International Journal of Renewable Energy Resources 1 (2011) 1-6

HEAT EXTRACTION FROM GRADIENT LAYER USING EXTERNAL HEAT EXCHANGERS TO ENHANCE THE OVERALL EFFICIENCY OF SOLAR PONDS

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ABSTRACTS

A salinity gradient solar pond is a combined solar collector and thermal energy storage system. In past heat has been successfully extracted from the Lower Convective Zone (LCZ) from working ponds for industrial process heat applications. Previous theoretical studies have shown that by extracting heat from the NCZ and LCZ together will improve the efficiency of a solar pond to 30% from the present 20% efficiency. This paper discusses concept designs and theoretical predictions of heat extraction from the Non Convective Zone (NCZ) using an external heat exchanger. Here, two methods of heat extraction from different levels within the NCZ using external heat exchanger are presented. A schematic drawing of the first method that uses pumps (forced convection) for heat extraction has been presented. The second method uses thermosyphon effect to transfer the heat from different levels in NCZ to the binary fluid. This paper presents theoretical modeling and experimental results for thermosyphon based heat extraction method. Later theoretical predictions and experimental results have been compared. This investigation shows good prospects for application of this system for heat extraction from NCZ of large solar ponds.

Keywords: Salinity-gradient solar pond; heat extraction; thermosyphon; heat exchangers.

NOMENCLATURE

$\dot{m}$	Mass flow rate of water (kg s^{-1})
C_p	Specific heat (J/kg.^{o}C)
ρ	Density (kg/m^3)
β	Thermal expansion (K^{-1})
Pr	Prandtl number
μ	Dynamic viscosity (N.s/m^2)
k	Thermal conductivity (W/m. ^{o}C)/ Losses
g	gravity (m^2/s)
ΔP	Pressure difference (kPa)
f	Losses due to friction
Re	Reynolds number
Nu	Nusselt number
A	Area (m^2)
D	Diameter of thermosiphon and heat exchanger coil(m)
d	Diameter of copper tube (m)
L	Length of thermosiphon pipe and heat exchanger coil (m)
l	Length of copper tube (m)
V	Velocity surrounding heat exchanger coil (m/s)
u	Velocity in thermosiphon pipe (m/s)
Th	Temperature of hot water (^{o}C)
Tc	Temperature of cold water (^{o}C)
h	Convection heat transfer coefficient (W/m^2.^{o}C)
Exp	Experiment
Theo	Theoretical

Subscripts

1	thermosiphon
2	heat exchanger coil
i	inlet
o	outlet
c	cold
h	hot
s	heat exchanger coil surface
d	copper tube

1. INTRODUCTION

A solar pond is a low-cost solar collector with long term thermal storage capability. It utilizes a large body of saline water with increasing density gradient from top to bottom that absorb solar radiation and stores the thermal energy (Kaushika, 1984). The solar pond technology has been used in Israel (Tabor and Doron, 1990) and USA (Swift et al., 1987; Xu et al., 1993) for power generation and low-temperature process heat applications respectively. Solar ponds have been simple and low cost solar energy system for a relatively longer period of time and they can be an alternative to fossil fuel for generation of electricity at locations close to saline water source like sea, a mining site and salt effected areas like in north Victoria, Australia. Figure 1 shows the schematic of conventional solar pond which consists of three regions (surface zone, gradient and lower zone). The surface or upper convective zone (UCZ) is a homogeneous layer of low salinity brine or fresh water. The gradient layer or non-convective zone (NCZ) constitutes a thermally insulating layer that contains a salinity gradient to suppress convection currents. The lower convective zone (LCZ) is a thermally convective zone with a homogeneous salt concentration and the brine in this region is much hotter than that in the top zone. Conventionally heat has been extracted from solar ponds solely from the LCZ by two common methods. The first method is by withdrawing the hot brine through an external heat exchanger as experimented in El Paso, Texas (Xu et al., 1993), Kutch in India (Kumar and Kishore, 1999), Beith Ha'arava in Israel (Tabor and Doron, 1990) and Singapore (Kho et al., 1991). The second method is by using an in-pond heat exchanger, where the cold water is used as heat transfer fluid as demonstrated at Pyramid Hill, Victoria (Leblanc et al., 2011) and Marshad in Iran (Jaefarzadeh, 2006) . Tabor (Tabor, 1981) has discussed both of these methods in his review of solar pond technology, and stressed that both method have convenient practical merit.

