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# Experimental and theoretical investigation of an evacuated tube solar water heater incorporating wickless heat pipes

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**Abstract** The present work involved an extensive outdoor performance testing program of a solar water heating system that consists of four evacuated tube solar collectors incorporating four wickless heat pipes integrated to a storage tank. Tests were conducted under the weather conditions of Baghdad, Iraq. The heat pipes were of 22 mm diameter, 1800 mm evaporator length and 200 mm condenser length. Three heat pipe working fluids were employed, ethanol, methanol, and acetone at an inventory of 50% by volume of the heat pipe evaporator sections. The system was tested outdoors with various load conditions. Results showed that the system performance was not sensitive to the type of heat pipe working fluid employed here. Improved overall efficiency of the solar system was obtained with hot water withdrawal (load conditions) by 14%. A theoretical analysis was formulated for the solar system performance using an energy balance based iterative electrical analogy formulation to compare the experimental temperature behavior and energy output with theoretical predictions. Good agreement of 8% was obtained between theoretical and experimental values.

**Keywords:** Wickless heat pipe; Fill charge; Electrical analogy; Evacuated tube solar collector; water heater; Performance

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## Nomenclature

| | | |
|---|---|---|
| $A$ | – | surface area |
| $c_w$ | – | specific heat of water |
| $D$ | – | diameter |
| $F$ | – | shape factor |
| $g$ | – | gravitational acceleration |
| Gr | – | Grashof number |
| $h$ | – | heat transfer coefficient |
| $h_{fg}$ | – | latent heat of vaporization |
| $I$ | – | solar insolation |
| $K$ | – | number of time intervals |
| $k$ | – | thermal conductivity |
| $L$ | – | length |
| $l_a$ | – | adiabatic section length |
| $M$ | – | mass of water (tank, load) |
| $m$ | – | mass |
| $\dot{m}$ | – | mass flow rate |
| Pr | – | Prandtl number |
| $Q_o$ | – | overall collected heat |
| $Q_u$ | – | overall useful energy gain |
| $q$ | – | heat |
| $R$ | – | thermal resistance |
| $r$ | – | heat pipe radius |
| Re | – | Reynolds number |
| $T$ | – | temperature |
| $T_a$ | – | ambient temperature |
| $t$ | – | time |
| $U_{loss,absorber}$ | – | absorber loss heat transfer coefficient |

### Greek symbols

| | | |
|---|---|---|
| $\alpha$ | – | absorptivity |
| $\beta$ | – | inclination angle from the horizontal, degree |
| $\Delta$ | – | difference |
| $\delta$ | – | characteristic diameter |
| $\delta_{insulation}$ | – | insulation thickness |
| $\varepsilon$ | – | emissivity |
| $\eta_o$ | – | overall efficiency |
| $\mu$ | – | dynamic viscosity |
| $\nu$ | – | kinematic viscosity, $= \mu/\rho$ |
| $\rho$ | – | density |
| $\sigma$ | – | Stefan–Boltzman constant |
| $\tau$ | – | transmissivity |

### Subscripts

| | | |
|---|---|---|
| $a$ | – | absorber |
| $b$ | – | bulk |

| | | |
|---|---|---|
| *co* | – | condenser |
| *co-total* | – | equivalent of the four condenser sections |
| *conv* | – | convection |
| *ev* | – | evaporator |
| *eff* | – | effective |
| *g* | – | glass absorber tube surface |
| *go* | – | glass absorber surface of the selective coating |
| *g-ev* | – | between the absorber inner surface and the evaporator surface |
| *i* | – | in |
| *l* | – | load |
| *load* | – | removed out by the load water |
| *loss,absorber* | – | radiative heat loss from the absorber glass tube surface to the outer glass tube |
| *loss,tank* | – | losses through the storage tank insulation |
| *m* | – | mean |
| *mf* | – | final mean value |
| *ms* | – | starting mean value |
| *o* | – | out |
| *p* | – | pipe |
| *rad* | – | radiative |
| *sat* | – | saturation |
| *stored* | – | stored in the storage water |
| *T* | – | equivalent between four condensers and the storage water |
| *t* | – | tank |
| *tot* | – | total |
| *v* | – | vapor |
| *water* | – | storage water surrounding the condenser section |
| *w,e* | – | evaporator wall |
| *w,c* | – | condenser wall |

# 1 Introduction

Heat pipes are thermal conductors of very high thermal conductance; they are being used in solar collectors as heat absorbers and conductors for several advantages. They have lower parasite pumping requirements, better freeze tolerance, no corrosion problems and a thermal diode benefit. Heat pipe solar collectors are modular in design; it is a single unit facilitating the assembly of a collector array composed of multiple heat pipes. The incorporation of heat pipes with flat plate collectors was investigated in many papers [1–5]. The utilization of the Owens–Illinois type evacuated glass tube solar collectors in the thermosyphon solar water heating systems is bounded by many disadvantages such as collectors destruction by hail and the corresponding solar system failure, low performance with loading (hot water withdrawal) originated from its low response and the corresponding low water temperature. Heat pipes are proposed to overcome these limitations. Most investigations on evacuated tube heat pipe solar collectors were carried out with conventional evacuated glass tubes of the Corning type. Walker *et al.* [6] investigated the thermal performance of a solar water heating system of 360 evacuated tube heat pipe solar collectors. Van der Aa proposed a solar collector comprising a metal heat pipe with the evaporator section within an evacuated glass envelope [7]. The absorber plate is connected to the heat pipe evaporator section. Ng *et al.* [8] proposed a simple energy balance based electrical analogy technique for the solar system components with suitable simplifying assumptions to compare the steady-state performance of two types of commercial evacuated tube heat pipe solar collectors under the climatic conditions in Singapore. Huang and Tsuei [9] and Praene *et al.* [10] also used this technique. Joudi and Al-Joboory investigated the thermal performance of individual evacuated tube heat pipe solar water heaters comprising small storage tanks and taking into consideration the effect of heat pipe evaporator length and diameter with different kinds of heat pipe working fluids [11]. They recommended solar heat pipes of 22 mm diameter and 1800 mm evaporator length.

Many papers [12–16] deal with evacuated tube solar water heaters incorporating wickless heat pipes with different working fluids and applications by using continuous flow configuration systems, where the process fluid is circulated through the system header *via* a circulation pump.

The use of wickless heat pipes with the Owens–Illinois type evacuated glass tubes to form evacuated tube heat pipe solar collectors connected to a storage tank for domestic applications has been found scarce in the available literature, a solar hot water system of four evacuated tube heat pipes integrated into a storage tank has not been cited in the literature.

The present work investigates experimentally the performance of an evacuated tube heat pipe solar water heater incorporating an array of four wickless heat pipes inside evacuated tube solar collectors of the Owens–Illinois type. The heat pipes were selected from outdoor tests of individual heat pipe solar systems [11]. The system was operated at various weathers. The effects of load conditions and heat pipe working fluids on the performance were assessed experimentally from outdoor tests. A simplified theoretical model based on electrical analogy was formulated to predict, theoretically, the performance of solar hot water system.

## 2 The experimental solar hot water system

The experimental evacuated tube-heat pipe solar water heating system, shown in Fig. 1, incorporated an array of four evacuated glass tube heat pipe units. Heat pipes of 1800 mm evaporator length and 22 mm diameter were used in this system. The tubular solar collectors' type and model were Owens–Illinois and 47-58-1800-YCF respectively [17].

Figure 1: The evacuated tube heat pipe solar water heating system.

Heat pipe evaporator sections were introduced into the evacuated glass tubes and aligned at the tube centerlines, while the condenser sections of heat pipes were inserted into the storage tank to transport the collected solar energy into the storage water and were fixed by rubber stoppers. Aluminum reflector was placed behind the evacuated glass tubes to increase the heat collection. The storage tank was made of a 0.8 mm thick galvanized steel of dimensions (660 mm × 350 mm × 350 mm) with a capacity of 60 liters. An insulated 120 l constant level water supply tank was used with the hot water removal (loading) tests to replenish consumed solar heated water. The assembled solar water heating system was supported on a south facing steel stand at a tilt angle of 45°, at which optimum heat pipe performance [11] and represents a good winter tilt angle for solar collectors in Baghdad at 33.3° N latitude angle.

