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PARAMETRIC ANALYSIS OF DOUBLE GLAZED FLAT PLATE SOLAR COLLECTOR FOR HEAT LOSE OPTIMIZATION

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Abstract

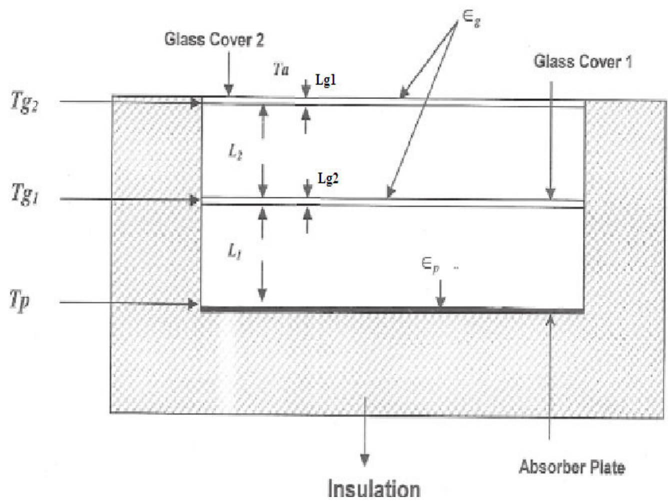
The present work aims at the study of effect of estimation of air properties by two sets of correlation on the estimation of top heat loss coefficient of double glazed flat plate solar collector. The natural convection heat transfer between configuration correlations of Holland & Buchberg was used.. This comparison of air properties effects on "over all heat loss coefficient" by Buchberg $\{(U_t)_b\}$ & Holland $\{(U_t)_h\}$ has been studied by changing the emissivity of absorber plate e_p . In present work an numerical approach is made to reduce the heat losses in flat plate solar collector by using various material quoting in absorber plate. The material quoting having various value of emissivity is used in order to minimize the over-all heat lose coefficient.

Introduction

The performance of a solar collector be as high as economical practical, design and operating factors that the value of $U_L(T_p-T_a)$. The greater the energy absorption in the metal surface and lower heat loss from the surface, the higher is the useful recovery. If an unglazed absorber plate is used, the heat loss coefficient to the atmosphere U_L of 30 to 60 W/m^{2}ºC is so large that an absorber temperature of 15 to 30 ºC above atmospheric temperature is the maximum achievable under full solar radiation of 1000 W/m^2. Under these conditions, no useful heat is delivered from the collector because the heat loss is as large as the solar heat observed when a fluid is circulated through the collector, no useful heat output requires an even lower delivery temperature. Unless a low temperature application is involved, such as swimming pool heating, heat losses must, therefore, be required.

To reduce the rate of radiation an convection losses one or more transparent surfaces such as glass, are placed above the absorber surface. One layer of glass can transmit as much as 92% of solar radiation striking it, while greatly reducing the heat loss UL. This reduction is due to the suppression of convection losses by interposing a relatively stagnant air layer between absorber plate and glass, and by absorption of long wave thermal radiation emitted by the hot metal absorber surface. the combined heat loss coefficient can be reduced to 5 to 10 W/m^2 by the use of one glass cover.

The heat loss coefficient can be reduced further by using a second transparent cover with an air space between the two surfaces. Two convection barriers are then present, as well as two surfaces impeding radiation loss coefficient in the range of 3-4 are then typically obtained.



Solar flat plate collector Radiation losses can be decreased by other techniques such as by reducing the radiation emitting characteristics of the absorber. Thermal radiation emitted by the absorber plate may also be reduced by reflecting it downward from the lower glass cover by employing an infrared reflecting coating on the glass. A very thin, optically transparent layer of tin oxide or indium oxide deposited on the glass reflects thermal radiation back to the absorber plate. This coating absorbs some of the solar radiation, however, so the reduced thermal loss is largely offset by reducing solar energy input to the absorber plate.

The foregoing discussion has been concerned with methods for reducing U_L the heat loss coefficient. By so doing, the total heat loss is minimized and the collector efficiency is increased. It is evident that the losses also decrease as the difference between average plate temperature and air temperature decreases. The ambient (outside) air temperature is uncontrollable factor, but the fact that it varies with time and with geographic location means that collector efficiency also depends on these factors. It is clear also, that a collector is more efficient at lower plate temperatures than at high temperatures. But plate temperature depends on the temperature of the fluid being circulated in contact with the plate, the rate of fluid circulation and the type of fluid. Fluid temperature depends on the condition elsewhere in the heat utilization system, whereas the other factors depend on the collector design, operating conditions, solar energy input, and atmospheric temperature. The thermal losses from the collector to the ambient by reducing conductive, convective and radiation losses.

Conductive losses that occur through the back and sides of the collector can be reduced by using sufficiently thick layer of thermal insulation. The main problem in the front, where heat is conducted from the absorber plate through the air layer between the plate and the transparent covers and an out to the ambient air. The increase in the thickness of the glass cover reduces the losses upto a certain limit, where further increases allow significant natural convection. The natural convection transfers heat at higher rates than does conduction ; this leads to higher, rather than lower, heat losses. Alternatively several transparent panels could be used to create a number of narrow air gaps. This however reduces the transmission of solar energy to the absorber. The single panels collector is the most efficient when the absorber temperature is not much higher than that of the outer cover plate (transmittance dominating over heat losses) but becomes rapidly less efficientas this temperature difference increases. Therefore, high temperature collectors require two transparent covers.

Convective losses separate into internal convective losses from the absorber plate to the outer cover pane, and external losses from the outer cover pane to the atmosphere or ambient air. In the absence of the wind, the external convective losses are caused by the natural convection. Even low winds, however dominate convection when they occur. Although means could and should be introduced to reduce to the external convective losses, it would be most useful to reduce the internal losses, and thus to reduce the temperature of the outer cover plate. While the available information on natural convection in vertical air gaps is not conclusive the convection is very small and comparable in its effect to the conduction for small Grashof or Rayleigh numbers, and it becomes significantly greater for larger values of these numbers. The nature of the convection also depends on the specific boundary conditions and geometric aspects of the enclosure. Besides the maintenance of narrow air gaps to decrease convection, cellular structures are:

(i) The absorber plate :
(ii) Increase the thermal conductivity of he space between the absorber and the cover plate
(iii) add to the cost of the collector

Evacuation of the space between the absorber and the cover plate practically eliminates convective losses. Radiation losses from the absorber to the ambient can be reduced by a spectrally selective coating on the absorber plate. These coatings have a high absorptivity in the solar spectrum, but have a substantially lower emissivity usually of the order of one-tenth, in the infra red spectrum, in which most absorber plates radiate. The selective absorbers thus decrease heat losses and increase collector efficiency.