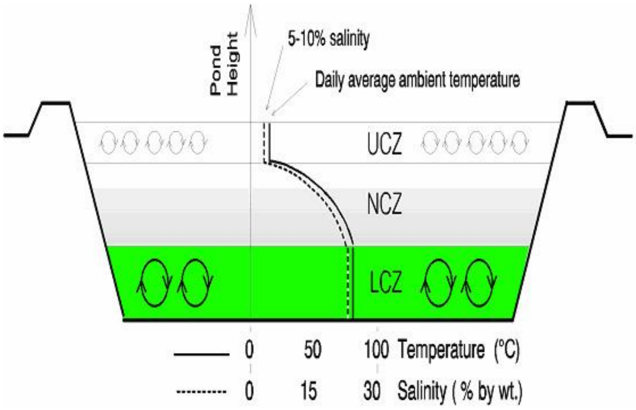


Fig. 1 The schematic of salinity gradient solar pond

The main concerns with solar ponds is the thermal performance to store and deliver the heat which is in average of 15-20% (Wang and Akbarzadeh, 1982). Many researchers have studied ways of improving the thermal performance of solar ponds. This could be done by improving the water clarity to allow better penetration of sunlight to the storage zone (Gasulla et al., 2011; Wang and Yagoobi, 1995), putting a reflector to reflect additional solar radiation into solar pond (Aboul-Enein et al., 2004) and by improving the method of heat extraction from LCZ (Akbarzadeh et al., 2005).

An alternative method of extracting heat from solar pond was investigated theoretically by Andrews and Akbarzadeh (Andrews and Akbarzadeh, 2005) with the aim of increasing the overall energy efficiency of collecting solar radiation, storing and delivering heat. In this method, heat is extracted from the NCZ as well as, or instead of, from the LCZ. The theoretical analysis showed that, heat extraction from the gradient layer has the potential to significantly reduce heat losses to the surface and consequently increase the overall energy efficiency of the solar pond by up to 50%. (Leblanc et al., 2011) have carried out an experimental study of the heat extraction process proposed by (Andrews and Akbarzadeh, 2005) at RMIT University, Victoria, Australia for a period of two months by using an in-pond heat exchanger made of polyethylene pipe. Results indicate that heat extraction from the gradient layer increases the overall efficiency of the solar pond by up to 55% compared with the conventional method of extraction from the LCZ. The result shows convection currents in the experimental study were found to be localised, further experimental studies using an external heat exchanger (which would be more practical for medium- to large-scale solar ponds) needed to be conducted. The similar study has been conducted by(Ould Dah et al., 2010) both numerically and experimentally for the 0.64 m^2 mini solar pond in Tunisia, where the result showed the improvement of the solar pond performance.

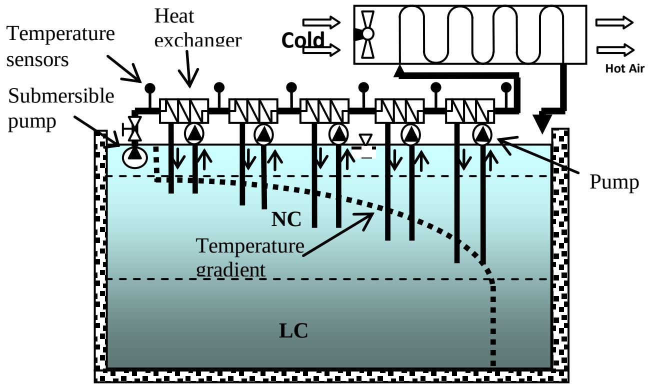


Fig. 2 Proposed heat extraction system from NC

Fig. 3 A 50m^2 experimental salinity gradient solar pond at RMIT University

Figure 2 shows the concept design for an active method of heat extraction from the NCZ of the experimental solar pond at RMIT University as shown in Figure 3. In this method brine is drawn through external heat exchangers using small pumps and re-injected back into the NCZ at the same level as shown in Figure 2. This research paper tries to prove the concept of using thermosyphon system for heat extraction from NCZ of a solar pond. This paper further presents the theoretical modeling and experimental results for thermosyphon heat extraction system. Later the experimental results from the prototype of thermosyphon heat extraction system have been compared with the theoretical prediction for computer model validation.

