THE EFFECT OF OPERATING CONDITIONS ON THE PERFORMANCE OF FLAT-PLATE THERMOSYPHONIC SOLAR WATER HEATER

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ABSTRACT

It is very important to eliminate the user risks by means of a certain performance guarantee by the contractor . A simple method is introduced for predicting the approximate change in the system performance parameters due to a change in an operating condition and also justifying a suitable mathematical model which can describe the system behavior in the case of no load condition on the basis of restricted information . The model is verified by comparing its results with the published experimental work . The results showed that the instantaneous and storage efficiencies depend mainly on the design parameters , while the effect of operating factors must be taken into consideration .

KEYWORDS

Solar energy; heating ; thermosyphonic systems ;modeling ,design.

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INTRODUCTION

It is very important to eliminate the user risks by means of a certain performance guarantee by the contractor . Also it is very important for the user to have a simple method to compare between many offers on the basis of his own conditions . This needs to perform an easy method to predict the approximate change in the system performance parameters due a change in one or more of the operating conditions . For the manufacturer a suitable mathematical model which can describe the system behavior on the basis of restricted information and optimize its design according to the user requirements (by solving the reversed problem) is needed . Such work for a thermosyphonic system is not found in literature .

A considerable amount of work has been carried out in attempting to predict the thermal performance of the solar water heaters. The more notable ones and those relevant to the present work are discussed below:

Ong performed two studies [2] and [3] to evaluate the thermal performance of a solar water heater. In the first study [2] he considered the linear model,while in the second study [3] he improved his first theoretical model by dividing the system into sections . Ong's work appears to be the first detailed study dealing with the thermosyphonic systems ,but due to the very basic scheme of finite difference he used, the time step is very small (0.6 minute) which is impractical for a long term simulation .

Shitzer and others [4] ,[5] performed two studies . The first study [4] deals with the method of solving the differential energy equation and the coupled momentum equation to obtain the steady state linear and non-linear temperature distributions to predict the volume flow rate variation. The second study [5] provided the needed experimental information about the system behaviour,which support the assumptions given by Ong [3].They measured thermosyphonic flow by using a thermal dissipation tracing method which introduced an error about 1.0 mm of water at a Reynold's number of 1100 [8]. The mass flow measurements in [5] were found to be approximately 30 % higher than the theoretical results during the period of steady flow near the solar noon . Shitzer and others [5] , also observed that the flow rate fluctuated considerably even when other factors were constant and suggested that this may be due to hydrodynamic instability in the collector -storage tank system.

Haung and others [7] presented a simplified simulation program following Close's model [1] . The number of sections in the tank was taken 5 only and the used time step was 15 minutes.

Morrison and others [8] compared theoretical predictions of flow rate in thermosyphonic solar collectors with experimental measurements obtained by using a laser doppler anemometer.

Tsiligirls [14] presented a simple computer model suitable for design of large solar water heating systems of forced circulation type .

The present work aims to study the change of system parameters due to changes in one or more of operating conditions that affect the thermosyphonic system performance parameters , described by the manufacturer .

The Change of System Performance Parameters :

It is very important to predict approximately the resultant percentage change in the value of system parameters due to the change of an independent parameter or more . A system parameter (S_t) is a parameter which describes a system property behaviour during sunshine hours , such as the rate of heat gained by the flat plate collector Qg , the rate of heat removed from the collector Qr ; the thermosyphonic head Hth, The thermo- syphonic mass flow rate M ,the instantaneous (collector) efficiency ζ_c, the system storage efficiency , ζ_{ov} , the water temperature rise through the collector ΔT_c and the mean tank temperature TT . The change of any of system performance parameters depend on how much operating conditions deviates from the manufacturer conditions .

The operating conditions can be regarded as independent parameters (X) , each has its own independent effect on every system performance parameter, such as the solar intensity I_{Tt} , the transmissivity of glass cover system , τ , the flat-plate heat loss factor ,Hc , the friction factor ,fric, the Nusselt number , Nu ,the wind velocity based on 1 m/sec ,wind ,and the fouling thermal resistance , foul .

A system performance parameter (S_t) is a time dependent variable . For simplifying the problem , its value as near as possible to its maximum point is taken as a reference point and is assumed time independent . The value of a system parameter (S)- at its reference time - is a function of the independent variables ($x_1, x_2, x_3 \ldots$) i.e.

$$S = S (x_1 , x_2 , x_3 , \ldots\ldots x_n) \qquad (1)$$

Let w_s be the change in a system parameter and w_1, w_2 , $w_3, \ldots w_n$ be the changes of the independent variables .The system parameter change will be :

$$dS = \frac{\partial S}{\partial x_1} dx_1 + \frac{\partial S}{\partial x_2} dx_2 + . \frac{\partial S}{\partial x_n} dx_n \qquad (2)$$

The magnitude of this change :

$$w_s = \left[\sum_1^n \left(\frac{\partial S}{\partial x_i} dx_i \right)^2 \right]^{\frac{1}{2}} \qquad (3)$$

The relative system parameter change form is :

$$\frac{dS}{S} = \left[\sum_1^n \left(\frac{\partial S}{\partial x_i} \frac{dx_i}{x_i} \frac{x_i}{S} \right) \right] \qquad (4)$$

and its relative magnitude :

$$\Delta_s = \frac{w_s}{S} = \left[\sum_1^n \left(\frac{\partial S}{\partial x_i} \frac{x_i}{S} w_i \right)^2 \right]^{\frac{1}{2}} \qquad (5)$$

or

$$\Delta_s = \left[\sum_1^n \left(F_i \; w_i \right)^2 \right]^{\frac{1}{2}} \qquad (6)$$

where

$$\Delta_s = \frac{w_s}{S} \; , \; w_i = \frac{dx_i}{x_i} \; , \; F_i = \frac{\partial S}{\partial x_i} \frac{x_i}{S} \qquad (7)$$

F_i could be defined as the relative change in a system parameter resultant due to a unit relative change in an independent parameter i , while keeping all other variables constant .

w_i represents the relative change in the independent parameter i The resultant uncertainty , Δ_s is the relative change in a system parameter resultant due to relative change in many independent parameters .

CALCULATION OF SYSTEM PERFORMANCE PARAMETERS :

The layout of a typical thermosyphonic solar water heater system and the hypothetical temperature distribution of the flow circuit are represented schematically in Fig. 1 . Although the system has few components,its performance depends on many factors which are varied with time. The model follows the same assumptions as in [6] . A small time interval is taken during which the time dependent variables experience a very small variation,thus the problem can be treated during the mean time interval as a steady state problem.

