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The Variation Effect of Collector Tilt Angle on the Heat Pipe Solar Collector Performance

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Abstract- The thermal performance of FPHPSC has been studied experimentally and theoretically. The collector consists of copper absorber plate, single glass cover, glass wool insulation and aluminum case, ten wickless heat pipes of 12.7 mm inner diameter. The experimental studies of collector performance have been performed on four different CTA (22°, 32°, 45° and 60°). The relation between the solar intensity and the collector energy losses are also discussed. The collector are tested under the climate condition of Kut city (32.602 latitude and 45.752 longitude). It is investigated that the increasing of CTA increase the efficiency of collector up to certain limit after that the efficiency decreases with the increasing of tilt angle of collector. A computer program which is based on VISUAL BASICE language (version 6) is made for the theoretical analysis. The experimental results agree reasonably with the theoretical predictions.

Key words: collector tilt angle, heat pipe flat plate solar collector, heat pipe solar collector, heat pipe

تأثير تغير زاوية ميل المجمع الشمسي على اداء المجمع الشمسي الذي يعمل بمبدأ الانبوب الحراري

عبد السلام داود، رواء علاء حسين

ألخلاصة: أجريت دراسة نظرية وعملية لتحديد ألأداء الحراري لمجمع الطاقة الشمسية مسطح يعمل بمبدأ الانبوب الحراري . ويتكون المجمع من لوح امتصاص من النحاس ،غطاء زجاجي منفرد، عازل من الصوف الزجاجي ، صندوق من الألومنيوم ، وعشرة أنابيب حرارية بقطر داخلي (12.7 mm) .أجريت الدراسات التجريبية على أداء المجمع بأستخدام أربعة زوايا ميل للمجمع عن المستوي الافقي(tilt angles) هي (٢٢ °، ٣٢ °، ٤٥ ° ،٦٠) درجة. و تمت دراسة العلاقة بين شدة الاشعاع الشمسي وكمية الخسائر الحرارية للمجمع تحت حالة المناخ لمدينة الكوت /جمهورية العراق(32) درجة شمالا. اظهرت ألنتائج بأن زيادة زاوية ميل المجمع تزيد كفاءة المجمع الى حد معين بعدها تبدأ كفاءة المجمع بالنقصان مع استمرار زيادة زاوية ميل المجمع. تم بناء برنامج للحاسوب يعتمد على لغة (VISUAL BASIC) الاصدار السادس لغرض التحليل النظري لأداء المجمع أتفقت النتائج لتجريبية بشكل معقول مع التوقعات النظرية.

I. INTRODUCTION

The HP is a device with a very high thermal conductance [1]. The HP consists of sealed container, a wick structure and small quantity of working fluid. The HP container is divided into three parts: the evaporator section, the adiabatic section and the condenser section. A HP may be with or without adiabatic region [2]. Basically the working fluid inside the HP will be vaporize when a heat source is applied to the evaporator section externally. The resulting vapor pressure leads the vapor to transfer through the adiabatic section to the condenser section. The vapor gives its latent heat of vaporization to the heat sink and condenses in the condenser section. The condensate is pumped back to the evaporator section either by the capillary pressure force as in the conventional heat pipes or by the gravity force as in the wickless HP (thermosyphon).The authors in [2] analyzed both numerically and experimentally the thermal behavior of plate HPSC. The numerical analysis based on energy conversation equation, and it was assuming a qusi steady state condition. The experimental test was made on small scale collector of aperture area of 0.1 m^2 painted with selective coating. The results showed an over-all loss coefficient of 5.5 $W/m^2.k$, over all time constant of (6min and 50 sec.) and optical efficiency of (64%). The simulated and experimental results was in good agreement and the performance was better than that of convnential HP collector**.** An experimental study of the effect of L/di ratio of the HP on the demeanor of the HPSC is presented in[3] . (L/di is the ratio of the total length to the inner diameter of a heat pipe). Methanol was used as working fluid. The results showed that the collector of L/di ratio equal to 52.63 was obtained to be more efficient than the collector of L/di ratio equal to 58.82.. The experimental analysis showed a good instantaneous system efficiency (68%) with 12L/hr mass flow rate and 52.63 L/di ratio. The research in [4] investigated experimentally the effect of different cross section geometries of wickless HP and different heat pipes filling ratios on the HPSC. Three groups of thermosyphon HP having different geometries, namely (elliptical, circular and semi-circular) cross sections were manufactured. Each group was filled with three different filling ratios of distilled water (10%, 20% and 35%) of the total volume. Experimental results showed that the flat plate elliptical HPSC had better performance than circular HPfor low fill ratios. In [5] the researchers examined theoretically and experimentally the thermal performance of gravity assisted HPSC. Theoretical study is simulated by theoretical model based on Effectiveness-NTU method.