2. THEORETICAL MODELLING

Thermosiphon refers to a passive heat exchanger based on a temperature differential with gravity causing the liquids to circulate by density variations without the need for a mechanical pump. Tundee et al. (2010) and Singh et al. (2011) have applied the similar concept of gravity assisted heat exchanger in their experiments to remove the heat from solar pond and potentially produce the electricity by combining it with thermoelectric modules. The current modelling consists of 15 equations with 15 of the following unknowns were solved using a Visual Basic-Excel computer program.

$$\begin{pmatrix} u, V, h_i, h_o, Tc_o, Th_o, T_s, \Delta P, f_1, f_2, \mathrm{Re}_1, \mathrm{Re}_2, \mathrm{Re}_d, \\ Nu_o, Nu_i \end{pmatrix}$$

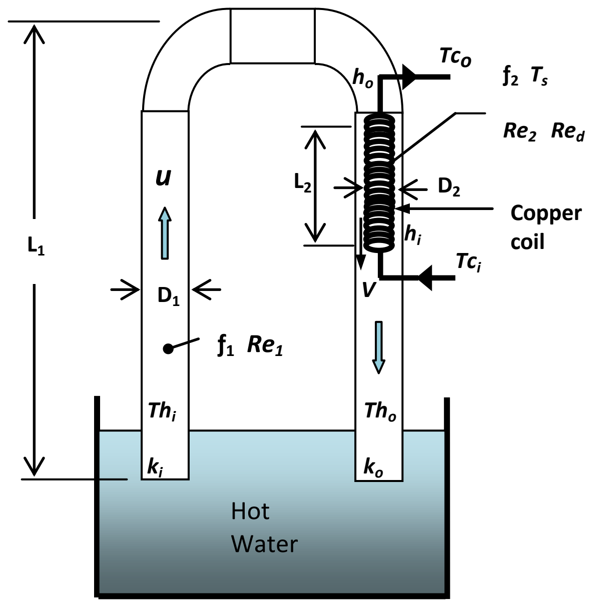


Fig. 4 Thermosiphon heat exchanger system schematic diagram

The following mathematical model of the system was derived to determine the performance of the heat exchanger, based on the thermosiphon heat exchanger system as shown in figure 4. The energy balance in the system is defined through equation (1), considering the heat flow into the cooling coil and the heat taken away by the cooling water by equation (2) and the heat transfer rate due to forced convection between the hot water outside the coil wall and inside the coil by equation (3).

$$\dot{m} C_p (Tc_o - Tc_i) = \rho u \frac{\pi}{4} D_1^2 C_p (Th_i - Th_o) \tag{1}$$

$$\dot{m} C_p (Tc_o - Tc_i) = \pi d l h_o \left[\left(\frac{Th_i + Th_o}{2} \right) - T_s \right] \tag{2}$$

$$\left[T_s \left(\frac{Tc_i + Tc_o}{2} \right) \right] h_i = \left[\left(\frac{Th_i + Th_o}{2} \right) - T_s \right] h_o \tag{3}$$

The difference of velocity due to the restriction by cooling coil in the system is given by;

$$uA_1 = V (A_1 - A_2) \tag{4}$$

The available pressure due to buoyancy and the pressure loss using Darcy's pressure drop correlation in the system are expressed by equations (5) and (6).

