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SYSTEM MODELING AND CHARACTERISTICS OF A HYBRID SOLAR COLLECTOR

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ABSTRACT

A validated model for simulation of thermosyphon solar collectors performance is introduced in this work. An empirical correlation for flow resistance in loop is used. The good consistency between exeriment and simulation has proved that this method is applicable in such thermosyphon problems. A new design of a hybrid solar collector has been presented and constructed with locally available materials. The new design has improved the collector efficiency from 60 percent to 68 percent and enhanced the mean tank temperature by 11.4°C at peak solar insolation.

INTRODUCTION

Numerous studies have been performed on thermosyphon solar water heating systems since they have the advantage of avoiding a water pump for circulating water in the collector. Most thermosyphon designs have been developed by trial and error, since the complexity of varying collector flowrate and a thermally stratified storage tank mean that the well established design methods for active systems can not be applied. Extensive research on the performance analysis of this collector was carried out during the last decade. However, a simple simulation method suitable for thermal performance prediction in a longterm scale is still limited.

Close (1967) first reported a model to describe the performance of the solar thermosyphon collector. With the concept of system mean temperature and the assumption of linear temperature distributions in absorber and tank, Close was able to calculate the performance of collector in a single day. However, a number of defects in Close's method still exist. For example, all the basic performance characteristic parameters of the absorber rely on theoretical calculations.

Mertol et al. (1981) designed a detailed loop model to study the performance of solar thermosyphon water heater with heat exchangers in storage tanks. This model is limited in its applications in long-term simulation. Stasa and Singh (1982) developed a computer programme for the design and analysis of thermosyphon solar collectors. Since no experimental verifications were made, the accuracy in long-term performance analysis by use of this programme needs to be further examined .

The enhancement of solar radiation, by plane reflectors, on flat-plate collectors is of great importance, due to its potential applications on various solar energy thermal devices. The enhanced input of solar energy finds applications, where higher-grade thermal output is to be achieved as in solar cookers (1980), solar water heaters,for heating water to specified higher temperatures etc. A comprehensive study on a system consisting of flat-plate collectors augmented by plane reflectors, is strongly needed.

The first aim of this study is to predict a model for simulation of the performance of any flat-plate solar collector and secondly to test the possibility of improving its performance by a low-cost technique such as the hybrid design

MATHEMATICAL MODEL

The typical thermosyphon collector can be divided into four parts: absorber, storage tank, riser and downcomer. The rise of water temperature inside the absorber due to the absorption of incident solar radiation gives rise to water uprising motion into the tank and generates a circulation loop through the whole collector.

System Modeling

The Hottel-Whillier-Bliss theory can applied to calculate the the water temperature distribution T_w in the absorber. Thus the following simple exponential relation results Deffie and Beckman (1980) .

$$T_w(x) = T_a + \frac{H\ F'(\tau\alpha)n\ Ka}{F'\ \ U_L} + [T_1 - T_a - \frac{H\ F'\ (\tau\alpha)n\ Ka}{F'\ \ U_L}$$

$$\exp\ (-\ \ F'\ U_L\ x\ /\ GC_p\ L\ ) \qquad (1)$$

where x is the absorber coordinate , T_a is ambient temper-ture, T_1 is the absorber inlet water temperature, H is the hourly mean global soalr radiation on horizontal surface, F' is the absorber geometric efficiency factor, $(\tau\alpha)_n$ is the transmittance-absorbtance product of absorber at solar noon, U_L is the overall heat loss coefficient of absorber, G is the mass flowrate per unit absorber area, C_p is heat capacity of water, L is the longitudinal length of absorber, and

$$K_a = 1 - b_o\ (\ 1\ /\ \cos\psi\ - 1\ ) \qquad (2)$$

F' $(\tau\alpha)_n$ and F' U_L are the absorber characteristic para-meters which can be obtained from solar-noon efficiency test following ASHRAE standard (93 -(1977). K_a is the absorber incident angle modifier which is related to the sun angle, (b_o is the coefficient of incident angle modifier

modifier which also can be obtained from absorber standard test. ψ is the solar incident angle normal to absorber. Equation (1) thus represents the approximate performance eqn of the absorber.