The evaluated parameters was the collector efficiency, the HP temperature and the cooling water outlet temperature The theoretical results suggest an optimum evaporator length to condenser length ratio of 8.25 to increase the amount of the useful heat and to absorb more heat flux. There was a good concurrence between the experimental and theoretical results.

II. Theoretical Analysis

Evaporator section of the HP is heated by the incoming collected solar radiation, therefore, the working fluid inside the HP is evaporated and travels upward to condenser section where it is condensed .The amount of heat energy transferred by the HP from the evaporator section to the condenser can be written as [6]:

$$Q_{hp} = (T_{hp} - \mathrm{T_{con}})\, \mathrm{A_{hp}} / (\sum R_{hp}) \tag{1}$$

$$\mathrm{U_{hp}} = 1/\sum \mathrm{R_{hp}} \tag{2}$$

$$\mathrm{Rhp = Rep + Rcp} \tag{3}$$

R_{ep}, R_{cp} can be calculated according to to Fourier's low for heat conduction through a cylindrical pipe as fallows [5] :

$$\mathrm{R_{ep}} = (\ln(\mathrm{d_o}/\mathrm{d_i}))/(2\pi \mathrm{K_p} \mathrm{L_e}) \tag{4}$$

$$\mathrm{R_{cp}} = (\ln(\mathrm{d_o}/\mathrm{d_i}))/(2\pi \mathrm{K_p} \mathrm{L_{con}}) \tag{5}$$

The amount of heat transferred between the condenser section of a single HP and the cooling water flowing over this section can be calculated as:

$$\mathrm{Q_{con}} = \mathrm{A_{con}}\, \mathrm{U_{con}}\, (\mathrm{T_{con}} - \mathrm{T_i}) \tag{6}$$

The heat energy transferred through the HP (eq. (1)) is equal to that extracted in the heat exchanger (eq. (6)), thus:

$$(\mathrm{T_{hp}} - \mathrm{T_{con}})\mathrm{A_{hp}}\, \mathrm{U_{hp}} = \mathrm{A_{con}}\, \mathrm{U_{con}}\, (\mathrm{T_{con}} - \mathrm{T_i}) \tag{7}$$

From eq. (7) T_{con} can be represented as

$$\mathrm{T_{con}} = \frac{\mathrm{T_i} + \mathrm{PT_{hp}}}{1 + \mathrm{P}} \tag{8}$$

$$\mathrm{P} = \frac{\mathrm{A_{hp}}\, U_{hp}}{\mathrm{A}_{con}\, U_{con}} \tag{9}$$

The over-all condenser heat transfer coefficient can be calculated as:

$$U_{con} = \frac{1}{\frac{t_p}{K_p} + \frac{1}{h_{con}}} \tag{10}$$

Where, h_{con} is The heat transfer coefficient between the outside wall surface of the condenser section and the water flow in the heat exchanger, it can be calculated as:

$$\mathrm{h_{con}} = (\mathrm{Nu} * \mathrm{K_w})/\mathrm{D_h} \tag{11}$$

According to the assumptions that the flow inside the heat exchanger is considered fully developed laminar flow .Also inside the condenser section the flow may be assumed thermally developed at constant heat flux condition. In conclusion Nusselt No. was given as constant value equal to [6]:

Nu=48/11

Figure (1) shows the single HP absorber plate with heat exchanger, the evaporator section the condenser section and the movement of the cooling water are also illustrated in this figure.