$$\Delta P = [\rho.g.\beta (L_1 - L_2)(Th_i - Th_o)] + \left[\rho.g.\beta.L_2 \left(Th_i - \left(\frac{Th_o + Th_i}{2} \right) \right) \right] \tag{5}$$

$$\Delta P = \left[f_1 \left(\frac{2L_1 - L_2}{D_1} \right) \frac{1}{2} \rho u^2 \right] + \left[f_2 \left(\frac{L_2}{D_2} \right) \frac{1}{2} \rho V^2 \right] + \left[k_i \frac{1}{2} \rho u^2 \right] + \left[k_o \frac{1}{2} \rho u^2 \right] \tag{6}$$

Where, the friction factors in fully developed flow in the thermosiphon and coil are given by equations (7) and (8).

$$f_1 = (0.79 \ln \mathrm{Re}_1 - 1.64)^{-2} \qquad 3000 < \mathrm{Re}_1 < 5x10^6 \tag{7}$$

$$f_2 = (0.79 \ln \mathrm{Re}_2 - 1.64)^{-2} \qquad 3000 < \mathrm{Re}_2 < 5x10^6 \tag{8}$$

The Reynolds numbers for the flow in thermosiphon and through the circular coil are given by equations (9) and (10).

$$\mathrm{Re}_1 = \frac{\rho u D_1}{\mu} \tag{9}$$

$$\mathrm{Re}_2 = \frac{\rho V D_2}{\mu} \tag{10}$$

The Nusselt numbers due to convection heat transfer from the coil wall and inside the copper tube are given by equations (11) and (12).

$$Nu_o = 0.913 \, \mathrm{Re}_1^{0.618} \, \mathrm{Pr}^{1/3} \tag{11}$$

$$Nu_i = 0.0243 \, \mathrm{Re}_d^{4/5} \, \mathrm{Pr}^{0.4} \tag{12}$$

Where, the Reynolds number inside the copper tube is;

$$\mathrm{Re}_d = \frac{4\dot{m}}{\pi d \mu} \tag{13}$$

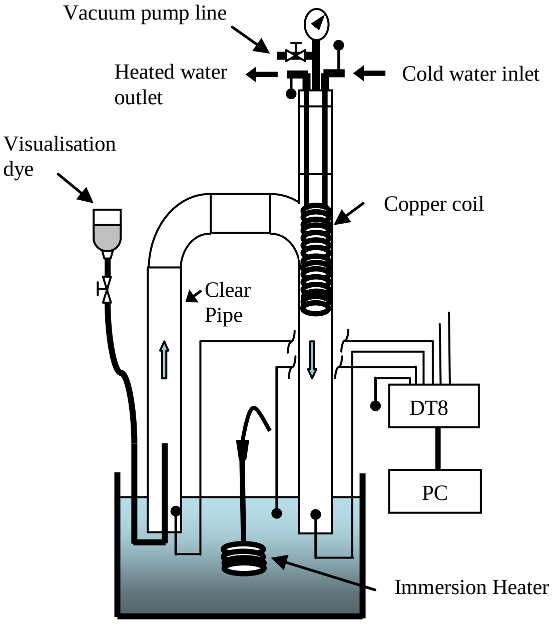
The convection coefficient heat transfer inside, h_i and outside, h_o of the copper tube are expressed from equation (14) and (15) respectively.