Considering the energy balance of a section number J in fig. 1 which has a water equivalent W_J and mean temperature ,T_J , after time interval , $\Delta\theta$, is given by :

$$W_J \frac{dT_J}{d\theta} = (Q_{j,i} - Q_{j,o}) \qquad (8)$$

and

$$\frac{\Delta T_J}{\Delta \theta} = \frac{1.}{W_J} (Q_{j,i} - Q_{j,o}) \qquad (9)$$

The finite difference scheme which was used in Ong [2],[3], and [6] is :-

$$T_{(K+1,J)} = T_{(K,J)} + \frac{\Delta \theta}{WJ} (Q_{(K,J),i} - Q_{(K,J),o}) \qquad (10)$$

In the present work the suggested finite difference scheme is:-

$$T_{(K+1/2,J)} = T_{(K,J)} + \frac{\Delta \theta}{2\ WJ} (Q_{(K,J),i} - Q_{(K,J),o}) \qquad (11)$$

$$T_{(K+1,J)} = T_{(K+1/2,J)} + \frac{\Delta\theta}{2 W_J}\left(Q_{(K+1/2,J),i} - Q_{(K+1/2,J),o}\right) \qquad (12)$$

where

for flat-plate collector (i.e $J = 1$)

$$(Q_{J,i} - Q_{J,o}) = A_c F_c (q_a - H_c (T_1 - T_a)) - \dot{M} C_p (t_{12} - t_{11}) \qquad (13)$$

for the upriser pipe (i.e $J = 2$)

$$(Q_{J,i} - Q_{J,o}) = - U_J (T_J - T_a) - \dot{M} C_p (T_2 - t_{12}) \qquad (14)$$

for any other section ($J \geq 3$)

$$(Q_{J,i} - Q_{J,o}) = - U_J (T_J - T_a) - \dot{M} C_p (T_J - T_{J-1}) \qquad (15)$$

The instantaneous efficiency , ζ_i , is a measure of the heat removed from the collector ,while the system storage efficiency , ζ_{ov} , is the ratio between the increase in thermal energy stored in the tank and the total sum of radiant energy falling on the collector surface up to the time considered.

$$\zeta_i = \frac{M^{*}\ Cp\{ t_{12} - t_{11} \}}{I_{Tt}\ A_c} ,$$

$$\zeta_{ov} = \frac{W_{wt} \{ T_m - T_m^{*} \}}{\int_{-Hr}^{H} A_c\ I_{Tt}\, d\theta}$$

$$\bar{T}_m = \text{mean tank temperature} = \frac{\sum_{J}^{JJ-1} \{ W_{VJ}\ T_J \}}{\sum_{J}^{JJ-1} W_{VJ}} \quad (16)$$

THERMOSYPHONIC FLOW RATE :

The thermosyphonic head which causes the flow to circulate is indicated by the shaded area on temperature height distribution and equals

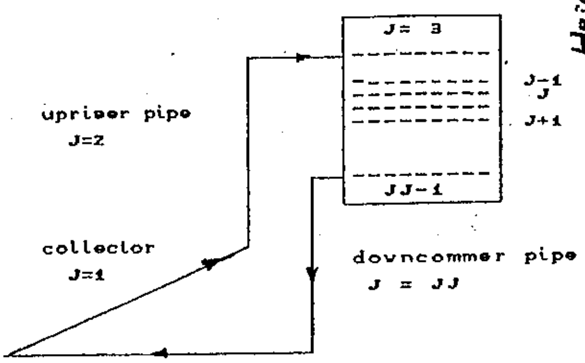
$\oint (1-\gamma(t))dz$. The area enclosed consists of :

1 - Positive area $Lc*\sin(B)*(1-\gamma(t1))$

2 - Positive area $(Z_{2} - Lc*\sin(B))*(1-\gamma(t2))$

3 - Negative area $\int_{z1}^{z2} (1-\gamma(t))dz$

4 - Negative area $(Z1\ )*(1-\gamma(t11))$



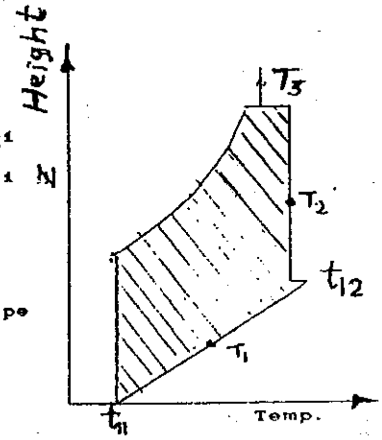
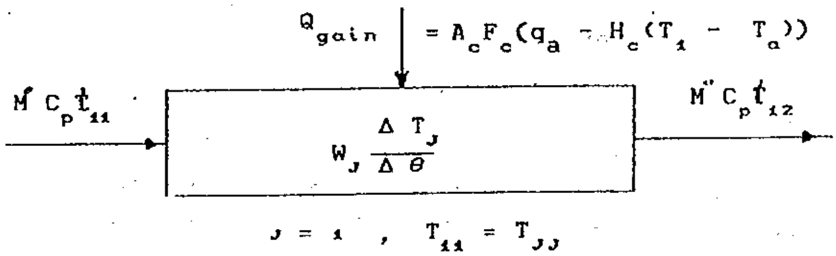
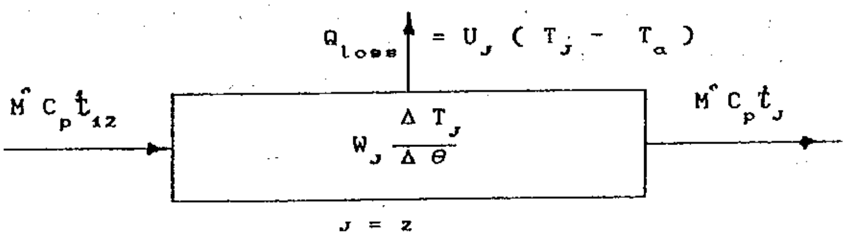


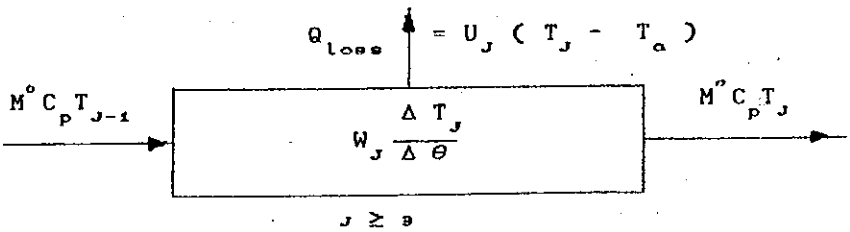
Fig.1-a Layout of a typical thermo-syphonic solar heating system. Fig.1-b Hypothetical temp.distribution



The flat-plate solar collector



The energy balance of the upriser pipe .



1-c The energy balance of the system sections .

This is balanced by the head loss due to friction in the collector pipes and headers and secondary losses through the system .The secondary head loss through a hydraulic resistance of length ℓ and inner diameter D and for fully developed flow is represented by an additional equivalent length $\Delta\mathcal{L} = C\ D\ /f$ where f is coefficient of friction through the concerned hydraulic resistance and C is the secondary head loss coefficient . The head loss for a fully developed laminar flow through a hydraulic resistance of actual length ℓ :

$$h = 128 * \mu *(\ell +\Delta\ell) /(g*\rho^2*\pi*D^4)* M^{\bullet}$$

RESULTS AND DISCUSSIONS

The specifications of the system mentioned in [5] and also the location parameters of the lab. were fed to the computer program as shown below :

The collector plate and pipes are made from steel of 1 mm thickness , insulated by the glass wool from one side and black painted and covered with a single commercial glass of thickness 2 mms . The pipes are of internal diameter 1.25 cm , length 168 cm and are placed with pitch 12.5 cm .The header length 1 m and inner diameter 18 mms The collector system consists of 2 parallel panels .The area of each panel 1.18 m2

The length of the downcommer pipe 2.75 m and of the upriser pipe 2 m with inner diameter for both 18 mms with wall thickness 1 mm and wall insulated with thickness 5 cm of glass wool . The storage tank is a cylinder of height 89 cm and volume 140 liters . The bottom tank section is 5 cm height . The top tank section is 1/3 of the whole tank . The bottom of the tank and the end of the solar collector are at the same level .