In order to measure temperatures of the solar system, calibrated thermocouples of type T were fixed at the desired locations (inner and outer surfaces of the tubular solar collectors, water inlet and outlet of the storage tank, ambient and the outer surface of the heat pipe evaporator section). All thermocouples were connected to a Lutron BTM-4205SD temperature recorder. To obtain the mean tank temperature (average storage water temperature), three thermocouples were located at three positions at different heights inside the storage tank and the mean tank temperature is taken as the average of the three temperatures. The hot water withdrawal flow rate was measured by a calibrated rotameter type flow meter. The experimental testing program was carried out in September, October, November, and December of 2008 in clear sky conditions on sunny days. The heat pipes of the system were all charged at 50% fill charge ratio and operated with ethanol, methanol, and acetone. Experimental data were recorded every half an hour interval along the operation period from 8:00 a.m. until 16:00 p.m., and every one hour interval along the operation period from 16:00 p.m. till 24:00 p.m. The solar system operation during the test data recording interval was considered to be steady and the recorded test data were assumed to be constant. Experiments were conducted with three load conditions (hot water withdrawal patterns); these are no water withdrawal, intermittent withdrawal of hot water, and continuous withdrawal of hot water. Experiments of the first loading pattern were conducted with three different amounts of storage water in the storage tank (storage capacities), these are 40, 50, and 60 l, respectively. In the experiments of the second loading pattern, hot water was withdrawn intermittently from the storage tank from 10:00 a.m. to 14:00 p.m. at the beginning of each hour, with seven removal quantities of 2, 3, 4, and 5 l within five minutes and 6, 8, and 10 l within ten minutes. In the third loading pattern hot water was withdrawn continuously along the operation period from 8:00 a.m. to 18:00 p.m. at five flow rates of 4, 5, 6, 7, and 8 l/h. The overall heat collection was obtained by measuring the starting and ending temperatures in the storage tank along the experimental operation time from 8:00 a.m. till 18:00 p.m. by

$$Q_o = M_t c_w \left(T_{mf} - T_{ms}\right), \tag{1}$$

where $M_t$ is the mass of water in the storage tank, $T_{ms}$ and $T_{mf}$ are the mean temperatures at the start and end of the period. The overall daily efficiency of the solar hot water system efficiency was then obtained from the equation

$$\eta_o = \frac{M_t c_w (T_{mf} - T_{ms})}{4 \displaystyle\int_0^{t_k} A_a I(t)\, \mathrm{d}t} \,, \tag{2}$$

while for the hot water withdrawal loading patterns, the overall heat collection was calculated from the following relation

$$Q_o = M_t c_w \left( T_{mf} - T_{ms} \right) + \int\limits_0^{t_k} m_l c_w \left( T_{lo} - T_{li} \right) \mathrm{d}t \,. \tag{3}$$

This equation can be simplified and re-arranged to the form

$$Q_o = M_t c_w \left( T_{mf} - T_{ms} \right) + M_l c_w \left( T_{mo} - T_{mi} \right) \,, \tag{4}$$

where $M_l = \sum \dot{m}_l^i \Delta t^i$, and $(T_{mo} - T_{mi}) = \sum\limits_{i=1}^{k} \dot{m}_l^i \left( T_{lo}^i - T_{li}^i \right) \Delta t^i \Big/ \sum \dot{m}_l^i \Delta t^i$, and $k$ is the number of recording intervals, $\Delta t^i$, during the loading period, and so the overall efficiency of the solar system could be obtained from

$$\eta_o = \frac{M_t c_w \left( T_{mf} - T_{ms} \right) + M_l c_w \left( T_{lo} - T_{li} \right)}{4 \displaystyle\int_0^{t_k} A_a I(t)\, \mathrm{d}t} \,. \tag{5}$$

The solar radiation intensity was measured and recorded by TES-1333 solar power data logging meter and calibrated by using the ASHRAE clear sky model [18].

# 3 Theoretical analysis of the steady-state performance

The steady-state performance of the evacuated tube heat pipe solar water heater is derived from a simple energy balance procedure similar to that presented by Ng *et al.* [8] and Praene *et al.* [10] for two commercial evacuated tube heat pipe solar collectors of the Corning design, attached to a manifold. Figure 2 illustrates schematically the heat flow into the components of the solar system.

As shown in the electrical circuit analogy to a single evacuated tube heat pipe solar water heating system in Fig. 3, the absorbed incident solar energy is distributed into useful energy gain, thermal losses and optical

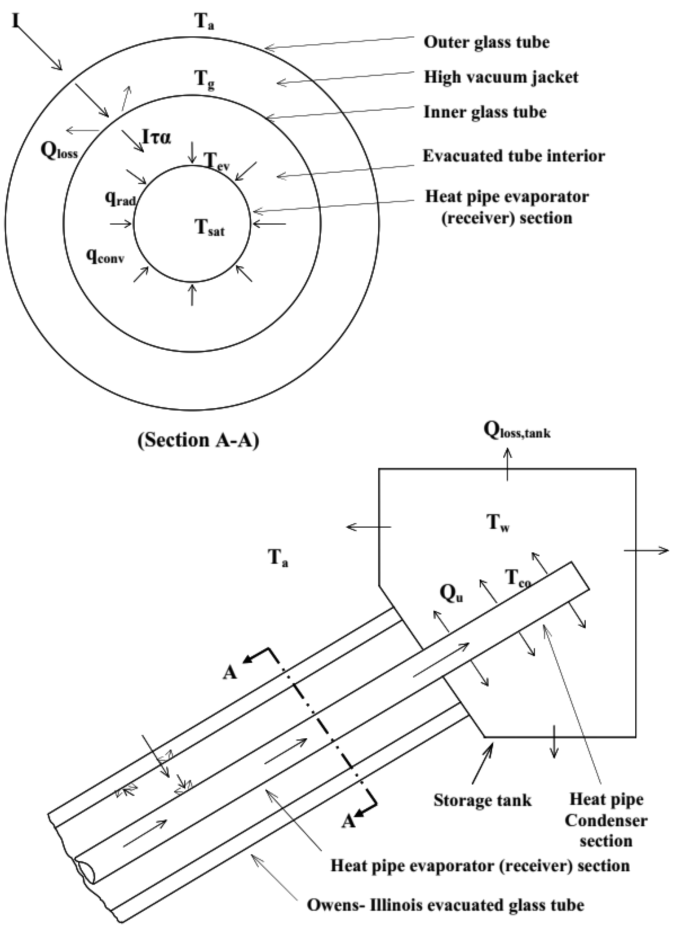


Figure 2: Side view of the evacuated tube heat pipe solar water heater.

losses. The incident solar radiation that falls on an absorber of surface area, $A_a$, is given by the product $I\tau\alpha A_a$. The product $\tau\alpha$ represents the optical efficiency when the radiation passes through the outer glass tube of the collector. On reaching the inner glass absorber tube, a part of the thermal energy is dissipated to the surroundings by radiative heat losses. These losses are represented as the overall heat loss coefficient, $U_{loss}$, multiplied by the difference between the glass absorber temperature, $T_g$, and the ambient temperature, $T_a$. A major part of the incident energy is transferred to the heat pipe evaporator section, which in turn transports this energy into the storage water tank via the heat pipe condenser section. This energy is then

