REVIEW OF TOP LOSS COEFFICIENT CORRELATIONS FOR FLAT PLATE COLLECTOR

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Abstract One of the principal parameter governing the efficiency of flat plate collector is its top-loss coefficient. The accurate determination of this parameter is a pre-requisite for performance evaluation of collector. There exists various approaches for determination of this parameter, varying from fundamental to empirical and analytical approach. A brief review of these approaches is presented here in.

Keywords: Top loss coefficient, flat plate collector, empirical correlations, analytical method.

INTRODUCTION

The determination of top-loss coefficient is a pre-requisite for performance evaluation of flat plate collector. This, however, involves the solution of (N+1) non-linear equations as suggested by Hottel & Woertz in 1942 [5]. The task becomes more difficult as it involves calculation of convective heat transfer coefficient from complex convective correlations as that of Hollands [2]. As an approximation to this iterative analysis, few empirical correlations and analytical methods have been proposed by various investigators which offers reasonably accurate values of top loss coefficients. Presented below is a brief review of such correlations and analytical methods.

Empirical correlations

As an approximation to iterative analysis, Hottel and Woertz [5] were the first to propose an empirical correlation. They in 1942, realised that repeatedly determining U_t & Q_t (top losses) as a function of operating parameters is a tedious task with iterative method, and hence developed an empirical correlation to enable quick calculation of U_t & Q_t as given below.

$$U_t = \left[\frac{N}{C\left(\frac{T_p - T_a}{N + f}\right)^{0.25}} + \frac{1}{h_w} \right]^{-1} + \frac{\sigma (T_p{}^2 + T_a{}^2)(T_p + T_a)}{(1/\in_p) + [(2N + f - 1)\in/_g] - N} \quad ....(1)$$

where,

U_t = Top loss coefficient, Btu/h-ft^2 -F

C = Coefficient whose value depends on collector tilt.

f = 0.76, 0.36 and 0.24 for wind speeds of 0, 10 and 20 mph.

T_a = Mean plate temp, in R,

T_a = Ambient temp, in R

$\in_p, \in_g$ = Emissivities of plate & glass, respectively.

N = No. of covers.

σ = Stefan Boltzmann constant.

h_w = Wind loss coefficient, Btu/h-ft^2 -F

Though their choice of convective and radiative heat transfer coefficients were some what arbitrary, the U_t values obtained by them were comparable to the values obtained from iterative approach. However, dependence of C on tilt angle & f on h_w was not well correlated by them. Further, the predictions of U_t with this correlations for selective absorber ($\in_p$ = 0.08 to 0.2) offers significant errors with respect to iterative method.

Klein [7] in 1973 developed a more accurate empirical relation for U_t using the values of convection coefficients and glass emissivities recommended by Tabor [16] in 1958. For a collector tilt of 45° from the horizontal, Klein suggests the relation of the same form as that of Hottel & Woertz [5] as:

$$U_t = \left[\frac{N^*}{0.165\left(\frac{T_p - T_a}{N^* + f^*}\right)^{0.31}} + \frac{1}{h_w} \right]^{-1} + \frac{\sigma (T_p - T_a)(T_p{}^2 - T_a{}^2)}{[\in_p + 0.05N^*(1 - \in_p)]^{-1} + [(2N^* + f^* - 1)/\in_g] - N^*} \quad ...(2)$$

where,

U_t = Top loss coefficient, Btu/h-ft^2 –F

N^* = 1, 1.85 and 2.65 for 1-glass, 2-glass and 3-glass, respectively.

h_w = Wind loss coefficient, Btu/h-ft^2-F,

$$f^* = 0.9556 - 0.211h_w + \left[1 + \frac{N^*-1}{N^*+3}\right]h_w^{\ 2} \quad .....(2.1)$$

T_p = Plate Temperature, R
T_a = Ambient Temperature, R.

Klein attempted to revise Hottel & Woertz correlation in two ways.

(i) He introduced N^* instead of N as effective covers
(ii) f* is presented as a function of h_w and N*.

However, the major limitation of this correlation is its predictive capability for tilt angles other than 45° and use of emissivities and convection correlations which today no longer are known to be the valid ones.