The solar collector is tilted by 35 degree with horizontal and south facing . The latitude of the lab. is 32 . The calculated collector sun rise time is 6 hr and the sunset time is 17.93 hr solar time . The date is 27 July i.e. day number 209 considering 1 January equal day number 1 .

In references [3]and [6] the number of tank sections and the time step for numerical calculations should be carefully selected to avoid numerical instability . The time interval was 0.6 minutes . The presented scheme of finite difference was tested as shown in fig. 2 . It is found in the present work that ,for time steps up to 4.6 minutes , the stability of solution can be achieved as shown in fig. 2 . The present simulation was carried out with 13 tank sections and the total number of sections was 16 and the time step was 2.4 minutes .

The maximum values of each of system performance parameters occur at a time depends mainly on the time of maximum solar intensity . Also , from many numerical experiments it is noted that the system parameter curves are not intersected at different values of an independent variable .Therefore each system parameter for a reference value of an independent variable could be characterized by its maximum value point or closer to it as given in Table 1 .

Table 1 The reference time of the system parameters

I_{Tt}	Qgo	Qr	Hth	FLOW	ζi	ζov	ΔTc	T_T
12.	11.	11	11.	13.	10.	12.	11.	17.

A series of numerical experiments were carried out for calculating numerically the factor F_i for the main effective independent variables and performing the change propagation through the system performance parameters ,each corresponding to its own reference point . The independent variables, x_i, which are considered in this work are :

1- The solar intensity I_{T} $(i = 1)$
2- The transmissivity of glass cover system , τ , $(i = 2)$.
3- The flat-plate heat loss factor ,Hc , $(i = 3)$.
4- The friction factor ,fric, $(i = 4)$.
5- The Nusselt number , Nu , $(i = 5)$.
6- The wind velocity based on 1 m/sec , wind , $(i = 6)$.
7- The fouling thermal resistance , foul

The effect of the previous independent variables on the system performance parameters had been investigated and presented through figures 3 - 15 . The effect of solar intensity relative deviation on the hourly variation of the thermosyphonic volume flow rate during sunshine hours is shown in fig. 3 . The effect of solar intensity relative deviation on the hourly variation of the instantaneous collector efficiency during sunshine hours is shown in fig. 4.

The effect of solar intensity relative deviation on the hourly variation of the storage efficiency rate during sunshine hours is shown in fig. 5 Although in each figure the curves are very close but the validity of the assumption that the reference value of a system performance is a time independent variable is clear .

The effect of solar intensity relative deviation on the absolute reference value of system performance parameters is demonstrated through figures 6 - 9 . Its effect on the rate of heat gained by the collector and also the rate of heat removed from the collector to upriser pipe is shown in fig. 6 .Also its effect on the thermosyphonic head generated due to temperature distribution through water paths in fig. 7 , on the thermosyphonic volume flow rate based on the inlet collector temperature , and on the instantaneous and overall efficiencies in fig. 9 . In all figures approximate linear relations are noted

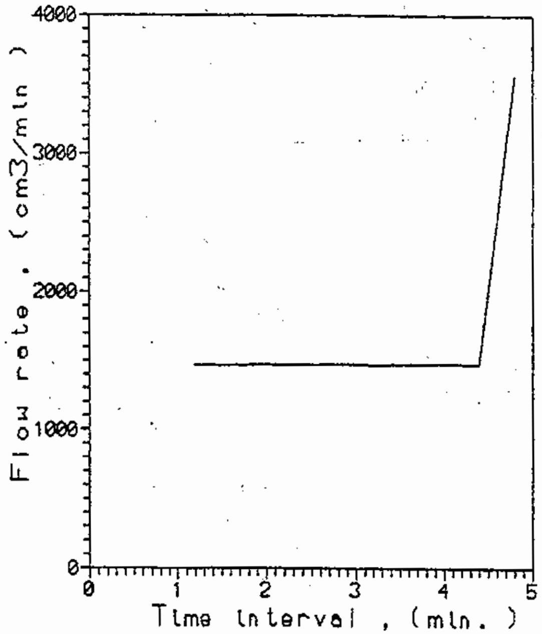


Fig. 2 The effect of time step in minutes on the calculated thermosyphonic flow at 1.pm

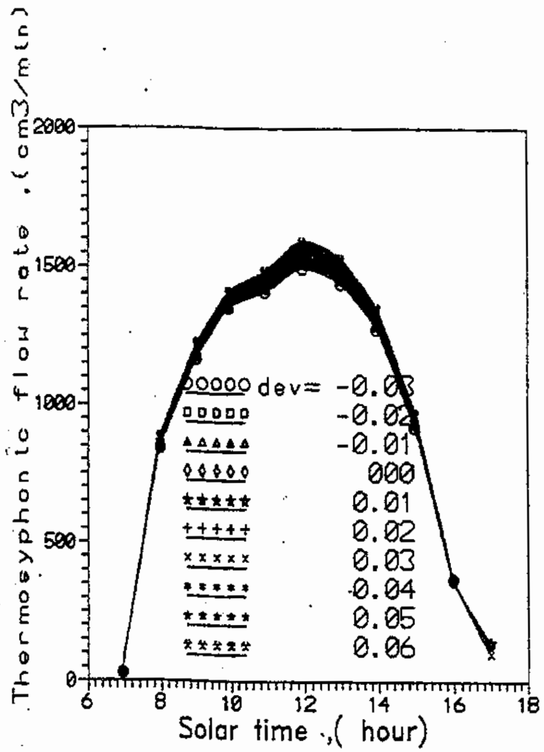


Fig.3 The effect of solar intensity deviation on the thermosyphonic volume flow rate ,cm^3/min.