Thermal radiation emitted by the absorber plate may also be reduced by reflecting it downward from the lower glass cover by employing an IR reflecting coating on the glass. An optically transparent, a very thin layer, tin oxide or indium oxide deposited on the glass will reduce radiation loss by reflecting it back to the absorber plate as stated earlier. The coating small fraction of the solar radiation, however, so the reduced thermal loss is largely offset by reduce solar energy input to the absorber plate.

LITRETURE REVIEW

Comparison Study of Solar Flat Plate Collector with Single and Double Glazing Systems……….. by ***H.Vettrivel and P.Mathiazhagan*** *- INTERNATIONAL JOURNAL of RENEWABLE ENERGY RESEARCH 2017*

In this research an approach is made to reduce the overall top loss heat transfer coefficient and improve the collector efficiency. The researcher introduced the double glaze system and optimized the space between the absorber plate to glass cover (1) and glass cover (2)were considered to analysis the overall top loss heat transfer coefficient (Ut).

In this work researcher fabricated The single and double glazing solar flat plate collectors with same dimensions and installed at a latitude angle of 12 degree facing towards N-S direction.

The result of this work shows that the efficiency of double glazing is higher compared to single glazing system with same solar intensity. The higher efficiency has obtained because of the overall top loss heat transfer coefficient was reduced in double glazing system.

Glass cover temperature and top heat loss coefficient of a single glazed flat plate collector with nearly vertical configuration. By -Suresh Kumar, S.C.Mullick **"Ain Shams Engineering Journal (2012)"**

In this work author had proposed An empirical relation for glass cover temperature of a single glazed flat plate collector for angle of tilt 60–90_ is proposed. Values of glass cover temperature obtained from empirical relation have been used for computation of top heat loss coefficient of collector. Analytical equation has been employed for estimation of top heat loss coefficient, U_t . The range of variables covered in the present analysis is 20 _C to 150 _C for absorber plate temperature, 0.1–0.95 for absorber coating emittance, 20–50 mm for air gap spacing, 60–90_ for collector tilt, 5–30 W/m K for wind heat transfer coefficient and _10 _Cto40_C for ambient temperature. The maximum absolute error in values of U_t is within two percent, in comparison to values obtained by numerical solution of heat balance equations, over the entire range of variables.

Parametric studies of top loss coefficient of double glazed flat plate solar Collector by -*Bisen, P.P Das, Rajeev Jain* (2011*)….MIT International journal of mechanical engineering vol. 1 aug 2011.*

In this research the author had done the study of effect of estimation of air properties by two sets of correlation on the estimation of top heat loss coefficient of double glazed flat plate solar collector by estimation glass cover temperatures .by using numerical solution technique this comparison of air properties effects on top heat loss coefficient U_t" has been wind heat transfer coefficient h_w using MATLAB. studied for given range of absorber plate temperature T_p ambient temperature T_a, and There is an increasing demand for the solar collectors, especially the flat-plate liquid solar collector. Therefore, an extensive research has been done to model the flat plate solar collectors operation and to predict the performance of different type's solar collector. This chapter presents a summary of the fundamentals as well as the state-of the- art research that has been conducted in the area of flat-plate solar collector modeling and performance prediction

3. PROPOSED WORK

The main objective of this research is to determine the various parameters on solar flat plate collector with double glazed by varying the thickness of glass cover with the change in ambient temperature.

- Overall heat transfer coefficient (U_L)
- Top heat loss coefficient (U_t)
- Bottom heat loss coefficient (U_b)
- convective heat transfer coefficient between plate & glass1 2 (h_{cpg1}).
- Convective heat transfer coefficient between glass 1 & glass 2 (h_{cg1g2}).
- Radiative heat transfer coefficient between plate & glass1.
- Radiative heat transfer coefficient between glass 1 & glass 2.
- Nusselt Number Space Between the first glass cover & second glass cover .
- Raleigh number of enclosed space between plate & first glass cover (R_{a1})
- Raleigh number of enclosed space between first glass cover & second glass cover (R_{a2})
- Thermal conductivities of air (K)
- Kinematic viscosities of the air (υ)
- PrandtlNumber (Pr)
- temperature of the first glass cover (T_{g1})
- temperature of the second glass cover (T_{g2})

According to[N. Akhtar , S. C. Mullick (2007)The geometrical data of the solar flat plate collector with double glazed is taken from the reference paper. And by taking the parameters from that paper has been carried out of that model. The parameters used in that paper are given below:-

VARIABLE	**RANGE OF COLLECTOR in reference paper**	**RANGE TAKEN IN PRESENT WORK**
Absorber plate temperature	373 K to 423 K	373 K
thickness of glass cover 1	0.004 m	0.001-0.010 m
thickness of glass cover 2	0.004 m	0.001-0.010 m
Inner air gap spacing	0.098 m	0.101-0.092 m
Outer air gap spacing	0.012 m	0.015-0.006 m
Wind heat transfer	5 W/m^2K TO 25 W/m^2K	5 W/m^2k

coefficeint		
Emissivity of absorber plate	0.9	0.9
Emissivity of glass	0.88	0.88
Ambient temperature	273 K TO 318 K	273-318 K

Estimation of air property correlation on the estimation of over all heat transfer coefficient of double glazed flat plate solar collector by estimating top heat loss, bottom heat loss, glass cover temperature, Rayleigh Number (Ra) and Nusselt Number (Nu) can be calculated by the relation suggested by the Buchberg et. al. [4] to estimate the natural convection heat transfer coefficient by using numerical solution technique. This comparison of air properties effects on top heat loss coefficient (Ut) has been studied for the given range of variables like, absorber plate Tp, ambient temperature Ta and glass cover temperatures Tg1 and Tg2, wind heat loss coefficient hw, emissivity of absorber plate Îp and glass cover Îg, spacing between the absorber plate and glass cover L1, L2, tilt angle of the collector B, number of glass covers N and thickness of glass covers Lg1, Lg2.

