EVALUATION OF HEAT LOSS COEFFICIENTS IN SOLAR FLAT PLATE COLLECTORS

Y. Raja Sekhar[1], K. V. Sharma[1] and M. Basaveswara Rao[2]
[1]Centre for Energy Studies, J. N. T. U. H. College of Engineering, Kukatpally, Hyderabad, India
[2]Department of Mechanical Engineering, S. V. I. T, Secunderabad, India
E-mail: yrssarma@gmail.com

ABSTRACT

Flat Plate Collector (FPC) is widely used for domestic hot-water, space heating/drying and for applications requiring fluid temperature less than 100°C. The absorber plate of the FPC transfers solar energy to liquid flowing in the tubes. The flow can takes place due to thermosyphon effect or by forced convection. However, certain energy absorbed by the plate is lost to atmosphere due to higher temperature of the plate. The collector efficiency is dependent on the temperature of the plate which in turn is dependent on the nature of flow of fluid inside the tube, solar insolation, ambient temperature, top loss coefficient, the emissivity of the plate and glass cover, slope, etc. The objective of the present work is to evaluate top loss coefficient considering these aspects of the flat plate collector both theoretically and experimentally. A test setup is fabricated and experiments conducted to study these aspects under laboratory conditions.

Keywords: flat plate collector, solar water heating, thermosyphon system, absorber plate emissivity, loss coefficient, efficiency.

INTRODUCTION

The performance of the solar thermal flat plate collector depends on the amount of solar insolation absorbed by the plate. The emissivity of the selective coated plate is usually around 0.1 and that of glass cover lies between 0.85 - 0.88. The major heat loss in the collector is from the top through the glass cover compared to bottom and side losses. The top loss coefficient from the collector is evaluated by considering both convection and radiation from the absorber plate to ambient. The thermal resistance circuit of the flat plate collector for single glass cover is shown in Figure-1.

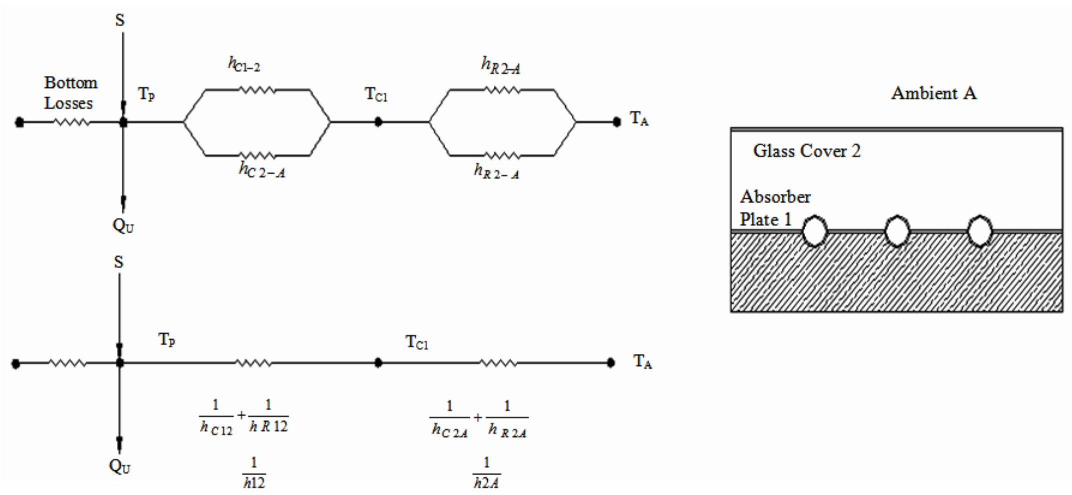

Figure-1. Thermal network diagram for single glass cover flat plate collector.