The effect of thermal stratification phenomenon in the storage tank was investigated by several researchers as (Close 1967 and Cabelli 1977). Assume that the tank can be divided into N fully mixed sections. Applying energy balance to the ith section and considering the heat conduction between two adjacent sections, we obtain a series of differential equations, for tanks without heating element inside it;

$$M_i C_p \frac{d\theta_i}{d t} = \alpha_i m C_p (T_3 - \theta_i) + \beta_i m_L C_p X (T_L - \theta_i) + (UA)_i (T_a - \theta_i)$$

$$+ \begin{cases} \gamma_i C_p (\theta_{i-1} - \theta_i), & \text{if } \gamma_i > 0 \\ \gamma_{i+1} C_p (\theta_i - \theta_{i+1}), & \text{if } \gamma_{i+1} < 0 \end{cases}$$

$$+ (K_f A_s N/h_t) [\epsilon_{Ni} (\theta_{i+1} - \theta_i) + \epsilon_{1i} (\theta_{i-1} - \theta_i)], \quad (3)$$

Where M_i is the water mass in the ith section, m is the mass flowrate in the thermosyphon loop, m_L is the mass flowrate of hot water load. θ_i is the water temperature in the ith section of tank , T_3 is the temperature at the entrance of the tank, T_L is the temperature of make-up water . $(U\,A)_i$ is the overall heat loss coefficient of its ~~section.~~ A_s is the cross-section area of tank. h_t is the height of water in tank . N is the number of tank sections. $\epsilon_{Ni} = \epsilon_{1i} = 1$ for i= 2,3,...., N-1. α_i, β_i and γ_i are the control functions defined as follows;

$$\alpha_i = \begin{cases} 1 , & \text{if } i = s_h \\ 0 , & \text{otherwise} \end{cases} \qquad (4)$$

where s_h is the number of tank section to which the water from the absorber (at T_3) is closest in temperature,

$$\beta_i = \begin{cases} 1 , & \text{if } i = s_L \\ 0 , & \text{otherwise} \end{cases} \qquad (5)$$

where s_L is the number of tank section to which the make-up water (at T_L) is closest in temperature,

$$\gamma_i = m \sum_{j=1}^{i-1} \alpha_j - m_L \sum_{j=i+1}^{N} \beta_j \qquad (6)$$

When the net mass flowrate through the tank is not small, the conduction term on the right-hand side of eqn (3) can be neglected.

The thermosyphon head D_{th} induced in the thermosyphon loop can be evaluated by integrating the specific gravity (SG) around the loop and the resultant eqn is

$$D_{th} = \int_{0}^{h} [\ SG\ (\theta) - SG\ (\ T\ )\ ]\ dz \qquad (7)$$

where for water $SG\ (T) = -4.05 \times 10^{-6}\ T^2 - 3.906 \times 10^{-5}\ T + 1.0002556$, h is the highest water level in tank from ground and z is the elevation.

Since the frictional head in the thermosyphon loop usually consists of two parts: one represents the pipe wall friction loss and the other represents the losses due to valves, bends and fittings, ect. in the loop (Huang 1980), it can be assumed that the following quadratic function can be used to correlate the flow resistance head D_f;

$$D_f = B_o\ \nu\ m\ +\ B_1\ m^2 \qquad (8)$$

where ν is the kinematic viscosity of water. The coefficients B_o and B_1 can be easily determined on site by directly measuring the variation of resistance head with flowrate.

