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THERMAL AND EXERGETIC ANALYSISOF SOLAR WATER HEATER

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ABSTRACT

Solar water heater is an important device utilizing solar energy for heating of water. Solar water heaters are conveniently designed on the basis of thermal energy considerations only .It has now been realized that special attention is required to the design to take into account the losses as well as the quality of input and output energy. Exergy based optimization of such systems ensures that the design yields an energy output of highest possible quality with minimum losses.

In this work, detailed second law analysis of the solar water heater has been carried out. A procedure has been laid out for the determination of exergetic efficiency as function of various influencing system and operating parameters. The effect of major system parameters namely number of covers, emmisivity of plate ,bond conductance and spacig between water carrying tubes on the exergy input and output has been determined corresponding to the range of operating parameters, namely, temperature rise parameter of the collector and the insolation .The optimum values of the system parameters have been determined that result in maximum value of exergetic efficiency for given values of operating parameters.

***Keywords*: Emmisivity, Bond Conductance, Covers, Exergy, Water Heater**

I. INTRODUCTION

A solar water heater represents the most important and most wide spread application of solar energy. Extensive research effort has gone into the various design and performance aspect of solar water heating systems. Most important components on which the investigations have been carried out with the objective of arriving at the optimum value of the sizes/ number/ characteristics of these components are Absorber plate: (Material and absorber coating),Transparent covers and Fluid Flow Network which includes Diameter of fluid tubes, their spacing and bonding of the tubes with the absorber plate.

Proper design of solar water heating system is important to assure maximum benefit to the user, especially for a large system. Designing a solar hot water system involve appropriate sizing of different components based on predicted solar insolation and hot water demand. Most of the studies have attempted to determine the optimal component sizes so that the water heating results in maximum thermal efficiency under the given operating conditions of the system. These operating conditions were normally the temperature rise of the fluid through the collector,

$\Delta T = T_o - T_i$. (1)

Here T_o is the fluid outlet temperature whereas T_i is inlet temperature which in most cases is very close or equal to the ambient temperature of the location and Insolation of the location where the collector will be located.

Some of the researchers have pointed out that in order to make the system perform in such a way as to result into minimum entropy generation is the better way of designing any heat exchange equipment.

A number of design methods for solar water heating systems have been proposed in the literature. These methods can broadly be classified into two categories, namely, correlation based methods and simulation based methods. Methods based on utilizability [10], F chart [18], /–F chart [16] etc. are prominent examples of correlation based methods. Different simulation programs such as TRNSYS [17]SOLCHIPS ([23], Lund and Peltola, 1992), etc. have been used to design solar hot water systems through detailed simulation approach. Application of utilizability method for solar hot water systems ([12]; [31]) depends on determination of constant critical radiation intensity [10]. Application of F charts ([2]; [6]; [34]) assumes a fixed collector loss and an average daily solar irradiation. Applicability of /–F chart method ([5]; [9]) is restricted due to the complexity involved in the calculation of utilizability. Detailed simulation models have been applied for design and optimization of solar hot water systems ([4]; [7] ; [11]; [32]; [27]; [1]; [20]; [3]; [29]; [15]; [8]). Different linear/nonlinear optimization techniques (Michelson, 1982; [30]) and evolutionary search algorithms ([30]; [22]; [15]) have also been applied. However, development, simulation, validation, and optimization of detailed mathematical model require significant time and effort. For a given type of solar collector-storage system, parameters such as total collector area, storage volume and solar fraction are important from the performance and optimization point of view. Existing methods identify a single design through optimizing an objective function, such as total annual cost, annualized life cycle cost ([13]), life cycle savings ([12]), payback period [28], internal rate of return ([12]), etc.

The exergy of a system is the maximum useful work possible during a process that brings the system into equilibrium with a heat reservoir. Exergy can be destroyed by irreversibility of a process. An exergy analysis (2^{nd} law analysis) is a very powerful way of optimizing complex thermodynamic systems. The term exergy was proposed by Rant in 1956, but the concept was developed by Gibbs in 1873. Now days, details of this concept can be found in thermodynamics text, several researchers have used this powerful method to optimize different thermodynamic parameters of power plant components. As a recent application of second law analysis, Saravanan*et. al.* used energy and exergy analysis to study the performance of a cooling tower.