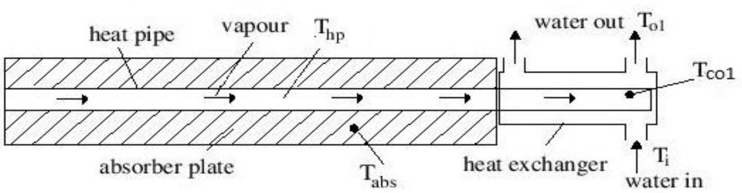

Fig..1: Single heat pipe absorber with heat exchanger

The useful heat is the amount of heat energy result from the differences between the heat energy absorbed and the heat energy lost to the ambient. The useful heat can be modeled According to the Hottel_Willer equation [7]:

$$\mathrm{Qu} = A_{a1} \mathrm{F}_R \left[H_t(\tau\alpha) - \mathrm{U}_c\left(\mathrm{T_{hp}} - \mathrm{T}_a\right)\right] \tag{12}$$

It can be assumed that all the useful heat transferred through the HP is extracted to the cooling fluid flowing in the heat exchanger, it can be written as follow:

$$\mathrm{Q}_{u1} = \dot{\mathrm{m}}\, \mathrm{Cp}\, (\mathrm{T_{o1}} - \mathrm{T}_i) \tag{13}$$

Equating Eq. (12) and Eq. (13) and by simplification. The result is:

$$T_{hp} = T_a + \frac{H_t(\tau\alpha)}{U_c} - \frac{T_{o1} - T_i}{(ntu)_{hp1}} \tag{14}$$

$$(ntu)_{hp} = \frac{Fr\ Aa1U_c}{\dot{m}\ Cp} \tag{15}$$

The vapour in the HP is almost at constant temperature because that the condensation and boiling heat transfer occurs at constant saturation temperature. Therefore its specific heat and capacity rate will be equal to infinity .Then as a consequence the capacity ratio will be equal to zero, therefore, the equation of Ɛ-ntu can be written as [8]:

$$\varepsilon_1 = 1 - e^{(-ntu)_{con}} \tag{16}$$

$$(ntu)_{con.} = \frac{A_{con}\ U_{con}}{(\dot{m}\ Cp)} \tag{17}$$

Effectiveness in general form is :

$$\varepsilon_1 = \frac{(To1-Ti)}{(T_{con}-T_i)} \tag{18}$$

From the above equation. To1 can be calculated as:

$$T_{o1} = Ti + \varepsilon_1 (T_{con} - T_i) \tag{19}$$

By substituting eq. (8) in eq.(19) ,the outlet condenser temperature from the first HP heat exchanger can be calculated as:

$$T_{o1} = T_i + \varepsilon_1 \frac{P}{(1+P)} \left(T_{hp} - T_i\right) \tag{20}$$

By substituting the value of T_{o1} from. (20) into equation (14) ,the HP temperature can be modified to the following form as shown in eq (23):

$$T_{hp} = \frac{\frac{H_t(\tau\alpha)}{U_c} + T_a + \frac{T_i\varepsilon1}{(ntu)hp}\left(\frac{P}{(1+P)}\right)}{1 + \frac{\varepsilon1}{(ntu)hp}\left(\frac{p}{(1+p)}\right)} \tag{21}$$

In the case of flat plate solar collector with n number of heat pipes eq. (20) can be modified to the form shown below:

$$T_o(n) = T_o(n-1) + \varepsilon n \frac{P}{(1+P)} \left(T_{hp}(n) - T_o(n-1)\right) \tag{22}$$

The cooling water exists from the end of the heat exchanger which overlap with the HP array at a temperature higher than the inlet temperature, it can be represented as:

$$T_o = T_i + E \frac{P}{(1+P)} \left(T_{hp} - T_i\right) \tag{23}$$

The overall effectiveness of n number of condensers and for equal values of effectiveness can be written as [6]:

$$E = 1 - (1-\varepsilon)^n \tag{24}$$

The average temperature of HP can be calculated as:

$$T_{hp} = \frac{\frac{H_t(\tau\alpha)}{U_c} + Ta + \frac{T_iE}{(NTU)hp}\left(\frac{P}{(1+P)}\right)}{1 + \frac{E}{(NTU)hp}\left(\frac{p}{(1+p)}\right)} \tag{25}$$