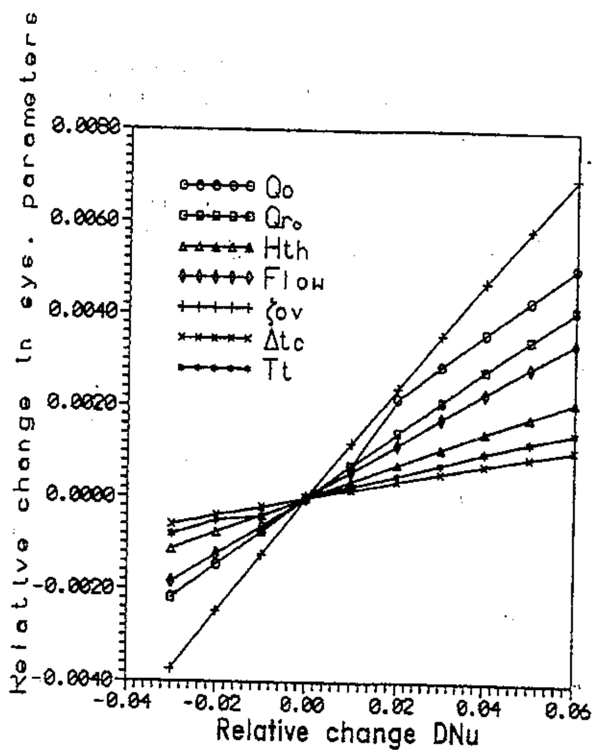


Fig.14 The dependence of relative change of main system performance parameters on relative change of Nusselt number of the flow through collector pipes

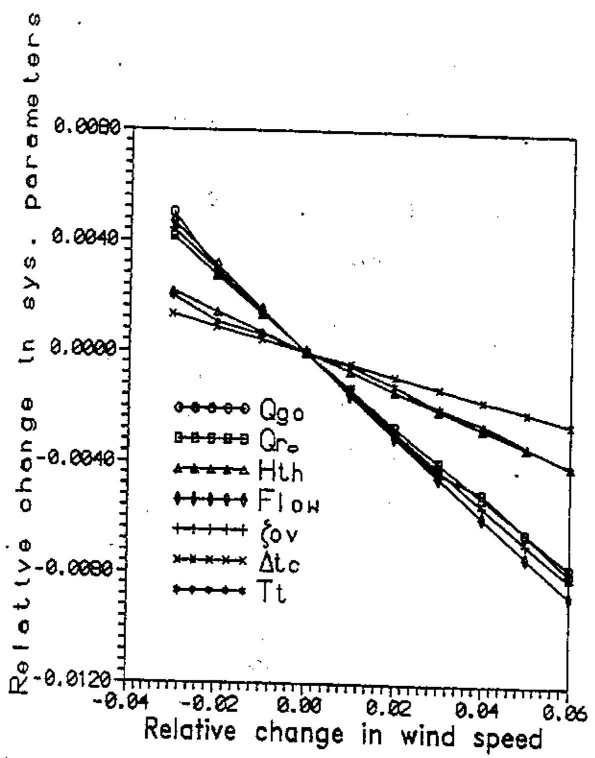


Fig.15 The dependence of relative change of main system performance parameters on relative change of wind speed based on 1 m/sec .

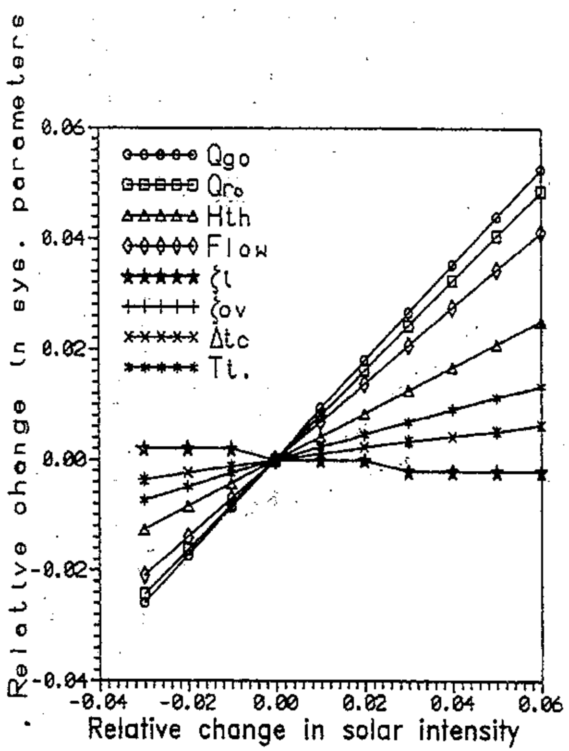


Fig.10 The dependence of relative change of main system performance parameters on relative change of solar intensity.

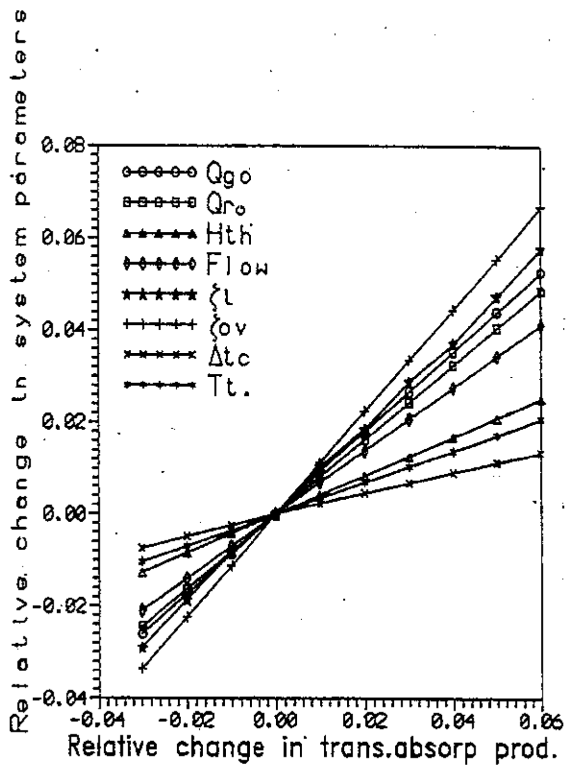


Fig.11 The dependence of relative change of main system performance parameters on relative change of transmissivity-absorptance product.

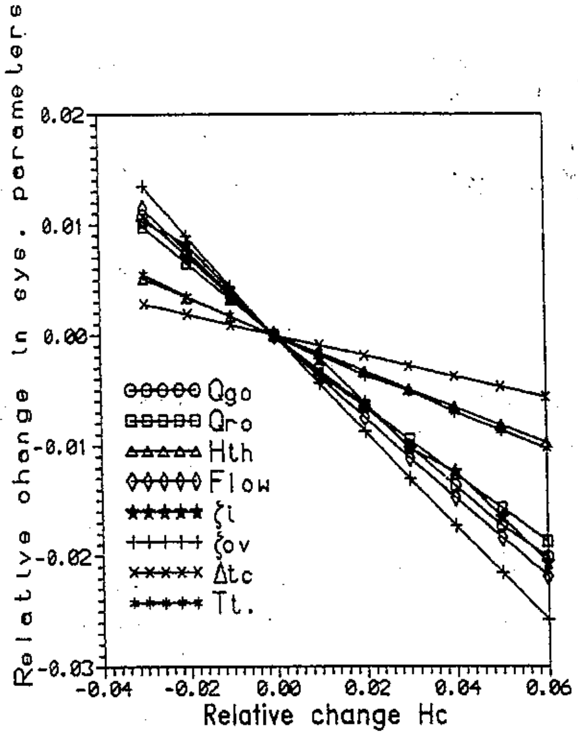


Fig.12 The dependence of relative change of main system performance parameters on relative change of flat plate heat loss coefficient.

