

THERMAL PERFORMANCE OF PARABOLIC TROUGH SOLAR COLLECTOR

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ABSTRACT

This work presents design, instrumented and test of a parabolic trough solar collector under Baghdad climate conditions (of latitude 33.33° N, of longitude 44.4° E). The parabolic trough solar collector consists of: a mirror matrix or tapes which work as reflective surface of (2m *1m), absorber copper tube (receiver), two axis tracking system. Water is used as a heat transfer medium. The setup is tested within clear days from June, to September 2017. The collector heat gain, efficacy and temperature of absorber were presented for absorber five different circulating mass flow rates of (0.15, 0.2, 0.3, 0.4, 0.5) lpm. The results show that the maximum thermal efficiency of the parabolic trough solar collector is 80.26%. The maximum outlet temperature of the absorber tube reaches 81°C at the noon when water flows at (0.15) lpm. The maximum obtained heat gain is (1619W) for (0.5) lpm flow rate of water.

KEYWORDS: Thermal parabolic trough solar collector, two axis tracking, solar radiation, Heat transfer, Characteristic factors.

الأداء الحراري لمجمع شمسي نوع قطع مكافئ خطي

كريمة اسماعيل عموري رند رشيد ساري

الخلاصة

هذا العمل تمثل بتصميم وتجهيز المنظومة باجهزة القياس والفحص لمجمع شمسي نوع قطع مكافئ (مركز) في ظل الظروف المناخية لبيدغداد العراق (خط عرض $33,3^{\circ}$ شمالا وخط طول $44,2^{\circ}$ شرقا). يتكون المجمع الشمسي من مصفوفات او الاشرطة من المرايا التي تعمل كسطح عاكس (1 م * 2 م)، انبوب سطح المستلم النحاسي ومنظومة تعقب ثنائية المحاور. استخدم الماء كوسيط لنقل الحرارة. تم اجراء التجارب في ايام مشمسة من شهر حزيران الى شهر ايلول 2017. تم عرض الحرارة المتجمعة المستفأة، كفاءة المجمع الشمسي، درجة حرارة انبوب السطح الماص لخمس قيم من معدل تدوير الماء والتي كانت (0,15 - 0,2 - 0,3 - 0,4 - 0,5) لتر/دقيقة. اظهرت النتائج المستحصلة من التجارب العملية ان اقصى كفاءة للمجمع الشمسي هي 80,26%. اقصى درجة حرارة خروج للمائع من انبوب الماص 81 درجة مئوية في منتصف الظهيرة عندما كان معدل تدوير الماء 0,15 لتر/دقيقة. اقصى حرارة متجمعة مستفأة تم الحصول عليها هي (1619 واط) نما كان معدل تدوير الماء 0,5 لتر/دقيقة.

Symbol	Description	Symbol	Description
A	Surface area (m ²)	F _r	Heat removal factor
A _a	Aperture area (m ²)	h _r	Radiation heat transfer coefficient (W/m ² . K)
A _r	Receiver area (m ²)	h _w	Convection heat transfer coefficient (W/m ² . K)
AL	Altitude location	h _s	Hour angle (degree)
C _p	Specific heat at constant pressure (kJ/kg. K)	I _b	Beam solar radiation (W/m ²)
C _r	Geometric concentration ratio	I _d	Diffuse solar radiation (W/m ²)
d _n	Days number from year	k	Thermal conductivity (W/m .K)
D	Diameter (m ²)	m	Mass flow rate (kg/s)
ET	Equation of time (minute)	T	Temperature (K)
f'	Collector efficiency factor	U _L	Heat loss coefficient (W/m ² .K)
f''	Collector geometry factor	V _w	Wind speed (m/s)
Greek Symbols			
ε	Emissivity	τ	Transmittance
η _{th}	Thermal efficiency	α	Absorptance
σ	Stefan-Boltzmann constant (W/m ² . K ⁴)	μ	Dynamic viscosity (kg/m. s)
α _s	The solar altitude angle (degree)	ρ	Density (kg/m ³)
δ _s	Sun's declination angle (degree)		
Dimensionless Groups			
Nu	Nusselt Number $\frac{hd}{k}$	Re	Reynolds number $\frac{\rho u_i d_i}{\mu}$
Pr	Prandtl Number $\frac{v}{\alpha}$		
Abbreviation			
LST	Local standard time	PTSC	Parabolic trough solar collector
NIR	Near Infrared	UV	ultraviolet

INTRODUCTION

The main principle of work of solar thermal Collection is that when sun rays is incident on a surface part of this rays is absorbed and causes an increase in the temperature of the surface. As the surface temperature increase, the surface loses heat due to due to different temperatures between the body and the surrounding. In the concentrators of solar, the reflective surface or other optical means was used to focus the incident sun rays onto an appropriate absorber, therefore solar concentrators increase the amount of incident solar energy on the absorber compared with that on the concentrator aperture. A solar concentrator is shown in figure (1) which consists mainly of an absorber or receiver, focusing device and tracking system for track the sun continuously. There are mainly three types of concentrating solar

thermal power (CSP) technologies, troughs which parabolic trough form mirror reflectors linearly focuses sun rays onto absorber tubes, Towers which consists of numerous heliostats to concentrate sun rays on a central receiver placed at the top of a tower and Dishes which parabolic dish form reflectors perfectly concentrate sun rays in two axes (focal point). Iraq is considered at the second level of solar radiation exposure. Solar energy exposure period in Iraq is available for a long period as shown in table (1) [NASA, 2008]. (Jeter, 1983) studied effects of geometrical on the thermal performance of Parabolic trough solar collector. The end-effect was analyzed. The results show the significance of end-effects particularly increases when short troughs are considered and elimination of this effect is important in obtaining test results. (Thomas, 1994) developed a sample structure of Parabolic trough solar collector (PTSC) to study its deflection and optical characteristics under various load conditions. In the absence wind tunnel facilities, the test gives sufficient information about the effect of wind load on the optical performance of a Parabolic trough solar collector (PTSC). (Thomas and Michael, 2005) developed a Parabolic trough solar collector similar in size to smaller-scale commercial modules for use in a South African solar thermal research program. The collector length is 5m, aperture width is 1.5m and rim angle is 82°. Two receivers were fabricated for comparative testing, including one enclosed in an evacuated glass cover. Peak efficiencies of 55.2% and 53.8% were obtained with the unshielded and glass-shielded receivers respectively. (Mutlak, 2011) studied the efficiency of parabolic trough solar collector. a matrix or pieces of mirrors to shape the surface of parabolic reflector (1.8 x 2.8m). Synthetic oil is used as working fluid. Three types of experimental tests have been achieved using, evacuated glass receiver, coated metallic and non-coated metallic. The experimental tests have been conducted in Baghdad (33.3° N, 44.4 °E) during selected days from October to December. The study show the heat loss coefficient of the evacuated receiver has been found 7.5 W/°C.m², while the heat loss coefficients were 18.3 W/°C.m² and 20.6 W/°C.m² for non-coated and coated metallic receivers, respectively, and the most heat losses occur through storage tank. The average thermal efficiency for the collector is approximately 61% when evacuated receiver is used. The heat losses which occurred at high Temperatures which decreased the average thermal efficiencies to 51% and 40% are respectively the selectively coated and non-coated metallic receivers. It was also found that the obtained characteristic curve of the tested parabolic trough collector is considerably higher than that of a typical collector owing to the lower thermal losses of the evacuated glass envelope.

