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AN EFFICIENCY OPTIMIZATION ANALYSIS OF FLAT-PLATE SOLAR COLLECTORS

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ABSTRACT

A simulation study is presented for the optimization of the thermal efficiency of the flat-plate solar collector, the relation between the envolved criteria(cover system,collector geometry) was presented , the configuration of cover system with possible different designs using convection suppressing devices(CSD) was considered, the result are presented in a non-dimensional form.

A heat transfer analysis for the collector is conducted and a model is developed, finally the steady state instantaneous thermal efficiency is evaluated for an uniform solar heat influx, the influence of cover system on efficiency is evaluated.

The objective of the present study, mainly concerning the cover system, is to present a model to the study and the prediction of the optimum efficiencies of flat-plate collectors via numerical simulation.

1. INTRODUCTION

The flat-plate solar collector is the most common device used for solar energy moderate temperature applications as residential heating and hot water supply. Recently the optimization of these collectors made it possible to regain a new fields of applications requiring high temperatures as food processing industries.

The common three methods used in intercepting solar energy are, flat-plate collectors, concentrating collectors(traking or stationary), and evacuate tubular collectors, the first represents the simplest and lowest in intial cost, an optomization design of this types can reach a high temperatures as high as 150°C [1],many works has been carried out to predict the collector efficiency, an early study of the factors affecting the efficiency factor of collectors was published by Whllier[2], a review of the performance analysis was presented in reference[3], a heat transfer analysis considering the geometry of the absorber, temperature and heat flow distribution is presented in reference[4],[5], in reference[6]a finite difference simulation method for transit operation is presented.

In fact a quick review of performance analysis of the flat-plate collectors leads to consider three main factors influencing the operation of flat-plate collectors, solar influx, cover system, and the collector heat removal factor, the first is dependant on the geographic location and the climatic conditions which are beyond the scope of this study, for the second and the third optimization is feasible. The cover system optimization study deals with the transmittance -absorbtance of cover, emittance of plate, heat loss by convection between plate and cover and eventualy it's reduction by a convection suppressing devices such as hony combs [7] [1]. The use of hony combs may significantly reduces the transmission of solar radiation but on the other hand makes the absorber surface emittance a most important variable.

The collector efficiency factor is dependant on the cover system, the absorber geometry, the influence of geometry was studied in [2], the thermal boundary layer development was studied in reference [5], which shows that the flow is thermaly established after a distance equal to 2D, also the thermosyphonic effect was considered.

In this study the envolved parameters are reduced to non-dimensional numbers to describe the thermal problem with objective to present a model to the study and the prediction of the optimum efficiency of flat-plate collectors, via numerical simulation.

2. THEORITICAL ANALYSIS

2.1. Cover System

The top loss coefficient was obtained following the procedure shown in Duffie, Beckman [3], the top loss coefficient is the sum of radiative heat loss from plate to cover , cover to ambient plus the convective heat loss from plate to cover, cover to ambient, the top loss coefficient could be written as :

$$U = \left[\frac{1}{H_{p-c} + H_{r\ p-c}} + \frac{1}{H_w + H_{r\ c-s}} \right]^{-1} \qquad (1)$$

In the analysis where convection suppression were considered, the Nusselt number was taken equal to unity, consequently the convection heat transfer coefficient from plate to cover H_{p-c} will simply equal to K/L, this leads to treat the plate and the hony combs as a single unit, the heat transfer will be by pure conduction assuming ideal efficiency of hony combs. In fact the use of convection suppressing devices (CSD), can modify the absorber emittance [7], in this theoritical analysis the absorber emittance is considered the same as without convection suppression

2.2. Heat Transfer Model

The flow is thermaly established after a distance y = 2D, which is small compard to the collector lenth, hence treating the the flow as thermaly established will not lead to a serious error in the heat balance.