Since the heat transfer in thermosyphon

collectors is in general a slow process, the inertia effects can be ignored and the momentum eqn for the loop then becomes;

$$B_o \nu m + B_1 m^2 = \int_0^h [\ SG(\theta) - SG(T)]dz \qquad (9)$$

Neglecting the thermal resistance due to the pipe wall, the overall heat transfer coefficient, U, based on the outer surface area of the pipe can be determined by the following eqn:

$$U = \frac{1}{d_o / h_i d + \dfrac{d_o \ \ln(d_o / d)}{2 K_t} + 1/h_o} \qquad (10)$$

where d_o and d are the outside diameters of the insulated and bare connecting pipes respectively, h_i is the convective heat transfer coefficient inside the pipes evaluated by the empirical eqn given by Sieder and Tate (1936) ,

K_t is the thermal conductivity of pipe insulating material, and h_o is the convective heat transfer coefficient outside the insulated pipe.

Equations (1), (3) and (9) can be simultaneously solved by computer to give temperature distribution and flowrate The instantaneous mean tank temperature $\bar{T}$ is defined as ;

$$\bar{T} = (1 / N) \sum_{i=1}^{N} \theta_i \qquad (11)$$

The overall efficiency is defined as ;

$$\eta = \frac{\int_0^{t_f} [\, \rho C_p M \frac{d\bar{T}}{dt} + \rho C_p m_L (\theta_1 - T_L) \,)\, dt}{\int_0^{t_f} H A_c \, dt} \qquad (12)$$

where t_f is the performance termination time , ρ is the water density and A_c is the absorber surface area.

EXPERIMENT

A hybrid solar collector is designed in the Solar Laboratory, Physics Department, Faculty of Science , Tanta University, and constructed with the help of the University Workshop. It consists of a single-glazed flat-plate absorber painted matt-black and two perpendicular edge to edge plane reflectors at the back and bottom of the absorber plate as shown in Fig.1.a. The F.P. absorber is of copper sheet-and-tube construction in which the headers constitute a negligible portion of the collector area and provide uniform flow to the risers as shown in Fig. 1.b. The collector is unshaded and no additional dust or dirt collects on the cover during the insolation period. The two headers are collecting tubes of 25 mm outside diameter and 21 mm internal diameter. The total harnessing area of the F.P. collector is

1 m^2. It possesses 10 collector tubes of 13 mm outside diameter and 11 mm inside diameter. The F. P. C. is mounted horizontally. The plastic upriser and downcomer are thermally insulated with foamed plastic.

The storage tank has been constructed from 1 mm thick galvanised iron sheet and has the capacity of 0.120 m^3. The uprised disscharges into the tank at a distance of 30 mm below its top, and the downcomer is connected at the level of 30 mm above the tank base.

The acquired data in these experiments include the hourly values of the following parameters: temperatures at various heighs in the system, the thermosyphonic flowrate, the insolation intensity on the plane of the collector and the ambient temperature. Copper-constantan thermojunctions are used to indicate the temepratures at eight equidistant vertical axial locations in the hot water store and at eight positions and four equidistant positions in the riser of the collector. An additional thermojunction, protected against exposure to sunlight and wind chill, is located in the outside air to measure the ambient temperature. These thermojunctions measure the local temeratures to within 0.8 °C.

The global insolation incident upon the collector surface is measured with an Eppley PSP pyranometer, mounted horizontally in the same plane as the collector. It is located on top of the storage tank, free from shadows and isolated from conduction sources. The water mass flowrate in the downcommer is measured using local thermal dissipation flowmeter. The optical and thermal characteristics of the natural convection water heating system under study are presented in Table (1).

RESULTS AND DISCUSSIONS

A series of single-day experiments were carried out under various operating conditions. Theoretical predictions were compared with actual performance data in case of the solar collector without utilizing the plane reflectors. To save computation time, the present simulation was carried out with five tank sections (N=5) and the time step of 15 min was found to be good enough for the desired accuracy in these applications.

Figure 2. presents the measured and predicted diurnal variations of the avarage water temperature in the tank ($\bar{T}$) for the solar collector without reflectors compared to average ($\bar{T}$) for the hybrid solar collector, the ambient temperature(T_a), and the global solar insolation (H). The diurnal variations of measured and predicted values of the mass flowrate ($\dot{m}$) in the thermosyphon loop are shown in Fig. 3 The comparison between measured and predicted water

temperature difference (ΔT) at inlet and outlet of the solar collector without reflectors during the daytime is plotted in Fig. 4. Figure 5. presents the comparison between the solar collector efficiencies in both cases of using the reflectors (η') and without reflectors (η) respectively.