The contribution of the present work can be expressed as: Using a mirror type of reflective surface for parabolic tough solar collector (mirror matrix or tapes is used as reflective surface) and to track the sun continuously, a two axis tracking system is used.

SOLAR RADIATION

The solar irradiation is defined as energy flowing from the sun simultaneously and constantly in all directions. The total radiation (I_T) incident on a surface consists of diffuse radiation (I_d) that been appreciable scattered by the atmosphere before reaching the surface and beam radiation (I_b) that crossed the atmosphere without being significantly scattered. The total solar radiation can be calculated as [Duffie & Beckman, 2005]:

$$I_T = I_b + I_d \quad (1)$$

$$I_b = I_o [a_o + a_1 e^{-(K \cdot AM)}] \quad (2)$$

$$I_o = I_{sc} [1 + 0.034 \cos \left(\frac{360 d_n}{365.25} \right)] \quad (3)$$

I_{sc} : solar constant and current accepted value from NASA is (1353 W/m²).

a_0 , a_1 and k : constants empirical parameters are given as [Duffie & Beckman, 2005]:

$$a_0 = 0.94 [0.4237 - 0.00821 (6 - AL)^2] \quad (4)$$

$$a_1 = 0.98 [0.5055 - 0.00595 (6.5 - AL)^2] \quad (5)$$

$$K = 1.02 [0.2711 - 0.01858 (2.5 - AL)^2] \quad (6)$$

(AM): the air mass which is equal to $\frac{1}{\sin \alpha_s}$.

EARTH – SUN ANGLES

Calculations performed for Baghdad, where ($L = 33.3^\circ$) is the local latitude, ($L_{local} = 44.4^\circ$) is the local longitude and ($L_{st} = 45^\circ$) is the L_{st} for the local time zone.

$$h_s = 15[ST-12] \quad (7)$$

To convert the local time to solar time used following equation according to, [Duffie & Beckman, 2005]:

$$ST = LST + 4(L_{st} - L_{local}) + ET \quad (8)$$

ET: Time equation in minutes can be calculated by [Spencer, 1971]:

$$ET = 9.87 \sin(2\beta) - 7.35 \cos(\beta) - 1.5 \sin(\beta) \quad (9)$$

$$\beta = 360 (d_n - 81) / 364 \quad (10)$$

Where (d_n) is the day number during the year ($1 \leq d_n \leq 365$).

The sun's declination angle δ_s can be calculated by [Cooper, 1969]:

$$\delta_s = 23.45 \sin \left[360 \frac{284 + d_n}{365} \right] \quad (11)$$

$$\sin(\alpha_s) = \sin(L) \sin(\delta_s) + \cos(L) \cos(\delta_s) \cos(h_s) \quad (12)$$

THERMAL ANALYSIS (ABSORBER TUBE)

As the circulated fluid (water) in the receiver absorbs energy, its temperature will increase which creates a difference in temperatures between the circulated fluid, absorber, glass cover and the surrounding ambient air. Heat losses from absorber to glass cover then to the surrounding ambient air are driven by this temperature difference. Figure (2) shows the heat transfer mechanisms for heat losses between the absorber, glass cover of the collector and the ambient environment.

The useful energy is calculated by the following equation.

Energy useful = Heat_{in} - Heat_{out}

$$Q_u = Q_{abs} - Q_{loss} \quad (13)$$

$$Q_{abs} = (\tau\alpha) I_b A_{ap} \quad (14)$$

I_b : Solar beam radiation intensity (W/m^2), is hourly measured each hour starting from 9:00 AM till 14:00 PM.

The actual useful heat gain is calculated as [Duffie & Beckman, 2005]:

$$Q_u = A_{ap} F_R \left[I_b (\tau \alpha) - \frac{U_L (T_{in} - T_{amb})}{C_r} \right] \quad (15)$$

where C_r is the concentration ratio calculated as:

$$C_r = \frac{\text{Aperture area } (A_{ap})}{\text{surface area of the absorber } (A_{abs})} \quad (16)$$

The heat loss factor (U_L) combines the thermal losses into one coefficient, U_L can be calculated by [Duffie & Beckman; 2005]:

$$U_L = \left[\frac{A_{abs}}{(h_w + h_{r,g-amb}) A_g} + \frac{1}{h_{r,abs-g}} \right]^{-1} \quad (17)$$

h_w is the factor of convection heat transfer owing to wind speed ($W/m^2.K$). It is calculated by [Kuman and Mullick]:

$$h_w = 5.7 + 3.8 V_w \quad (18)$$

$$h_{r,g-amb} = \varepsilon_g \sigma (T_g + T_{amb})(T_g^2 + T_{amb}^2) \quad (19)$$

$$h_{r,abs-g} = \frac{\sigma (T_{abs} + T_g)(T_{abs}^2 + T_g^2)}{\frac{1}{\varepsilon_{abs}} + \frac{A_{abs}}{A_g} \left(\frac{1}{\varepsilon_g} - 1 \right)} \quad (20)$$

The heat transfer from the absorber to the working fluid was characterized by turbulent or laminar flow conditions accordingly, so Reynolds number Re_f should be evaluated:

$$Re_f = \frac{4 \dot{m}}{\pi D_{abs,i} \mu_f} \quad (21)$$

In the present work the adopted mass flow rate is ranged (from 0.15 lpm to 0.5 lpm) according to equation (22):

$$m_r = \frac{\pi d_i^2}{4} L_r \rho_w \quad (22)$$

where ρ_w is density of water, The riser length (L_r) have been considered as (2 m), while its inner diameter (d_i) was (12.7mm) and (0.8 mm) thickness.

$$\dot{m} = \frac{m_r}{t} \quad (23)$$

t : Time taken as (1 minute).