4. METHODOLOGY

The top heat loss coefficient (U_t) is evaluated by considering convection and radiation losses from the absorber plate in the upward direction. For most of the applications, the flat- plate collectors are either single glazed or double glazed. Following analytical expressions has been proposed by samdarshi & Mullick (8) for the calculation of U_t of double glazed flat plate collectors.

$$U_t^{-1}=(h_{rpg1}+h_{cpg1})^{-1}+(h_{rg1g2}+h_{cg1g2})^{-1}+(h_{r2a2}+h_w)^{-1}+(L_{g1}+L_{g2})/K_g\ldots\ldots(1)$$

Upward heat loss from the absorber plate to the first glass cover

$$Q_t= (h_{cpg1}+h_{rpg1})(T_p-T_{g1})$$

And from the first glass cover to the second glass cover by

$$Q_t= (h_{g1g2}+h_{rg1g2})(T_{g1}- T_{g2})$$

And from the second glass cover to the atmosphere by

$$Qt= (h_w+h_{rg2a})(T_{g2}- T_a)$$

Where,

Heat transfer coefficient between absorber plate & first glass cover

$$h_{cpg1}=\frac{K1\times Nu1}{L1}$$

Radiative heat transfer coefficient between absorber plate & first glass cover [by DUFFIE]

$h_{rpg1} = \{\frac{\sigma}{\frac{1}{\varepsilon p}+\frac{1}{\varepsilon g}-1}\}\ (T_p^2+T_{g1}^2)(T_p+T_{g1})$

Heat transfer coefficient between absorber plate & first glass cover

$h_{cg1g2} = \frac{K2 \times Nu2}{L2}$

Radiative heat transfer coefficient between first glass cover & second glass cover [by DUFFIE]

$h_{rpg1} = \{\frac{\sigma}{\frac{2}{\varepsilon g}-1}\}\ (T_{g1}^2+T_{g2}^2)(T_{g1}+T_{g2}$

Radiative heat transfer coefficient between second glass cover & ambient [by DUFFIE]

$h_{rg2a2} = \sigma.\varepsilon_g.(T_{g2})^4-(T_{sky})^4\frac{(Tg2)4-(Tsky)4}{Tg2- Ta}$

Measurfable variables in a solar collector

T_p= Absorber plate temperature in °K

T_a = Ambient temperature in °K

ε_g= Emissivity of glass

ε_p= Emissivity of Plate

L_1 = Spacing between the absorber plate & first glass cover in meters

L_2 = Spacing between the first glass cover to the Second glass cover in meters

L_{g1}= Thickness of first glass cover in meters

L_{g2}= Thickness of second glass cover in meters

K_g= Thermal conductivity of glass cover materials W/m°K

Using numerical solution Method has been done. The properties of air with Mean temperature as independent variable are evaluated.

For most of the applications, the flat- plate collectors are either single glazed or double glazed. Following analytical expressions has been proposed by samdarshi&Mullick(2008) for the calculation of U_t of double glazed flat plate collectors.

$U_t^{-1}=(h_{rpg1}+h_{cpg1})^{-1}+(h_{rg1g2}+h_{cg1g2})^{-1}+(h_{r2a2}+h_w)^{-1}+(L_{g1}+L_{g2})/K_g$

Air Properties

For accurate prediction of collector performance it is necessary of the working fluid (air) to calculate the convective heat transfer coefficient. Properties of air evaluated at the arithmetic mean of the corresponding surface temperatures [BY SUKHATME]

$$T_{mpg1}=\frac{Tp+Tg1}{2}$$

And,

$$T_{mpg1}=\frac{Tg1+Tg2}{2}$$

The following relations for air properties are used to calculate Raleigh number.

Air Properties Correlations – I

Air property correlation –I for absorber plate & first glass cover [BY DUFFIE& BECK MEN]

Thermal conductivities of air are calculated by the relation given below

$$K_1= -3\times10^{-8}\,T_{mpg1}^{\;2} + 10^{-4}T_{mpg1}-4\times10^{-5}$$

$$K_2= -3\times10^{-8}\,T_{mg1g2}^{\;2} + 10^{-4}T_{mg1g2}-4\times10^{-5}$$

Kinematic viscosities of the air are calculated by the relation given below

$$\upsilon_1= [9\times10^{-5}T_{mpg1}^{\;2} + 0.040T_{mpg1}-4.17]\times10^{-5}$$

$$\upsilon_2= [9\times10^{-5}T_{mg1g2}^{\;2} + 0.040T_{mg1g2}-4.17]\times10^{-5}$$

Prandtl Number is calculated by the

$$P_{r1}=1.057-0.06\,\log T_{mpg1}$$

$$P_{r2}=1.057-0.06\,\log T_{mg1g2}$$

Calculation of Raleigh number

Raleigh number (Ra1) of enclosed space between plate & first glass cover and Raleigh number (Ra2) of enclosed space between first glass cover & second glass cover can be calculated by using air properties of corresponding enclosed space.

Raleigh number (R_{a1}) of enclosed space between plate & first glass cover is calculated by the relation given below

$$R_{a1} = \frac{9.8\times(Tp-Tg1)(L1)3(Pr1)}{\upsilon1*\upsilon1(\frac{Tp+Tg1}{2})}$$

Raleigh number (Ra2) of enclosed space between first glass cover & second glass cover is calculated by the relation given below

$$R_{a2} = \frac{9.8\times(Tg1+Tg2)(L2)3(Pr2)}{\upsilon2*\upsilon2(\frac{Tg1+Tg2}{2})}$$

Calculation of Nusselt Number

A review of correlations quantifying heat transfer has been carried out by Holland et. al. (6) and Buchberget.al.(4). They recommended use of the following correlations to calculate Nusselt Number.

According to Buchberg equation for Nusselt Number

The Nusselt Number has been calculated by the following conditions.