It can be seen from Figs. 2, 3 & 4 that the experimental results are found to be in good agreement with simulation. The slight deviation in mass flowrate calculation is attributed to the fluctuation of actual flowrate induced by rapid radiation variation during the time when the mass flowrate measurements were taken. As far as the mean value is concerned, the calculated mass flowrate is still consistent with the experimental results.

The new design of the solar collector as hybrid by using back and bottom reflectors, is developed in order to optimize η without tilting the F.P. absorber. It is clear from Fig. 2. that the average water temperature in tank ($\bar{T}$) has been increased accordingly during daytime. The rise in temperature ($\bar{T}$) was ranging between 11.4 °C at noon and 4.2 °C at 6 p.m. It is evident from Fig.5. that the maximum efficiency η_{max} for the solar collector without reflectors is $\sim$ 60 percent has been improved to be $\eta_{max} = \sim 68$ percent in case of hybrid solar collector.

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CONCLUSION

A validated model for simulation of thermosyphon solar collectors performance is introduced in this work. An empirical correlation for flow resistance in loop is used . The thermally stratified tank model developed by Close (1967) is used to compute the thermal performance in tank.

The approach described is simple and very quickly executed using a microcomputer. Its application is limited by the need for the mean diurnal insolation intensity to exceed a, usuall unknown, threshold level for a given system geometry. However, the collector temperature differential is mutually related to the collector steady-state heat loss coefficient, and if an attempt is made to impose an incorrect value of ΔT, for a given set of meteorological conditions, then usually no solution will be obtained. Furthermore, the number of passes associated with performance prediction can be readily calulated. The inaccuracy of the prediction increases as this decreases; the model is thus recommended only when $N > 4$.

The good consistency between experiment and simulation has proved that this method is applicable in such thermosyphon problems. The agreement between the flowrates predicted theoretically and observed experimentally was, however, described as satisfactory , because there were uncertainties in determining the values of several parameters. This was thought to be due to non-uniformities in the actual velocity and temperature distributions across the rises within the collector plate.

A height difference between the collector and the storage tank has the advantage of suppressing nocturnal reverse circulations and the ensuant heat losses from the store (Norton and Probert 1984) . The tendency to reverse circulation is inhibited by the height difference, because the contribution of the hot water in the store to the nocturnal buoyancy force becomes relatively small. Empirical equations have been derived (Norton and Probert 1983) to predict the variations of the minimum critical height between the collector base and the base of the store necessary in order to prevent entirely nocturnal reverse circulations, for ranges of encountered conditions.

The new design of the hybrid solar collector has improved the performance of the solar collector considerably. The existence of the perpendicular plane reflectors at the back and bottom of the F.P.C. has increased η_{max} from $\sim$ 60 percent to $\sim$ 68 percent and enhanced the mean tank temperature by 11.4 °C at peak slor insolation. The hybrid design has also the advantage of avoiding the tilting of the absorber or sun-tracking. If thermosyphon systems are to gain wider acceptance and become a more economic method for domestic hot-water production, it is imperative that they should be cheap, compact, robust and easy to install. Therefore, this hybrid design is a real step towards achieving this goal.

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Table (1) Optical and thermal characteristics of the natural convection water heating system under study

Collector absorber area	1 m^2
Single glazed	
Glazing transmissivity	0.91
Plate absorptivity	0.93
Collector heat loss coefficient U_L	5.3 W $m^{-2}\,{}^{\circ}K^{-1}$
Collector maximum efficiency $_{max}$	60 %
Hybrid collector maximum efficiency $_{max}$	68%
Tank capacity	0.120 m^3
Maximum average temperature $\bar{T}$ at 6 p.m.	43°C (without reflectors)
Maximum average temperature $\bar{T}$ at 6 p.m.	48 °C (hybrid)
Fall in the average water temperature of the tank during night	5 ° C
Loss coefficient of the tank	1.9 W $m^{-2}\,{}^{\circ}K^{-1}$
Density of water at 20 ° C	1000 Kg m^{-3}
Heat capacity of water at 20 °C	4200 J $Kg^{-1}\,{}^{\circ}K^{-1}$
Conductivity of water at 20 °C	0.61 W $m^{-1}\,{}^{\circ}K^{-1}$
Average wind speed	2 m s^{-1}