The governing equations of exergy balance as applied to solar collectors has been published by Bejan*et al.*. Recently Londono-Hurtado developed a model to study the behavior of volumetric absorption solar collectors (VASC) and the influence of the design parameters on the performance of the collectors. Their approach is based on the use of several dimensionless numbers, each of them having a clear physical significance, which play a key role in the analysis of the collector. The model is then used to conduct a thermodynamic optimization of VASC, which gives the optimal design parameters that maximize the exergy output of the heat extracted from the collector. Another notable study is the work of Luminosu*etal*. where they experimentally studied an air solar installation. They processed their experimental results through thermodynamic analysis, using energy and exergy methods to find the best flow rate of passing air.

II. METHODOLOGY

Thermal and exergetic performance of the solar water heater depends upon a number of system and operating parameters. System parameters include the collector size (Length & width), plate thickness, thermal conductivity of plate, number of covers, bond conductance, tube diameter, emissivity and transmittance of covers, bond conductance, tube diameter, tube spacing, insulation thickness, its thermal conductivity etc. Whereas the operating parameters include insolation, temperature rise, ambient temperature, winds velocity etc. These parameters can be categorized into:.

Fixed parameters: Those parameter which are not the major parameter that do not substantially influence thermal and exergetic performance.

Variable parameters:Those parameters that are proposed to be investigated and form a set of major influencing parameters that affect the performance.

Tables 1 & 2 respectively list the fixed and variable parameters. The range of variable parameters has been selected on the basis literature related to the design of solar water heater.

Table 1 List of Fixed Parameters

Area of collector (A)	1 m²
Insulation thickness (δ)	.05 m
Plate thickness (mp)	1.3 mm
Length of collector (L)	1m
Thermal conductivity of insulation (K)	.04 W/m K
Gap between absorber plate and glass cover (Lgp)	.025 m.
Wind velocity	2 m/sec.
Inlet temperature of water (Ti)	286 K
Ambient temperature (Ta)	284 K
Emmisivity of glass (ϵg)	0.72-0.88
Inside diameter of tubes (d)	0.0127 m
Bond Conductance	30 W/m K
Transmitanceabsorptance product (τα)	0.74-0.82
Thermal conductivity of plate (kt)	285 W/mK

Table 2 List of Variable Parameters

Emissivity of plate (ϵp)	0.75-0.95
Number of covers (N)	1-3
Spacing between the tubes (w)	0.025-0.2 (m)
Outlet temperature of water (To)	303-333 (K)
Solar radiation intensity (Itt)	500 & 1000(W/m²)

The system and operating parameters listed above affect the performance of the solar water heater, some to greater and others to smaller extent. Those parameters that have substantially affect to the greater extent need to be selected with greater care. The extent to which they affect can only be known on the basis of detailed study of the variation of these systems operating parameters on thermal & exegetic performance of the system. Following are the major performance parameters whose variations have been studied as the system and operating parameters are varied.

(i) Useful heat gain Qu

(ii) Over all loss coefficient Ul

(iii) Collector efficiency factor F'

(iv) Heat removal factor ,FR

(v) Thermal Efficiency ηth.

(vi) Exergetic efficiency η_{ex}

The overall heat loss coefficient (U_l) is determined from $U_l = U_t + U_b$[10]

$$U_t = \left[\frac{N}{\left(\frac{C}{T_{pm}}\right)\left(\frac{T_m - T_a}{N+f}\right)^{0.33}} + \frac{1}{h_w} \right]^{-1} + \left[\frac{\sigma\left(T_m{}^2 + T_n{}^2\right)\left(T_m + T_n\right)}{\frac{1}{\epsilon p + 0.0425N(1-\epsilon p)} + \frac{2N+f-1}{\epsilon_g} N} \right] \quad (2)$$

N=number of glass covers [10]

$$f = \left(\frac{9}{h_w} - \frac{30}{h_w^2}\right)\left(\frac{T_a}{316.9}\right)(1 + 0.091N)$$

$$C = \frac{204.429(\cos\beta)^{0.252}}{L^{0.24}}$$

The useful energy heat gain rate (Q_{u1}) is calculated from known values of T_a, I, U_l and assumed value of T_{pm} using :

$$Q_{u1} = \left[I(\tau\alpha) - U_l\left(T_{pm} - T_a\right)\right]A_P \quad (3)$$