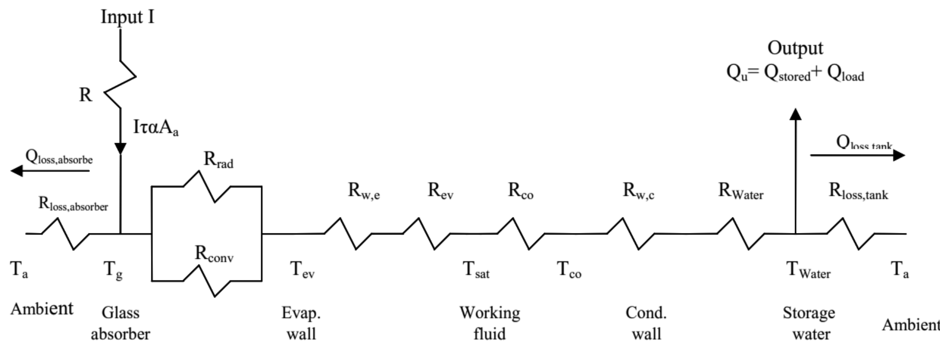


Figure 3: Electric network analogy to a single evacuated tube heat pipe solar water heating system.

extracted by the storage water that gives the useful heat, $Q_u$. Part of $Q_u$ is also lost to the ambient through heat leaks from the storage tank, as shown by the resistance term, $R_{loss,\ tank}$. The following simplifying assumptions are proposed to simplify the analysis and calculations:

1) steady-state conditions,
2) negligible convective and conductive heat losses from the inner absorber tube to the ambient due to the high vacuum envelope jacket surrounding the absorber tube,
3) outer glass tube temperature is at the ambient temperature,
4) negligible heat losses from the heat pipe evaporator surface,
5) absorbed solar energy is transmitted to the evaporator surface by natural convection and radiation in the radial direction,
6) average optical efficiency of 0.85 for the evacuated tube solar collector,
7) heat accumulation inside the collector is neglected,
8) all solar system component surfaces are considered isothermal,
9) phase change of the working fluid in the heat pipe occurs at constant temperature,
10) negligible vapor thermal resistance inside the heat pipe,
11) the effect of heat pipe end caps is negligible,
12) one dimensional heat flow in the solar system components,
13) water in the storage tank is fully mixed (negligible stratification),

14) negligible heat losses from pipe connections, valves, air vent and the probe,
15) evacuated tube heat pipe solar water heating system is divided into four single identical parallel systems,
16) heat pipe thermal diode property prevents heat flow back to the evacuated tube solar collector.

Energy balance for the outer glass tube gives $I - I\tau = q_{loss,\, absorber}$

$$I(1-\tau) = \varepsilon_{go}\sigma \frac{\left(T_g^4 - T_a^4\right)}{(T_g - T_a)} (T_g - T_a)$$

or

$$T_g = \left[T_a^4 + \frac{I\,(1-\tau)}{\varepsilon_{go}\sigma}\right]^{1/4}, \tag{6}$$

where: $T_a$ – ambient temperature, $T_g$ – glass absorber tube temperature, $I$ – solar insolation, $\varepsilon_{go}$ – glass absorber surface emittance of the selective coating (given as 0.044), $\sigma$ – Stefan–Boltzmann constant, $\tau$ – transmittance of the outer glass tube.

Energy balance for the inner glass absorber tube gives

$$\frac{T_g - T_{ev}}{R_{tot}} = I\tau\alpha - \frac{T_g - T_a}{R_{loss,\, absorber}}$$

or

$$T_{ev} = T_g - R_{tot}\left[I\tau\alpha - \frac{T_g - T_a}{R_{loss,\, absorber}}\right], \tag{7}$$

where $T_{ev}$ is the evaporator surface temperature and $\alpha$ is the absorptance of the glass absorber. Thermal resistance to radiative heat loss from the absorber glass tube surface to the outer glass tube is given as

$$R_{loss,\, absorber} = \frac{1}{U_{loss,\, absorber} A_{absorber}},$$

where $U_{loss,\, absorber}$ is the absorber loss heat transfer coefficient, given as

$$U_{loss,\, absorber} = \varepsilon_{go}\sigma \frac{T_g^4 - T_a^4}{T_g - T_a}.$$

Equivalent thermal resistance between the glass absorber tube inner surface and the evaporator surface is given for two parallel resistances as

$$R_{tot} = \frac{R_{rad}\,R_{conv}}{R_{rad} + R_{conv}}\,.$$

Here the thermal resistance to radiative heat exchange between the glass absorber tube inner surface and the evaporator surface is given as

$$R_{rad} = \frac{\dfrac{1-\varepsilon_g}{\varepsilon_g} + \dfrac{1}{F_{g\text{-}ev}} + \dfrac{1-\varepsilon_{ev}}{\varepsilon_{ev}}\dfrac{D_g}{D_{ev}}}{\sigma A_g\left(T_g^2 + T_{ev}^2\right)\left(T_g + T_{ev}\right)}\,,$$

where: $\varepsilon_g$ – emittance of the glass absorber tube inner surface, $\varepsilon_{ev}$ – emittance of the evaporator surface, $D_g$, $D_{ev}$ – diameters of the inner glass absorber tube and the evaporator, $A_g$ and $A_{ev}$ – surface areas of the inner glass absorber tube surface and the evaporator, $F_{g\text{-}ev}$ – shape factor between the absorber inner surface and the evaporator surface, given as $F_{g\text{-}ev} = D_{ev}/D_g$. Convective thermal resistance between the inner surface of the glass absorber tube and the evaporator surface is given as

$$R_{conv} = \frac{\ln\left(\dfrac{D_g}{D_{ev}}\right)}{2\pi L_g k_{eff}}\,,$$

where $L_g$ is the absorber tube length. The apparent (effective) thermal conductivity is given as [19, 20]

$$k_{eff} = 0.386\left(\frac{\mathrm{Pr}}{0.861 + \mathrm{Pr}}\mathrm{Ra}^*\right)^{1/4}\,,$$

where

$$\mathrm{Ra}^* = \frac{\left[\ln\left(\dfrac{D_g}{D_{ev}}\right)\right]^4}{\delta^3\left[D_g^{-3/5} + D_{ev}^{-3/5}\right]^5}\mathrm{Ra}_\delta\,, \qquad \mathrm{Ra}_\delta = \frac{g\beta\left(T_g - T_{ev}\right)\delta^3}{\nu^2}\mathrm{Pr}\,,$$

$$\beta = \frac{1}{T_f}\,, \qquad T_f = \frac{T_g + T_{ev}}{2}\,.$$

Energy balance for the heat pipe evaporator section wall

$$\frac{T_g - T_{ev}}{R_{tot}} = \frac{T_{ev} - T_{sat}}{R_{w,e} + R_{ev} + R_v}$$

or

$$T_{sat} = T_{ev} - \frac{R_{w,e} + R_{ev} + R_v}{R_{tot}} \left(T_g - T_{ev}\right), \tag{8}$$

where $T_{sat}$ is the saturation temperature of the working fluid, $R_v$ is the vapor thermal resistance, which is negligibly small compared with the other resistances. The thermal resistance depending on the phase change of the heat pipe working fluid inside the evaporator section, given as

$$R_{ev} = \frac{1}{h_{ev} A_{i,ev}}$$

and the evaporator wall resistance is given as

$$R_{w,e} = \frac{\ln\left(\frac{r_o}{r_i}\right)_{ev}}{2\pi L_{ev} k_{ev}},$$

where $h_{ev}$ is the average heat transfer coefficient in the evaporator section (calculated from the correlations for the working fluids), $L_{ev}$ is the length of evaporator section, $r_i$ and $r_o$ are the inner and outer radii of the evaporator wall, respectively, and $k_{ev}$ is thermal conductivity of the evaporator wall.

Energy balance for the heat pipe working fluid gives

$$\frac{T_{ev} - T_{sat}}{R_{w,e}} = \frac{T_{sat} - T_{co}}{R_{co}}$$

or

$$T_{co} = T_{sat} - \frac{R_{co}}{R_{w,e}} \left(T_g - T_{ev}\right), \tag{9}$$

where $R_{co}$ is the thermal resistance depending on the phase change of the heat pipe working fluid inside the condenser section, given as

$$R_{co} = \frac{1}{h_{co} A_{i,co}},$$

where $h_{co}$ is the heat transfer coefficient in the condenser section, given by Hussein, Muhammd and El-Asfouri [21] as

$$h_{co} = \left[0.997 - 0.334(\cos\beta)^{0.108}\right] \times \left[\frac{k_l^3\, \rho_l^2\, g\, h_{fg}}{\mu_l\, L_{co}\, (T_{sat} - T_{co})}\right]^{0.25} \left[\frac{L_{co}}{d_i}\right]^{\left[0.254(\cos\beta)^{0.385}\right]}, \tag{10}$$

where $\beta = 45^\circ$ is the inclination angle, $L_{co}$ is the heat pipe condenser section length, and $d_i$ is the inner diameter of the condenser section.