From figure (1), a heat balance for the collector tube will give:

$$\dot{m}\, C_p\, (T_f + dT_f) - \dot{m}\, C_p\, T_f - q\, dy = 0 \qquad (2)$$

Which can be reduced to:

$$\dot{m}\, C_p \frac{dT_f}{dy} = q \qquad (3)$$

Where q is the useful gain per unit length of the collector and equal:

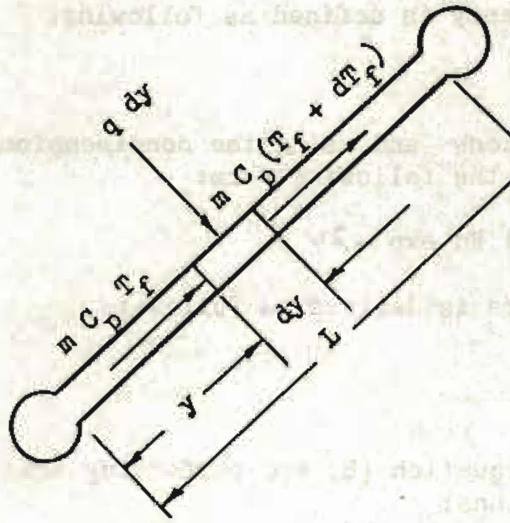

Fig.1. Cross section of absorber plate.

$$q = W\,Fd\,(\,S - U\,(\,T_f - T_c))\qquad(4)$$

With:

$$Fd = \frac{1/U}{W\left[U\left[D + (W - D)\,F\right] + 1/c_b + 1/\pi D\,H_f\right]}\qquad(5)$$

Where Fd is the collector heat removal factor and c_b is the bond conductance. Introducing the following nondimensional numbers:

$\varphi = W/R$ the ratio between the tube spacing and the tube diameter.
$\psi = y/R$ the dimensionless coordinate parallel to the axis of flow.
$P = m\,C_p / U R^2$ which is the ratio between the heat capacity per unit area of the tube and the convective, radiative heat loss through the cover. The ratio of the solar flux reaching the absorber and the heat capacity has the dimension of temperature ($\Delta = Qs\ R / m\,C_p$) and cosequently the non-dimensional temperature equal to T/Δ.

Introducing the non-dimensional notation in eqaution (3) :

$$d\theta_f/d\psi + \frac{Fd\,\varphi}{P}\,(\,\theta_f - \theta_a\,) = 1\qquad(6)$$

Or:

$$d\theta_f/\,d\psi + \lambda\,(\,\theta_f - \theta_a\,) = 1\qquad(7)$$

With $\lambda = Fd\cdot\varphi/P$

Integrate equation (7) with the following boundary conditions:

$y = 0$	$\psi = 0$	$\theta_f = \theta_{fi}$
$y = L$	$\psi = \psi$	$\theta_f = \theta_{fe}$

Equation (7) after integration becomes:

$$\theta_f = \theta_a + 1/\lambda + (\theta_{fi} - \theta_a - 1/\lambda) \exp -\lambda\psi \tag{8}$$

The instanteneous efficiency is defined as following:

$$\eta = \frac{n\ q\ L}{S\ n\ W\ L} \tag{9}$$

After the forgoing equations and using the nondimensional notations the efficiency can be written on the following form:

$$\eta = 1 - (\theta_{fi} - \theta_a) \ Fd \exp -\lambda\psi \tag{10}$$

The fluid mean temperature is defined as follows:

$$\theta_{f,m} = \frac{1}{\psi} \int_{\psi=0}^{\psi=\psi} \theta_{f,\psi} \ d\psi \tag{11}$$

Substituting for θ_f by equation (8) and performing this integration with the following boudary conditions:

$\psi = 0$ at $y=0$
$\psi = \psi$ at $y=L$

The fluid mean temperature will be as follows:

$$\theta_{f,m} = \theta_a + \frac{1}{\lambda} - \frac{1}{\lambda\psi} \ (\theta_{fe} - \theta_{fi}) \tag{12}$$

3. SOLUTION AND RESULTS

The previous equations are general, the solution may be obtained for different cases, since the calculations are of an exploratory nature a wide range of parameters has been adopted. The parameters which has been taken constants are:

- Solar heat influx at 500 W/ sq m.
- Ambient temperature at 20°C.
- Single cover collector.
- Cover glass emittance ϵ=0.86 ,which corresponding to window glass.