According to the absorber inlet thermo physical properties, the obtained Reynolds number is ranged (314.39 to 1047.99) which are within laminar fluid flow. Nusselt number of the working fluid (Nu_f) for laminar flow (if $Re_f < 2200$) and constant heat flux condition is equal to 3.7 [Thepa S, 1999].

The heat transfer coefficient (h_f) is evaluated:

$$h_f = \frac{Nu_f k_f}{D_{abs,i}} \quad (24)$$

The coefficient of heat loss based on the outside absorber diameter between the circulated fluid and the surrounding is called the overall heat transfer coefficient which evaluated as [Duffie & Beckman, 2005]:

$$U_o = \left[\frac{1}{U_L} + \frac{D_{abs,o}}{h_f D_{abs,i}} + \frac{D_{abs,o} \ln\left(\frac{D_{abs,o}}{D_{abs,i}}\right)}{2k_{abs}} \right]^{-1} \quad (25)$$

The efficiency factor of collector (F') can be calculated as:

$$F' = \frac{U_o}{U_L} \quad (26)$$

The value of correction factor or heat removal factor (F_R) is ($0 < F_R < 1$), it can be interpreted as the ratio of the actual useful heat collected by parabolic collector to that which would be collected if the entire absorbed surface is at the inlet temperature of the fluid, and it can be evaluated as [Duffie & Beckman, 2005]:

$$F_R = \frac{\dot{m}_f C_p}{A_{ap} U_L} \left[1 - e^{-\left(\frac{A_{ap} U_L F'}{\dot{m}_f C_p}\right)} \right] \quad (27)$$

The collector geometry factor is calculated as:

$$F'' = \frac{F_R}{F'} = \frac{\dot{m}_f C_p}{A_{ap} U_L F'} \left[1 - e^{-\left(\frac{A_{ap} U_L F'}{\dot{m}_f C_p}\right)} \right] \quad (28)$$

$$\eta_{th} = \frac{Q_u}{A_{ap} I_b} \quad (29)$$

The thermal efficiency of parabolic trough collector depends upon the operating conditions parameter and the parameters of concentrator design. The solar flux, inlet fluid temperature and the ambient temperature are operating conditions while the heat removal factor and heat loss coefficient are the design dependent parameters.

EXPERIMENTAL WORK

Parabolic reflector is a two axis linear parabolic shape reflector. The Parabolic reflector has the property of collecting incident rays along a single line focus. The absorber located along the focus line. In this work a parabolic reflector is designed and manufacture locally in the thermal research laboratory of University of Baghdad of College of Engineering Mechanical Dept. The Parabolic Reflector consists of several parts which are as follows,

- Stationary main base consists of cast iron tube with (0.12 m) diameter fixed on mechanical structure loader. This main base was fixed on the ground by hummer bolt.
- The moving base which has two motions, circular motion from (0° to 360°) and tilting motion. This part consists of cast iron ring with (0.5 m) diameter with two bearing brackets, two ball bearings and tilting motion base as shown in figure (3). The moving base was supported by stationary main base.
- The reflecting surface consists of many segments or pieces of mirrors of width (25 mm) and (1 m) length fixed by silicon sticker on the designed parabolic structure which formed of flat bar as shown in figure (4 a, b).

- The flat glass cover of thickness (3mm) was used to cover the reflector from the top. The function of this glass cover to isolate the reflector from the ambient conditions and to reduce the thermal losses.

To ensure that the beam radiation was collected at the focus, a laser pointer source is used which moved along a horizontal axis parallel with structure to fix the focus at the certain place as shown in figure (5). The final form of parabolic trough solar collector is presented in figure (6). The specifications of this reflector are given in table (2). The focal length of this reflector is calculated using equation (30) [Lifang et al., 2011]:

$$f = \frac{\left(\frac{l}{2}\right)^2}{4 d_p} \quad (30)$$

f : Focal length (m).

d_p : Depth of parabolic (m).

l : Length of parabolic (m).

The absorber (riser) is a copper tube of (12.7 mm) ID, (0.8mm) thickness and (2300 mm) in length with thermal black matt coating. (200 mm) of its length is used as an absorber and the remained used for connection.

RESULTS AND DISCUSSION

The thermal performance of the parabolic trough solar collector was analyzed under different climatic conditions of Baghdad for different water mass flow rates (0.15, 0.2, 0.3, 0.4 and 0.5 lpm). A figure (7) presents the distribution of measured solar radiation when a concentrated solar collector is tested on (7th June, 4th July, 27th Aug., 13th September, 2017). These figures show that the solar radiation rises from first day hour to reach its maximum value at solar noon, and then ceased to exist at sun set. The maximum solar radiation was (995 W/m²) at (12:00 PM) on 7th July-17 and decreases after that. Figure (8) shows that the measured ambient temperature on 8th August was higher than that on 7th June, 4th July and 13th September, it rises from (39.8 °C) at (9:00 AM) to (49.4 °C) at (13:00 PM).

Figures (9 a, b, c, d) reports the history of the average temperature of absorber surface, surface reflector (mirror) and glass cover of solar collector for different water flow rates (0.15, 0.3, 0.4 and 0.5 lpm). It's found that the average temperatures of the absorber surface on 7th June for (0.15 lpm) water mass flow rate was 79.8 °C at (12:00 PM) as highest value, due to highly incident solar radiation and high ambient temperatures on these days (970 W/m²), (46.7 °C). Figure (10 a, b, c, d) illustrates the history of water inlet and outlet temperature for different cases of water flow rates. The outlet temperature increased with time to reach 81 °C within one hour (12:00 AM) to (13:00 PM) on June, July, August and September, and then decreased after that, because the solar radiation decreased at these hours.