The mass flow rate (*m*) is calculated from:[10]

$$m = Q_{U1}/c_p * (T_O - T_i) \quad (4)$$

Flow velocity is determined from:

$$V = \frac{m}{\frac{\pi}{4} * d^2} \quad (5)$$

Where flow reynold number is determined from mass flow rate in the individual tubes.

$$Re = \frac{\rho * V * d}{\mu} \quad (6)$$

The plate efficiency factor F' is calculated using:[36]

$$\mathbf{F'} = \frac{1/U_L}{W\left[\frac{1}{\pi D h_{f1}} + \frac{m_1}{\pi D K_f} + \frac{1}{C_b} + \frac{1}{U_l\,[D + (W-D)F]}\right]} \quad (7)$$

f = fin efficiency factor = $tan\ h\ (a(w-d)/2)/a(w-d/2))$

$$\mathbf{a}^2 = \frac{U_l}{K_p m_p} \quad (8)$$

The collector heat removal factor, $F_R$is calculated from [10]

$$F_R = \frac{mC_p}{A_p U_i}\left[1 - \exp\left\{-\frac{F' U_t A_p}{mC_p}\right\}\right] \quad (9)$$

The useful heat gain is calculated from the inlet air temperature and heatremoval factor (F_R) as:[10]

$$\mathbf{Q_{u2} = A_p F_R [I(\tau\alpha) - U_l(T_i - T_a)]} \quad (10)$$

The values of useful heat gain Q_{u1} *(*calculated in step 4) and Q_{u2} are compared. The thermal efficiency is calculated from useful heat gain Q_u, which isaverage of Q_{u1} and Q_{u2}, as:

$$\mathbf{\eta th = Q_u / (I_{tt} * A)} \quad (11)$$

The exergy inflow associated with the solar irradiation on the solar collector surface is calculated:[10]

$$\mathbf{E_s = I_{tt} * A\left(1 - \frac{T_a}{T_{sun}}\right)} \quad (12)$$

Where T_{sun} = 5777 K.

The net exergy flow of water can be calculated as:[36]

$$\mathbf{E_n = I_{tt} * A * \eta th * \eta c - P_m(1 - \eta c)} \quad (13)$$

Exergetic efficiency is calculated as: [10]

$$\eta_{II} = \frac{E_n}{E_s} \quad (14)$$

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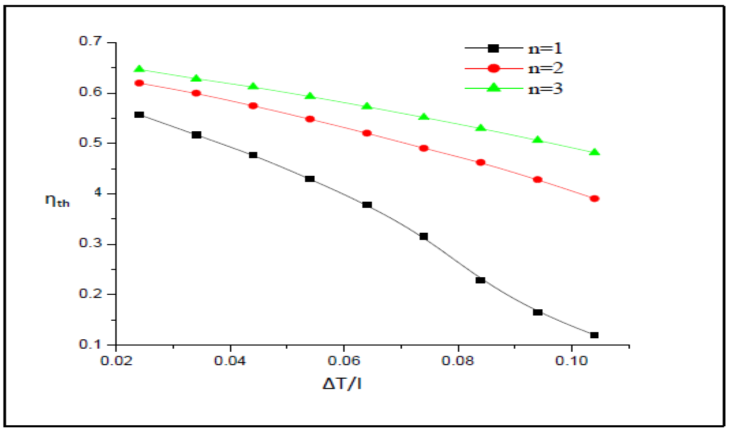
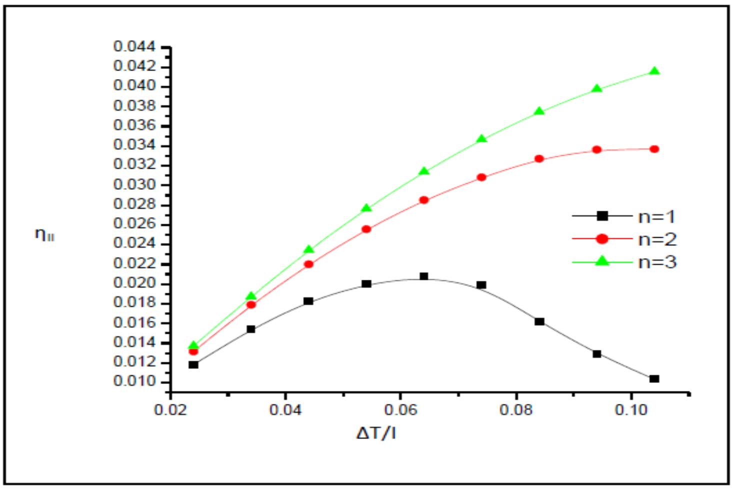
III. RESULTS AND DISCUSSION