Two values for absorber plate emissivity were considered(0.90, 0.12), when a convection suppression is considered the Nusselt number is taken equal to unity, the effect of (CSD) on the plate emittance is neglected, hence four different cases are present:

a) Cover system with convection suppression.
b) Cover system with selective absorber surface (ϵ = 0.12).
c) Cover system with convection suppression and selective absorber surface
d) Cover system with plate emittance ϵ=0.9, without convection suppression

The above listed four cases are considered at two conditions of cover spacing, the first is 30 mm and the second is 60 mm to investigate the cover spacing influence on the collector performance.

The top loss coefficient is evaluated for the above condotions, figures (2),(3), shows the top loss coefficient for different plate mean temperature, for the different cases studied ,figures (4),(5)show the relation between heat capacity to heat loss ratio and the dimensionless plate mean temperature. It should be noted the ratio between heat capacity and heat loss involves collector's geometry since it includes the tube diameter and the mass flow rate,the

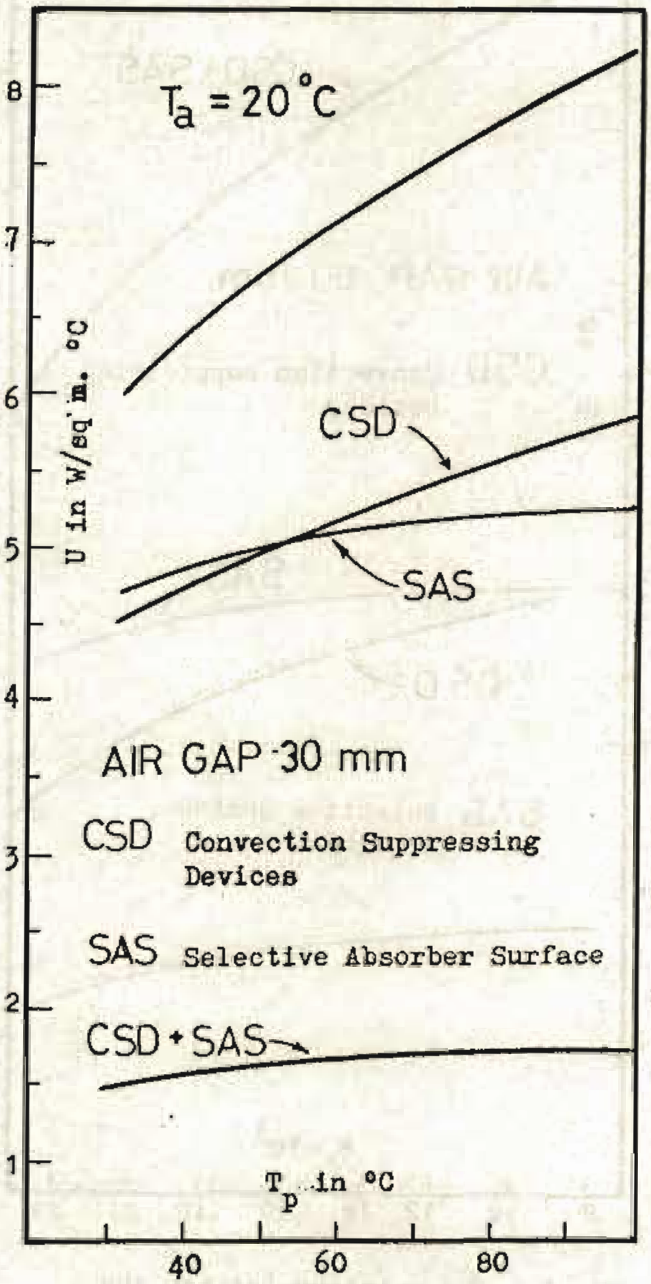

Fig.2. The top loss coefficient function of plate mean temperature, air gap 30 mm.