The useful heat gain Q_u is determined by obtaining value of mass flow rate and the measured inlet and outlet temperatures of circulated fluid. Figure (11) shows the relation between the variation of beam solar radiation (I) and the useful heat gain (Q_u) on 7th June. It is noted that the useful heat gain increases, reaches a maximum value (1581 W) at noon and then decreases because that the incident beam radiation decreases in this period. This means that the useful heat gain is followed the solar radiation. Figure (12) shows the relation between useful heat gain and inlet working fluid temperature. The useful heat gain decreased when the inlet working fluid increased than (42 °C) for water working fluid because temperature difference with the ambient surroundings increases which cause an increase in thermal losses. Figure (13) shows the relation between useful heat gain and mass flow rate of working fluid. The maximum value of useful heat gain is (1619 W) for (0.5 lpm) water mass flow rate and minimum value of

(1494 W) when the mass flow rate equal to 0.15 lpm. It found an increasing in the fluid mass flow rate entail increased the useful heat gain. Figure (14) shows the relation between the variation of beam solar radiation (I_b) and the thermal efficiency on 3th July. It is noted that increasing of the solar radiation causes a little decrease in the thermal efficiency. Figure (15) show the relation between thermal efficiency and mass flow rate of working fluid. The indicated maximum thermal efficiency is (80.26%) for (0.5 lpm) water mass flow rate and minimum value of (78.6 %) when the mass flow rate equal to (0.15 lpm).

The heat loss coefficient was generally dependent on the heat transfer coefficients. The heat transfer coefficients determined by measured value of ambient, glass cover and absorber surface temperatures. Figure (16) shows the variation of the heat loss coefficient along the absorber with time. It is found that the heat loss coefficient increases with time and reaches its maximum value of (8.1) at (12:00 PM) then decreases due to the decrease of the glass cover and absorber temperatures.

The heat removal factor is depended on mass flow rate of working fluid, fluid specific heat and heat loss coefficient. Figure (17) illustrates the variation of the heat removal factor with the mass flow rate for water working fluid. It can be seen that the heat removal factor increases with an increasing in the mass flow rate.

Figure (18) reports the variation of the heat removal factor with time for different days. It is found that the heat removal factor decreases with time and reaches minimum value at (12:00 PM) then increased due to that the heat loss coefficient decreased after the noon.

CONCLUSIONS

The following conclusions can be extracted:

- 1- Solar tracking system of two-axis has proved the reception of a higher energy gain and improved collector operating efficiency.
- 2- The experimental results have shown the maximum thermal efficiency of parabolic trough solar collector PTSC is (80.26%).
- 3- The maximum outlet temperature of the absorber tube was (81 °C) for solar radiation equal to (995 W/m²) and ambient temperature (47.5 °C).
- 4- The suitable mass flow rate for parabolic trough solar collector under studied equal to 0.15 lpm.

Table (1) solar exposure period in Iraq [NASA, 2008]

<i>Quantity.</i>	<i>Solar Exposure period</i>
Sunny hours	4100 hours
Sunny days	333.6 days
cloudy days	31.4 days

Table (2): Description of the Reflector

Reflector	Description
Appearance	Parabolic trough
Length	2 m
Aperture width	1.05 m
Depth of parabolic	0.27 m
Aperture area	2.1 m ²
Rim angle	92.2°
Mode of tracking	Two - axis
Concentration ratio	26.3
Thickness of glass cover	3 mm
Focal point	0.253 m

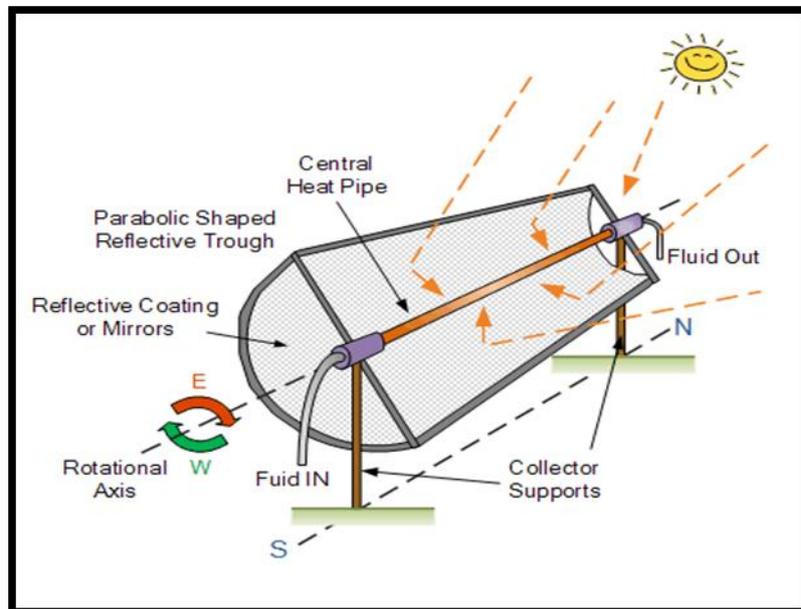


Figure (1): Component of the solar concentrator.

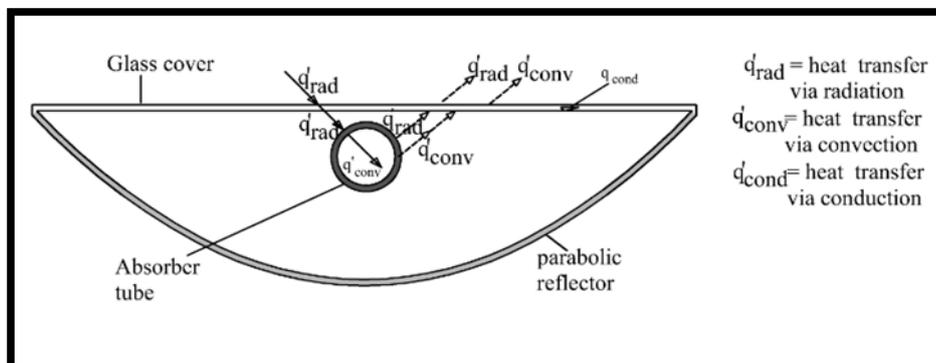


Figure (2): Heat transfer mechanisms for the concentrated collector.

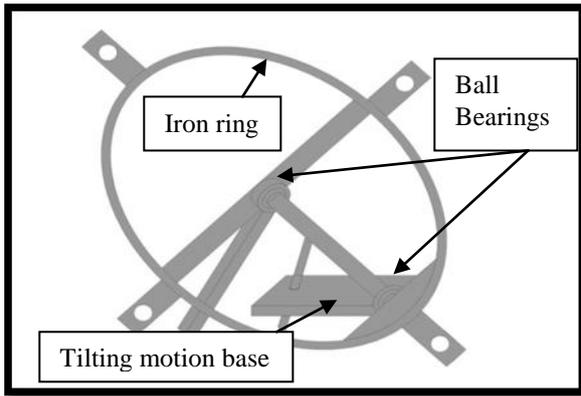


Figure (3): Moving base connected to the arm of driving motor.