The number of transfer unit for n number of HP and condenser section can be calculated as:

$$(NTU)_{hp} = n(ntu)_{hp} \tag{26}$$

The outlet water temperature (eq(23)) can be used to calculate the collector theoretical efficiency as follow:

$$\eta_{theo} = \frac{\dot{m}\ c_p\ (T_o - T_i)}{H_t\ A_c} \tag{27}$$

III. EXPERIMENTAL SET UP

The FPHPSC is shown in Figure (2), which is consisted of copper absorber plate of copper absorber plate (155 cm length, 100 cm width and 0.1cm thickness), single glass cover of (0.6 cm thickness), glass wool insulation and aluminum case. Ten wickless heat pipes of 12.7 mm inner diameter, and 185 cm length 1.34cm outer diameter length (155 cm represent the evaporator section and 30 cm represent the condenser section) are welded on the surface of the of the absorber plate by using silver welding filler they together painted with black paint. Closed loop system was designed and connected with the solar collector to obtain test result. This loop is an auxiliary system consisted of (water storage tank, water pump, water filter, stand of the tank and piping network). The measuring tools used in this work are digital thermocouples type k, thermocouples type k, thermocouple reader, selector switches, digital solar power meter, hot wire anemometer. The heat pipes of collector cooled with cooling water at 0.06 kg/sec mass flow rate.

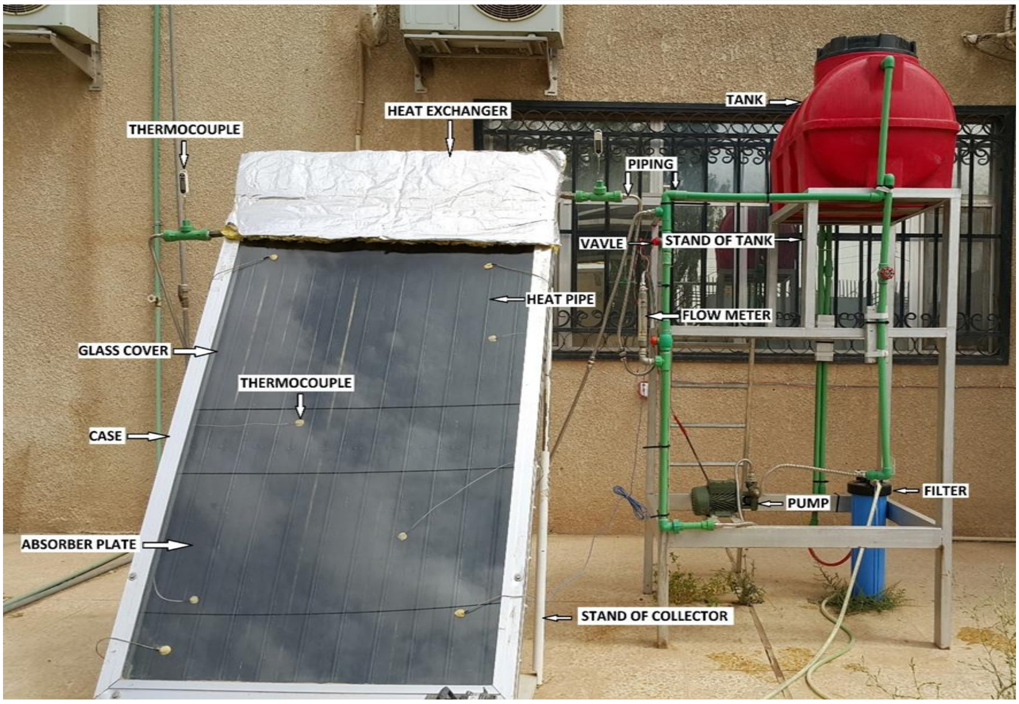

Fig.2.photography of the heat pipe heat exchanger system

IV. RESULT AND DISCUSSION.

Figure (3) shows the variation of the FPHPSC tilt angle on the collector efficiency during solar time for 0.06 kg/sec mass flow rate. It obvious from the figure that the collector efficiency increased with the increasing of the CTA up to certain limit then further increasing of tilt angle causes reduction in the collector efficiency. Basically an increasing of the CTA enhances the bouncy force on the up traveling vapor and the gravity force on the down coming liquid inside the HP. The tilt angle effects also on the incidence angle of solar radiation received by the solar collector .Table I shows the variation of the incidence angles with tilt angles used in the experimental work at the midday time of 29/03/2016 in Kut city.