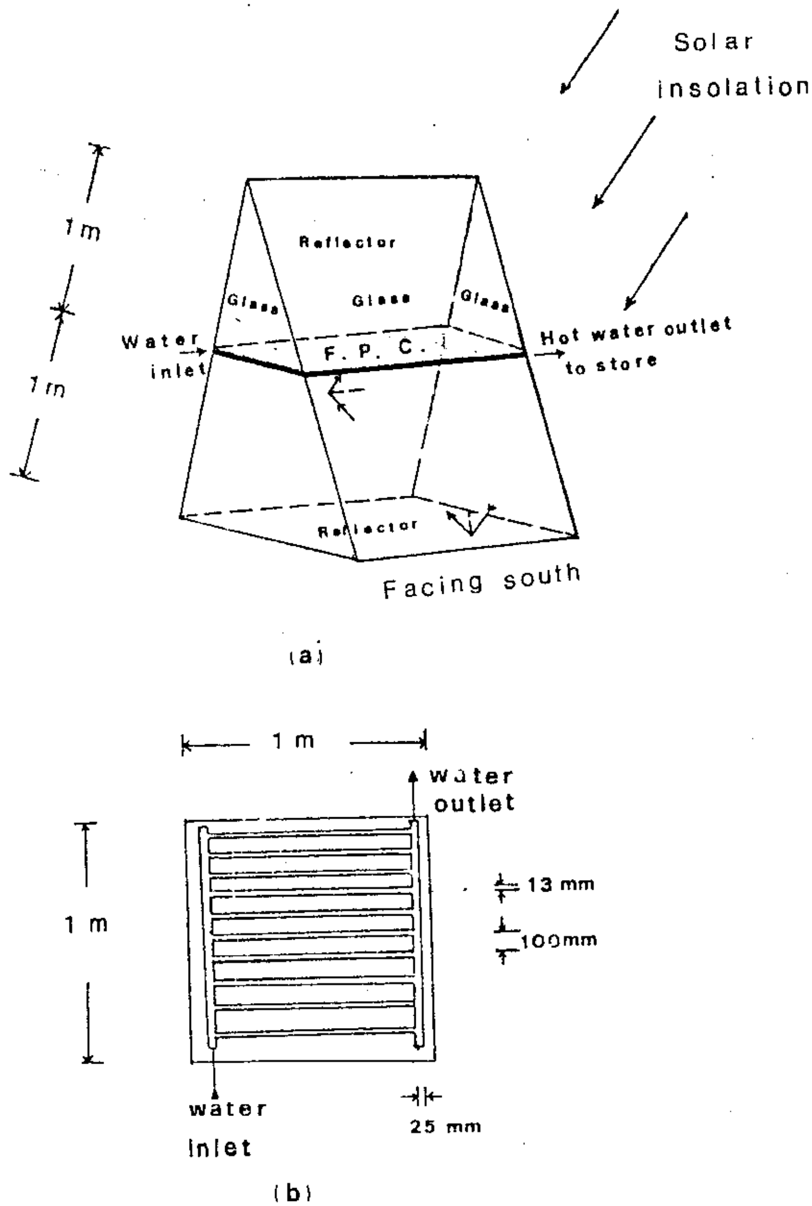

Fig. 1

a. Schematic of the hybrid solar collector.

b. Schematic of the F. P. C.