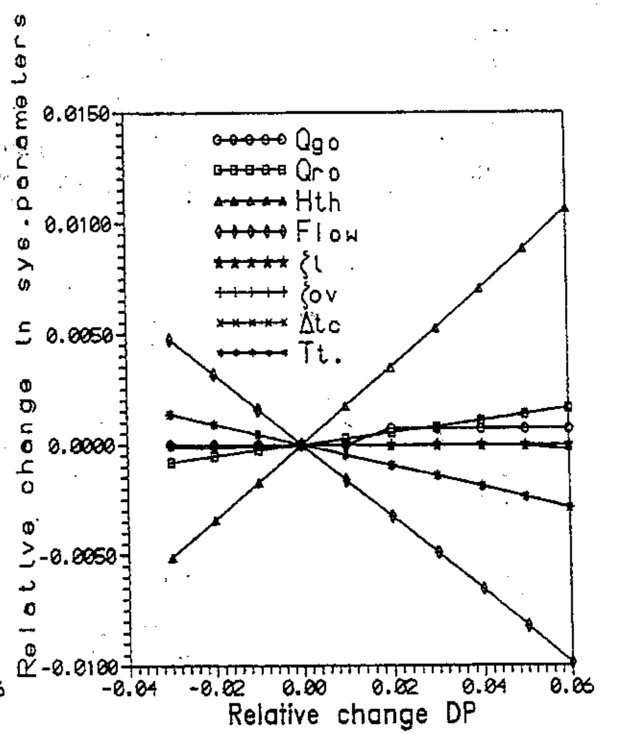


Fig.13 The dependence of relative change of main system performance parameters on relative change of pressure drop through system paths.

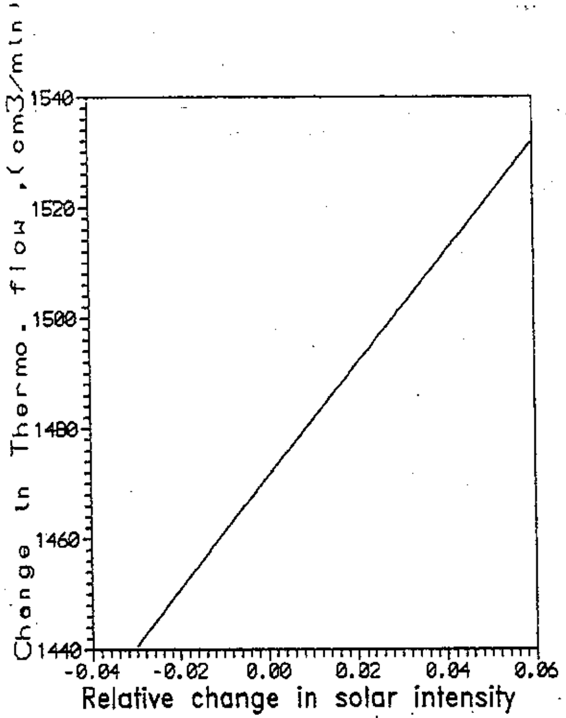


Fig.8 The change of rate of thermosyphonic flow generated due to relative change of solar intensity

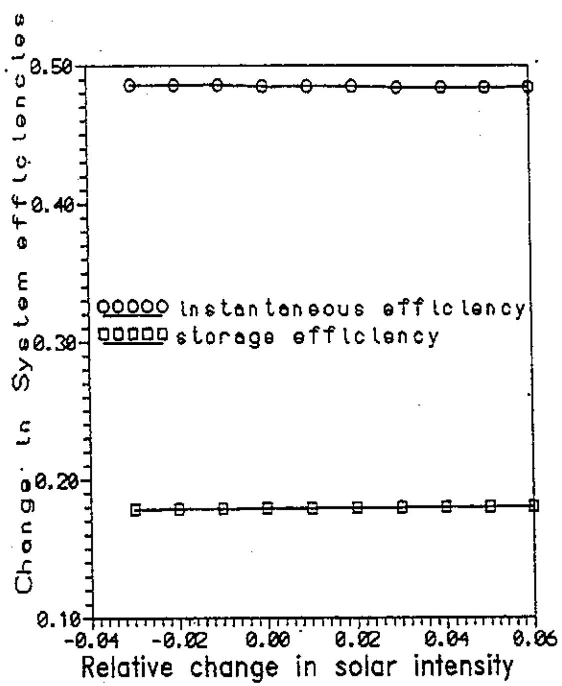


Fig.9 The change of instantaneous and storage efficiencies due to relative change of solar intensity

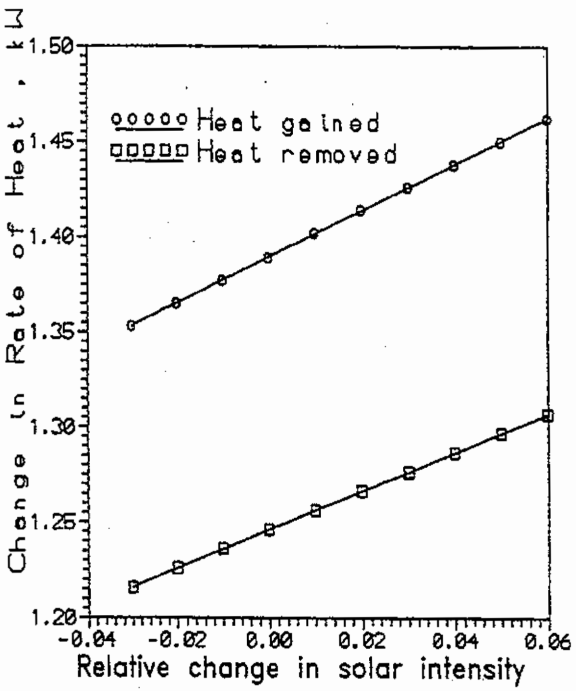


Fig. 6 The change of rate of heat gained by the collector and the rate of heat removed from the collector due to relative change of solar intensity

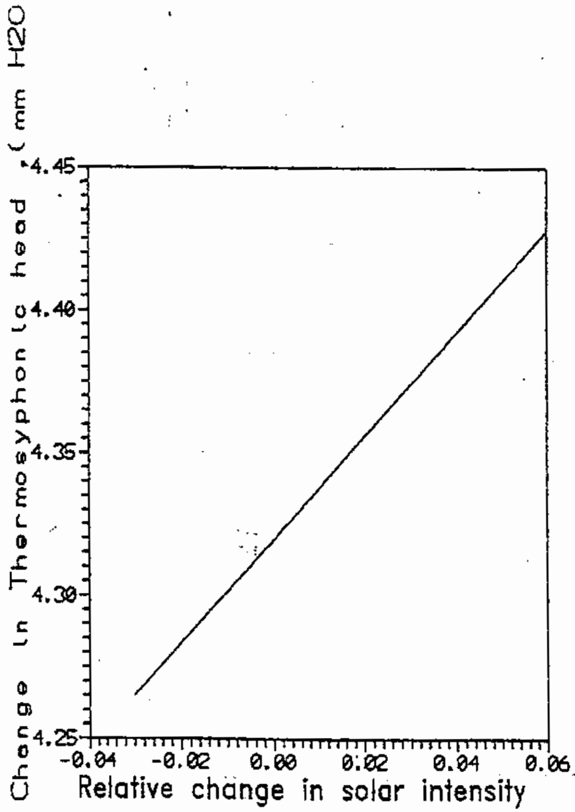


Fig.7 The change of rate of thermosyphonic head generated due to relative change of solar intensity