$$Nu_i = \frac{h_i d}{k} \tag{14}$$

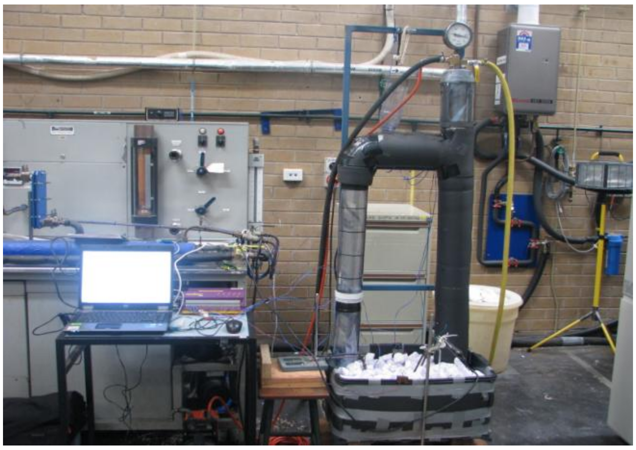
$$Nu_o = \frac{h_o D_2}{k} \tag{15}$$

3. Experimental Study

An experimental study was conducted to measure the actual performance of a thermosiphon heat exchanger. The arrangement is shown in Figure 5. The water will rise to fill the U-shape siphon as the vacuum pump is turned on. The vacuum pump is shut off when the water reached the desired level, where the pressure is approximately -10 kPa. The data acquisition system is switched on for five minutes to check the stability of all the temperatures before the heater is turned on. After five minutes, the cold water valve is opened to allow tap water to flow through the heat exchanger coil and the flow rate in is measured. As the tank water gets hotter, it starts to rise due to the natural convection process, and as it flows through the heat exchanger, it will lose some heat making it colder. Therefore it will fall back to the tank, and the cycle is repeated. The outcome from this process is that the temperature of the cold water outlet will rise to create a temperature difference between inlet and outlet. A monitoring system using a Data-Taker (DT800) data acquisition system was used to allow continuous temperature measurements for six different locations. Thermocouples were placed at the inlet and outlet of the cold water; inlet and outlet of the hot water on the siphon; in the tank; and next to the rig to measure the ambient temperature.



(a)



(b)

Fig. 5 (a) Schematic diagram of thermosiphon heat exchanger (b) Thermosiphon heat exchanger experimental set-up

A flow visualisation technique, adopted from (Lim, 2000), was used to measure the velocity of the hot water in the U-shaped thermosiphon by injecting dye into the system. The dye is injected, while the water is circulating in the system, to the siphon pipe through a small tube. A Canon digital camera was used to capture the movement of the dye in the video mode. A stopwatch was used to record the time as the dye travel passed by the marked scale with 0.1 m interval on the clear pipe section. By analysing the series of video obtained, the distance travelled by the dye during a particular time was calculated in order to determine the flow velocity in thermosiphon.

4. RESULTS AND DISCUSSION

Theoretically the system is initiated with input values of the physical geometry, the water properties with relevant coefficient of loss factor and the temperature of hot water and cooling water. The system has been modelled with the effect of cooling water mass flow rate through the heat exchanger coil of 0.033 L/s, 0.04 L/s, 0.05 L/s, 0.067 L/s and 0.1 L/s. The experiments have been carried out accordingly applying the same mass flow rates. As a result, the input temperatures of hot and cold water were obtained as the system is reaching the steady states as shown in table 1. These temperatures becomes an input values to the theoretical modelling.

Table 1 Temperatures outcome with variation of cooling water mass flow rates.

Cooling water flow rate (L/s)	Thi, ^{o}C	Tci, ^{o}C
0.033	40.11	17.06
0.04	38.28	16.6
0.05	36.66	16.89
0.067	34.05	16.72
0.1	32.34	16.55

The obtained results from theoretical and experimental were compared and are shown in figure 6 to figure 8. Figure 6 shows the temperature differences of cold and hot water for theoretical and experimental respectively. Temperature difference of cold water are getting decreased with the increased of the cooling water mass flow rates, approximately 16 ^{o}C for flow rate of 0.033L/s and reduced to around 5 ^{o}C at flow rate of 0.1 L/s theoretically. This phenomenon clearly due to the effect of the mass flow rates, where the lower the flow rates will caused more heat to be absorbed by the cooling water inside the coil and vice versa. The results from experiments shows slightly higher temperature on the output, but it curve proved the similar phenomenon is achieved. The hot water temperature difference is remains constant approximately of 1.2 ^{o}C with the variations of cooling water mass flow rates, for both experimental and theoretical. This condition is due to the effect of the constant hot water temperature in the tank, where the return hot water will gradually gain the heat again before it re-enter the thermosiphon heat exchanger.