For (Nu_1) Space Between the absorber plate & first glass cover is calculated by the relation,

If Ra1<1708 Then

$Nu_1=1$

If $(Ra_1>1708)\&(Ra_1<5900)$, then

$Nu_1= 1+1.446[1-\frac{1708}{Ra1}]$If $(R_{a1}>5900)\&(R_{a1}<9.23\times10^4)$, then

$Nu_1= 0.229(R_{a1})^{0.252}]$ If $(Ra_1>9.23\times10^4)\&(Ra_1< 10^6)$, then

$Nu1= K_1\times(Ra_1)^{0.285}]$

For (Nu_2) Space Between the first glass cover & second glass cover is calculated by the relation,

If $Ra_2<1708$ Then,

$Nu_2=1$

If $(Ra_2>1708)\&(Ra_2<5900)$, then

$Nu_2= 1+1.446[1-\frac{1708}{Ra2}]$

If $(Ra_2>5900)\&(Ra_2<9.23\times10^4)$, then

$Nu_2= 0.229(Ra_2)^{0.252}$

If $(Ra_2>9.23\times10^4)\&(Ra_2< 10^6)$, then

$Nu_2= K_2\times(Ra_2)^{0.285}$

CALCULATION FOR NATURAL CONVECTION & RADIATION HEAT TRANSFER COEFFICIENTS

The convective heat transfer coefficient (hcpg1, hcg1g2) and radiative heat transfer coefficient (hrpg1, hrg1g2, hrg2a2) has been calculated by the following relations given below

Relation for convective heat transfer coefficient between plate & glass1.

$$h_{cpg1}=\frac{K1\times Nu1}{L1}$$

Relation for Radiative heat transfer coefficient between plate & glass1.

$$h_{rpg1}=\{\frac{\sigma}{\frac{1}{\varepsilon p}+\frac{1}{\varepsilon g}-1}\}\,(T_p{}^2+T_{g1}{}^2)(T_p+T_{g1})$$

Relation for convective heat transfer coefficient between glass 1 & glass 2.

$$h_{cg1g2}=\frac{K2\times Nu2}{L2}$$

Relation for Radiative heat transfer coefficient between glass 1 & glass 2.

$$h_{rpg1}=\{\frac{\sigma}{\frac{2}{\varepsilon g}-1}\}\,(T_{g1}{}^2+T_{g2}{}^2)(T_{g1}+T_{g2})$$

Relation for Radiative heat transfer coefficient between glass 2 & atmosphere.

$$h_{rg2a2}=\sigma.\varepsilon_g.(T_{g2})^4-(T_{sky})^{4\frac{(Tg2)4-(Tsky)4}{Tg2-Ta}}$$

Calculation of Top Heat Loss Coefficient

For most of the applications, the flat- plate collectors are either single glazed or double glazed. Following analytical expressions has been proposed by samdarshi & Mullick(8) for the calculation of U_t of double glazed flat plate collectors.

$$U_t^{-1}=(h_{rpg1}+h_{cpg1})^{-1}+(h_{rg1g2}+h_{cg1g2})^{-1}+(h_{r2a2}+h_w)^{-1}+(L_{g1}+L_{g2})/K_g$$

Where,
L_{g1}= Thickness of the first glass cover,
L_{g2}= Thickness of the Second glass cover,
K_g= Thermal conductivity of glass,

Overall heat transfer coefficient (U_L) can be calculated by following correlations

$U_L=U_t+U_b+U_s$
Where Bottom heat loss (U_b)

$$U_b=\frac{K}{L1}$$

If side heat loss coefficient (U_s) is assumed to be constant.
$U_L=U_t+U_b$

RESULT AND DISCUSSION

A way to describe to overall heat transfer co-efficient of double glazed plate solar collector using numerical evaluation. The result have been represented in graphical manner. And tables also produce to elaborate the graphs. The present study shows the result between change in emissivity of absorber plate and ambient temperature. The overall heat transfer coefficient conserving the a set of air property correlations, on over all heat transfer coefficient has been discussed

In this study

Top heat loss co-efficient U_t is calculated from: equation

$$U_t^{-1} = (h_{rpg1}+h_{cpg1})^{-1}+(h_{rg1g2}+h_{cg1g2})^{-1}+(h_{r2a2}+h_w)^{-1}+(L_{g1}+L_{g2})/K_g$$

&

$$\frac{1}{Ut} = \frac{1}{\mathrm{Upg1}} + \frac{1}{\mathrm{Ug1g2}} + \frac{1}{\mathrm{Ug2a}} + \frac{Lg1+Lg2}{Kg}$$

&

$$\frac{1}{Ut} = \frac{1}{(\mathrm{hcpg1+\ hrpg1})} + \frac{1}{(\mathrm{hcg1g2+hrg1g2})} + \frac{1}{(\mathrm{hw+hrg2a})} + \frac{Lg1+Lg2}{Kg}$$

Radiative heat transfer coefficient is calculated from equation

$$\mathrm{hrpg1} = \left\{\frac{\sigma}{\frac{1}{\varepsilon p}+\frac{1}{\varepsilon g}-1}\right\} (T_p^2+T_{g1}^2)(T_p+T_{g1})$$

Convective heat loss coefficient is calculated from equation –

$$\mathrm{hcpg1} = \frac{\mathrm{K1\times Nu1}}{\mathrm{L1}}$$

For natural convection nusselt number is calculated from equation

According to Holland relation

$$Nu_1 = 1+1.44[1-\frac{1708}{Ra1}] +[(\frac{Ra1}{5830})^{1/3}-1]$$

If $(\frac{Ra1}{5830})^{1/3}‹1$,

then$Nu_1 = 1+1.44[1-\frac{1708}{Ra1}]$

According to Buchberg relation

If $Ra_1<1708$ Then $Nu_1=1$.

If $(Ra_1>1708)\&(Ra_1<5900)$, then

$$Nu_1 = 1+1.446\left[1-\frac{1708}{Ra1}\right]$$

If $(Ra_1>5900)\&(Ra_1<9.23\times10^4)$, then

$$Nu_1 = 0.229(Ra_1)^{0.252}$$

If $(Ra_1>9.23\times10^4)\&(Ra_1< 10^6)$, then

$$Nu_1 = K_1\times(Ra_1)^{0.285}$$

The raileigh number is calculated from equation

$$Ra = \frac{9.8\times(Tp-Tg1)(L1)^3(Pr1)}{V1^2(\frac{Tp+Tg1}{2})}$$

The glass cover temperature is calculated by the equations

$$T_{g2} = T_a + h_w^{-.04}(0.0012T_p+0.37\varepsilon_p-0.146)(T_p-T_a).$$

$$Tg1 = T_p - (0.7-0.34\varepsilon_p)(T_p-T_2).$$

The two sets of air properties correlations used in present work. The Overall heat transfer coefficient is computed by varying the emissivity of absorber plate material and the ambient temperature with this Property Correlation

VARIABLE	RANGE
T_P	373 K
B	0 deg
L_{g1}	0.004 m
L_{g2}	0.004 m
L_1	0.098 m
L_2	0.012 m
h_w	5 W/m^2k
ϵ_p	0.93,0.91,0.90 – 0.12,0.11,0.10
ϵ_g	0.88
T_a	273-318 K