3.1 Exergetic v/s Thermal efficiency



W = .025
εp = 0.85
εg = 0.80
Cb = 30
τα = 0.78
kt = 285
d = .0127

Fig.3.1.1 Thermal Efficiency at I = 500 W/m²



W = .025
εp = 0.85
εg = 0.80
Cb = 30
τα = 0.78
kt = 285
d = .0127

Fig.3.1.2 Exergetic Efficiency at I = 500 W/m²

Fig3.1.1 and 3.1.2 explains the variation of thermal and exergetic efficiency as function of temperature rise parameter keeping all other parameters fixed. These figures indicate the comparative difference between two most important performance parameter as a function of design parameter. As a result ,two distinct points emerged from the observations.

As from the figures ,dependence of thermal andexergetic efficiency parameters on temperature rise parameters appears to be diametricallyopposite to each other. The thermal efficiency is maximum corresponding to a very low value of temperature rise parameter (or in other words when the temperature of output fluid is lowest) whereas the exergetic efficiency has a lowest value corresponding to this condition. It is well known that when output temperature is very low, the absorber plate temperature is lowest leading to lowest amount of heat losses to the environment , thus converting maximum amount energy absorbed by the absorber plate into useful heat gain and resulting in highest thermal efficiency . However, this energy is at the lowest value of Carnotefficiency and hence very small amount of work output or exergy.

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Besides, when temperature rise is minimum, the flow rate of fluid has to be very large to collect even a small amount of energy (useful thermal gain). This will result in very high friction losses because friction losses have a cubic power relationship with fluid velocity .This combination of low exergy gain from thermal energy output and high amount of friction losses results in very low (or even negative, sometimes) value of exergetic efficiency when temperature rise parameter is very low.

This leads to the conclusion that “under low temperature rise conditions, the collector collects maximum possible amount of verylow gradeenergy which from the stand point of second law is worst condition.

As the value of temperature rise parameter is increased thermal efficiency comes down whereas the exergetic efficiency increases. As can be seen from the discussion given above that although thermal energy gain begins to decrease, the quality of energy begin to improve because of an increase in Carnot efficiency. This benefit is derived from the condition that the absorber plate absorbs energy at higher temperature resulting in lower amount of energy generation and thus results in better second law performance of the system.

3.2 Effect of Spacing Between the Tubes

Spacing between tubes carrying heat transfer fluid (water) indicates how effective the contact between the tubes and the absorber plate is; a higher spacing indicates relatively lesser contact area between the tube plate for a fixed value of tube diameter. Thus a higher spacing should represent a proper performance as can be seen from Fig 3.2.1 where exergetic efficiency respectively has been plotted as function of operating parameters to reveal the effect of inter-tube spacing. Decreasing the spacing from value of 0.2 m to 0.025 m is seen to cause an increase of exergetic efficiency from .027 to 0.034. Fig 3.2.2 have been prepared to show the effect of change of spacing for an insolation value of 1000 W/m^2 where a corresponding change of spacing is seen to change the value of exergetic efficiency from .042 to 0.047.From these plots it can be noted that for betterexergetic performance, the lowest value of spacing is best in the entire range of temperature rise parameter.