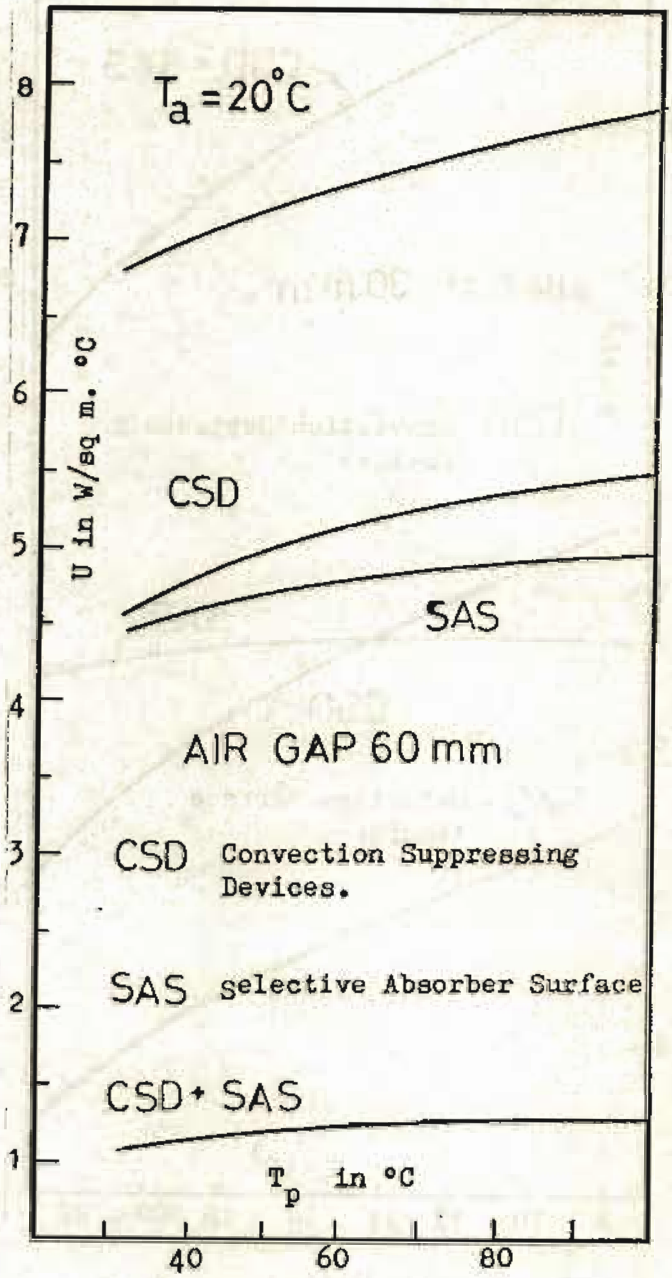

Fig.3. The top loss coefficient function of plate mean temperature, air gap 60 mm.

The collector geometry is also included in the dimensionless temperature, since it includes the heat removal factor, tube space to radius ratio and the solar heat influx.

The thermal efficiency was evaluated for the different cases described, the efficiency is evaluted for different inlet water temperatures, figure (6) shows the collector thermal efficiency function of the nondimension number at different water inlet temperatures. In fact λ groups heat capacity to heat loss ratio, collector heat removal factor and the tube space to radius ratio, this presentation eliminates the plate mean temperature and the different cover system configurations, which are included implicitly in λ.