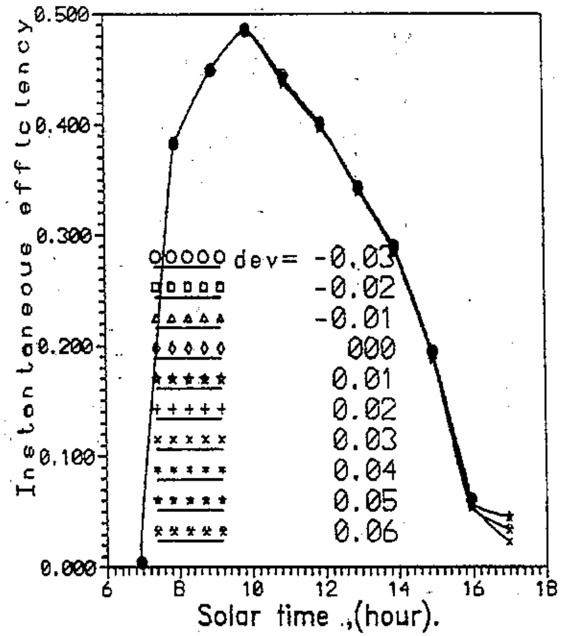


Fig.4 The effect of solar intensity deviation on the instantaneous collector efficiency .

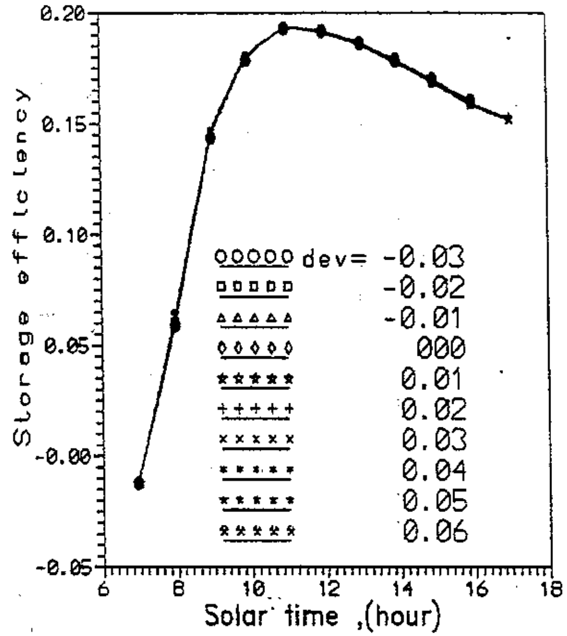


Fig.5 The effect of solar intensity deviation on the system storage efficiency .

but with different slopes. This observed linearity is due to the small interval of relative change of I_{Tt} .

The effect of relative change of many independent variables on the relative change of performance parameters at their reference values is shown through figures 10 - 15 . The slope of a line represents the factor F defined by eq. 7

The results are summarized in Table 2. .

Table 2 Parameter F

Δ_s	F_{ITt}	F_{τ}	F_{Hc}	F_{fric}	F_{Nu}	F_{vind}	F_{foul}
Δ_{Qgo}	.9359	.9359	-.3600	.0000	.0720	-.1440	.0000
Δ_{Qro}	.8126	.8126	-.3196	.0273	.0722	-.1315	.0007
Δ_{HTH}	.4219	.4219	-.1660	.1736	.0376	-.0679	.0008
Δ_{FLOW}	.6920	.6920	-.3761	-.1614	.0579	-.1537	-.0005
Δ_{η_i}	.0000	1.031	-.2062	.0000	.2062	.0000	.0000
$\Delta_{\eta_{ov}}$	.1152	1.116	-.4379	-.0466	.1207	-.1419	-.0007
Δ_{DTC}	.2347	.2347	-.0941	.1783	.0189	-.0429	.0000
Δ_{TT}	.3520	.3505	-.1833	-.0017	.0272	-.0478	-.0014

In Table 2. ,the nil values of F_i means that the system parameter S is not sensitive to the variation in the factor x_i . If a system parameter has a value S (the rate of heat gained by the collector Q_{go} as an example) and an independent variable (the solar intensity as an example) is changed by + 5 % as an example , then the maximum value which occurs at the reference time 11 hr solar time is expected to increase by $F_i*\Delta_i*S_o$ = o.9359*0.05 * Qgo and the new value will be , $(1.0+F_i*\Delta_i)*S_o=(1.0+\ 0.9359*0.05)*Q_{go}$

If the change in more than one parameter then the change will be

$$\Delta = \sqrt{(F_1*\Delta_1)^2 + (F_2*\Delta_2)^2 + (F_3*\Delta_3)^2}$$

and the new value will be $(1.0 + \Delta)* S$

From Table 2.it is noted that the wind velocity has a great effect on the calculated thermosyphonic flow. . In this work ,it is assumed that the system is new i.e the fouling resistance could be neglected, also the wind velocity assumed 1. m/sec which is reliable in many times in summer (the wind speed is not mentioned in [5]). Also,it is noted the importance of daily maintenance of the glass cover and cleaning it from time to time. The heat loss coefficient has a great effect on the mean tank temperature . Also, for the wind effect ,the designer should take into account the wind effect as shown in Table 2 .

The above table could be used also for analyzing the reasons of deviation between the experimental measurements and the calculated results .

Fig.s 16 - 24 show the comparison between the model predictions and experimental measurements in[5], from which a good qualitative agreement is noted. The calculated solar intensity is less than the measured one as shown in Fig. 16 at noon with about 10 % ,which is referred to the method itself.
Fig. 18 shows the comparison between the collector inlet and outlet temperatures and the measured ones . The comparison between the measured water temperature variations at the top of the storage tank ,and the calculated ones is shown in fig. 20 . It is noted that the predicted mean temperatures are less than the measured ones . This is referred mainly to the error in calculating the intensity of solar radiation as shown in table 2 .

Fig. 24 shows the variation of the calculated rate of heat gain and also the comparison between the calculated rate of heat removed from the collector and that calculated from the experimental measurements . The comparison shows a very large deviation (with about 50 % at 11 am.). The source of such large deviation due to (by the help of table 2) the overestimation of solar intensity and under-estimation of collector top heat loss coefficient .

In this work other values for equivalent lengths of the local hydraulic resistances due to bends and T-sections are chosen from [13] which are too smaller than that used in [6] . Fig. 19 shows the comparison between the predicted water temperature rise ,ΔT_c, and that obtained from the experimental work. The predicted ΔT_c is higher than the measured one specially around noon .

From Fig.23 the maximum error at noon is about 30 % with respect to the mean measured value of the thermosyphonic volume flow rate ,but away the noon time the error decreases . It should be mentioned that the experimental results show scattering values for the thermosyphonic flow + 30 % ,- 20% of the mean value and for the instantaneous efficiency ± 10 % , also the error of the flow measuring method is about 1mm of water at Re = 2100 (noting that the maximum thermosyphonic head is about 6.8 mms of water as shown in Fig. 10) which means that the measured flow is not very accurate . Any how , the representation of ΔT_c becomes better than [6] while at the same time the representation of the thermosyphonic volume flow rate becomes worse .The other variables has not notable changes especially the rate of heat removed from the collector. This explains the previous note .