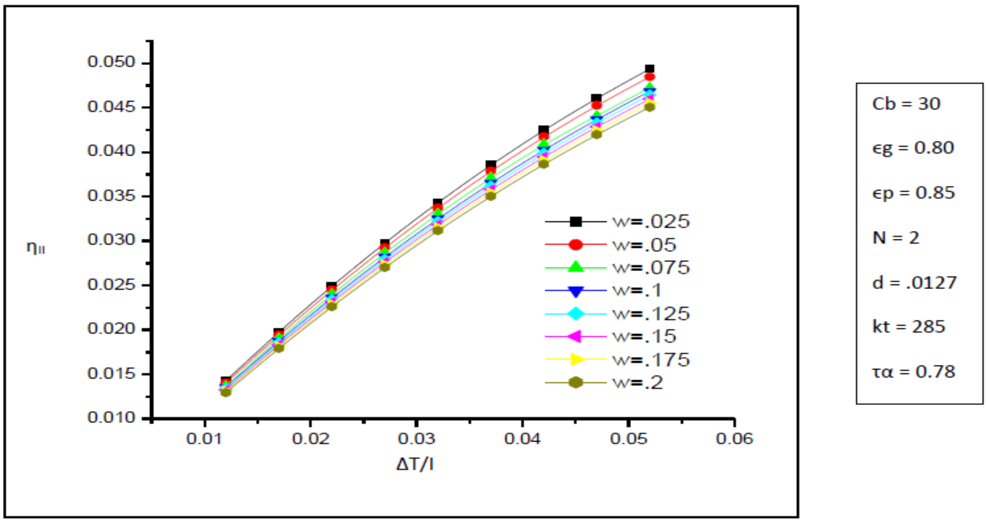

Fig 3.2.1.Effect of Spacing Between Tubes on Exergetic Efficiency at I = 500 W/m²

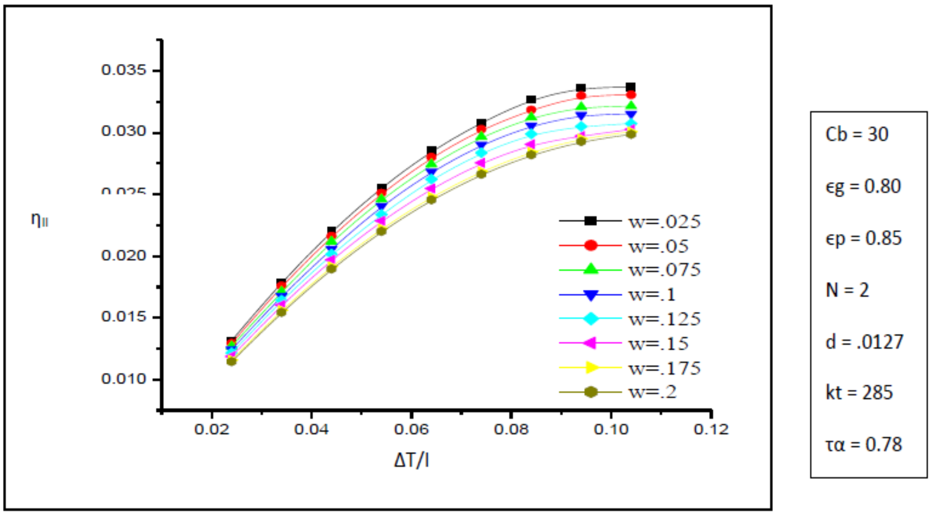

Fig.3.2.2 Effect of Spacing Between Tubes on Exergetic Efficiency at I = 1000 W/m²

3.3 Effect of Emissivity of Absorber Plate

Figs 3.3.1& 3.3.2 show the effect of emissivity of absorber plate on the exergetic efficiency. The values of exergetic efficiency have been plotted as function of temperature rise parameter andplate emissivity corresponding to two selected values ofinsolation namely 500 W/m² and 1000 W/m². It is seen from Figs 3.3.1 and 3.3.2 that the quantum of changes is not seen to be substantially influenced by the values of insolation. These effects are seen to be negligible in the case of exergetic efficiency.

