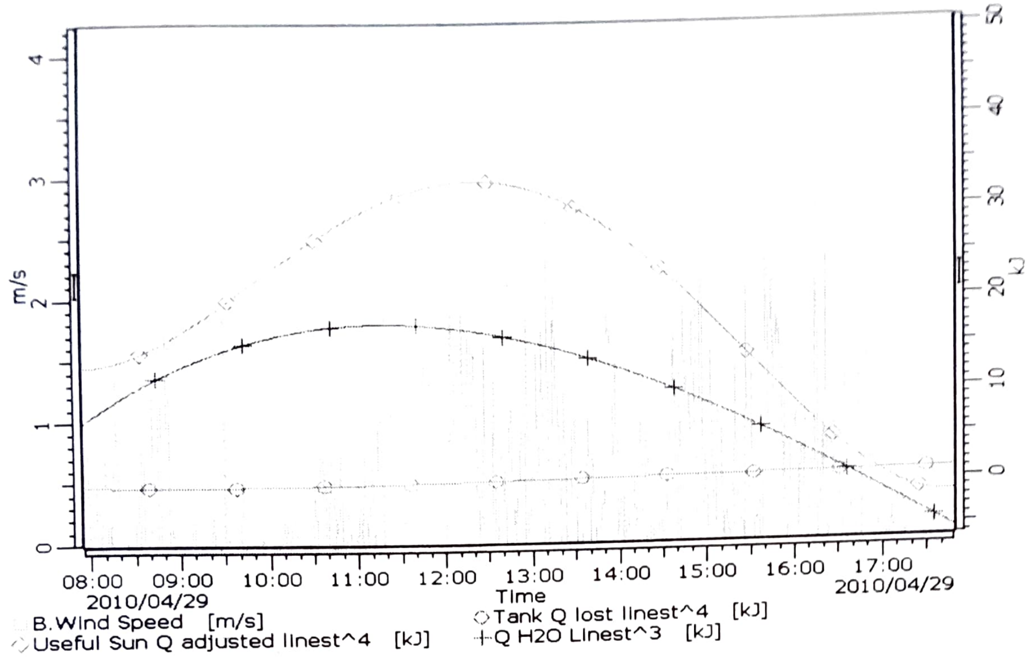
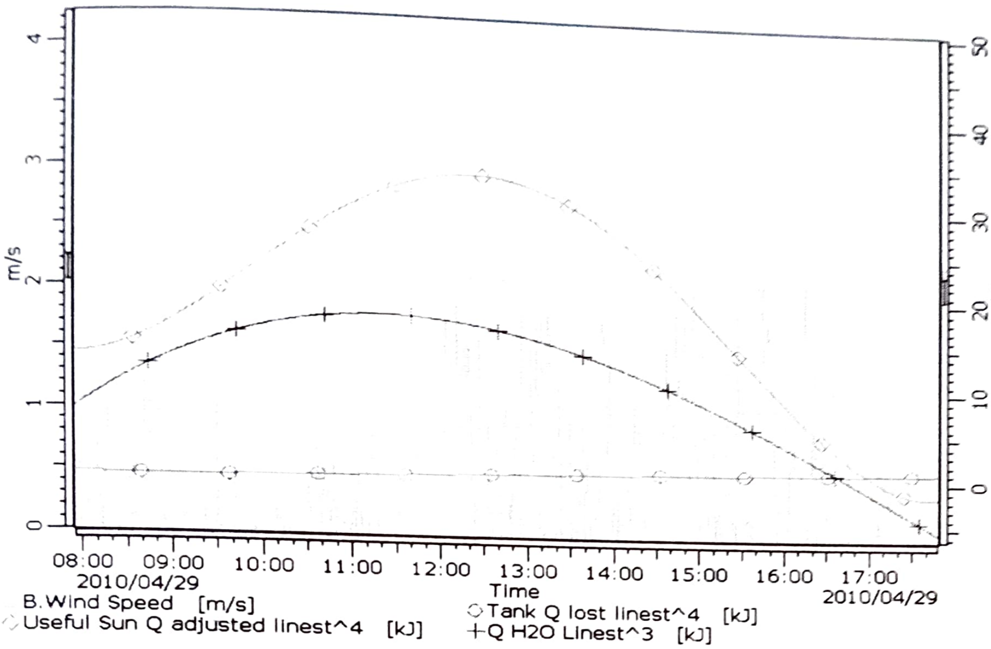