Fig. 23 shows the comparison between the calculated instantaneous efficiency , ζ_i ,and that obtained from the experimental measurements in [5]. The predicted instantaneous thermal efficiency is always higher

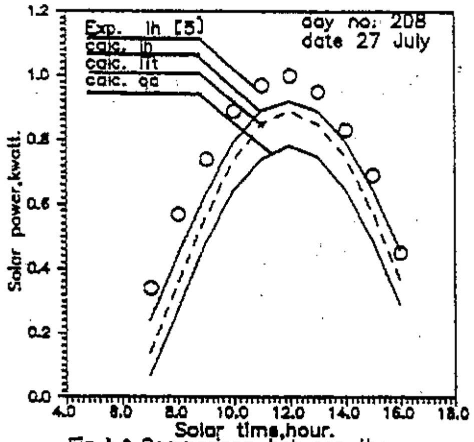


Fig.16 Comparison between the measured and the calculated values of solar power.

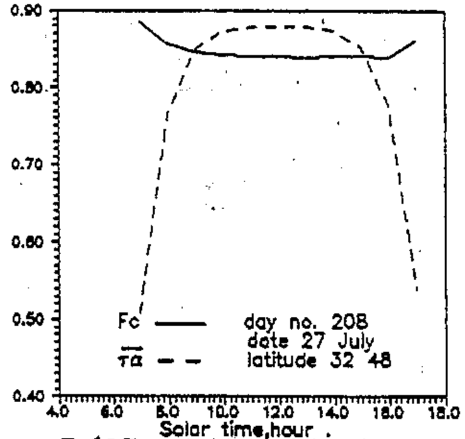


Fig.17 The calculated variation of transmittence–absorptance and flat plate efficiency factor.

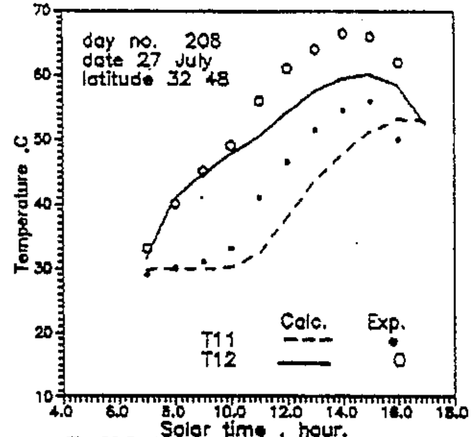


Fig.18 Collector inlet & outlet flow temperatures compared by that obtained by Zvirin [5].

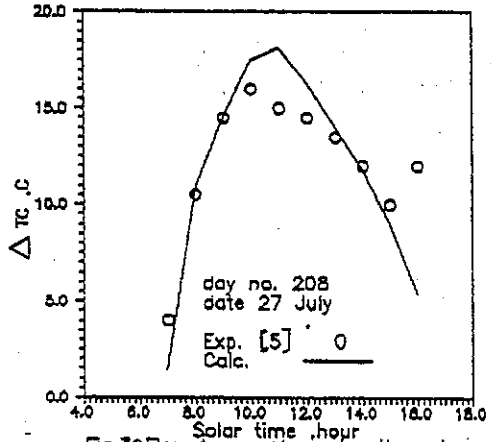


Fig.19 Flow temperature rise through the collector compared by that obtained by Zvirin [5].

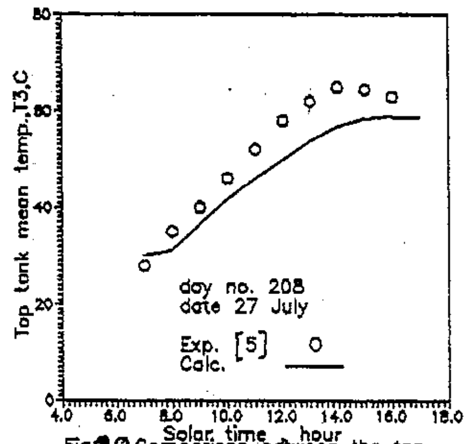


Fig.20 Comparison between the top tank temperature and that measured by Zvirin [5].

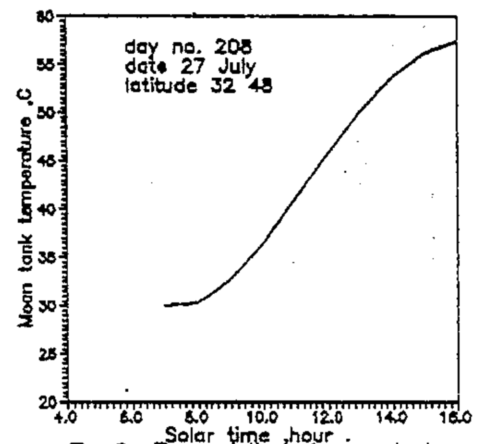


Fig. 21 The variation of mean tank temperature according to experimental set up data [5].

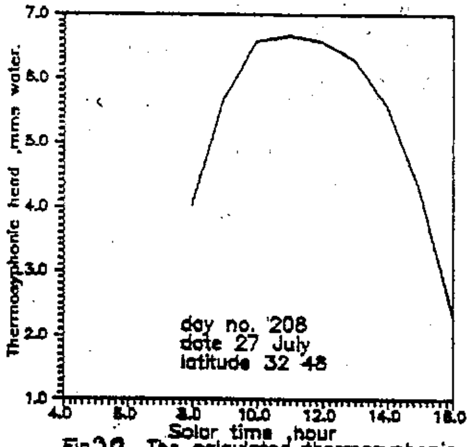


Fig.22 The calculated thermosyphonic head according to Zvirin [5] set up data in mms water.

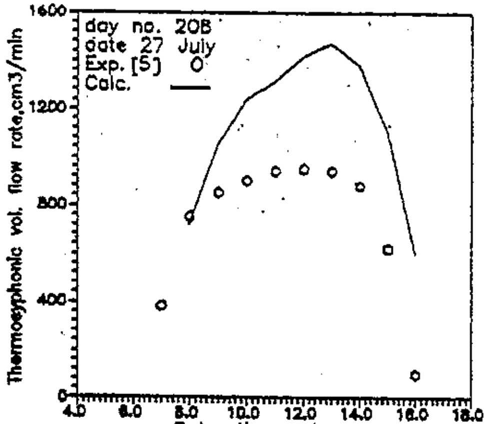


Fig.23 The thermosyphonic volume flow rate according to Zvirin [5] experimental set up data .

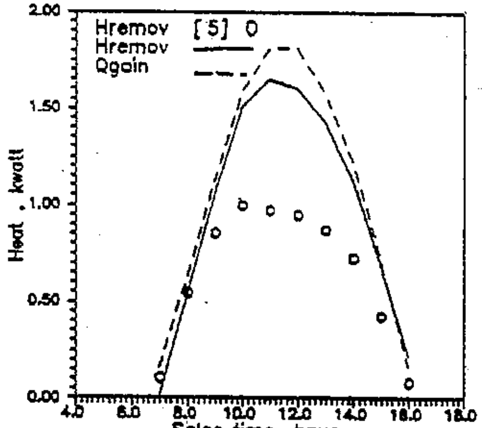


Fig.24 The heat gained (Qgain) and the heat removed (Hremov) from the collector .