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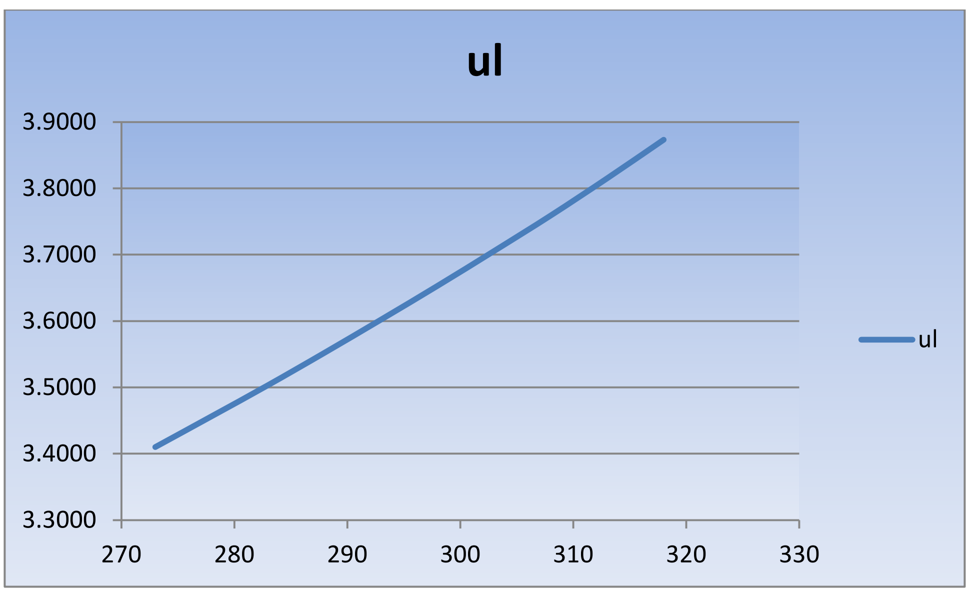
Change in overall heat transfer coefficient by changing the ambient temperature T_a.for different absorber plate material.

TABLE:-1 Absorber plate material Black Tar Paper (e_p= 0.93)

S. No.	tp	ta	tg1	tg2	ut	ub	ul
1	373	273	347.6381	306.918	3.0825	0.3275	3.4100
2	373	278	348.9061	310.223	3.1285	0.3280	3.4565
3	373	283	350.1743	313.527	3.1750	0.3285	3.5035
4	373	288	351.4424	316.831	3.2233	0.3290	3.5523
5	373	293	352.7104	320.135	3.2728	0.3295	3.6023
6	373	298	353.9785	323.439	3.3232	0.3300	3.6532
7	373	303	355.2466	326.743	3.3751	0.3305	3.7056
8	373	308	356.5174	330.047	3.4280	0.3310	3.7590
9	373	313	357.7828	333.351	3.4836	0.3315	3.8151
10	373	318	359.0509	336.665	3.5412	0.3320	3.8732

Graph : 1:- The overall heat transfer coefficient with ambient temperature for absorber plate material black tar paper(e_p= 0.93)



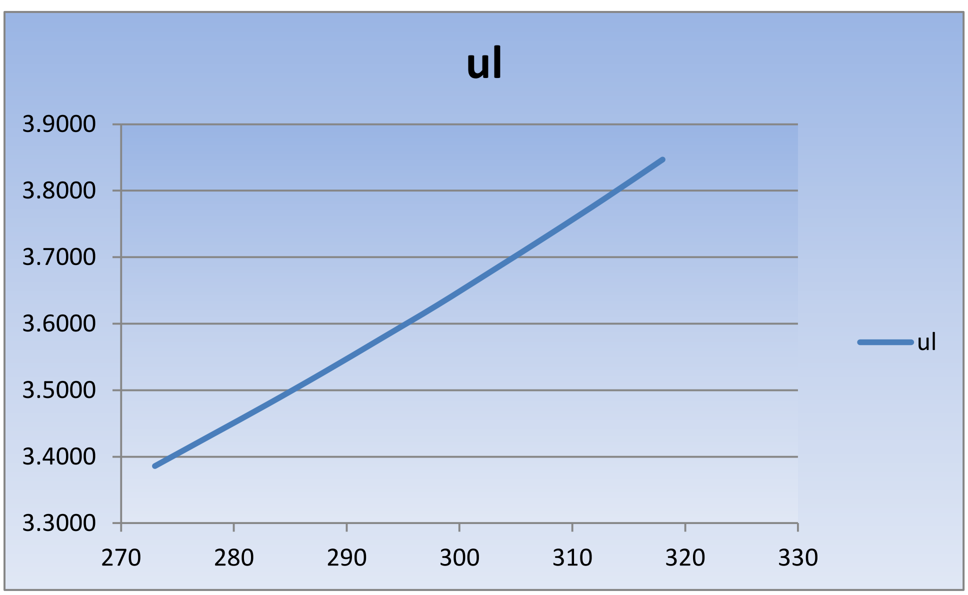
Graph 1.1 represents the changes in overall heat transfer coefficient by changing the ambient temperature T_a. (Black Tar Paper)

The increase in the ambient temperature T_a effects on the value of overall heat transfer coefficient increases.at **273k- 3.4100 to 318 k-3.8732.**

TABLE:-2 Absorber plate material white paint (e_p= 0.91)

S. No.	tp	ta	tg1	tg2	ut	ub	ul
1	373	273	347.037	306.53	3.0588	0.3272	3.3860
2	373	278	348.335	309.853	3.1047	0.3277	3.4324
3	373	283	349.633	313.177	3.1510	0.3280	3.4790
4	373	288	350.932	316.501	3.1990	0.3283	3.5273
5	373	293	352.229	319.824	3.2480	0.3292	3.5772
6	373	298	353.528	323.148	3.2980	0.3296	3.6276
7	373	303	354.825	326.472	3.3503	0.3304	3.6807
8	373	308	356.123	329.754	3.4036	0.3309	3.7345
9	373	313	357.422	333.118	3.4583	0.3314	3.7897
10	373	318	358.720	336.441	3.5150	0.3319	3.8469

Graph 2:- The overall heat transfer coefficient with ambient temperature for absorber plate material white paint (e_p= 0.91)