In 1975, Klein [8] modified his earlier correlation and introduced a new parameter 'C' which physically accounted for convection losses from the top of absorber plate and was a function of tilt angle. This correlation is also of the same form as that of Hottel & Woertz [5]:

$$U_t = \left[\frac{N}{\left(\frac{C}{T_p}\right)\left(\frac{T_p - T_a}{N+f}\right)^{0.33}} + \frac{1}{h_w}\right]^{-1} + \frac{\sigma(T_p + T_a)(T_p^{\ 2} + T_a^{\ 2})}{[\in_p + 0.05N(1-\in_p)]^{-1} + [(2N+f-1)/\in_g] - N} \quad ..(3)$$

where,
U_t = Top loss coefficient,W/m^2-K
T_p = Plate temperature, K,
T_a = Ambient temperature, K.
f = $(1-0.04h_w + 0.0005\ h^2_w)$ (1+0.091N) ……….…(3.1)
C = 365.9 (1-0.00883 β+ 0.0001298β^2) ……………..(3.2)
β = Collector tilt angle, Deg.
h_w = 5.7 + 3.8 V W/m^2-K…………….……..…(3.3)
V = Wind velocity , m/s.

Range of applicability of this correlation is stated to be :

$320 < T_p < 420$ K
$260 < T_a < 310$ K
$0.1 < \in_p < 0.95$
$0 \le V \le 10$ m/s.
$0 \le \beta \le 90°$

This revised equation certainly took care about tilt angle variations, but it was observed that the parameters 'C' did not predict the expected behaviour of the convective loss with tilt angle and validity was found to be restricted to 0° ≤β≤34°only.

To have better agreement of U_t values with iterated ones, Klein [3,12] proposed another equation as:

$$U_t = \left[\frac{N}{\left(\frac{C}{T_p}\right)\left(\frac{T_p - T_a}{N+f}\right)^{e}} + \frac{1}{h_w}\right]^{-1} + \frac{\sigma(T_p + T_a)(T_p^{\ 2} + T_a^{\ 2})}{[\in_p + 0.0591Nh_w]^{-1} + [(2N+f-1+0.133\in_p)/\in_g] - N} \quad ..(4)$$

U_t= Top loss coefficient of collector, W/m^2- K.
F=(1+0.089h_w-0.1166$h_w \in_p$)(1+0.07866N)……..…(4.1)
C = 520(1-0.000051β^2) ………………………..… (4.2)
for 0° < β< 70° For β> 70°, use β=70°

$$e = 0.43\left[1 - \frac{100}{T_p}\right] \quad ....(4.3)$$

h_w = wind loss coefficient
= 5.7 + 3.8 V W/m^2 K ……………….……..…..(4.4)
V = Wind velocity, m/s

This correlation predicts better results for U_t but it lacks a satisfactory agreement with iterated U_t values in following instances.

(i) The assumption that the value of C and hence U_t does not change beyond β=70° is not very sound and contradicts the experimental finding of Holland et. al. [6].
(ii) It shows agreement for highly selective ($\in_p$ =0.1) and non-selective (∈=0.95) regimes, but differ significantly in the moderately selective regimes. (∈=0.4 to 0.7)
(iii) Further, this equation assumes equivalent black body sky temperature, T_s equal to the ambient temperature.

Agarwal and Larson [1] modified the Klein's [8] correlation for a correct variation of 'C' with 'β'. For β varying from 0° to 90°, this yields the expected decrease of C and hence U_t with increase in tilt angle. Their correlation takes a form as :

$$U_t = \left[\frac{N}{\left(\frac{C}{T_p}\right)\left(\frac{T_p - T_a}{N+f}\right)^{0.33}} + \frac{1}{h_w}\right]^{-1} + \frac{\sigma(T_p + T_a)(T_p^{\ 2} + T_a^{\ 2})}{[\in_p + 0.05N(1-\in_p)]^{-1} + [(2N+f-1)/\in_g] - N} \quad ....(5)$$

where
U_t = Top loss coefficient, W/m^2-K
T_p , T_a = Plate and ambient temperatures, respectively, K
C = 250 [1-0.0044(β-90)]………… …1………(5.1)
f = (1-0.04 h_w+0.0005 h_w^2) (1+0.091N) … ……(5.2)
h_w = 5.7 + 3.8 V W/m^2-K…………....……..…..(5.3)
V = Wind velocity, m/s

This equation also assumes equivalent black body sky temperature, T_s, equal to ambient temperature.

Agarwal & Larson [1] claimed that regardless of how 'h_w' is computed, their correlation is valid up to h_w =40 W/m^2-K.