The overall top heat loss coefficient is a function of various parameters which include the temperature of the absorber plate, glass cover and ambient, emissivity of absorber and glass cover, spacing between the absorber and glass cover L, tilt angle of the collector β, number of glass covers, etc given by

$$U_t = f\left(\varepsilon_p, \varepsilon_g, T_p, T_a, T_g, N, L, \ldots\right)$$

$$U_t = \left[\frac{N}{\left(\frac{C}{T_{pm}}\right)\left(\frac{T_{pm}-T_\infty}{M+f}\right)^{0.33}} + \frac{1}{h_W}\right]^{-1} + \left[\frac{\sigma\left(T_{pm}^2+T_\infty^2\right)\left(T_{pm}+T_\infty\right)}{\frac{1}{\varepsilon_p+0.05M\left(1-\varepsilon_p\right)}+\frac{\left(2N+f-1\right)}{\varepsilon_g}N}\right]$$

Where $f = \left(1.0-0.04h_W+0.0005h_W^2\right)\left(1.0+0.091M\right)$ and $C = 365.9\left(1.0-0.00883\beta+0.0001298\beta^2\right)$

Tabor [1] has modified the equation of Hottel and Woertz [2], for the estimation of energy transferred to the glass cover. Klein [3, 4] used the modified equation of Tabor [1] for estimating the loss coefficients. Pillai and Agarwal [10] discussed the influence of various factors on the efficiency of solar collectors and concluded that at low solar insolation in the range of 200-600 W/m^2 double glazed collectors are superior to single glazed. FPC with selective absorber coatings having two glass covers is stated to give optimal performance. Garg and Rani [7] calculated the overall heat loss coefficient and the collector efficiency under different conditions such as the absence of cover, with single and double glazing under different ambient conditions, tilt angles, wind speeds, emissivity of both glass cover and absorber plate. The objective of the present work was to evaluate theoretically and compare with the experimentally obtained the heat loss coefficient from flat plate collector. The effect of other significant parameters was evaluated by conducting the experiments on the experimental setup.

EXPERIMENTAL SETUP

The schematic of the experimental setup is shown in Figure-2 for the estimation of top loss coefficient. To investigate the heat loss coefficient of a closed loop thermosyphon system, indoor tests were performed under different operating conditions under natural and forced circulation. The top loss coefficients obtained under laboratory conditions can be extended to estimate under actual operating conditions. The usual nine-fin tube array in a conventional FPC is replaced with a prototype assembly of one-third size of actual FPC i.e. 300 x 600 mm^2 area having 3 fins of 600 mm length with inlet and outlet headers. Glass wool insulation is provided at the bottom of the absorber tubes. The fin tube assembly consists of a copper tubes of 12 mm dia of thickness 18 SWG forming part of the conventional FPC fin tube array.

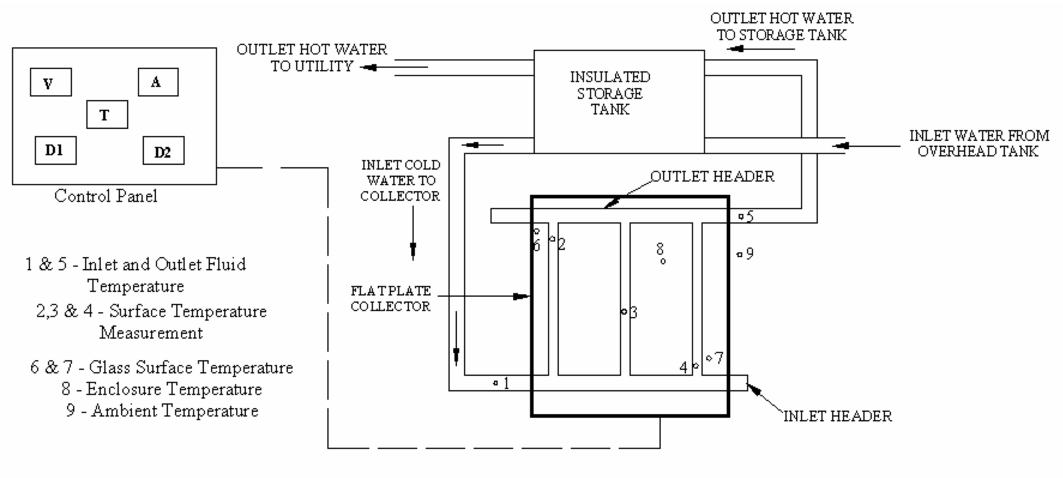

Figure-2. Experimental setup.