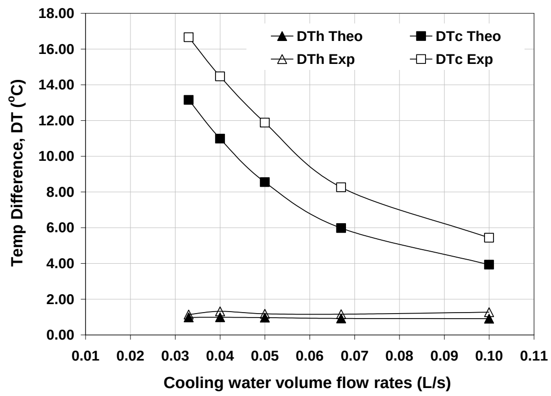


Fig. 6 Temperature difference of cooling water and hot water

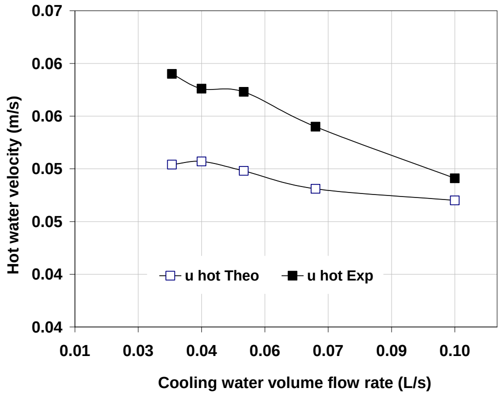


Fig. 7 Variation of hot water velocity in thermosiphon

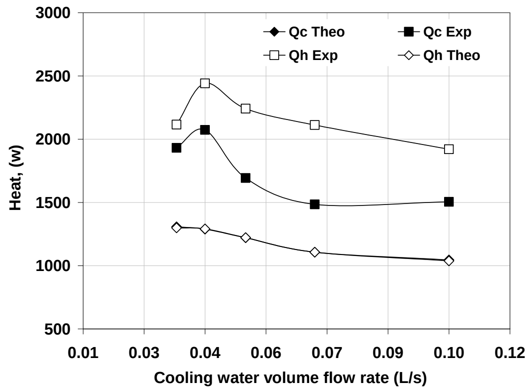


Fig. 8 Energy Extracted from variation of cooling water mass flow rate

From table 1 and figure 7 it can be seen that the velocity of hot water in the thermosyphon increases with increased temperature difference between hot water simulating NCZ and cooling water. This increase in the velocity of hot water will help enhance the heat transfer rates and hence for a constant heat extraction rates the required heat transfer area can be reduced for higher temperature regions of NCZ. The amount of heat extracted by the cooling water and the heat removed from the hot water theoretically and from experiment is shown in figure 8. Apparently, the theoretical results show the balance of the heat transfer phenomenon with heat which has been extracted by the sooling water and heat that had been removed from the hot water. This is to prove the validity of the modelled equations for this thermosiphon system. But, the experimental result shows a larger difference between heat extracted by cooling water and the heat removed from the hot water, approximately 8%-12% compared to theoretical result. The results are proportional to the obtained velocities as discussed previously. This is understandable, as velocity is one of the parameters to determine the quantity of the energy produced. Collectively, all results in figure 6, 7 and 8 show a good agreement between theoretical and experiments. The sensitivity of velocity of hot water, u though with small changes may lead to large discrepancy on the heat extracted by cooling water and heat removed from the hot water. As an example, with the difference of 0.001 m/s of hot water velocity with regards to the cooling water mass flow rate of 0.033 L/s and 0.04 L/s resulted a difference of 142 W of heat removed from the hot water as shown in figure 8.

5. CONCLUSIONS

The concept design and method of using thermosyphon heat extraction system has been proved through theoretical and experimental analysis in this paper. The theoretical modeling of the system has shown the validity of the system where the balance of the energy was obtained from the governed equations. Later the experimental results from the prototype of thermosyphon heat extraction system have been compared with the theoretical prediction for validation. It has shown a good agreement with the theoretical modeling, where the velocity of hot water in the thermosyphon increases with increased temperature difference between hot water and cooling water. This phenomenon will enhance the heat transfer rates and provide the guidance to optimize the design of the thermosyphon heat exchanger for higher temperature. In future validated computer model can be used for design of a NCZ thermosyphon heat extraction system from a $50m^2$ solar pond at RMIT University.

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