Malhotra et. al. [9] modified Klein's equation taking into account variation of heat transfer coefficient with air gap spacing, 'L' between two parallel plates and proposed an empirical relation for 'f' expressing it as a function of the number of glass covers, wind heat transfer coefficient and ambient temperature. They expressed factor 'f' as function of (1/hw) since it should level to zero as h_w tends to infinity. The correlation given by them is presented below :

$$U_t = \left[\frac{N}{(204.429/T_p).[L^3\cos\beta(T_p - T_a)/(N+f)]^{0.252}.L^{-1}} + \frac{1}{h_w}\right]^{-1} + \frac{\sigma(T_p^2 + T_a^2)(T_p + T_a)}{[\in_p + 0.0425N(1-\in_p)]^{-1} + [(2N+f-1)/\in_g] - N} \ldots\ldots\ldots\ldots(6)$$

Here U_t is in W/m^2-K and all temperatures in K and f is given by :

$$f = \left[\frac{9}{h_w} - \frac{30}{h_w^2}\right]\left[\frac{T_a}{316.9}\right][1 + 0.091N] \ldots\ldots\ldots\ldots(6.1)$$

Malhotra et. al. calculated equivalent black body sky temperature from Swinbank's relation [15]

T_s=0.0552 $T_a^{1.5}$……………...…………….……....(6.2)

Analytical methods

Empirical equations for the top loss coefficient may have acceptable accuracy in certain ranges of variables, but caution has to be exercised in their application over certain other ranges of variables. An improved technique has been proposed recently by Mullick and Samdarshi [10] for calculation of the top loss coefficients of a flat plat collector with a single glazing. The method enables the calculation of top loss coefficient by analytical equation in place of the empirical equations employed until recently. As a result, the computational errors of the approximate methods are reduced by nearly an order of magnitude. The following equation in the analytical form has been recommended for a collector with a single glazing.

$$U_t^{-1} = \left[h_{pc1} + \frac{\sigma(T_P + T_C)(T_p^2 + T_C^2)}{1/\in_p + 1/\in_g - 1}\right]^{-1} + \left[h_w + \frac{\sigma\in_g (T_1^4 - T_s^4)}{(T_1 - T_a)}\right]^{-1} + \frac{t_g}{k_g} \ldots\ldots\ldots\ldots(7)$$

Here U_t is in W/m^2-K and all temperature are in K, t_g & k_g are thickness and thermal conductivity of glass covers & T_c is cover temperature in K. The convective heat transfer coefficient may be determined from Hollands correlation [6] or by an approximate equation as follows :

$$h_{p-c} = \frac{12.75[(T_p - T_c)\cos\beta]^{0264}}{(T_p - T_c)^{0.46}.L^{0.21}} \ldots\ldots\ldots\ldots(7.1)$$

The glass temperature T_1 is obtained by an empirical relation that expresses T_1 as a function of the basic variables; absorber plate temperature, absorber plate emissivity, wind heat transfer coefficient and the ambient temperature & sky temperature as given below:

Case : 1

T_s = 0.0552 $T_a^{1.5}$ (Swinbank [15])

$$T_c = T_a + h_w^{-0.42}[0.6336\in_p - 0.6547 + \frac{T_p}{346} - 1.16\exp\{-0.072\,(T_p - T_a)\}](T_p - T_a) \ldots(7.2)$$

Case : 2

T_s = T_a

$T_c = T_a + h_w^{-0.38}[0.567\in_p - 0.403 + T_p/429](T_p - T_a)$.... 7.3

Samdarshi & Mullick [11] extended their work for double glass cover system and presented their equation as :

$$U_t = \left[h_{pc1} + \frac{\sigma(T_p^2 + T_1^2)(T_p + T_1)}{1/\in_p + 1/\in_g - 1}\right]^{-1} + \left[h_{c1c2} + \frac{\sigma(T_1^2 + T_2^2)(T_1 + T_2)}{1/\in_g + 1/\in_g - 1}\right]^{-1} + [h_w + \sigma\in_g .(T_2^2 + T_a^2)(T_2 + T_a)]^{-1} + 2t_g / k_g \ldots(8)$$

Here U_t inW/m^2-K and all temperature are in K, T_1 & T_2 are first & second glass cover temperatures and h_{pc1} & h_{c1c2} are either determined from Holland's correlation [6] or given by :

$$h_{pc1} = \frac{4.66[(T_p - T_1).\cos\beta]^{0.27}}{(T_p + T_1)^{0.31}.L^{0.21}} \ldots\ldots\ldots\ldots(8.1)$$

$$h_{c1c2} = \frac{4.66[(T_1 - T_2).\cos\beta]^{0.27}}{(T_1 + T_2)^{0.31}.L^{0.21}} \ldots\ldots\ldots\ldots(8.2)$$

They have also given empirical equations for determination of glass cover temperature T_1 and T_2 as ..