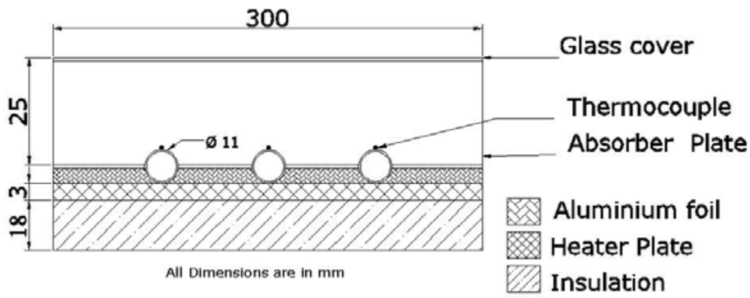

Figure-3. Cross sectional view of the experiment setup.

The copper fin measuring 11.5cm width 24 SWG thick with selective coating applied. The effective area of the fin-tube assembly is approximately 0.24m^2. To measure the temperature of water inside the tube for applied heat flux, three thermocouples to read the temperature variations are located diagonally on the absorber surface. The junctions of the thermocouples are soldered to prevent corrosion during operation and affect the output values. Holes are drilled at suitable locations in the headers to measure the inlet and outlet temperature of water. Two thermocouples are placed on the glass surface with the help of epoxy binder exactly parallel to the inlet and outlet header. One thermocouple is placed in the spacing between absorber surface and glass cover. All the thermocouples used in the present setup are chosen such that they measure temperature upto 120^{0}C. To simulate solar insolation falling on the collector, flat strip heating is located below the collector plate assembly. The heater plate being flat and the fin tubes being curved, the gap of 5 to 6 mm is filled with 14-micron aluminum foil strip is packed to ensure uniform transfer of heat. The strip heater is connected to a dimmerstat provided in the control panel. The complete assembly of the prototype FPC is placed on an adjustable stand of 1.2m x 0.5m x1.5m such that the inclination of the assembly can be varied. A specially fabricated double walled cylindrical water tank of 50 liter capacity made of mild steel and 18 SWG thick is installed over a stand for circulation of water to the FPC. The tank is stuffed with 2-inch thick rockwool insulation material between the walls. The water supply to the assembly is made with the help of inlet and outlet copper header tubes of 1-inch dia provided at both the ends. These headers are connected to the water tank with the aid of 1 inch PVC flexible pipe. To study the effect of heat flux on the mass flow rate and heat transfer rate to fluid the heat flux is varied in the range of 200 to 1000W/m^2 with the dimmerstat located in the control panel. In addition, the control panel consists of a Voltmeter, Ammeter, power on/off switch, Digital temperature indicator, 12 channel female pin to connect the thermocouple. The different thermocouples fixed on the fin tube assembly are connected to the indicator panel suitably by means of a male connector. The temperatures from different thermocouples are noted manually at specified time intervals until steady state values are obtained. As the mass flow rate of water is very low, the flow rate is measured manually by collecting water with the aid of a measuring jar.

ANALYSIS AND CALCULATION OF LOSS COEFFICIENTS

Assuming one-dimensional heat flow considering thermal capacity and temperature drop across the glass cover, iterative procedure is performed to estimate the top loss coefficient with the range of variables as shown in Table 1.

Under steady state conditions, the heat loss from the absorber plate to the glass cover is given by

$$Q_l = \left[U_t A_p \Delta T\right]$$

U_t, is the heat loss coefficient due to both convection and radiation from the absorber plate and glass cover as shown in Fig. 1 of the thermal resistance network diagram.

Therefore, $U_t = \dfrac{1}{h_{12}} + \dfrac{1}{h_{2A}}$

The convective heat transfer coefficient between the absorber plate and glass cover is calculated from the Nusselt number obtained using equation given by Hollands *e al.*, [2].

$$Nu = 1 + 1.44\left[1 - \frac{1708}{Cos\beta.Ra}\right]\left[1 - \frac{Sin(1.8\beta)^{1.6}.1708}{Cos\beta.Ra}\right] + \left[\left(\frac{Cos\beta.Ra}{5830}\right)^{1/3} - 1\right]$$

Where $Ra = \dfrac{g\beta(\Delta T)d^3}{\upsilon^2} \times \Pr$

From the Nusselt number the overall top loss coefficient U_t is calculated at different tilt angles β and Ra.