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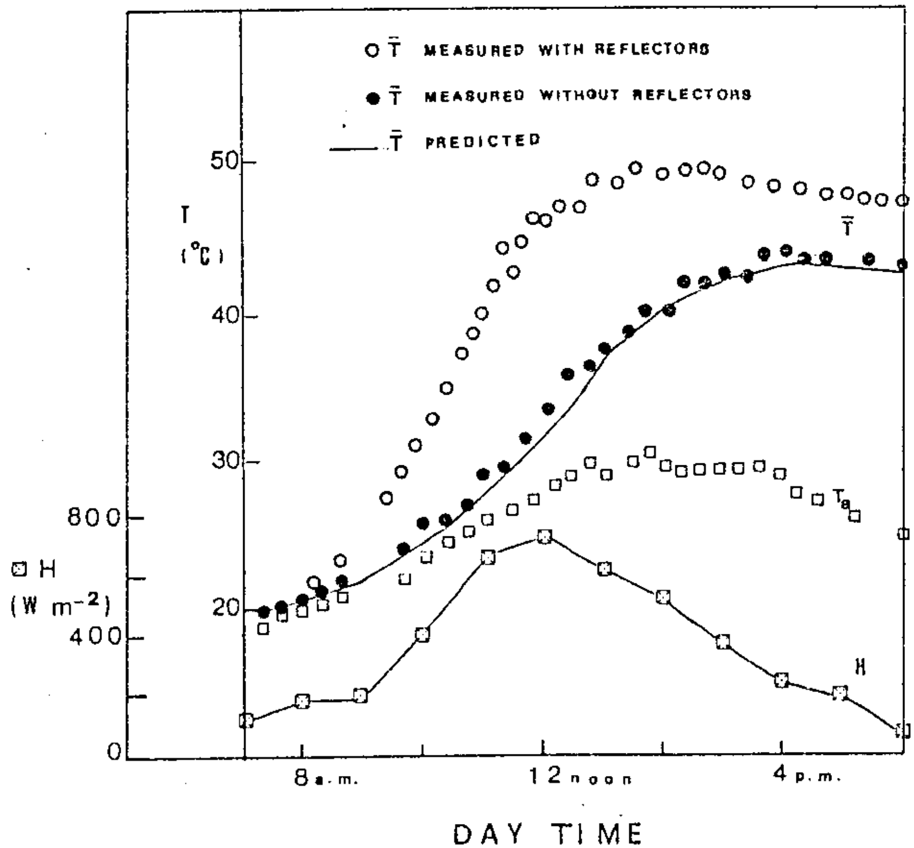

Fig. 2

Diurnal variation of measured and predicted $\bar{T}$ for the solar collector without utilizing reflectors, compared to $\bar{T}$ for the hybrid solar collector with presentation of T_a and H.