Energy balance for the heat pipe condenser section wall gives

$$\frac{T_{sat} - T_{co}}{R_{co}} = \frac{T_{co} - T_{water}}{R_{w,co} + R_{water}}$$

or

$$T_{water} = T_{co} - \frac{R_{w,c} + R_{water}}{R_{co}} \left(T_{sat} - T_{co}\right), \tag{11}$$

where $T_{water}$ is the storage water temperature. The condenser wall thermal resistance is given as

$$R_{w,co} = \frac{\ln\left(\frac{r_o}{r_i}\right)_{co}}{2\pi L_{co} k_{co}},$$

where $L_g$ is the length of condenser section, $k_{co}$ is the thermal conductivity of the condenser wall, $r_i$ and $r_o$ are the inner and outer radii of the condenser wall. The thermal resistance of the storage water surrounding the condenser section is given as

$$R_{water} = \frac{1}{h_{water}},$$

where $h_{water}$ is the average film heat transfer coefficient in the storage tank, given as follows [19, 20, 22, 23]:

- for no-load condition, the Churchill correlation is recommended as

$$h_{water} = \frac{k}{D_{co}} \left\{ 0.6 + 0.387 \left( \frac{\mathrm{Gr\,Pr}}{\left[1 + \left(\frac{0.559}{\mathrm{Pr}}\right)^{\frac{9}{16}}\right]^{\frac{16}{9}}} \right)^{\frac{1}{6}} \right\}^2, \tag{12}$$

- for loading conditions, the same references recommended the correlation of Churchill and Bernstein, with $D_{co}$ – the condenser outer diameter, as

$$h_{water} = \frac{k}{D_{co}} \left\{ 0.3 + \frac{0.62\mathrm{Re}^{\frac{1}{2}}\mathrm{Pr}^{\frac{1}{3}}}{\left[1 + \left(\frac{0.4}{\mathrm{Pr}}\right)\frac{2}{3}\right]^{\frac{1}{4}}} \left[1 + \left(\frac{\mathrm{Re}}{282000}\right)^{\frac{5}{8}}\right]^{\frac{4}{5}} \right\}. \tag{13}$$

Figure 4 shows the electrical circuit diagram analogous to the evacuated tube heat pipe solar water heating system with four evacuated tube heat

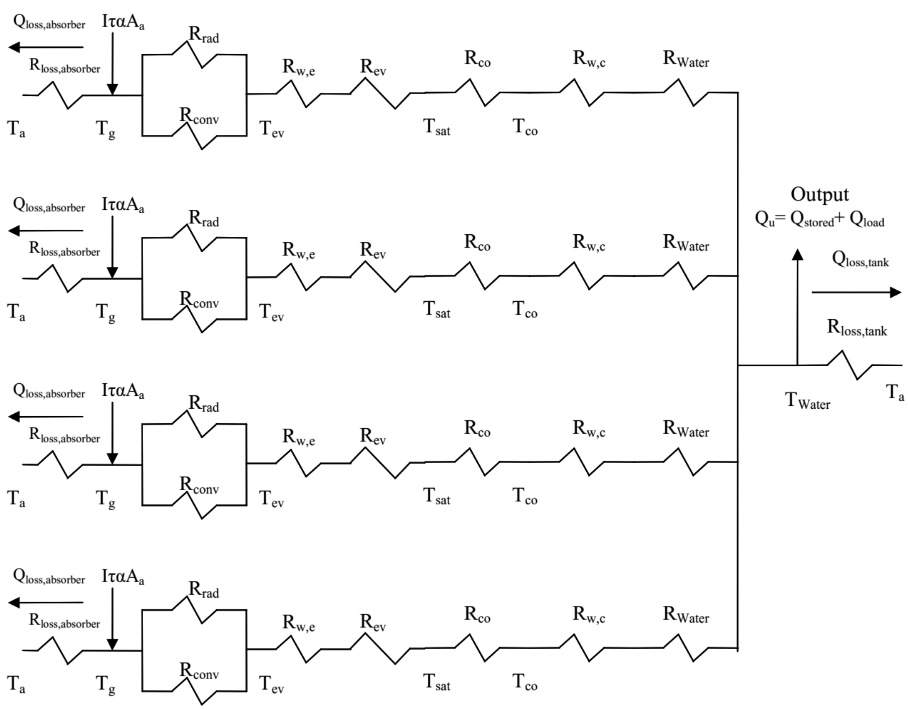


Figure 4: Electric network analogy to evacuated tube heat pipe solar water heater with four evacuated tube heat pipe assemblies.

pipe assemblies and a 60 l capacity storage tank is considered to be composed of four identical single units, each of 15 l storage capacity, connected in parallel. This is electrically represented by four parallel identical networks connected to the same common point (the storage tank); since it is already assumed that the storage water is fully mixed. Therefore, Eq. (11) can be re-written for this system as follows:

$$T_{water} = T_{co} - \frac{R_T}{R_{co_{total}}} \left( T_{sat} - T_{co} \right), \tag{14}$$

where $R_T$ is equivalent thermal resistance between the condenser and the storage water of the four units, calculated as $R_T = \left( R_{w,c} + R_{water} \right)/4$, and $R_{co_{total}}$ is the equivalent thermal resistance of the four condenser sections, calculated as $R_{co_{total}} = R_{co}/4$.

Energy balance for the storage tank yields the useful energy gain as

$$Q_u = \frac{T_{co} - T_{water}}{R_{w,co} + R_{water}} - \frac{T_{water} - T_a}{R_{loss,\,tank}}, \tag{15}$$

where the thermal resistance through the storage tank insulation is given as $R_{loss,tank} = 1/U_{loss}A_{tank}$, where $U_{loss} = k_{insulation}/\delta_{insulation}$. In case of loading, this useful energy gain is the summation of the energy stored in the storage water and that extracted with the load water. Therefore,

$$Q_u = Q_{stored} + Q_{load}\,, \tag{16}$$

where $Q_{stored}$ is the heat stored in the storage water, and $Q_{load}$ is the heat removed by the load water.

To calculate the storage water temperature, a marching iterative procedure is used, as illustrated in Fig. 5. The inputs are $I$, $\tau$, $\sigma$, $T_a$, $T_{in}$, $\alpha$, $\varepsilon_g$, $\varepsilon_{go}$, $\varepsilon_{ev}$, $D_g$, $D_{ev}$, $D_{co}$, $L_g$, $L_{ev}$, $L_{co}$, $d_i$, $k_{ev}$, and $k_{co}$.