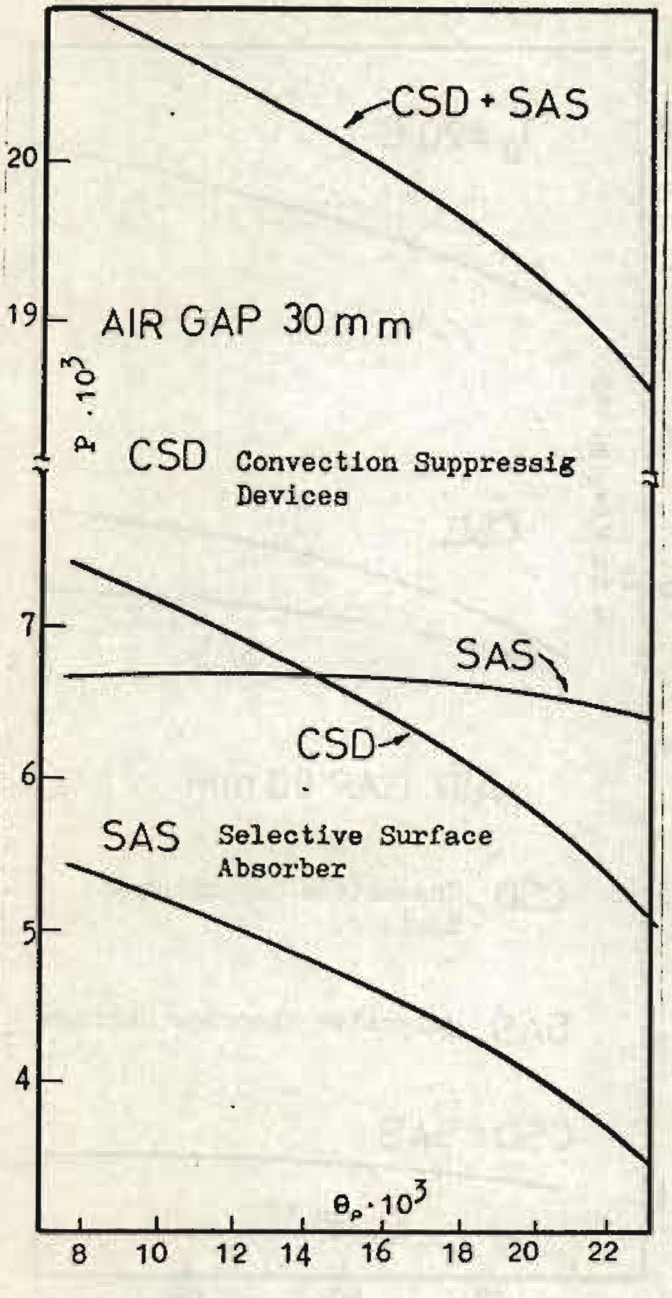

Fig.4. The relation between the heat capacity to heat loss ratio and the dimensionless plate mean temperature.