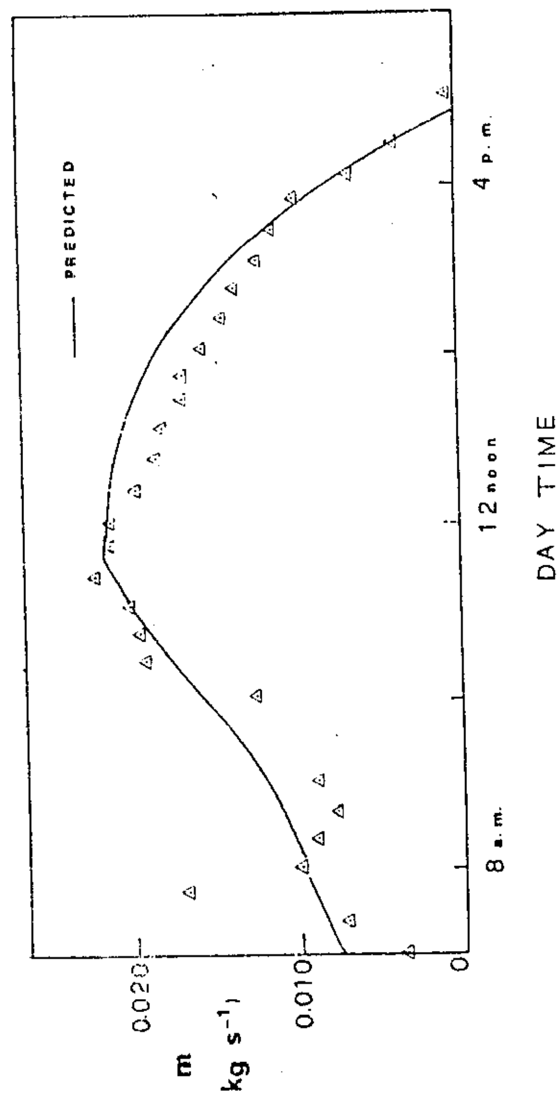

Fig.3 Comarison between diurnal variations of measured and predicted values of m.

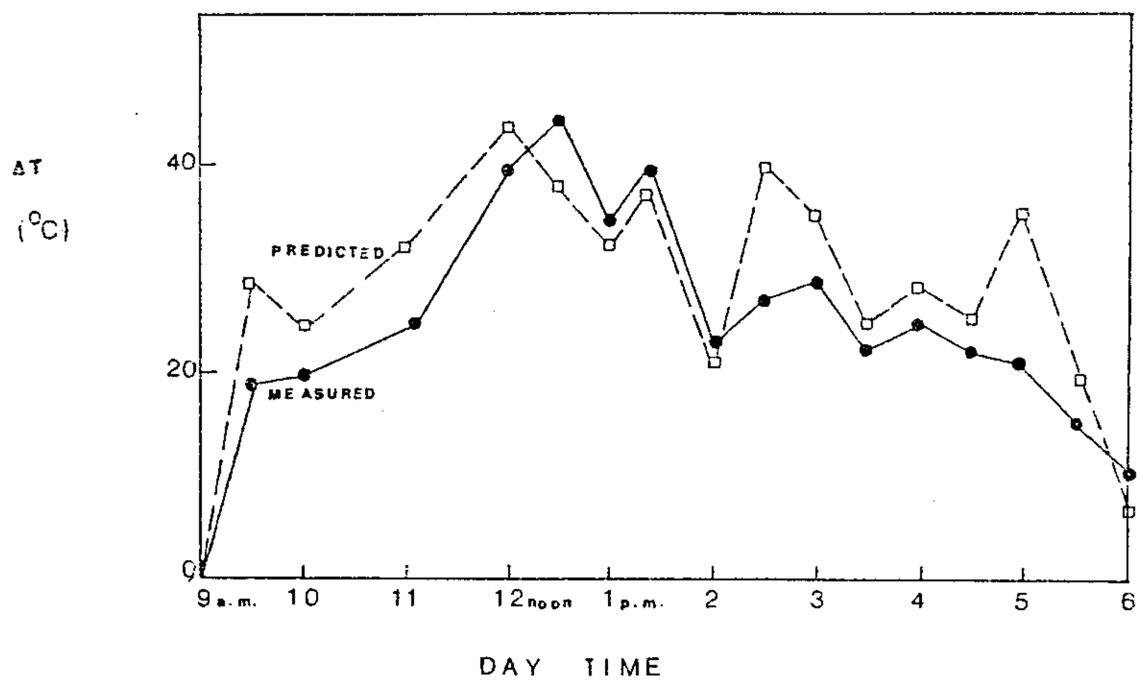

Fig. 4 Comparison between measured and predicated water temperature difference (Δ T) at inlet and cutlet of the solar collector for the daytime (without reflectors).