TABLE I. THE VARIATION OF THE TILT ANGLES WITH INCIDENCE ANGLES.

Tilt angle (deg.)	22	35	45	60
Incidence angle (deg.)	7.4	3.22	13	31

The influence of tilt angle on the performance of the FPHPSC are discussed below:

- At 22° tilt angle the incidence solar radiation angle have low value (7.4°) that means the solar intensity becomes closer to be orientated normally on the collector's absorber plate this leads to greater solar energy which is received by the collector. But the heat pipes at 22° tilt angle position near to horizontal plane which leads to certain difficulties for the condensate to return from the condenser section to the evaporator section .Thus the evaporation and condensation cycle will be decelerated and there will be less utilization of the solar energy.

- At 32° tilt angle the incidence angle is at value of (3.22°). In other word the solar radiation become closer and closer to be ordinated normally on the collector absorber plate surface which is improve the collector efficiency. But the lower value of the tilt angle still makes the difficulty on the condensate returning.
- At 45° tilt angle of the collector the value of the incidence angle (13°) more than the values of the incidence angle for the two situation mentioned above .But at (45°) tilt angle the condensate can be return quickly to the evaporator section that means an acceleration of the evaporation-condensation cycle and better utilization of the absorbed solar radiation. The experimental results shows generally that the optimum collector efficiency occurs at this tilt angle.
- At 60° tilt angle, the condensate will be move from the evaporator section to the condenser section easily .But the solar radiation will be fallen on the collector with very high value of incidence angle. That means great deviation from normal incidence and this effects adversely on the on the solar collector efficiency.

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Figure (4) shows the variation of solar intensity with collector energy losses at different mass flow rate and tilt angle along the solar time of the day .It is clear from figures that the collector energy losses ($Q_{loss} = U_c^* (T_{pl}-T_a)$ W/m^2) behave like the solar intensity curve. Because the high solar heat flux causes an elevated value of absorber plate temperature and collector loss coefficient. This eventually leads to high collector energy losses. The comparison is made between the theoretical and experimental instantaneous efficiency for different values of cooling water mass flow rates and solar CTAs .Figure (5) shows the validation between the results. The comparison results shows acceptable values with good agreement.

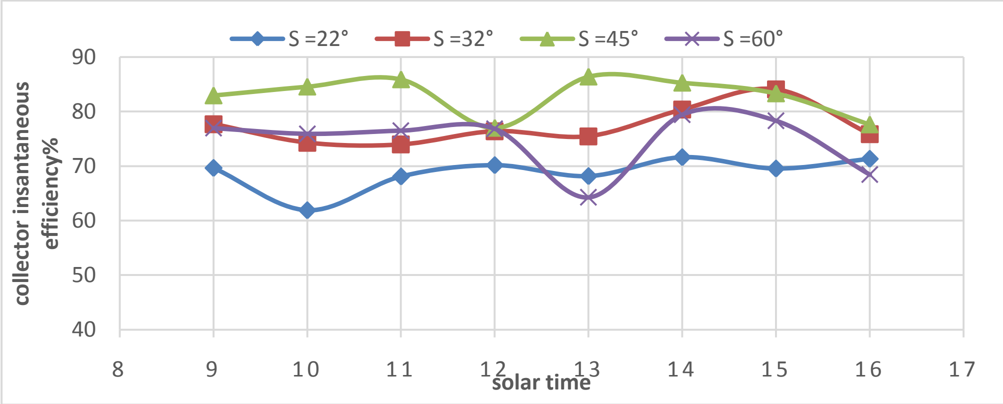
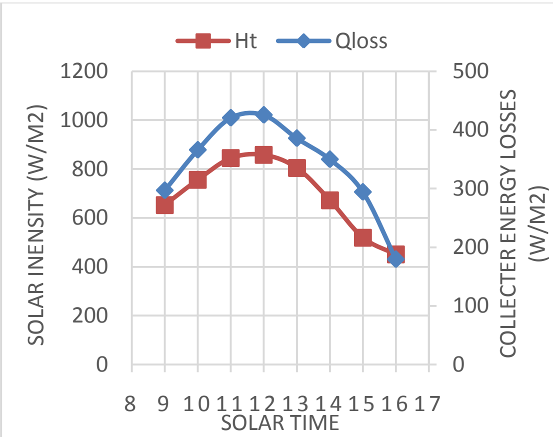
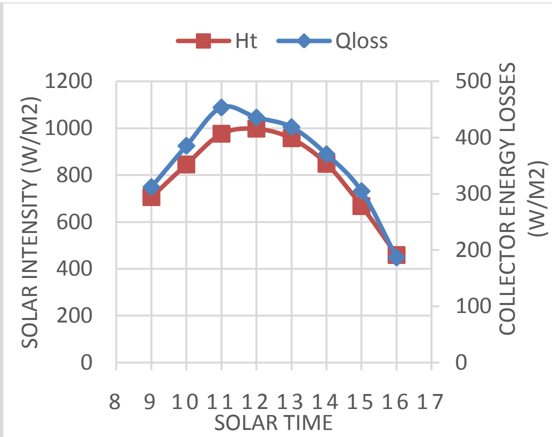