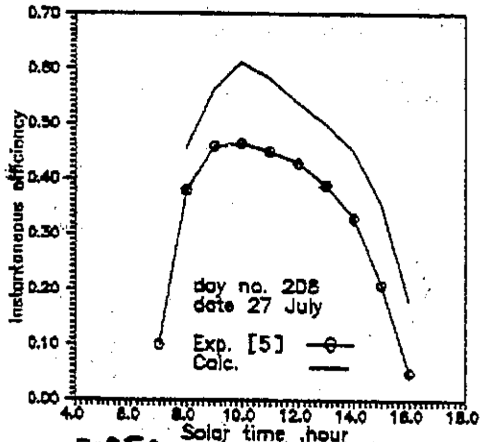


Fig.25 Comparing the calculated instantaneous efficiency and that in Zvirin [5] .

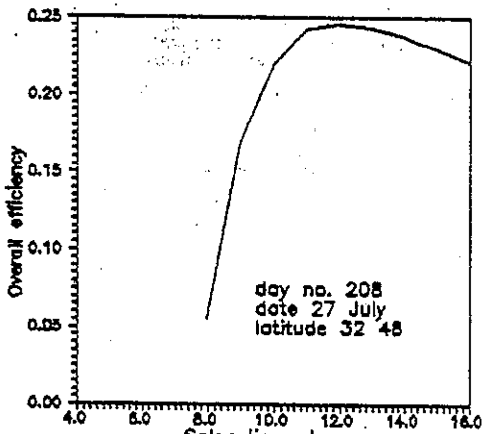


Fig.26 The calculated system overall efficiency according to Zvirin [5] experimental set up data .

than that deduced from the experimental measurements by about 20 % at 10. am solar time. From table 2 this could be referred mainly to underestimation of the value of collector top heat loss factor .

CONCLUSION & RECOMMENDATIONS

1- For the user it is very important to perform an economical study to compare between different offers according to his own operating conditions . Also for the manufacturer , it is very important to eliminate or decrease the risks of guarantee according to the user's operating conditions . The present study gives a useful tool to fulfill these requirements .

2- If the user has to install the collector system in a windy or dusty place , he must take into account the negative effect of these factors in the economical study . The present method gives the user and also the manufacturer a good mean to do that

3- The model results agree very well qualitatively during the main part of the day with the experimental results of ref [5].

4- It is required to design more accurate method to measure the thermosyphonic mass flow rate .

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NOMENCLATURE

Ac = surface area of flat plate collector , m2
B = collector tilt angle with horizontal
Cp = water specific heat capacity , W/kg/C
F_c = flat-plate efficiency factor as defined in [2]and [6] .

F_{ITt} the relative change in a system parameter resultant due to a unit relative change in the solar intensity I_{Tt}

F_T the relative change in a system parameter resultant due to a unit relative change in the transmissivity-absorptivity product

F_{Hc} the relative change in a system parameter resultant due to a unit relative change in the collector heat loss coef.

F_{fric} the relative change in a system parameter resultant due to a unit relative change in the pressure drop through the system .

F_{Nu} the relative change in a system parameter resultant due to a unit relative change in the Nusselt number of the flowing liquid

F_{wind} the relative change in a system parameter resultant due to a unit relative change in the the wind speed based on 1 m/s

F_{foul} the relative change in a system parameter resultant due to a unit relative change in the tube thermal resistance as a result of fouling deposits .

H_c - heat loss coefficient factoras in [12], W/m2/C

I_{Tt} - the total solar intensity calculated as in [11], W/m2

J - section index .

Lc - collector tube length , m

$M^{\bullet}$ - the thermosyphonic mass flow rate ,kg/sec

U_J - overall heat transfer coefficient for section number J , W/m2/C

q_a - total heat absorbed by the blackened plate., $= \overline{\tau.\alpha}\, I_{Tt}$, W/m2

Q_g - heat gained by the collector $= A_c F_c \{q_a - H_c(T_1 - T_a)\}$, W

Q_{go} - Rate of heat gained by the collector at its ref.time, W

Q_r - rate of heat removed from the collector =

$= M^{\bullet} C_p (t_{12} - t_{11})$, W

Q_{ro} - Rate of heat removed from the collector at its ref. time

T_a - the ambient temperature , C.

t_{11}, t_{12} - the inlet and outlet collector temperatures ,C

T^*_m - the initial system temperature.

W_{wJ} -mass of water inside section no.J

W_J -water equivalent of section no.J including the water inside it , the wall metal , and the insulation , W/C

w_i a relative change in the independent parameter i

Z_1 - the vertical distance between the storage tank outlet and the collector inlet , m

Z_2 - the vertical distance between the storage tank inlet and the collector outlet , m

$\overline{\tau\alpha}$ - transmittance -absorptance product as in [10]

$\gamma(t)$- water specific gravity at temperature t

Δ_s the relative change in a system parameter resultant due to relative change in many independent parameters .

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عنوان البحـــــث

(تأثير ظروف التشغيل على أداء نظام التسخين الشمسى ذو السريان الطبيعى)

ملخص البحـــــث:

بالنسبة للمستهلك فإنه من المهم جدا تجنب وتقليل عناصر المخاطرة من استخدام السخان الشمسى عند إجراء دراسات الجدوى الإقتصادية وعند المقارنة بين أنظمة التسخين الأخرى بواسطة ضمانات الأداء المعطاه من الطرف الثانى (إما المنتج نفسه أو الموزع) فى هذا البحث قدمت طريقة بسيطة لحساب التغيرات المتوقعة فى أداء النظام نتيجة لإختلاف ظروف التشغيل الفعلية عند الظروف التى تم التصميم عليها وبالتالى تقليل عنصر المخاطرة سواء بالنسبة للمستهلك أو الطرف الثانى . أيضا فى هذا البحث تم تحسين طريقة الحل الرياضى للنموذج النظرى لتوقع أداء نظام التسخين الشمسى ذو السريان الطبيعى فى ظروف اللاحمل باستخدام معلومات محدودة – وقد أوضحت النتائج أن كفاءة التخزين والكفاءة اللحظية للنظام تعتمد بصورة أساسية على المواصفات التصميمية للنظام بينما يجب الأخذ فى الإعتبار تأثير تغير ظروف التشغيل على مواصفات الآداء الأخرى (مثلا إذا ما أضطر الى وضع الجهاز فى مكان سرعة الهواء فيه عاليه أو فى مكان قريب من محالج القطن أو نسبة التيار فيه عالية) – أوضحت الدراسة أن الطريقة المقترحة يمكن استعمالها أيضا فى تحليل أسباب الإختلاف بين النتائج المعملية والنتائج النظرية وتم توضيح أنه ينبغى تصميم وسيلة أخرى لقياس معدل التدفق الطبيعى خلال النظام أكثر دقـــــــة.