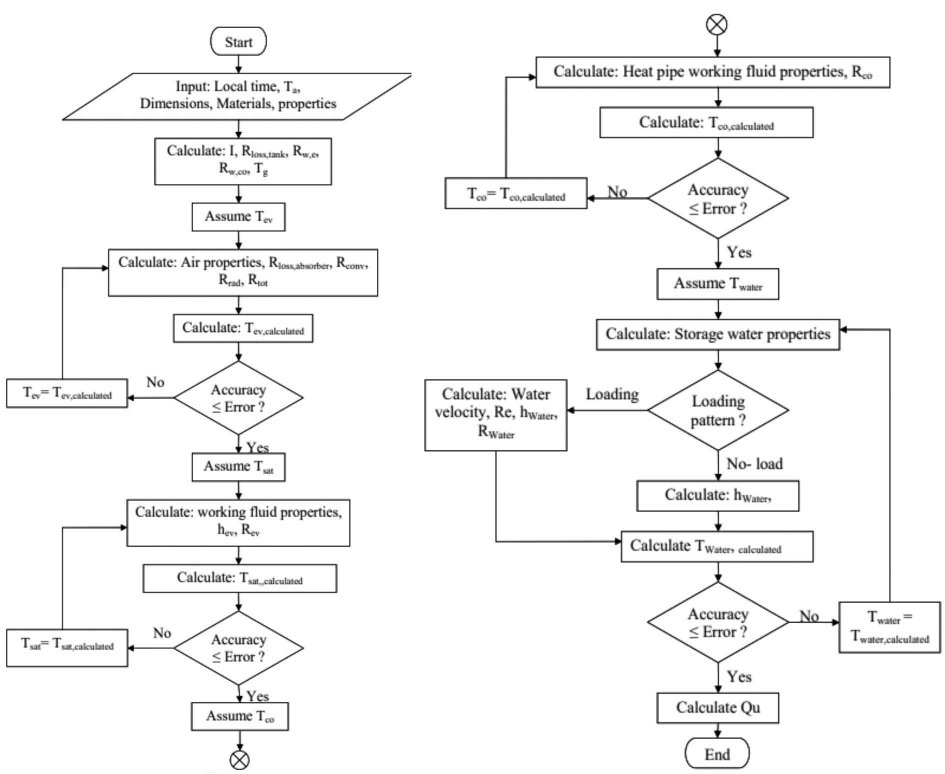


Figure 5: Flow chart of the iterative calculation procedure.

# 4 Discussion of results

The selection of heat pipes for this system was based on the outdoor testing results of the individual solar systems [11]. The design details and experimental program of this system are given in Tab. 1. The experimental results

Table 1: Characteristics and experimental outdoor testing program of the evacuated tube heat pipe solar water heating system.

<table>
<tr><th colspan="3">Components and parameters</th><th>Specifications</th></tr>
<tr><td colspan="3">Collector orientation</td><td>45°, south facing</td></tr>
<tr><td colspan="3">Number of collectors</td><td>4</td></tr>
<tr><td colspan="3">Heat pipes incorporated</td><td>4 wickless heat pipes without adiabatic section</td></tr>
<tr><td colspan="3">Evaporator length</td><td>1800 mm</td></tr>
<tr><td colspan="3">Condenser length</td><td>200 mm</td></tr>
<tr><td colspan="3">Diameter</td><td>22 mm</td></tr>
<tr><td colspan="3">Working fluids</td><td>ethanol, methanol and acetone, at 50%<br>fill charge ratio of the evaporator volume</td></tr>
<tr><td rowspan="8">Evacuated glass tubes [17]</td><td colspan="2">Type</td><td>Owens–Illinois type,<br>model 47-58-1800-YCF</td></tr>
<tr><td colspan="2">Material</td><td>high quality borosilicate glass</td></tr>
<tr><td colspan="2">Dimensions</td><td>length 1800 mm,<br>outer tube $\phi$ 58 mm,<br>inner tube $\phi$ 47 mm,<br>glass thick. 1.6 mm</td></tr>
<tr><td colspan="2">Vacuum</td><td>0.0133 Pa</td></tr>
<tr><td rowspan="4">Absorber</td><td>Coating</td><td>graded aluminum nitride/ aluminum</td></tr>
<tr><td>Transmittance</td><td>0.93</td></tr>
<tr><td>Absorptance</td><td>$> 95\%$</td></tr>
<tr><td>Emittance</td><td>7–8% (at 80 °C)</td></tr>
<tr><td colspan="3">Maximum storage capacity</td><td>60 l</td></tr>
<tr><td colspan="3">Reflector</td><td>aluminum sheets reflector</td></tr>
<tr><td colspan="3">Weather conditions</td><td>clear sky</td></tr>
<tr><td colspan="3">Load conditions</td><td>no load, intermittent load,<br>and continuous load</td></tr>
<tr><td colspan="3">No load condition</td><td>storage capacities of 40, 50, and 60 l</td></tr>
<tr><td colspan="3">Intermittent load condition</td><td>removal of 2, 3, 4, 5, 6, 8, and 10 l of hot water at the start of each hour from 10:00 a.m.−14:00 p.m.</td></tr>
<tr><td colspan="3">Continuous load condition</td><td>continuous hot water removal along the operation period with flow rates of 4, 5, 6, 7, and 8 l/h</td></tr>
</table>

on individual evacuated tube heat pipe solar systems revealed that, higher storage temperatures were recorded, at no load conditions, with small storage capacities, corresponding to higher stored heat and overall efficiency obtained with large storage capacities. Therefore, the storage tank of this system was designed to hold up to 60 l of water. This storage volume is expected to fulfill the hot water demand for 1 to 3 persons.

Figure 6 compares the mean tank temperature variation under various weather conditions. The differences in clear sky temperatures are attributed to the differences in solar insolation and the initial tank temperature. However, the system continued operation even under severely bad weathers. This is due to the effect of the employed evacuated tubes and the heat pipes that are capable of transporting large amounts of energy even at very small temperature differences. The variation of mean tank temperature with the storage capacity of hot water is shown in Fig. 7. The trend is similar to that of individual heat pipe solar systems [11]. The storage water temperature rises with time to reach maximum value at nearly 16:00 p.m. then decrease slightly at the end of the operation period. The mean tank temperature is higher with decreased storage capacity. The maximum mean tank temperatures attained were 78.4 °C, 75.5 °C and 72.8 °C at 40, 50, and 60 l storage capacities, respectively. The overall stored energy of the system increased with the larger storage capacity from 7135.6 kJ to

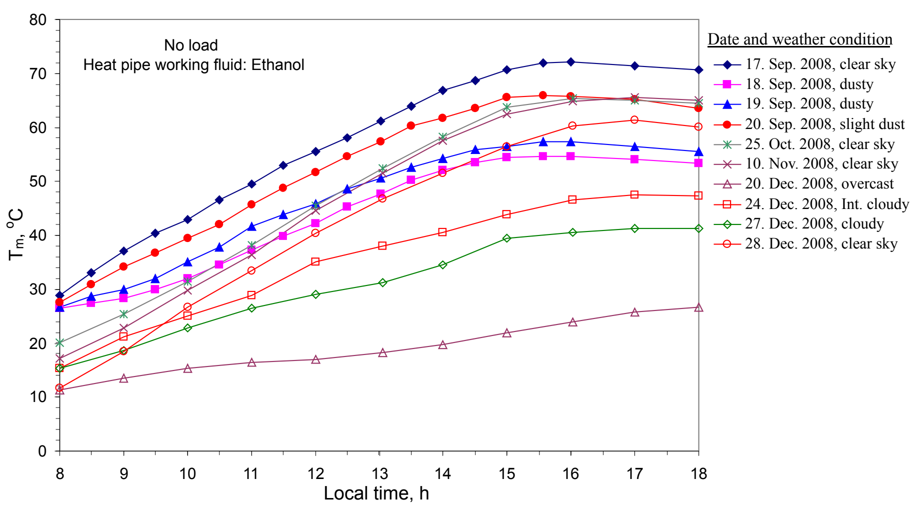


Figure 6: Variation of mean tank temperature of evacuated tube heat pipe solar water heating system under various weather conditions.

9358 kJ with increased storage capacity which also increased the overall efficiency from 50.06% to 71%, as shown in Fig. 8. However, a disproportionally low increase in overall efficiency for the 50 l storage capacity reveals that, at no load condition, the thermal losses from the storage tank are not proportionally related to the storage capacity.

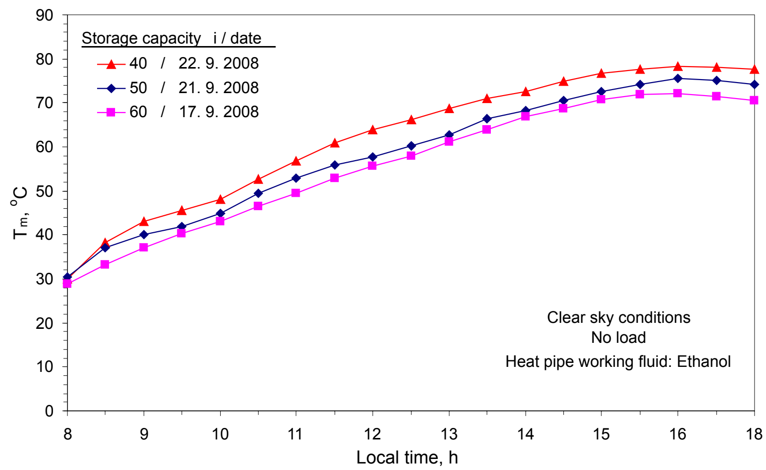


Figure 7: Variation of the mean tank temperature of the evacuated tube heat pipe solar water heating system at various storage capacities.