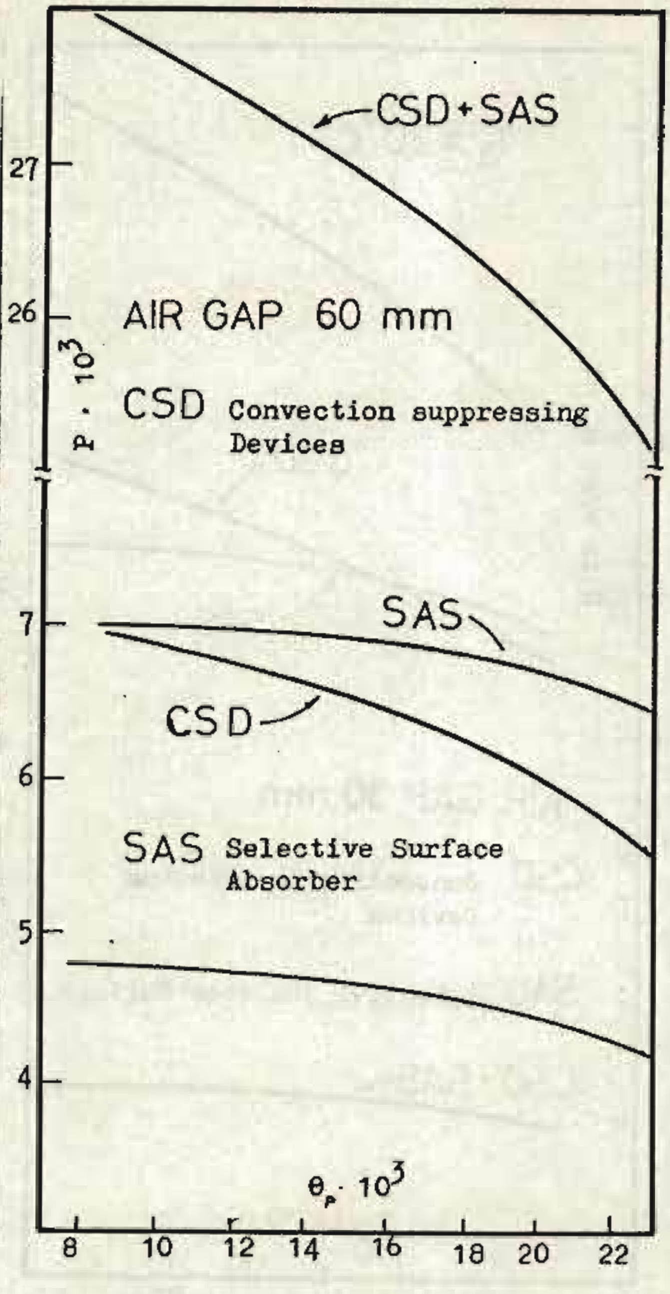

Fig.5. The relation between the heat capacity to heat loss ratio and the dimensionless plate mean temperature.

This nondimensional presentation permits the comparison and the prediction of thermal efficiency of different collectors provided that λ is known.

4. DISCUSSION

In dealing with optimization, there are many criteria to be taken into consideration, certainly maximizing the performance with respect to single creterion may reduce the performance with respect to another, bearing in mind that finally the economical consideration has not to be omitted, a compromise should be reached .