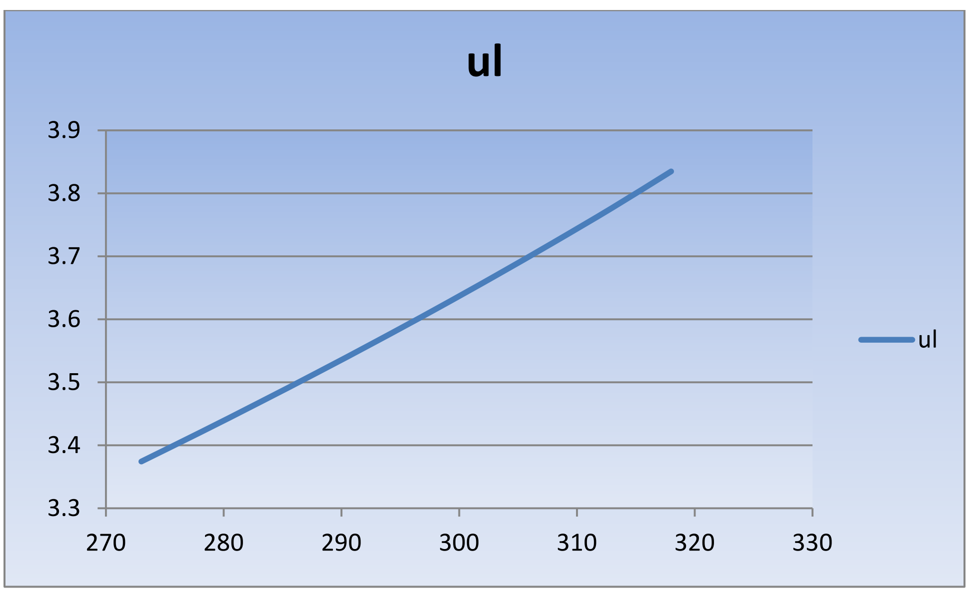
Graph 1.2 represents the changes in overall heat transfer coefficient by changing the ambient temperature T_a. (white paint)

The increase in the ambient temperature T_a effects on the value of overall heat transfer coefficient increases.at 273K –3.3860 to 318k-3.8469

TABLE;-3 Absorber plate 3M Velvet black paint (e_p= 0.90)

S. No.	tp	ta	tg1	tg2	ut	ub	ul
1	373	273	346.734	306.335	3.046885	0.32715	3.374035
2	373	278	348.0476	309.6691	3.092623	0.3276	3.420223
3	373	283	349.3609	313.023	3.139345	0.3282	3.467545
4	373	288	350.6742	316.3355	3.187121	0.3287	3.515821
5	373	293	351.9875	319.6687	3.236041	0.3292	3.565241
6	373	298	353.3008	323.0019	3.286215	0.3297	3.615915
7	373	303	354.614	326.3351	3.337786	0.3303	3.668086
8	373	308	355.9273	329.6683	3.39094	0.3308	3.72174
9	373	313	357.2406	333.0015	3.44548	0.33135	3.77683
10	373	318	358.5539	336.3347	3.502739	0.33187	3.834609

Graph 3:- The overall heat transfer coefficient with ambient temperature for absorber plate material 3M Velvet black paint (e_p= 0.90)



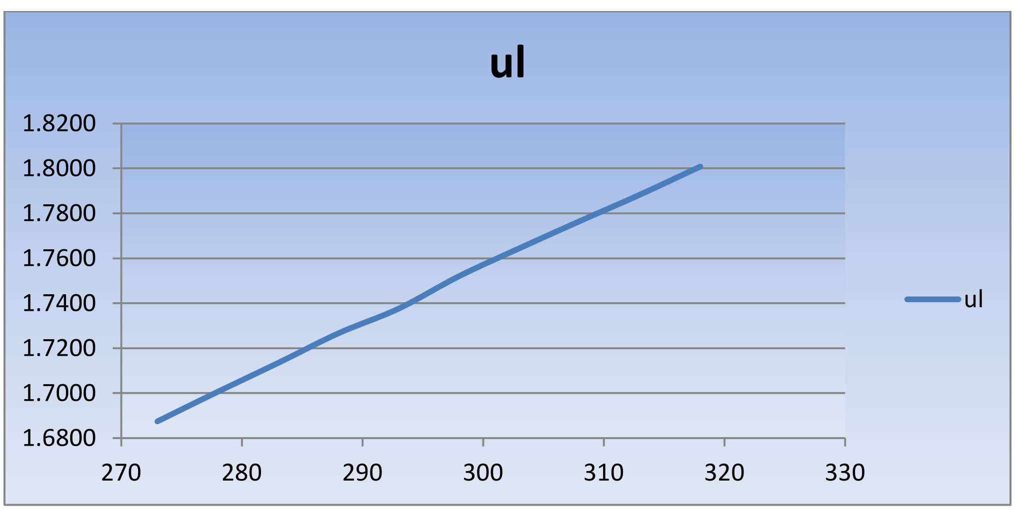
Graph 1.3 represents the changes in overall heat transfer coefficient by changing the ambient temperature T_a. **3M Velvet black paint**

The increase in the ambient temperature T_a effects on the value of overall heat transfer coefficient increases.at 273K –3.374035to 318k-3.834609

TABLE;-4 Absorber plate material black nickel on galvanized iron (e_p= 0.12)

S. No.	tp	ta	tg1	tg2	ut	ub	ul
1	373	273	319.061	291.176	1.3714	0.3160	1.6874
2	373	278	321.758	295.260	1.3835	0.3171	1.7006
3	373	283	324.455	299.398	1.3952	0.3182	1.7134
4	373	288	327.152	303.449	1.4074	0.3192	1.7266
5	373	293	329.849	307.540	1.4174	0.3202	1.7376
6	373	298	332.546	311.632	1.4306	0.3214	1.7520
7	373	303	335.249	315.722	1.4419	0.3226	1.7644
8	373	308	337.940	319.814	1.4529	0.3236	1.7765
9	373	313	340.637	323.905	1.4636	0.3247	1.7883
10	373	318	343.233	327.956	1.4751	0.3257	1.8008

Graph 4:- The overall heat transfer coefficient with ambient temperature for absorber plate material black nickel on galvanized iron (e_p= 0.12)