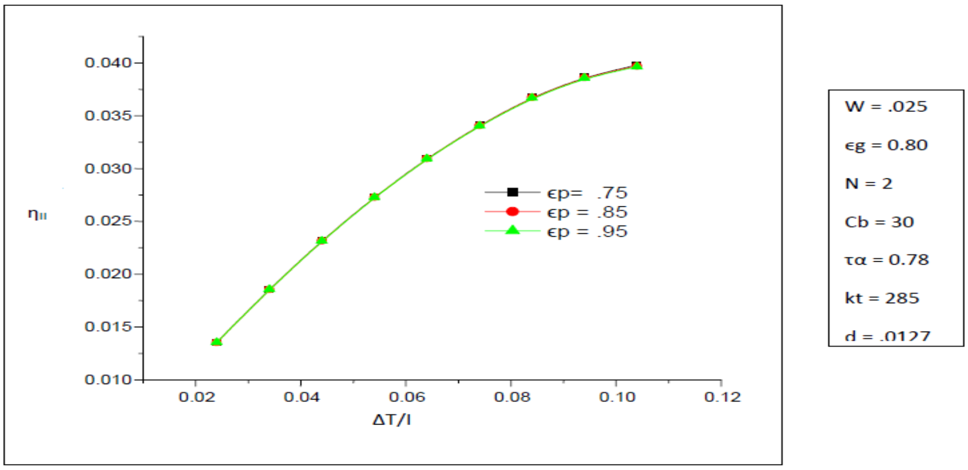

Fig.3.3.1. Effect of Emissivity of Plate on Exergetic Efficiency at I = 500 W/m²

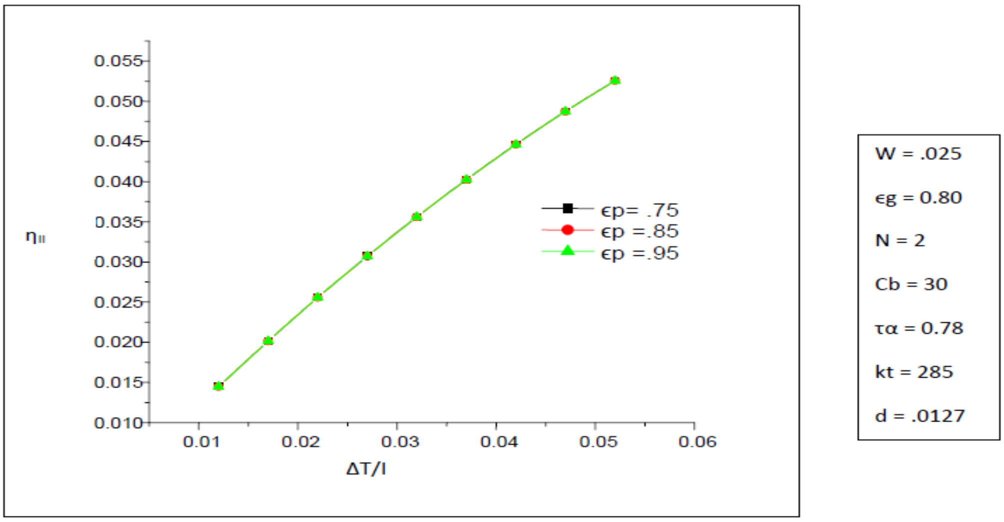

Fig.3.3.2 Effect of Emissivity of Plate on Exergetic Efficiency at I = 1000 W/m²

IV. CONCLUSION

Solar water heater performance has been analyzed using simulation technique to investigate the effect of major system design parameters, namely, absorber plate characteristics, number of covers and their quality, characteristics related to carrier fluid flow system in the thermal and exergetic performance of the solar collector.

The effect of variation of values on major design parameters on the exergetic efficiency has been investigated. Major conclusion of the investigations is given below:-

1) It is seen that although in general performance of the system improves with increasing number of covers but the value of temperature rise parameter must be taken into account while deciding the number of covers. If the system is to be operated for lower value of temperature rise parameters, any of the values can be accepted but for a higher value of temperature rise parameter, a higher value of number of covers is essential from exergetic consideration.
2) Exergetic efficiency is seen to improve with decreasing values of the inter-tube spacing; effect appear to be small for lower values of temperature rise parameter , but it can be recommended that lowest possible value should be used for design and increases with increase in temperature rise parameter.
3) Emissivity of the absorber plate does not appear to influence the performance to a large extent.
4) There is not any significant effect of bond conductance on exergetic efficiency. Hence this parameter is not of much conceren while exergetic performance is required to be improved.

V. NOMENCLATURE

Nu =Nusselt number (hd/k)

Ra= Rayleigh number, Gr.Pr

Gr=Grashof number

Pr= Prandtl number

h =convective heat transfer coefficient, −2 −1 Wm K

w_h= Wind loss coefficient, −2 −1 Wm K

N-=number of glass covers

L =spacing between the absorber plate and glass cover

T=Temperature, °C

T_{pm}=Absorber Plate mean Temperature, °C

T∞ =Ambient Temperature, °C

ΔT- Temperature difference between enclosed surfaces

U_l = Overall Loss Coefficient, W/m² K^-1

U_t= Overall Top Loss Coefficient, W/m² K^-1

η_{II} = Second law efficiency

VI. GREEK SYMBOLS

ε = Emissivity

σ =Stefan–Boltzman constant, −2 −4 Wm K

β' =Film Temperature, C 0

μ =Viscosity of air

υ =kinematic viscosity of air

α =Thermal diffusivity of air

δ_p= plate thickness [m]

β= Collector tilt angle

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