The radiative heat transfer coefficient varies with the wind loss coefficient and the temperature difference between the glass cover and the ambient.

Table-1. Range of variables.

Variables	Range
Ambient temperature, T_∞	273–318 K
Air gap spacing, L	8–90 mm
Absorber plate temperature, T_{pm}	323–383 K
Absorber plate emittance, ε_p	0.1–0.95
Wind heat transfer coefficient, h	10–30 W m K^{-1}
Collector tilt angle, β	20–60^0

RESULTS AND DISCUSSIONS

The variation of absorber and glass cover temperature for two values of plate emissivity is shown in Figure-4. As the emissivity of absorber plate ε_p increases, the top loss coefficient also increases. The experimental data is shown for the measured values of glass temperature T_P along with theoretical predictions shown in Figure-5. The effect of ambient temperature on the efficiency of the system is shown in Figure-6. The efficiency is found to increase with increase in ambient temperature due to reduction in heat loss from the system. Figure-7 shows the effect of plate emissivity ε_p, on system efficiency. It can be observed increase in ε_p is to dissipate more heat to atmosphere and consequent reduction in efficiency of the system. The effect of wind loss coefficient h_w, on the efficiency is shown in Figure-8. As the wind loss coefficient H_W increases, more amount of heat is dissipated to atmosphere and consequently lower efficiency can be expected. The analysis has been conducted on FPC with a tilt angle of 20^0 with the horizontal.

The effect of inclination angle β is shown in Figure-9. It can be observed that there is no significant effect of tilt angle on the top loss coefficient.

CONCLUSIONS

Theoretical and experimental analysis is performed on a flat plate collector with a single glass cover. It can be concluded that the emissivity of the absorber plate has a significant impact on the top loss coefficient and consequently on the efficiency of the Flat plate collector. The efficiency of FPC is found to increase with increasing ambient temperature. There is no significant impact of tilt angle on the top loss coefficient.

ACKNOWLEDGEMENT

The Authors acknowledge the financial support for funding the fabrication of the experimental setup from JNTUHCE, JNTU, Hyderabad.

Nomenclature

A_p Area, m^2
A Ammeter
D Dimmerstat
Nu Nusselt number (hd/k)
Ra Rayleigh number, Gr.Pr
Gr Grashof number
Pr Prandtl number
h convective heat transfer coefficient, $Wm^{-2}K^{-1}$
h_w Wind loss coefficient, $Wm^{-2}K^{-1}$
N number of glass covers
L spacing between the absorber plate and glass cover
T Temperature, ^0C
T_{pm} Absorber Plate mean Temperature, ^0C
T_∞ Ambient Temperature, ^0C
ΔT Temperature difference between enclosed surfaces
U_l Overall Loss Coefficient, $Wm^{-2}K^{-1}$
U_t Overall Top Loss Coefficient, $Wm^{-2}K^{-1}$
V Voltmeter

Greek symbols

ε Emissivity
σ Stefan–Boltzman constant, $Wm^{-2}K^{-4}$
β' Film Temperature, ^0C
μ Viscosity of air
υ kinematic viscosity of air
α Thermal diffusivity of air
δ_P plate thickness [m]
β Collector tilt angle

Sub scripts

a ambient
p absorber plate
g glass cover

REFERENCES

[1] H. Tabor, Radiation. 1985. Convection and conduction coefficients in solar collectors, Bull. Res. Council of Israel. 6C. pp. 155-176.

[2] K. G. T. Hollands, T. E. Unney, G. D. Raithby and L. Konicek. 1976. Free convective heat transfer across inclined air layers, J. Heat Transfer, Trans.ASME. 98(2): 189-193.

[3] Hottel HC, Woertz BB. 1942. The performance of flat-plate solar heat collectors. Trans ASME. 64: 94-102.

[4] Klein SA. 1975. Calculation of flat-plate collector loss coefficients. Sol. Energy.17: 79-80.

[5] Klein SA. In: Duffie JA, Beckman WA. 1991. Solar engineering of thermal processes. New York: Wiley. p. 260.

[6] Agarwal VK, Larson DC. 1981. Calculation of top heat loss coefficient of a flat-plate solar collector. Sol Energy. 27: 69-71.