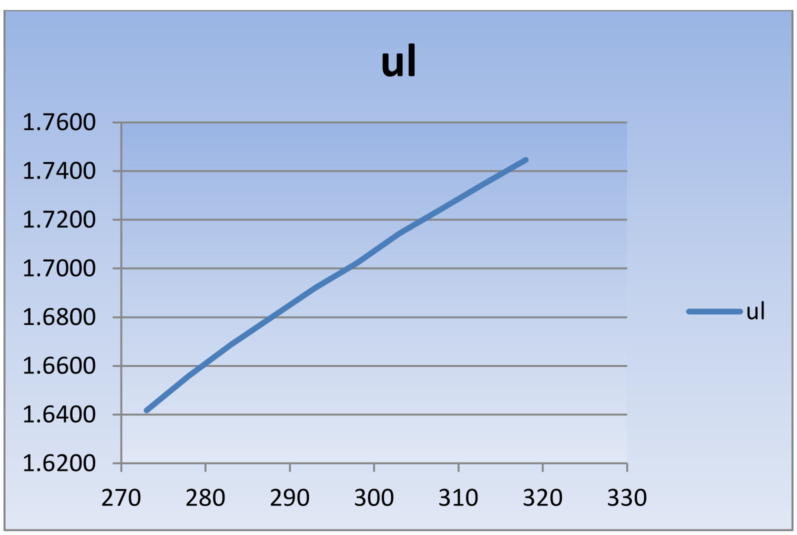
Graph 1.4 represents the changes in overall heat transfer coefficient by changing the ambient temperature T_a. (black nickel on galvanized iron)

The increase in the ambient temperature T_a effects on the value of overall heat transfer coefficient increases.at 273K –1.6874 to 318k-1.8008

TABLE:-5 Absorber plate material Cu on aluminum (e_p= 0.11)

S. No.	tp	ta	tg1	tg2	ut	ub	ul
1	373	273	318.654	290.981	1.3259	0.3158	1.6417
2	373	278	321.3716	295.082	1.3390	0.3170	1.6560
3	373	283	324.088	299.183	1.3507	0.3180	1.6687
4	373	288	326.806	303.284	1.3613	0.3190	1.6803
5	373	293	329.5634	307.384	1.3718	0.3202	1.6920
6	373	298	332.24	311.489	1.3810	0.3213	1.7023
7	373	303	334.958	315.587	1.3919	0.3224	1.7143
8	373	308	337.675	319.657	1.4015	0.3230	1.7245
9	373	313	340.392	323.788	1.4107	0.3240	1.7347
10	373	318	343.109	327.889	1.4196	0.3250	1.7446

Graph 5:- The overall heat transfer coefficient with ambient temperature for absorber plate material **Cu on aluminum (e_p= 0.11)**



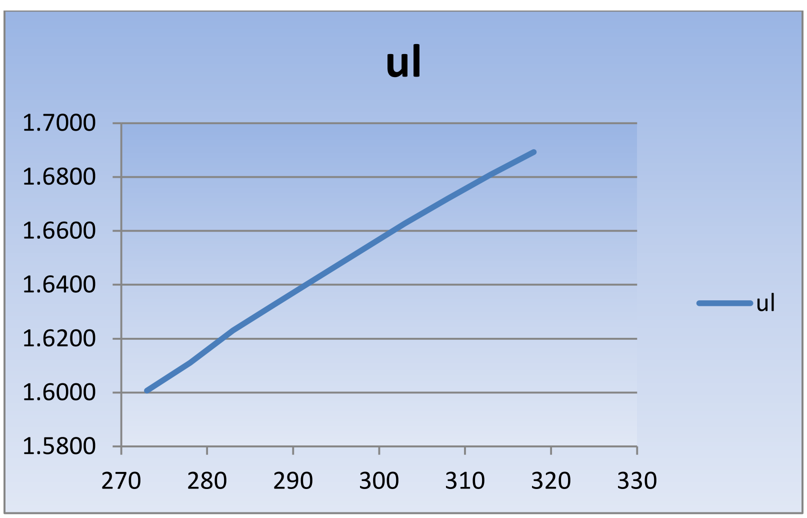
***Graph 1.5 represents the changes in overall heat transfer coefficient by changing the ambient temperature T_a.*(Cu on aluminum)**

The increase in the ambient temperature T_a effects on the value of overall heat transfer coefficient increases.at 273K –1.6417to 318k-1.7446

TABLE:-6 Absorber plate material Black crome (e_p= 0.10)

S. No.	tp	ta	tg1	tg2	ut	ub	ul
1	373	273	318.246	290.787	1.2850	0.3157	1.6007
2	373	278	320.984	294.897	1.2950	0.3160	1.6110
3	373	283	323.721	299.008	1.3050	0.3180	1.6230
4	373	288	326.459	303.189	1.3140	0.3190	1.6330
5	373	293	329.196	307.229	1.3230	0.3200	1.6430
6	373	298	331.934	311.340	1.3319	0.3210	1.6529
7	373	303	334.672	315.450	1.3404	0.3223	1.6627
8	373	308	337.41	319.561	1.3486	0.3234	1.6720
9	373	313	340.147	323.672	1.3564	0.3245	1.6809
10	373	318	342.885	327.783	1.3637	0.3256	1.6893

Graph 6:- The overall heat transfer coefficient with ambient temperature for absorber plate **material Black crome (e_p= 0.10)**



Graph 1.6represents the changes in overall heat transfer coefficient by changing the ambient temperature T_a. (Black crome)