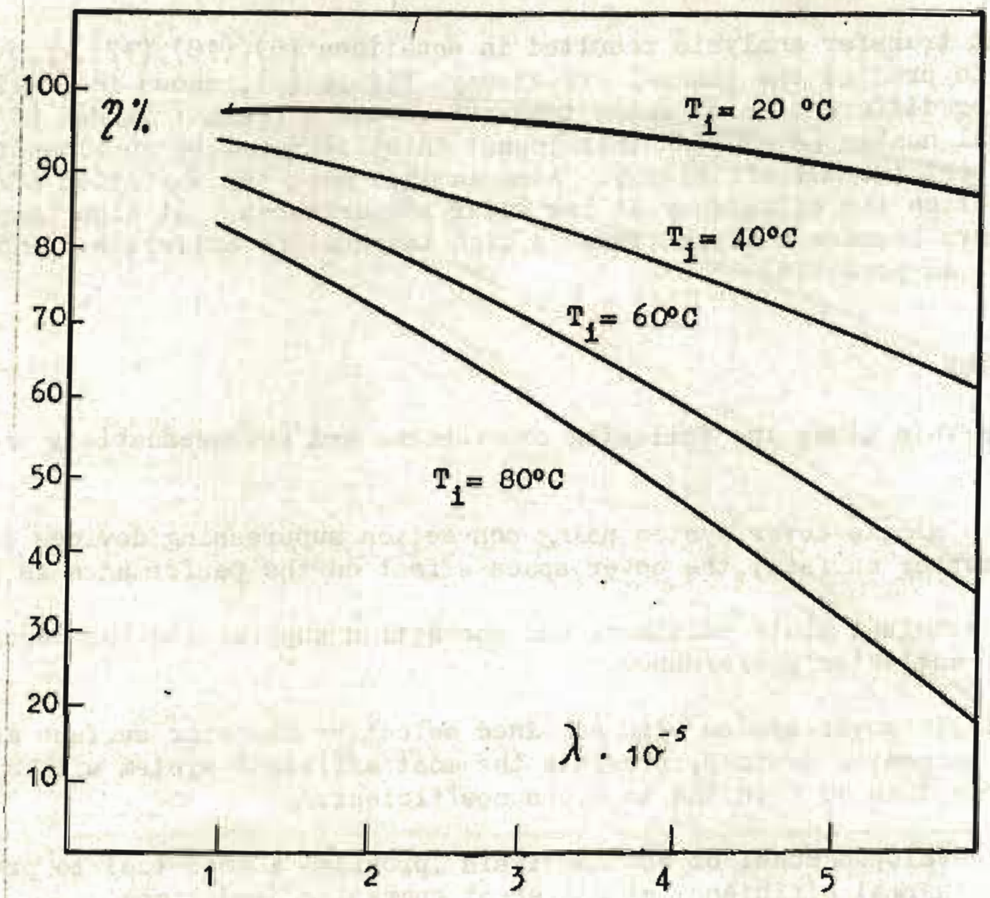

Fig.6. The relation between λ and the thermal efficiency at different inlet temperature.

In this optimization analysis the emphasised criteria are, cover spacing, absorber surface and the reduction of heat loss by covection, as follows:

Cover spacing. As it may be seen from figures (2),(3), that the collector performance is not highly infuenced by the cover spacing, in the particular case of convection suppression or selective surface absorber, the cover spacing has minor effect on the collector performance . This results were confirmed by Gani et al [1], figure (4),(5) also confirm this conclusion. The combination of convection suppression and selective absorber surface is infuenced by the cover spacing , but it could be neglected since the the top loss coefficient is very small in this case.

Selective surface absorber. The reduction of radiation losses has a great influence on the collector performance as it could be seen from figures (2),(3) the reduction of emissivity can lead to a reduction of about 25% in the top loss coefficient in the case of low emittance factor (ϵ =0.12).

Convection suppression. The influence of convection suppression on the collector performance is approximately equivelant to the influence of the emissivity of the absorber plate. However in general the combination of convection suppression and low emissivity of absorber surface yield the lowest loss coefficient and the highest heat capacity to heat loss ratio. Also it may be noticed from figures (2),(3), that using these combination flatten the curve and hence becomes slightly influenced by the plate mean temperature.

The heat transfer analysis resulted in equations (8),(10),(12), provides a good tool to predict the thermal efficiency. Figure ( ), shows the collector efficiency for different inlet water temperature and different values of the nondimensional number ($\lambda = \varphi Fd/P$) this lowest inlet temperature and lowest λ provide the best thermal efficiency. Also in this case the variation of λ has a small eefect on the efficiency at low water temperatures, at high temperatures the curve becomes steeper ,thus in high temperature collector λ should be kept small as possible.

5. CONCLUSION

Based on this study the following conclusions and recommendations are offered:

1. For a single cover system using convection suppressing devices or with selective absorber surface, the cover space effect on the performance is minor.

2. The absorber plate emittance and convection suppression has major effect on the collector performance.

3. A single cover system with combined selective absorber surface and convection suppressing devices, presents the most efficient system with a reduction of more than 70 % in the top loss coefficient.

4. The developed model of heat analysis ,provides a good tool to predict the collector thermal efficiency at different operating conditions.

ACKNOWLEDGEMENTS

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NOMENCLATURE

C_p Water specific heat in Kcal/Kg. °C.
D Tube diameter in meter.
Fd Collector heat removal factor.
F Standard fin efficiency.
H Heat transfer coefficient W/sq m °C.
K Thermal conductivity W/m°C.
L Collector length in meter.
m Water mass flow rate in Kg/Hr.
P Dimensuoless number presenting the ratio between the heat capacity per unit area of the tube and the heat loss through the cover.
Qs Solar radiation reaching the plate W/ sq m.
R Tube radius in meter.
S Solar influx W/sq m.
T Temperature in °C.
U top loss coefficient in W/sq m °C.
y Coordinate parallel to the flow in meter.
W Space between tubes in meter.

Greek Symbols

Φ Ratio between tube spacing and radius.
ψ Dimensionless coordinate parallel to the flow.
η Collector instataneous thermal efficiency
λ Nondimension number
θ Dimensioless temperature .
ϵ Emittance.

Subscripts

a Ambient.
c Cover.
e Exit.
f fluid.
i Inlet.
p Plate.
r Radiation.
s Sky.