Figure 4.2 Plots of wind speed, useful sun heat, tank heat lost and water heat gain regressed

Sun Out Heat Loss

During nighttime, when the collector was not receiving solar radiation, the tank continued to lose heat to the ambient air, and with the thermosyphons now functioning as fins, the collector was then heated by the tank water through conduction, and the collector losing heat to the ambient. This is illustrated in Figure 4.3. The plot of the total heat loss from the tank which is the sum of heat losses from tank sides and ends and through the thermosyphons during no sun, appeared to be the same.



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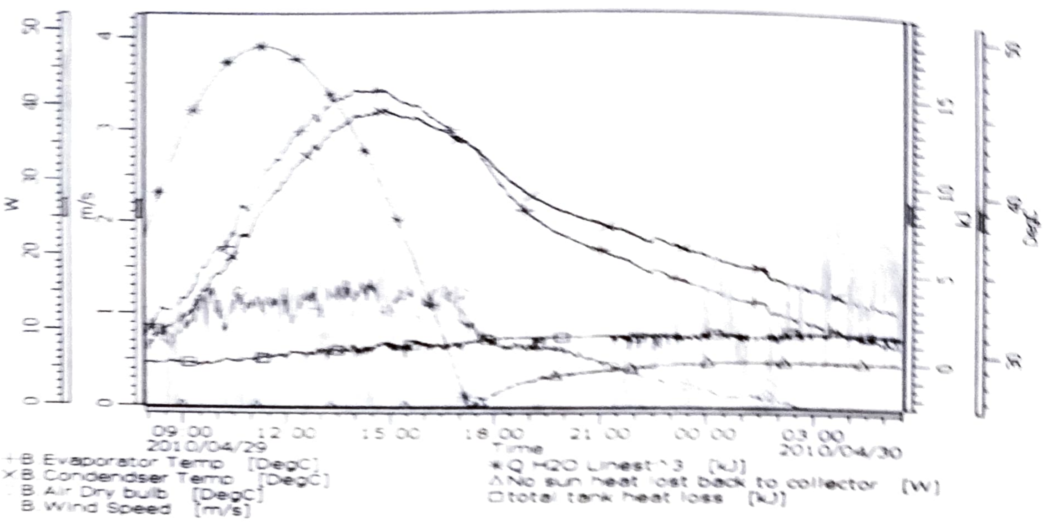

Figure 4.3 Plot of total tank heat loss don't seem to be affected by sun down.

Efficiency

The efficiency was computed as the ratio of the heat it delivered to the heat it absorbed. The plot of the said ratios is plotted on Figure 4.4. The actual plot of said water heat gain is shown along with the regressed plot and water temperature. It can be seen that water temperature rises faster with the useful heat increasing and the water temperature just maintains with the useful heat decreasing.