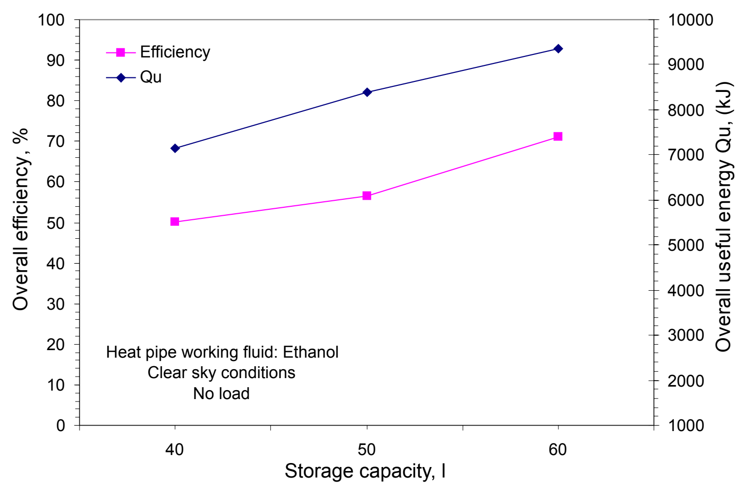


Figure 8: Effect of storage capacity on the overall useful energy and efficiency of the evacuated tube heat pipe solar water heating system.

The influence of intermittent hot water withdrawal on the storage water temperature throughout the operation period with the seven removal quantities is shown in Fig. 9. The trend is similar to that observed previously with the individual heat pipe solar systems [11]. The mean tank temperature increases gradually with time from the beginning of operation period, drops suddenly after each withdrawal process, and increases slightly after removing the load, then increases continuously after the last withdrawal process until the end of the operation period where maximum value is reached. The sudden temperature drop of the storage temperature ranged from 0.5 °C to 5.6 °C, depending on the amount of hot water withdrawn and is attributed due to the replenishment amount of cold water entering into the storage tank. The maximum values of the mean temperature, $T_m$, attained at the end of operation period were ranging from 59.4 °C to 69.1 °C, depending on the hot water removal quantities and the supply water temperature which is a good system response to loading. The variation of daily overall useful energy gain and efficiency with intermittent loading was from 11561.3 kJ and 63.5% to 12013.84 kJ and 72.9%, as shown in Fig. 10. A decrease is observed with removal quantities higher than 4 l due to a high drop in the mean temperature caused by the entry of cold mains water, as was discussed previously. Lower mean tank temperatures than

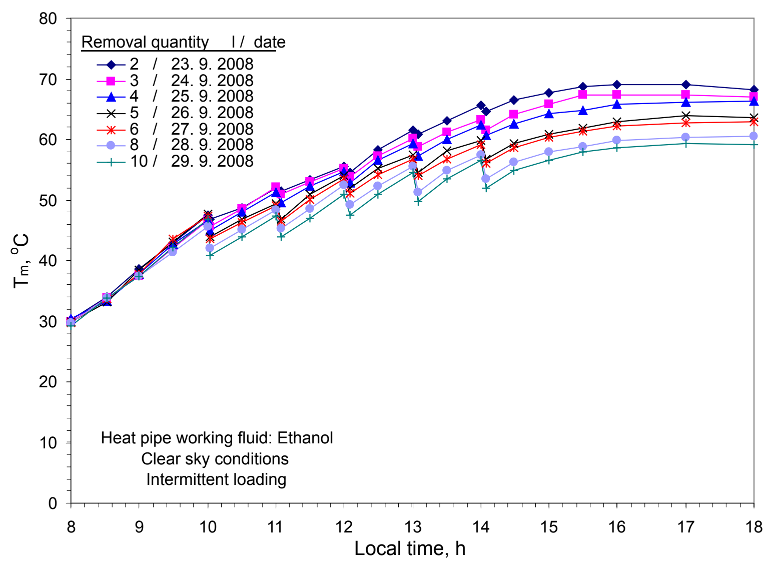


Figure 9: Effect of intermittent loading on the mean tank temperature variation of the evacuated tube heat pipe solar water heating system.

with no load were attained with loading. Higher daily overall useful energy and efficiency by 28.4% and 8.4%, respectively than no load condition were attained with intermittent load condition.

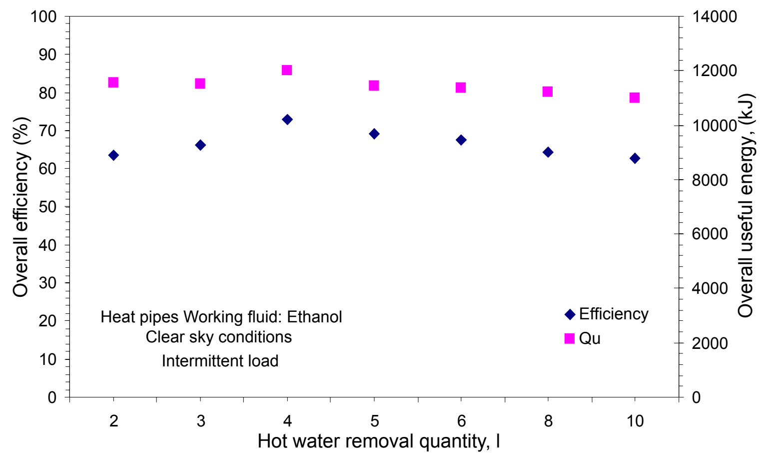


Figure 10: The effect of intermittent loading on the overall useful energy and efficiency of the evacuated tube heat pipe solar water heating system.

Figure 11 shows the variation of the mean temperature at various continuous hot water removal rates. The trend is similar to that observed previously. Maximum values of the mean temperature ranged from 55.8 °C to

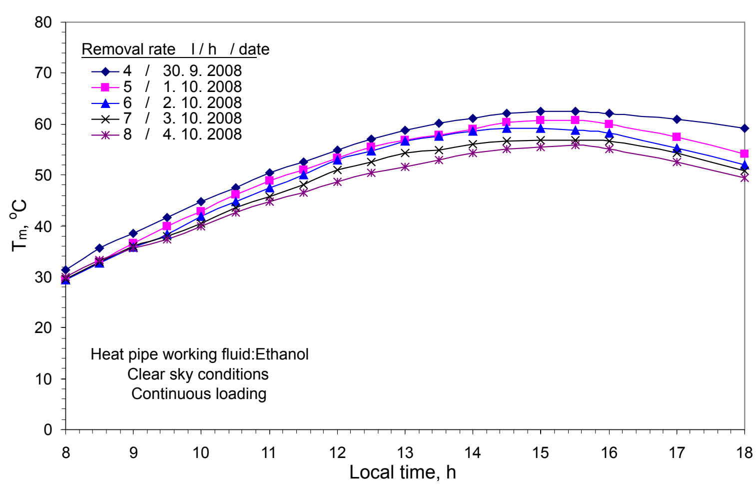


Figure 11: The effect of continuous loading on the mean tank temperature variation of the evacuated tube heat pipe solar water heating system.

62.4 °C, depending on the hot water removal rate. The mean tank temperature decreased with increased removal rate because of the corresponding continuous replenishment of cold water entering the storage tank. The overall useful energy and the overall efficiency varied from 10900 kJ and 72.3% to 11520 kJ and 79.5% with removal rates, as shown in Fig. 12. Thus, the system shows better performance with hot water withdrawal conditions than without hot water withdrawal. Higher overall useful energy gain and overall efficiency by 23.1% and 58.8%, respectively were attained by the system with hot water withdrawal. This is attributed to the advantage of evacuated tube heat pipe solar collectors that have higher response and energy transport capability.