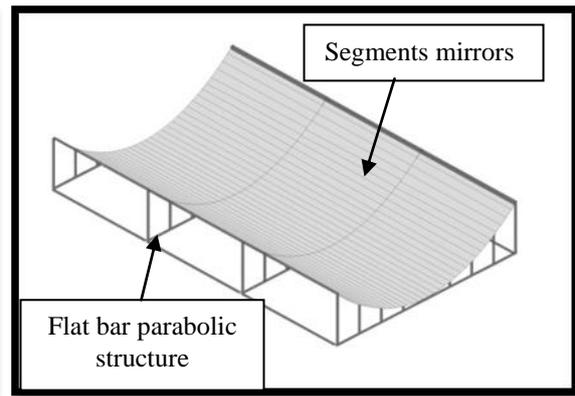


Figure (4 a): Parabolic Reflector.



Figure (4 b): Parabolic Reflector which used in the experimental work.

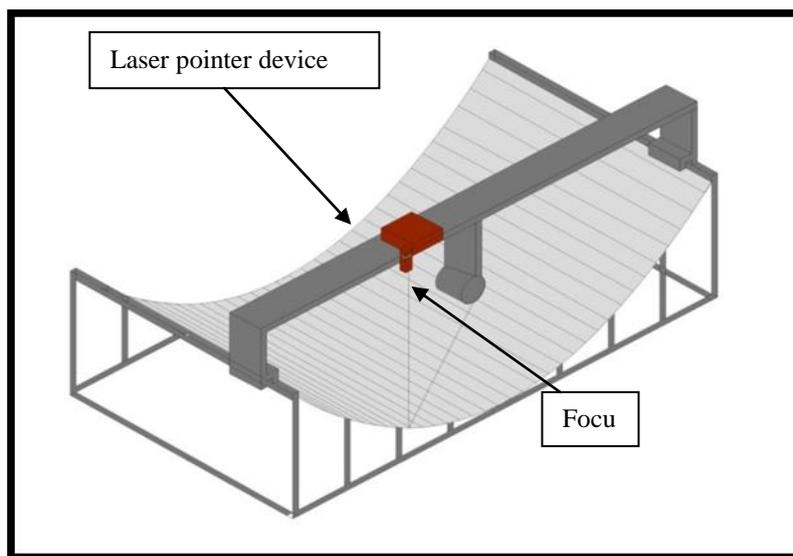


Figure (5): Apparatus for characterizing slop error of parabolic reflector mirrors.

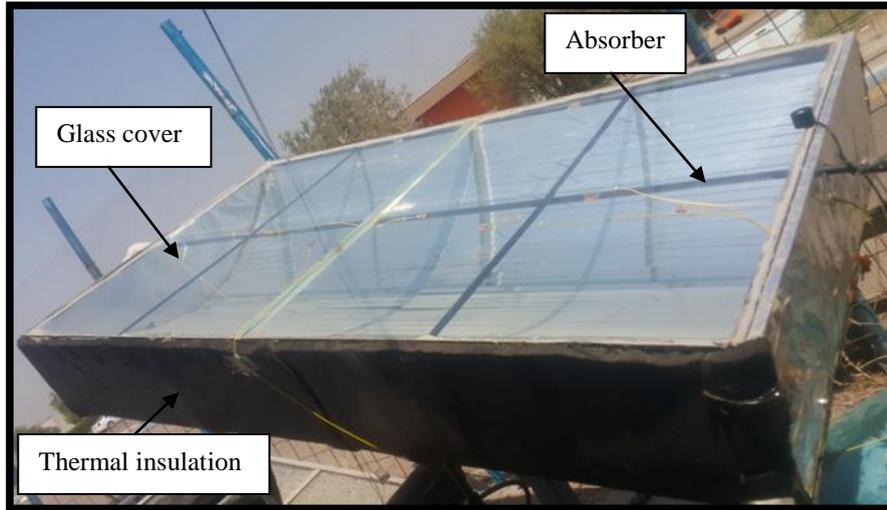


Figure (6): Final form of parabolic reflector.

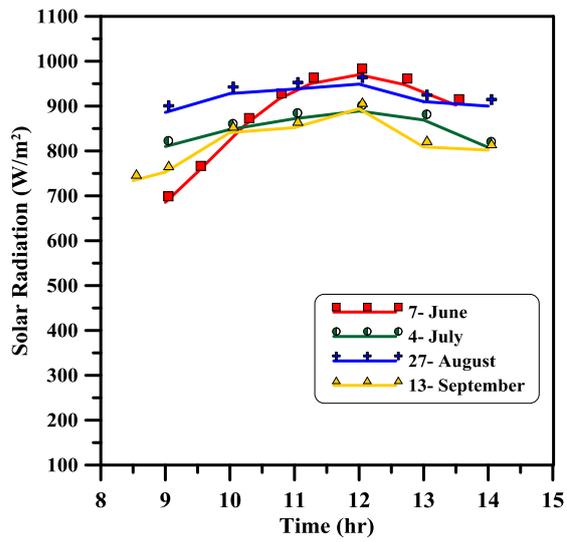


Figure (7): Variation of solar radiation with local time for different days and months.