$T_1 = T_p - [0.7 - 0.34\in_p]\,[T_p - T_2]$…...…………..….(8.3)

$T_2 = T_a + h_w^{-0.4}[0.0021T_p + 0.57\in_p - 0.146][T_p - T_a]$..…...(8.4)

In this analysis Samdarshi & Mullick [11] has taken equivalent black body temperature of sky to be same as ambient temperature.

Mullick and Samdarshi [14] extended their analytical method for N glass covers. The top loss coefficient, U_t, of the flat plate collector with N glass covers can be written in terms individual heat transfer coefficients as :

$$Ut^{-1} = (h_{pc1} + h_{rp1})^{-1} + (h_{c12} + h_{r12})^{-1} + \ldots\ldots\ldots\ldots\ldots\ldots + (h_w + h_{rNa})^{-1} + Nt_g/k_g \ldots\ldots(9)$$

In this case, they believed that the temperature difference between any two adjacent parallel plates, ΔT can be assumed equal for estimation of the convective heat transfer coefficients. This assumption simplifies the analytical equation (9), as all the inter-glass convective heat transfer coefficients can be equated approximately each equal to h_{cgg}. Further, it is found that empirical estimation of the temperature of top glass cover, T_N, is quite enough to compute individual heat transfer coefficients to reasonable accuracy.

The radiative heat transfer coefficients between successive glass covers may also be taken as approximately equal provided they are evaluated at the mean temperature of the glass cover system. However, The radiative heat transfer coefficient between the absorber and the first glass cover is distinctly different. This coefficient cannot be taken equal to the glass cover as its value is strong function of the absorber emittance. The resulting simplified equation in the analytical form is :

$$\frac{1}{U_t} = \left(h_{pc1} + \frac{\sigma(T_p^{\,2} + T_1^{\,2})(T_p + T_1)}{1/\epsilon_p + 1/\epsilon_g - 1} \right)^{-1} + \left(h_{cgg} + \frac{\sigma(T_1^{\,2} + T_N^{\,2})(T_1 + T_N)}{2/\epsilon_g - 1} \right)^{-1} (h_w + \sigma\epsilon_g (T_N^{\,2} + T_a^{\,2})(T_N + T_a))^{-1} + [Nt_g / k_g] \ldots\ldots(9.1)$$

where,

$$h_{pcl} = \frac{4.66(\Delta T \cos\beta)^{0.27}}{(T_1 + T_N)^{0.31} L^{0.21}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots(9.2)$$

$$h_{pcl} = \frac{4.66(\Delta T \cos\beta)^{0.27}}{(T_p + T_1)^{0.31} L^{0.21}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots(9.3)$$

$$\Delta T = \frac{T_p - T_N}{N} = \frac{T_p - T_a}{N + f} \ldots\ldots\ldots\ldots\ldots\ldots\ldots(9.4)$$

and

$$f = [5/h_w]^{1/2} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(9.5)$$

or

$$f = (0.0004T_p + 0.48\epsilon_p + 0.5h_w^{-0.5} - 0.5) \ldots.(9.6)$$

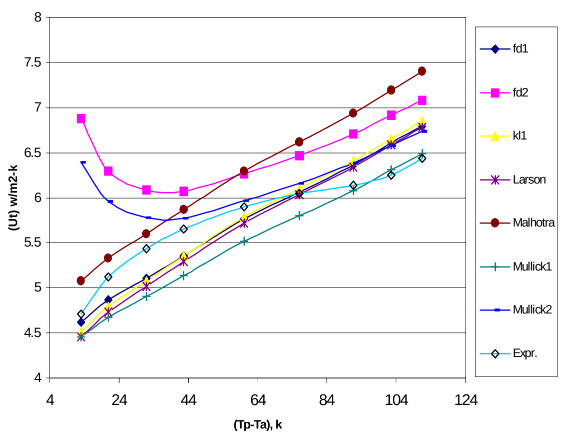
The top heat loss factors obtained by this proposed analytical equation are compared with the numerical solution of the heat balance equations. The results are within ±7 % of the numerical solution over the entire range of variables. On the other hand the errors resulting from the widely used semi-empirical correlation can be as high as ±33 %. The analytical equation (9) can accurately predict the variation of the top heat loss factor with plate temperature, plate emmisivity, number of glass covers, tilt angle, air gap spacing, ambient temperature as well as wind velocity. The analytical method has been composed in a manner that they can be simplified for single glass cover (N=1) without adversely affecting their accuracy. However, it is worth to mention over here that the assumption of equal ΔT values is not good for a collector with a selective coating on the absorber.

Comparative Assessment

An attempt is made to make the comparative assessment of existing semi-empirical and analytical method for determination of U_t with experimental results.