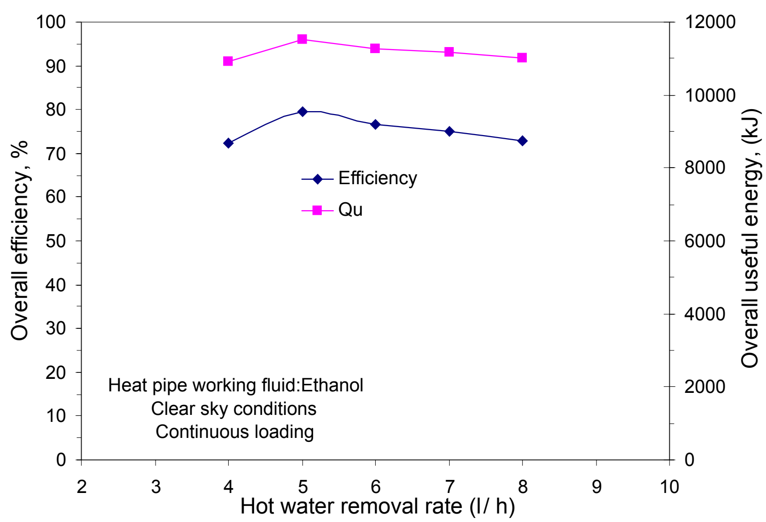


Figure 12: Effect of the continuous load on the overall useful energy and efficiency of the evacuated tube heat pipe solar water heating system.

To investigate the effect of heat pipe working fluid on the solar system performance, the system heat pipes were evacuated and recharged with methanol and acetone. Experiments were then repeated with each working fluid, with the three hot water withdrawal (loading) patterns. Figures 13, 14, and 15 compare the effect of heat pipes working fluids on the variation of mean tank temperature at the three load conditions. At no load condition and 60 l storage capacity (Fig. 13), the trend is similar for both methanol and acetone to that with ethanol. There is a slight difference in the maximum value of the mean temperature which was 72.8 °C, 73.4 °C, and 71.2 °C for ethanol, methanol and acetone respectively. The slight difference is attributed to the slight difference in the physical properties of the working fluids. For intermittent load condition with a removal quantity of 4 l of hot water (Fig. 14), the maximum values of the mean temperature were 65.5 °C, 67.4 °C and 66.6 °C with ethanol, methanol, and acetone, respectively, which are slightly lower than values attained with no load as expected. The variation of mean tank temperature for continuous load con-

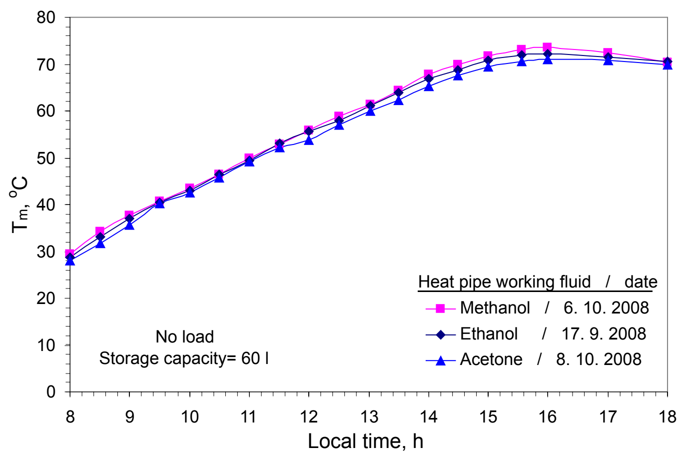


Figure 13: Effect of working fluid on the mean tank temperature at no load for the evacuated tube heat pipe solar water heating system.

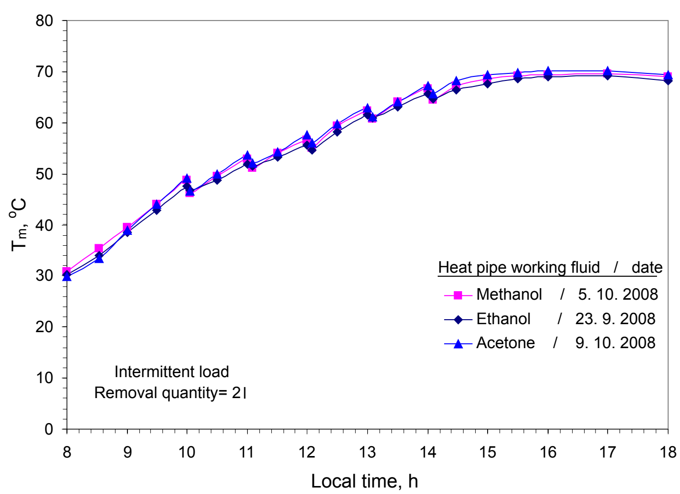


Figure 14: Effect of working fluid on the mean tank temperature at intermittent load for the evacuated tube heat pipe solar water heating system.

dition with hot water removal rate of 5 l/h (Fig. 15) with the three heat pipe working fluids shows similar trend and the maximum values of the mean temperature attained were 60.7 °C, 61.5 °C and 61 °C with ethanol, methanol and acetone, respectively. These are lower values than values for intermittent loading because of the continuous load. Figure 16 shows the overall useful heat gain of the system with the three working fluids. Only slight differences are observed due to the small differences between physical properties of these fluids. This result agrees with the experimental results

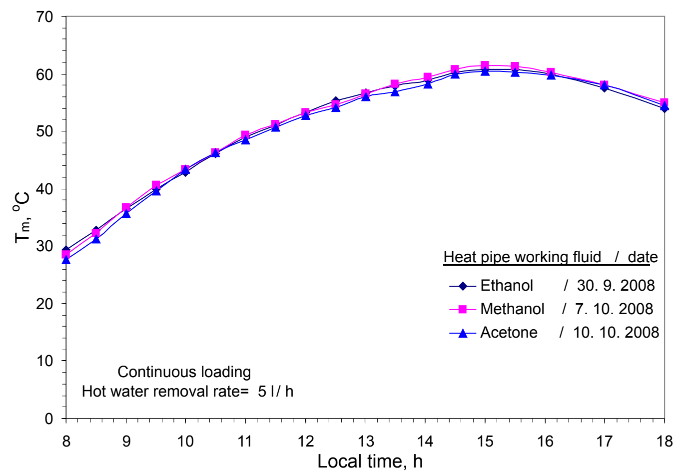


Figure 15: Effect of working fluid on the mean tank temperature at continuous load for the evacuated tube heat pipe solar water heating system.

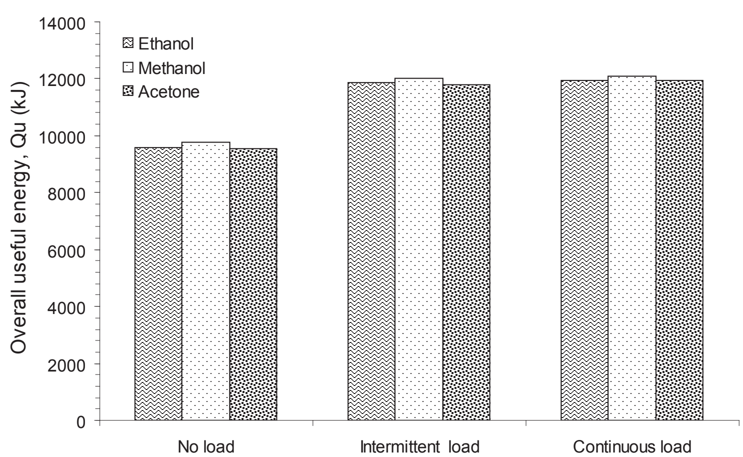


Figure 16: Overall useful energy of the heat pipe with the three working fluids.

obtained by Chun *et al.* [3]. It was also observed that, with a wide range of working fluids, the system performs better with loading conditions than without loading condition. Comparatively high storage water temperature rises were attained corresponding to 60 °C to 85 °C attained by other systems incorporating heat pipe arrays of more than 20 heat pipes [1, 3, 24, 25], and other conventional evacuated tube solar hot water systems not incorporating heat pipes [26–29].