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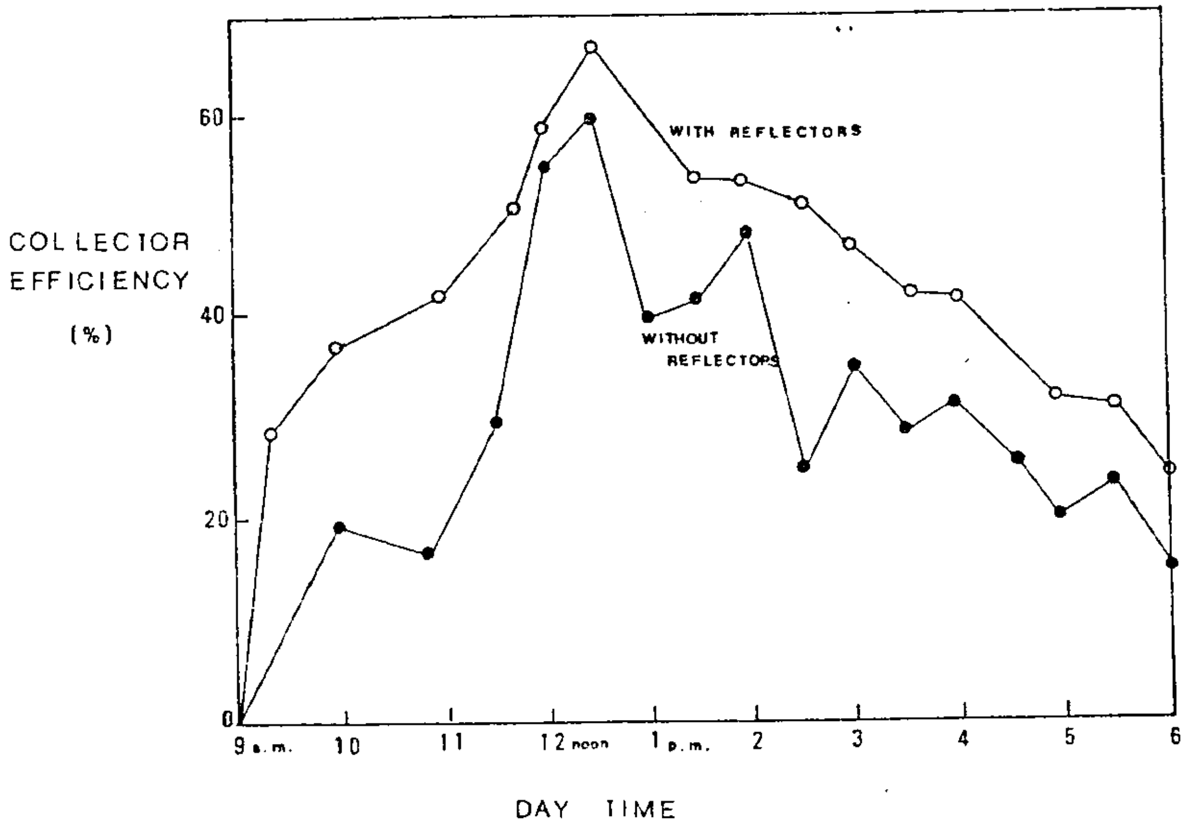

Fig.5

Comparison between the solar collector efficiencies; (i) without using reflectors , and (ii) as hybrid solar collector.

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النموذج النظرى لاستنتاج الكفاءة والخواص المميزة لمجمع شمسى مشترك

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قدمنا فى هذا البحث نموذجا نظريالاستنتاج كفاءة أداء مجمع شمسى من نوع الضخ الحرارى الذاتى Thermosyphon ، وقد أدخلنا معادلة لتقدير مقاومة الانسياب فى دورة المياه للمجمع الشمسى. وكان للاغاق الكبير الذى حصلنا عليه بين القياسات العملية والتنبؤات النظرية باستخدام هذا النموذج أثره الكبير فى امكان استخدام هذه التقنية لتقدير كفاءة الأداء نظريا بالنسبة لأى مجمع شمسى من هذا النوع . كما تم وضع تصميم مبتكر وكذلك تنفيذ " مجمع شمسى مشترك " يجمع بين السخان المسطح والمركزات الشمسية فى جهاز واحد (ذلك بالمواد الخام المتاحة محليا . ولقد كان لادخال هذا التصميم أثره فى تحسين كفاءة الأداء للمجمع الشمسى من ٦٠٪ الى ٦٨٪ كما أن متوسط درجات الحرارة للمياه داخلى الخزان قد زاد بمقدار ٤و١١°م عند القيمة العظمى لكمية الاشعاع الشمسى الساقط .