An experimental research collector, with single glass cover and plate size as 1.21 x 0.7 m is designed & fabricated. It is heated from the bottom by reflective heaters of 1 kW capacity to give uniform heat flux. The temperature at heater plate, absorber plate, glass cover, sides, edges & covers are measured using 48 Chromel Alumel thermo couples. A complete energy balance is carried out under different input condition and U_t is determined experimentally.

Fig. 1 : Comparative Assessment of existing Correlation with experimental values of Ut (Ep=0.9,Ec=0.88,u=0,N=1 **σ= 5.667 E-8 w/m2-K4**)



fd1=Fundamental 1 for (Ts=Ta)
fd2=Fundamental 2 for (Ts=0.0552 *Ta $^{1.5}$)
Kl-1=Klein's equation 1
Kl-2=Klein's equation 2
Larson = Agarwal and Larson's equation
Malhotra= Malhotra's equation
Mullick1= Samdarshi and Mullick's equation for (Ts=Ta)
Mullick2= Samdarshi and Mullick's equation for (Ts=0.0552 *Ta $^{1.5}$)
Exper= Experimental Results

Table :1 Comparative assessment of fundamental,semi empirical and analytical Methods for varying plate emittance (T_p=70°C,T_a=25°C, $\in_c$=0.88, u_2=3 m/s., L=22.5 x 10^{-3}m, β=0, σ=5.667 x 10^{-8} W/m^2k^4).

Sr. No	Plate Emittance	∈p=0.9	∈p=0.70	∈p=0.12
1.	***Fundamental :*** Case-1: (T_s=Ta)	6.9448	6.310	3.850
	Case-2 (Ts=0.0552 $T_a^{1.5}$)	7.408	6.8222	4.1579
2.	Klein-1 (3)	6.881	6.3056	3.8497
	Klein-2 (5)	6.743	5.7172	3.7772
3.	Agarwal & Larson (8)	6.774	6.1983	3.7424
4.	Malhotra (9)	7.282	6.6516	4.0372
5.	Mullik (11) Case-1 (Ts=Ta)	6.6145	6.01969	3.6914
	Case-2 (Ts=0.0552 $Ta^{1.5}$)	7.035	6.42237	4.0654

Fig. 1 gives this comparative assessment. The variation of U_t as a function of (T_p-T_a) is shown in this figure. It is clearly observed that all the methods do not converge to a single curve. This is probably due to use of different correlation for h_w & assumption of infinite plates for computing radiant heat exchange. It is also interesting to note that use of Swinbank [15] correlation to compute T_s results in unrealistic U_t values in lower (T_p-T_a) range. This cautions the use of Swinbank's [15] correlation for estimation of sky temperature. Further, if one looks into the literature, there is no sincere efforts made to evaluate U_t experimentally with such simulated experiments and a majority of correlations are based on numerical exercise.

A similar comparison is made to study the effect of plate emissivity [Table-1] and it is observed that there do not exist a unique value of U_t. Each method gives different value of U_t and hence it may be stated that there is a need to evolve an experimentally validated correlation for U_t.

CONCLUSIONS

There exists good amount of empirical correlations and analytical method for determination of top loss coefficient for flat plate collector. Each has its own merit and demerit. Analytical methods are considered to be superior as compared to empirical correlations. However these methods, too, involves certain degree of empirism in terms of convection correlations and are based on few assumptions which may not be true in actual collectors. It is surprising to note that very little or almost negligible literature is available on experimental determination of top loss coefficients for real life collectors. It is felt that experimental validation of existing empirical and analytical approaches is an immediate need and probably, there exists a need to develop a more logical, experimentally validated correlation for determination of top loss coefficient of collector.

NOMENCLATURE

f = ratio of thermal resistance from glass to atmosphere,
to that between absorber and glass.
h = heat transfer coefficient (W/m^2-K)
h_c = convective heat transfer coefficient (W/m^2K)
h_r = radiative heat transfer coefficient (W/m^2K)
h_w = wind heat transfer coefficient (W/m^2-K)
k = thermal conductivity of air (W/m-K)
k_g = thermal conductivity of glass (W/m-K)
L = air gap spacing (m)
N = number of glass covers
T_p = temperature of absorber plate (K)
T_a = temperature of ambient air (K)
T_s = equivalent temperature of sky (K)
t_g = thickness of glass cover (m)
U_t = top heat loss coefficient (W/m^2 -K)
$\in_p$ = emmisivity of absorber plate
$\in_g$ = emmisivity of glass cover
T_i = temperature of i^{th} cover

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