[7] H. P. Garg and U. Rani. 1980. Loss Coefficients from Solar Flat-Plate applied Energy. 7: 109 -117

[8] Malhotra A, Garg HP, Palit A. 1981. Heat loss calculation of flat-plate solar collectors. J Thermal Eng (J Indian Soc Mech Eng). 2(2): 59-62.

[9] Garg HP, Datta G. 1984. The top heat loss calculation of flat-plate solar collectors. Sol Energy. 32: 141-3.

[10] P. K. C. Pillai and R. C. Agarwal, An Analytical Approach for Otimising a set of α and ε values for Solar Energy Applications; Energy Conv. and Mgmt. 20: 205 to 212

[11] Mullick SC, Samdarshi SK. 1988. An improved technique for computing the top heat loss factor of flat-plate collector with a single glazing. ASME J Sol Energy Eng. 110: 262-7.

[12] Samdarshi SK, Mullick SC. 1991. An analytical equation for top heat loss factors a flat-plate solar collector with double glazing. ASME J Sol. Energy Eng. 113: 117-22.

[13] Akhtar N, Mullick SC. 1999. Approximate method for computation of glass-cover temperature and top heat-loss coefficient of solar collectors with single glazing. Sol Energy. 66(5): 349-354.

ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

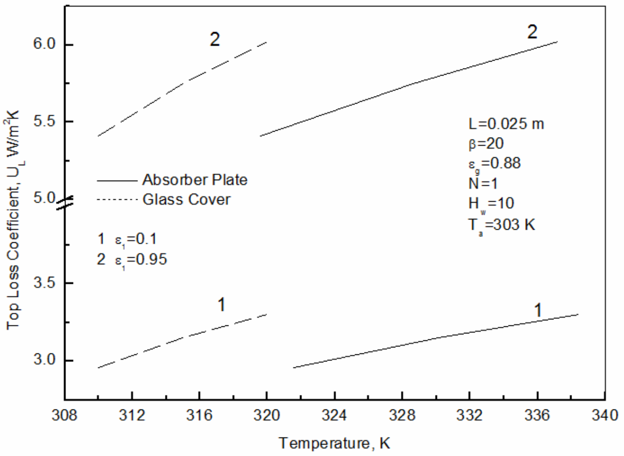

Figure-4. Variation of top loss coefficient with absorber plate and glass cover temperature for different plate emissivity.

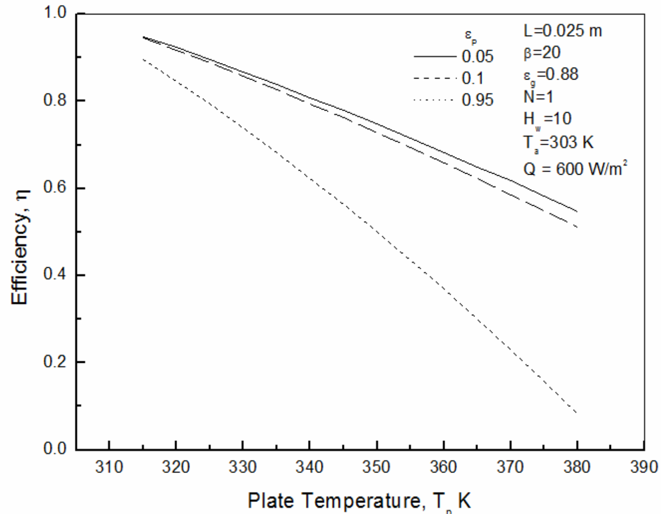

Figure-7. Effect of emissivity of absorber plate (ε_p) on efficiency.

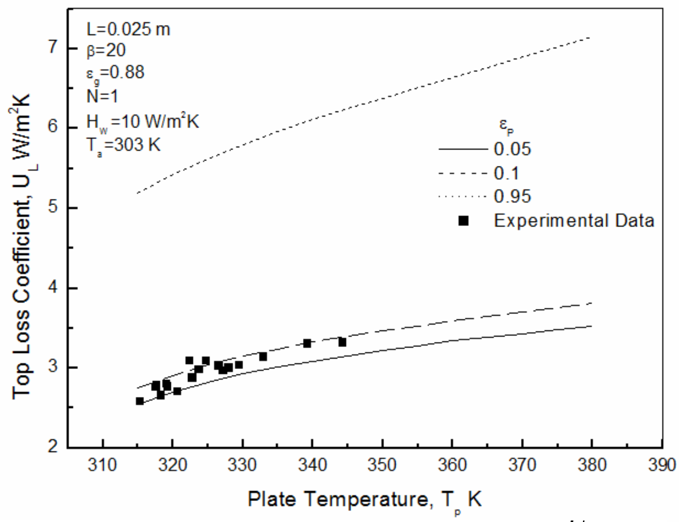

Figure-5. Variation of top loss coefficient (U_L) with absorber plate temperature.