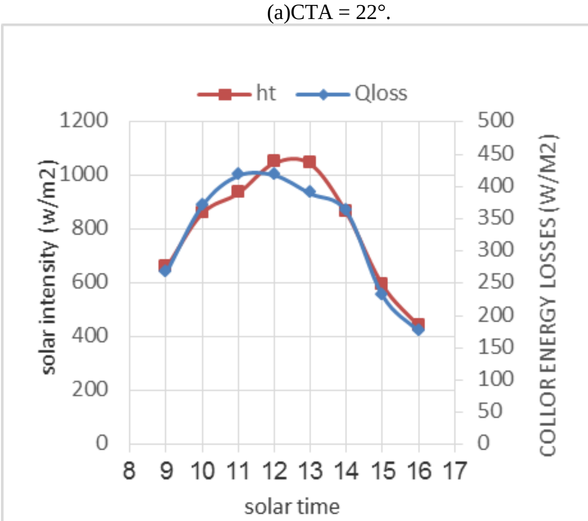
Fig..3.variation of collector instantaneous efficiency with solar time at different CTA and at 0.06 kg/s cooling water mass flow rate.



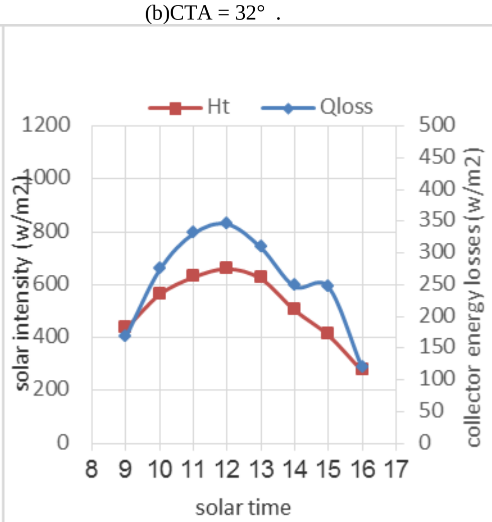
(a)CTA = 22°.



(b)CTA = 32° .



(c) CTA = 45 °.



(d) CTA = 60°

Fig.4. variation of solar intensity and collector losses with solar time at different collector tilt angle and at 0.06 kg/s cooling water mass flow rate

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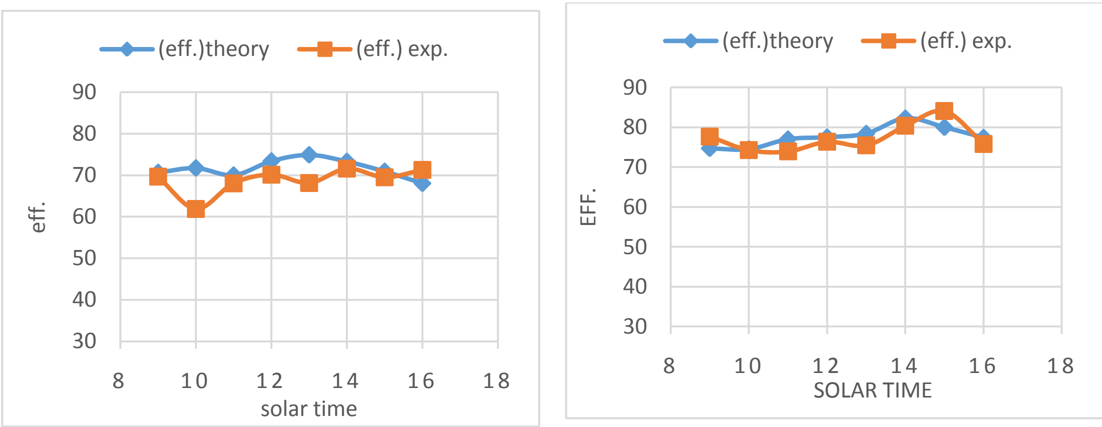