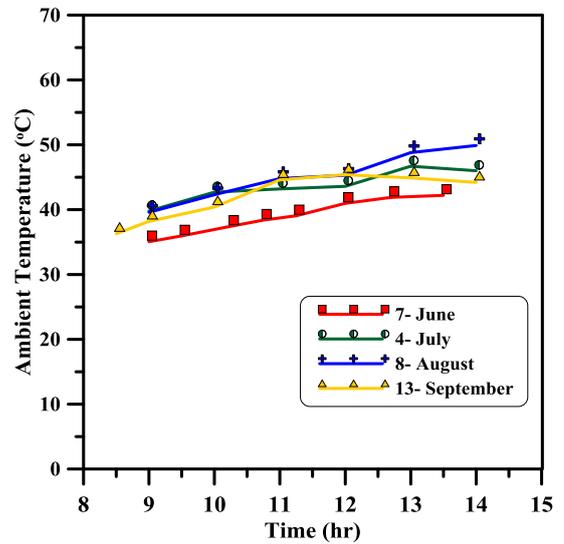
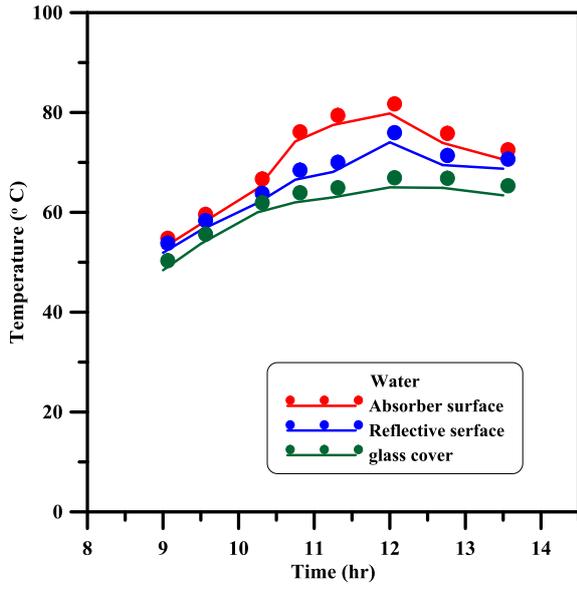
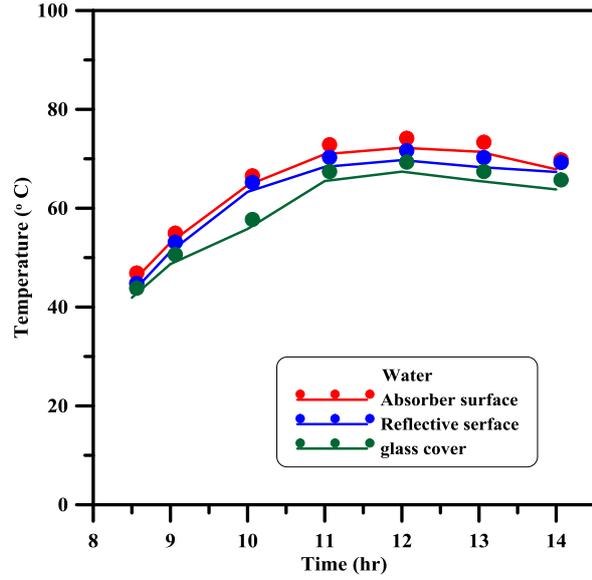


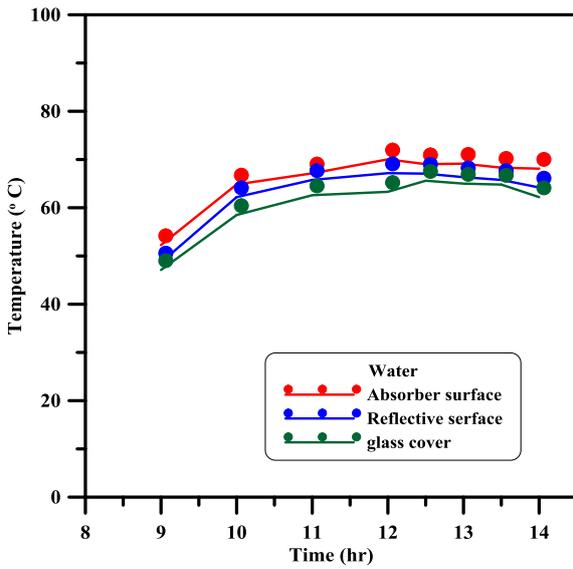
Figure (8): Variation of ambient temperature with local time for different days and months.



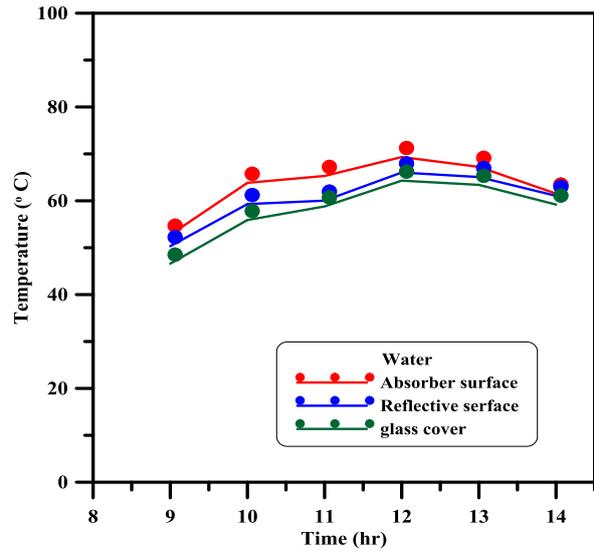
(a)



(b)



(c)



(d)

Figure (9): Variation of absorber surface, reflector surface and glass cover temperatures for different water mass flow rate a) 0.15 lpm, b) 0.3 lpm, c) 0.4 lpm, d) 0.5 lpm).