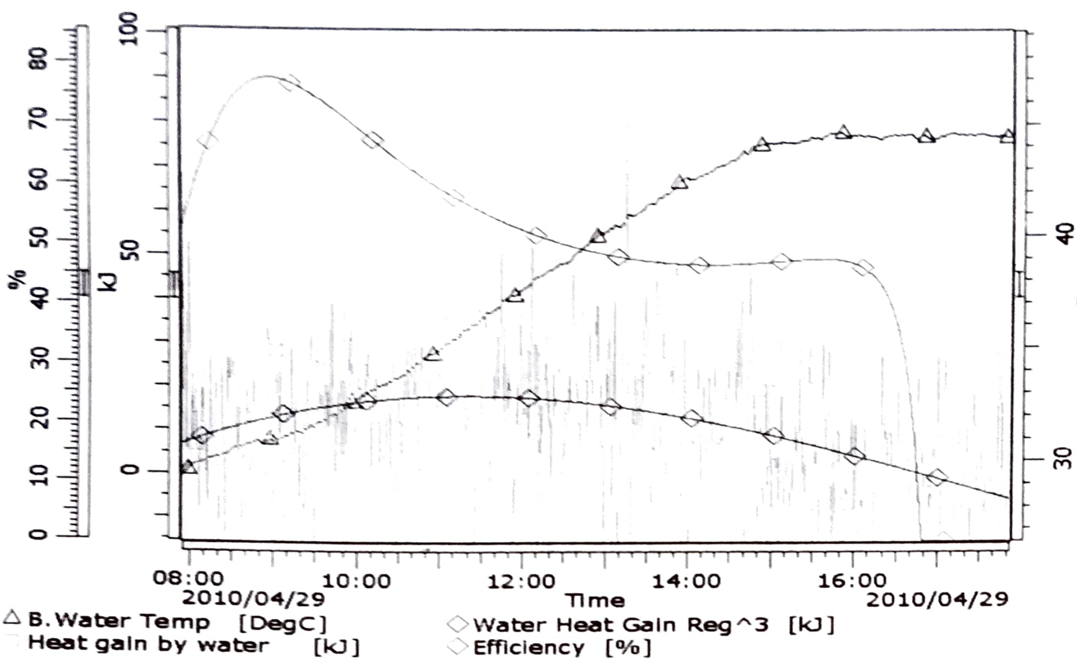

Figure 4.4 Water Temperature, Water Heat Gain (differential), Water Heat Gain (Regressed) and efficiency

It is interesting to note that the efficiency computed was highest before 10 am. This was because with the water temperature was low and the thermosyphon condenser temperature was high. The high temperature difference induces highest heat transfer rate. The steep drop of the efficiency is also notable in Figure 4.6 at around 4:30 pm. This happened with the water heat gain falling to zero and is evidenced with water temperature almost horizontal. The water heat gain extending to negative after 4:49 pm meanst the useful heat could no longer compensate with the tank heat loss, thus, the water started to lose heat.

Conclusion

As Figure 4.4 shows, the temperature of water varied directly with the insolation. The rather small difference of the temperatures between the tank water and the thermosyphons in Figures 4.1 and 4.3 clearly indicated that the thermosyphons performed well and are effective in transferring heat from the sun to the tank water. The ratio of heat transferred to the water to the useful heat reached to as high as 90% at around 9 am (Figure 4.4). This encouraging result can provide fundamental data for the adoption of the thermosyphon technology in applications requiring enhanced thermal conductance.

The night time or no-sun data however, reveal that those thermosyphons cannot keep the heat stored in the tank water. This is because during night time or no-sun condition, the thermosyphon, with the condenser section still protruding inside the tank containing the hot water, then served as fins extending to the collector. Although not necessarily functioning as thermosyphons, the copper tube (thermosyphon casing) is still a good heat conductor.

The heat losses from tank through the thermosyphons during night time add up to the total heat loss from the tank. The plot of the total heat loss from the tank illustrated the slow yet constant increase even through the night.

Recommendations

This study showed that the copper-water thermosyphons are effective heat transfer mechanism in water heating. Solar water heater engineers or designers can adapt the use of this type of thermosyphons in their designs or projects. When applied as solar water heater in the same manner, however, because the tank failed to keep the water hot during night time it is therefore recommended that the hot water be decanted at around 3 pm to a separate insulated tank so as to keep it hot until used.

The following maybe considered for further research:

a. Characterizing a thermosyphon with different fluid inventories.
b. The use of absorber plate with the thermosyphons
c. Life cycle analysis of a thermosyphon in terms of continuous heating, and in terms of cycles of heating up and cooling down.

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