(a)CTA = 22°. (b)CTA = 32°.

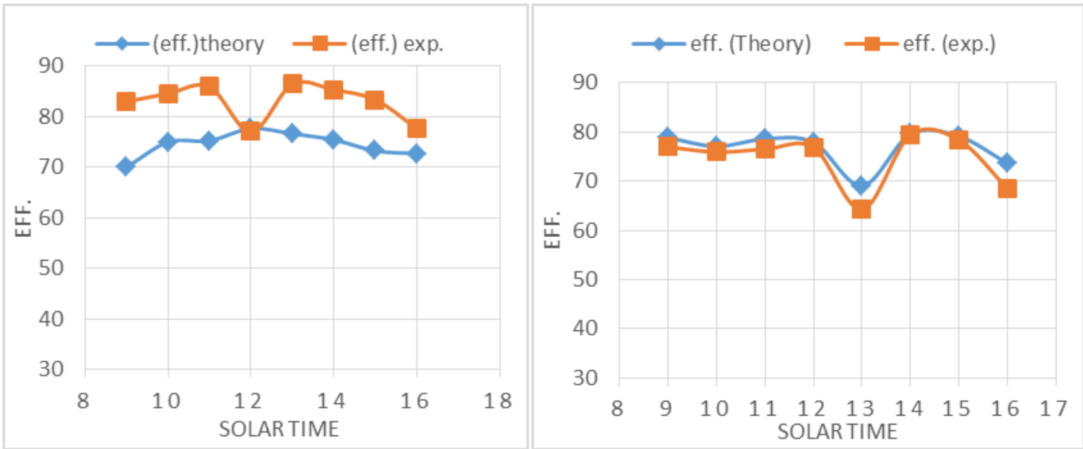

Fig.. 5. Comparison between experimental and theoretical collector efficiency and at 0.06 kg/s cooling water mass flow rate.

V. Conclusion

This work has exhibited both experimental and theoretical study of the FPHPSC increasing the tilt angle with constant mass flow rates leads to increasing the useful energy gain and solar collector efficiency up to certain limit. After that the collector efficiency decreases with the increasing of tilt angle at constant mass flow rates. The thermal collector losses increase with the increasing of both the temperature differences between the collector absorber plate and ambient and the over-all loss coefficient.

Nomenclature

Latin symbols

A: area, m_2

C_p: specific heat capacity, J/kg K

D: diameter, m

E: overall effectiveness

F_r: solar collector heat removal factor.

h :convection heat transfer coefficient, $W/m^2.°C$

H_t: solar insolation on tilt angle W/m^2

K: thermal conductivity, W/m.°C

t: thickness m

T: temperature °C

U_c: over all heat transfer rate coefficient, $W/m^2.°C$

Greek samples

Ɛ: Effectiveness of heat exchanger.

η: Efficiency of solar collector.

(τα): Transmissivty_ absorptivity product

Subscripts

a1: absorber for a single heat pipe

L: length, m
ṁ: cooling water mass flow rate, kg/s
n: number of heat pipes
ntu: number of transfer unit for one heat pipe
NTU: number of transfer units for n heat pipe
Nu: Nusselt number
Q: heat transfer rate W
R: thermal resistance K/W
pl: plate
Theo: Theoretical
u: useful
w: water

a: ambient
con: condenser section
e: evaporator section
h: hydraulic
hp: heat pipe
i: inlet, inside
o: outlet, outside
p: pipe

Abbreviations

CTA: collector tilt angle
HPFPSC: heat pipe flat plate solar collector
HPSC: heat pipe solar collector
HP: heat pipe

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