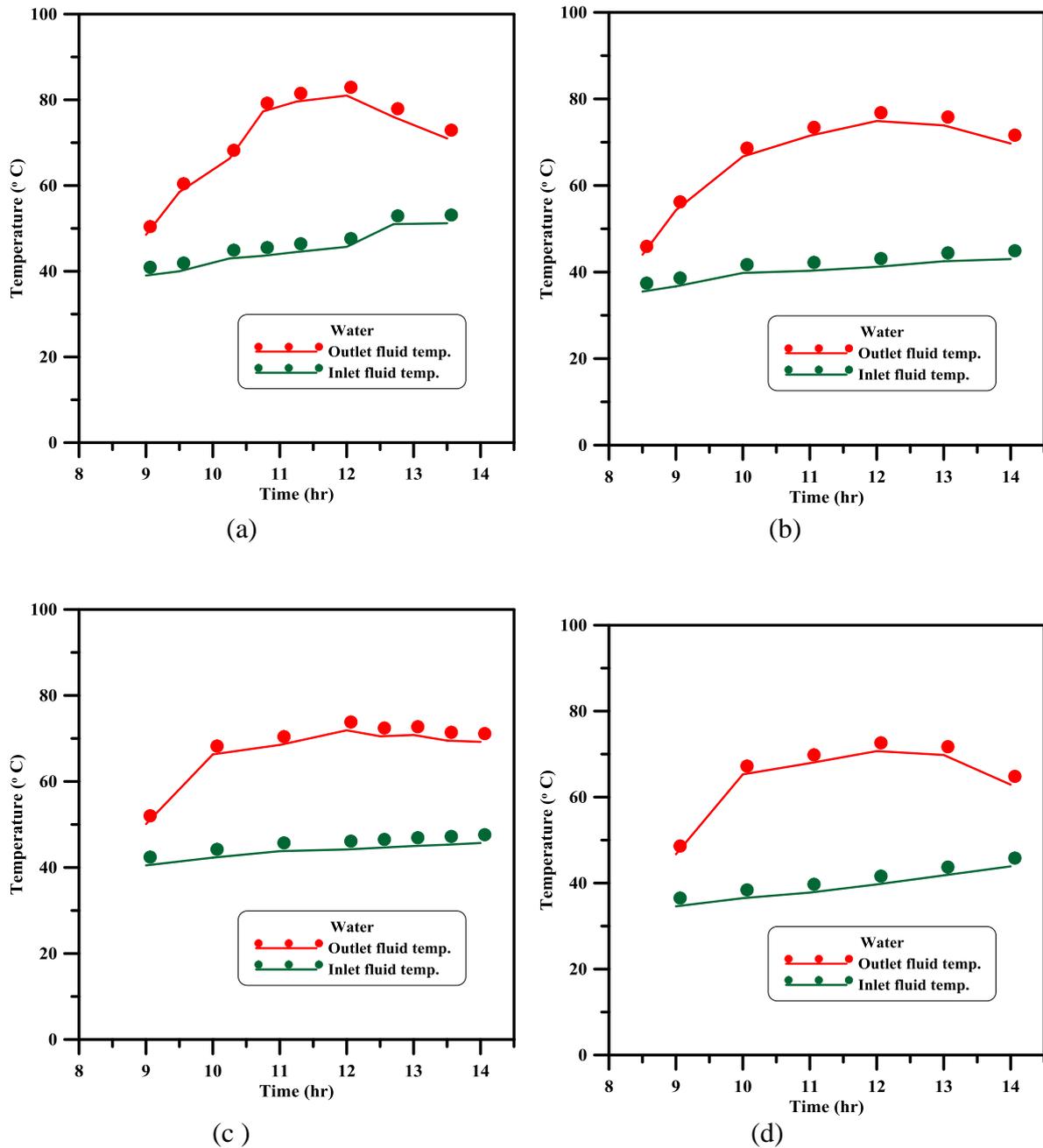


Figure (10): Variation of absorber inlet and outlet temperature for different water mass flow rate a) 0.15 lpm, b) 0.3 lpm, c) 0.4 lpm, d) 0.5 lpm.

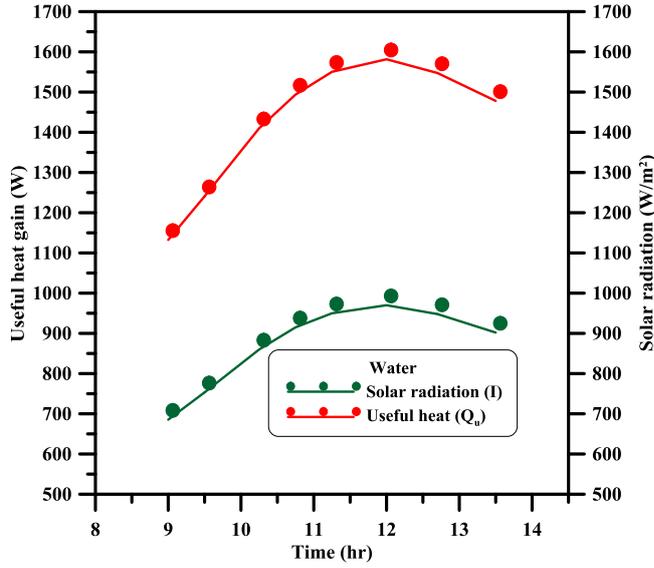


Figure (11): Variation of solar radiation and collector useful heat gain with local time on 7th June.

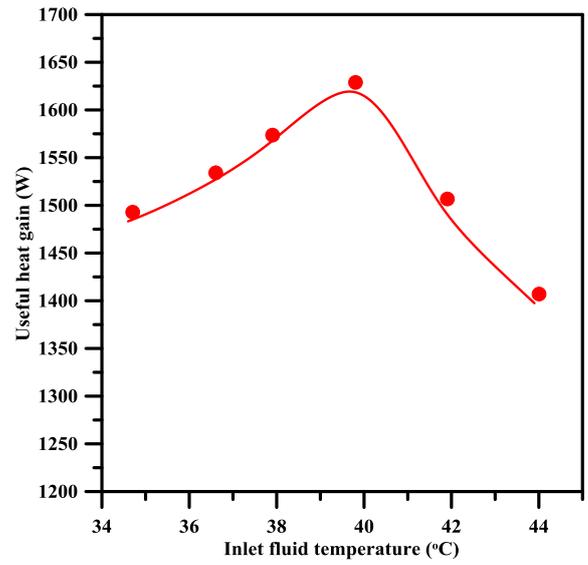


Figure (12): Variation of useful heat gain with the inlet temperature on 6th July.

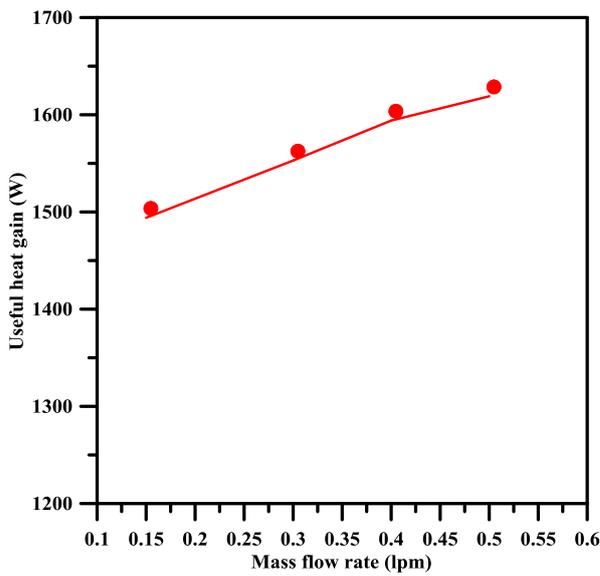


Figure (13): Variation of useful heat gain with the fluid mass flow rate.