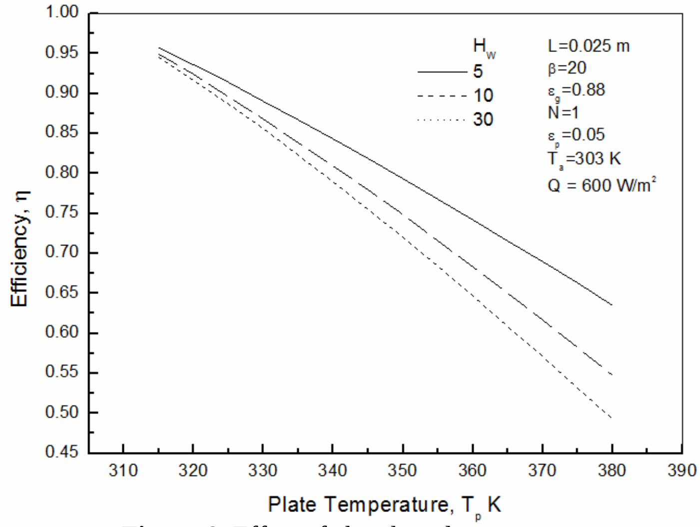

Figure-8. Effect of absorber plate temperature on efficiency for different values of h_W.

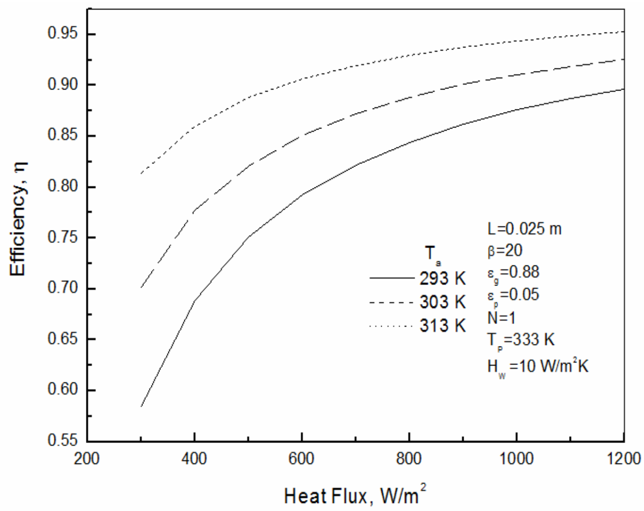

Figure-6. Effect of efficiency on heat flux for different ambient temperatures.

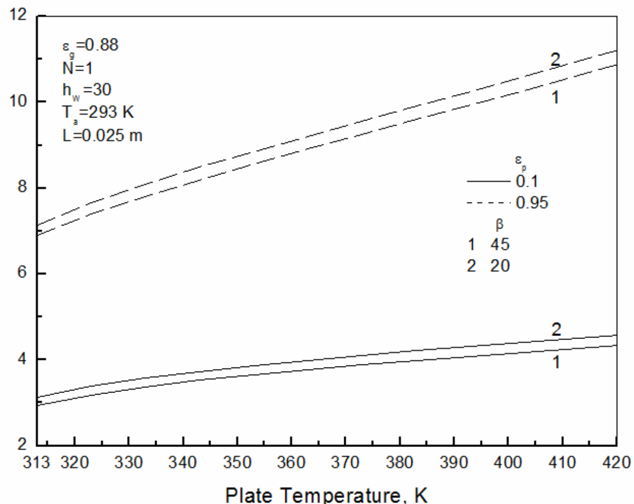

Figure-9. Effect of top loss coefficient on absorber plate temperature for different emissivities of absorber plate.