An accurate theoretical analysis of the evacuated tube heat pipe solar water heating system [10] requires application of energy balance to each element of the system components, since each element has its own heat capacity. Account must be taken for both longitudinal and azimuthal variation of temperatures in each system component, and the effect of the solar insolation, ambient temperature, wind speed and the heat accumulation inside the evacuated tube heat pipe solar collector must be considered. Such an analysis requires complicated iterative solution procedures. Therefore, simplifying assumptions were considered at the expense of the accuracy of results. The performance of the solar hot water system under study was obtained theoretically by the electrical analogy technique. The analysis is similar to other models [28, 10] proposed for the evaluation of commercial solar hot water systems incorporating heat pipes, using conventional evacuated glass tubes and attached to a manifold.

The results are shown in Fig. 17 for no load condition and 60 l storage capacity, Fig. 18 for intermittent load condition with 4 l removal quantity

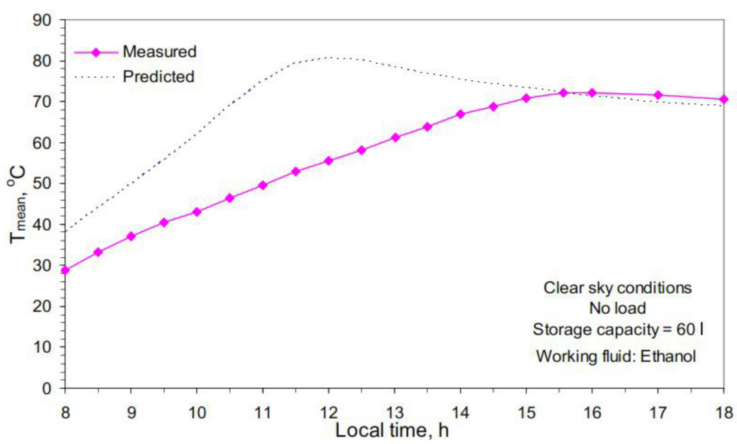


Figure 17: Comparison of measured with predicted mean tank temperature variation of the evacuated tube heat pipe solar water heater with no load.

and Fig. 19 for continuous load condition with 5 l/h removal rate. A difference is noticed between predicted and measured temperatures but the trend is similar for the three load patterns. This difference is attributed to many reasons. The model assumes that all of the absorbed solar energy is transferred directly to the storage tank *via* heat pipe action and neglects heat losses from the system components except the storage tank. Any heat accumulation in the evacuated glass tube was neglected. Therefore, the mean tank temperatures are observed to increase with increased solar radiation, reach maximum values in the period of maximum solar radiation, and then decrease with decreased solar radiation, i.e., time lag is not evident in the model. Whereas, the measured temperature variation reveals that a heat accumulation exists in the evacuated tube solar collector. Also, the theoretical assumptions neglect the temperature gradients of the system components. The heat transfer correlations themselves may contribute to the differences. Since correlation equations are based on experimentally determined data, they do not always provide very accurate predictions of heat transfer coefficients. Errors as high as 25% are not uncommon [20].

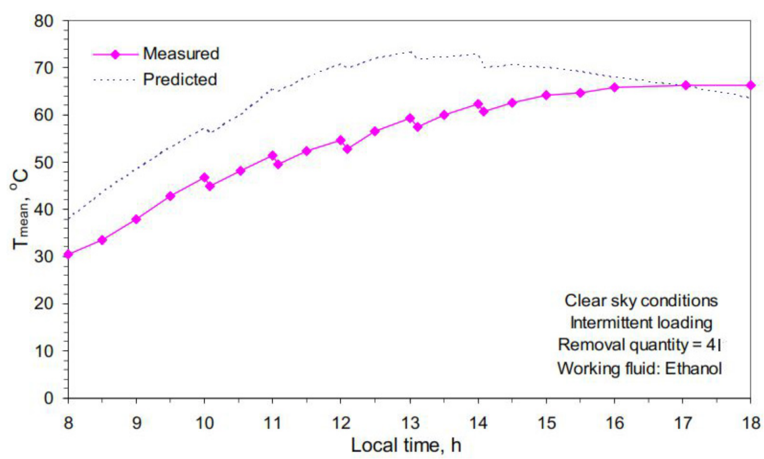


Figure 18: Comparison of measured with predicted mean tank temperature variation of the evacuated tube heat pipe solar water heater with intermittent load condition.

The differences in the overall useful energy between experimental and theoretical results of 8.5%, 8.3%, and 8% with no hot water withdrawal, intermittent and continuous hot water withdrawal patterns, respectively, as shown in Fig. 20.

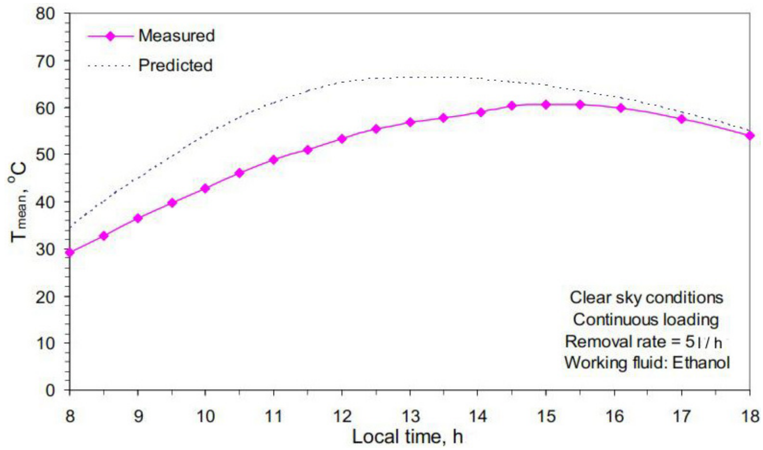


Figure 19: Comparison of measured with predicted mean tank temperature variation of the evacuated tube heat pipe solar water heater with continuous load condition.

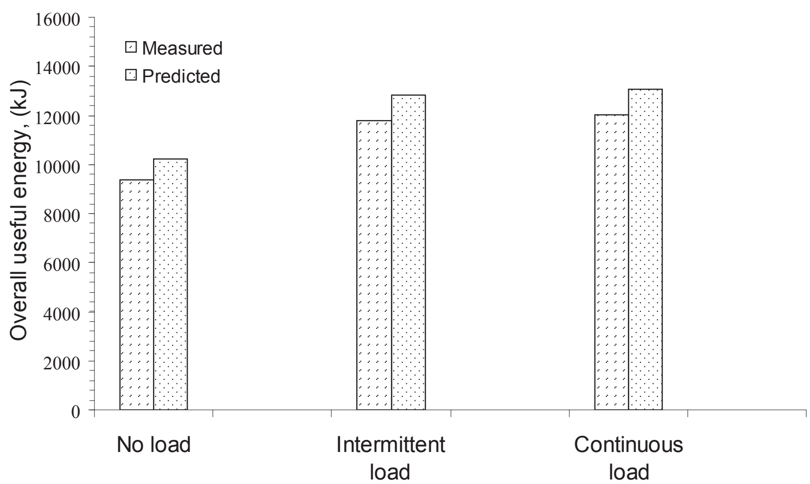


Figure 20: Predicted and measured overall useful energy for the evacuated tube heat pipe solar water heating system.

# 5 Conclusions

1. Water temperature increased with decreased storage capacity, and the overall daily energy stored and the overall efficiency increased with decreased storage water capacity.

2. Higher overall useful energy and overall efficiency and lower water temperature were obtained with load conditions than with no load condition.

3. The system performance was not significantly affected by the type of heat pipes working fluid for the range of heat pipe working fluids used in the present work.

4. Differences in storage tank temperature between predicted and measured values with similar trend for the three load patterns.

5. An agreement of 8% is obtained between predicted and measured values of the overall useful energy.

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