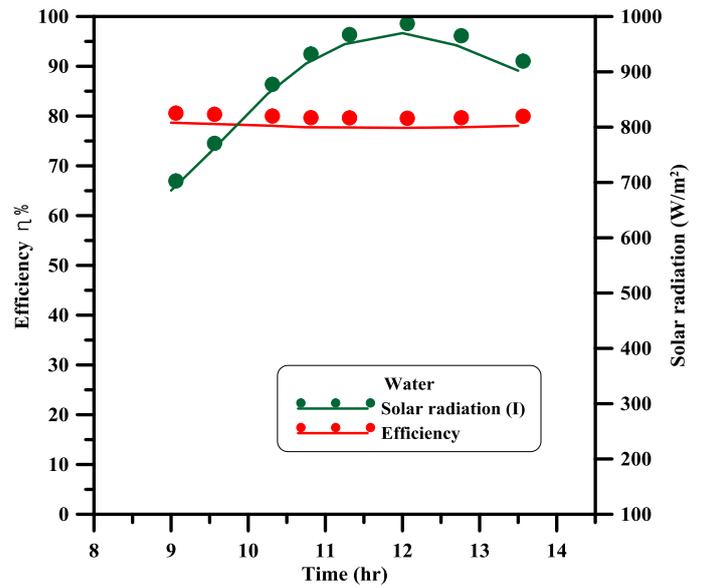


Figure (14): Variation of solar radiation and thermal efficiency with local time.

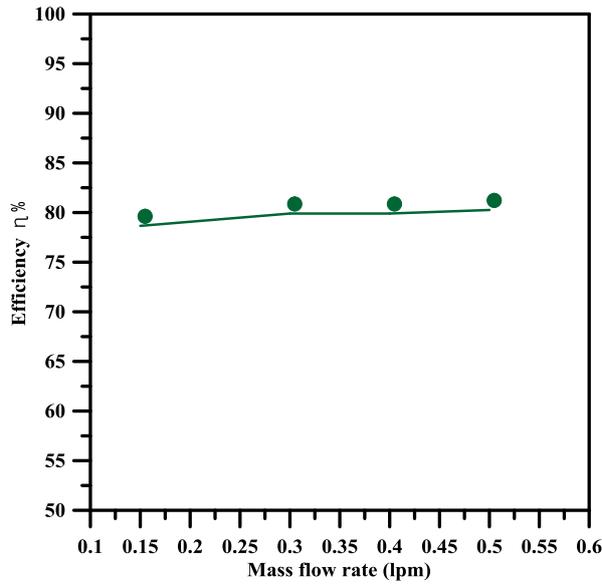


Figure (15): Variation of thermal efficiency with the fluid mass flow rate.

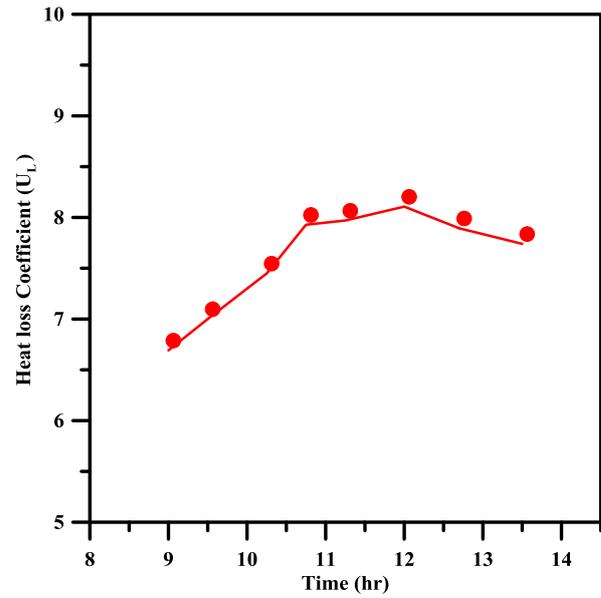


Figure (16): Variation of the heat loss coefficient along the absorber with time on 7th June.

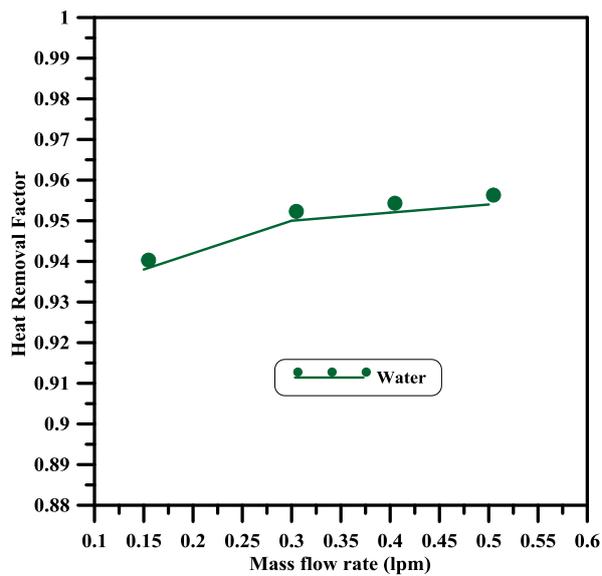


Figure (17): Variation of the heat removable factor with the mass flow rate for water working fluid.

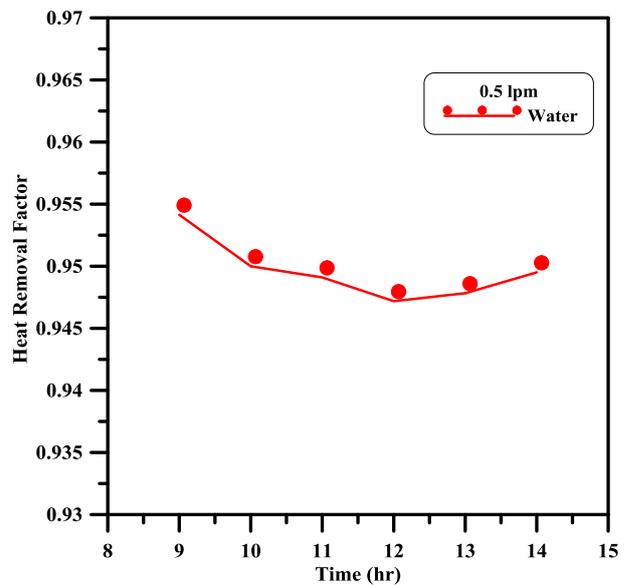


Figure (18): Variation of the heat removal factor with time 30th July.

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