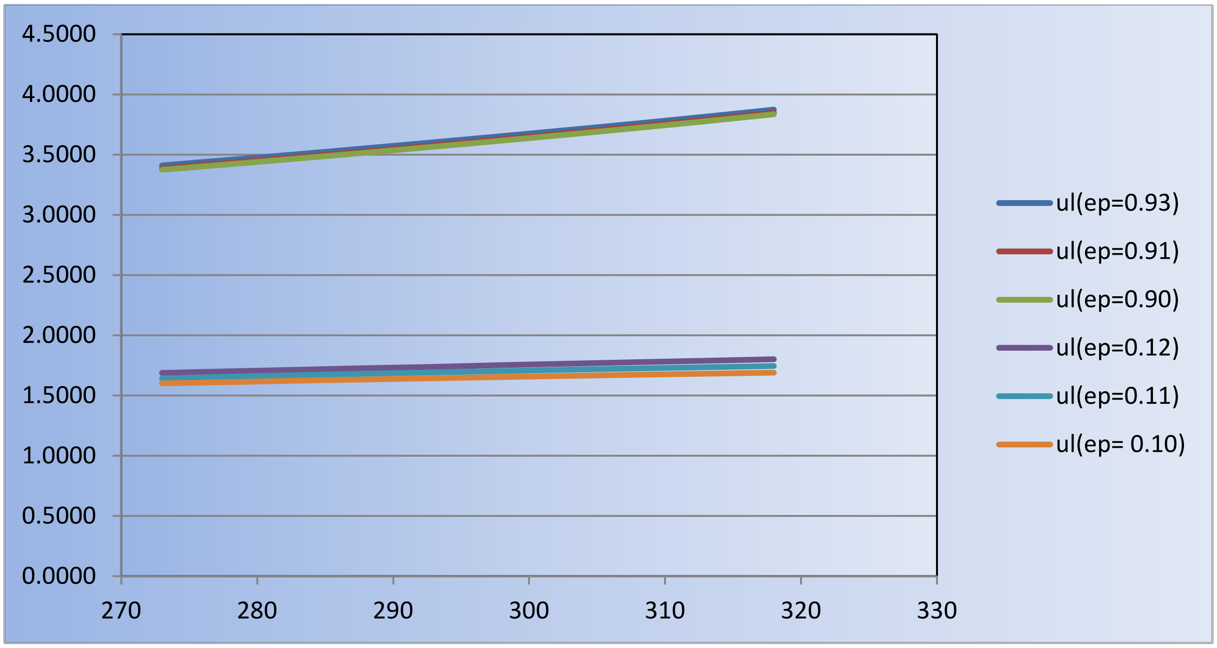
The increase in the ambient temperature T_a effects on the value of overall heat transfer coefficient increases.at 273K –1.6007 to 318k-1.6893

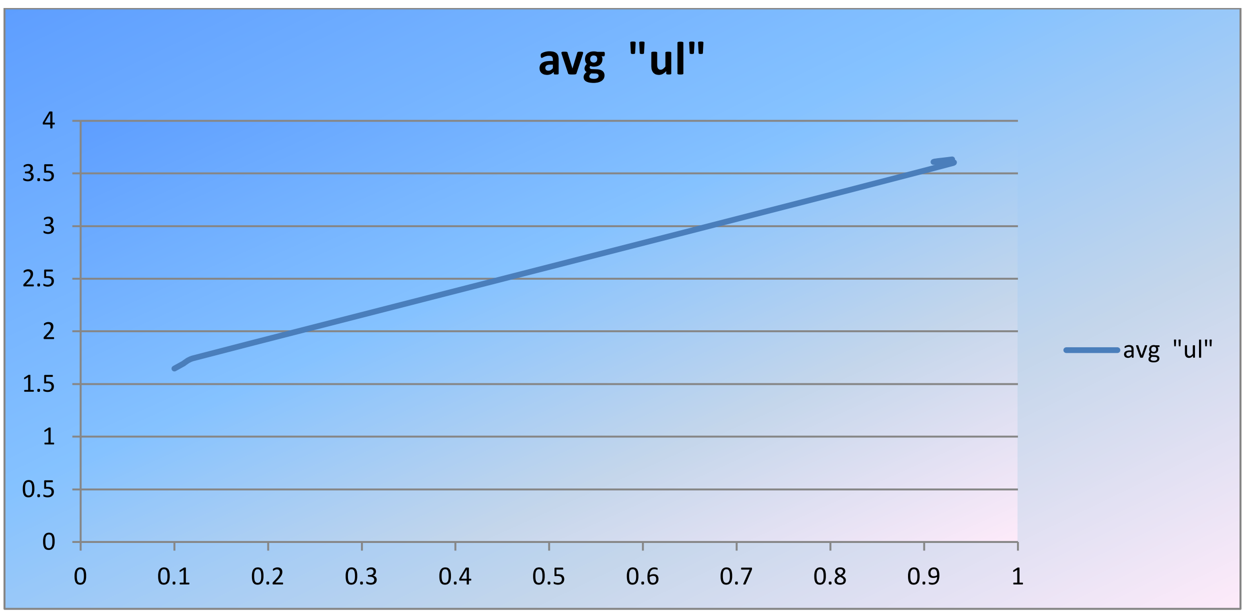
Change in overall heat transfer coefficient by changing the Emissivity for different absorber plate material.

TABLE:-7 Emissivity for different absorber plate material with changing the ambient temperature

ta	ul(ep=0.93)	ul(ep=0.91)	ul(ep=0.90)	ul(ep=0.12)	ul(ep=0.11)	ul(ep= 0.10)
273	3.4100	3.38595	3.374035	1.68743	1.6417	1.6007
278	3.4565	3.4324	3.420223	1.7006	1.6560	1.611
283	3.5035	3.479	3.467545	1.7134	1.6687	1.623
288	3.5523	3.5273	3.515821	1.7266	1.6803	1.633
293	3.6023	3.5772	3.565241	1.7376	1.6920	1.643
298	3.6532	3.6276	3.615915	1.75201	1.7023	1.6529
303	3.7056	3.6807	3.668086	1.76443	1.7143	1.6627
308	3.7590	3.7345	3.72174	1.7765	1.7245	1.672
313	3.8151	3.78965	3.77683	1.7883	1.7347	1.6809
318	3.8732	3.84687	3.834609	1.8008	1.7446	1.6893
avg "ul"	3.6331	3.6081	3.5960	1.7448	1.6959	1.6469

Graph 6:- The overall heat loss coefficient with ambient temperature for all absorber plate material **taken.**





The above graphs shows the effect of overall heat loss coefficient for different absorber plate materials.The graph shows that overall heat loss coefficient increase when we use material having emissivity(0.93-0.90) and overall heat loss coefficient decrease when we use material having emissivity(0.1-0.1).

CONCLUSION

An analytical study has been conducted to evaluate the overall heat loss coefficient of double glazed flat plate solar collector. In this evaluation of heat loss from the solar collector, a set of correlation have been used for estimation air properties. By the variation in the emissivity of absorber plate and ambient temperature, air property correlations respectively. For natural convection heat transfer between configuration correlations of Buchberg was used. The effects of variation in emissivity of absorber plate on overall heat transfer coefficient has been evaluated by correlations suggested by Buchberg. In comparison of air properties effects on over all heat loss coefficient has been studied for the taken emissivity range of absorber plate material by different coating, with the changes in ambient temperature. The present study will help to calculate overall heat loss coefficient for a solar flat plate collector with the double glazed system. In this study I found that the absorber plate material coating of "***Black crome***" with emissivity **0.10 and absobstivity 0.93** gives less heat loss and gives better performance of double glazed solar flat plate collector.

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