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EXISTING CORRELATIONS OF TOP HEAT LOSS FACTOR OF FLAT PLATE SOLAR COLLECTORS AND IMPACT ON USEFUL ENERGY

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ABSTRACT
The calculation of heat loss from the collector to its surroundings is required for the design or simulation of the performance of solar collectors. In the present work numerical solutions of the heat balance equations for single glazed flat plate collectors are found using computer program. The value of top heat loss factor (U_t) obtained from the numerical solutions of heat balance equations are compared to the values obtained by using the various correlations available in the literature. The percentage error in the calculation of top heat loss factor (and hence the useful energy) of a single glazed flat plate solar collector obtained by using different correlations of U_t as compared to the numerical solutions of heat balance equations are compared for the wide range of variables: 323-423 K in absorber plate temperature, 273-318 K for ambient temperature, 10-50 mm in air gap spacing, 5-45 W m^{-2} K^{-1} for wind heat transfer coefficient, 0-70º collector tilt angle and absorber plate emittance 0.1- 0.95. The study shows that the empirical relations of U_t predict the values close to numerical solutions only for certain assumed conditions and cause large errors in the calculation of top heat loss factor and useful energy for other range of variables.

Key words: Flat plate solar collectors, Top heat loss factor, Useful energy

1. INTRODUCTION
The useful energy of the collector is defined as the actual amount of energy that is available after meeting the heat losses. The calculation of heat loss from the collector to its surroundings is required for the design or simulation of the performance of solar collectors. The top heat loss coefficient, U_t determines the energy lost from the absorber to ambient by a combined process of convection and radiation between the absorber plate and the glass cover along with conduction across the glass cover thickness followed by forced convection due to wind and radiation losses between the outer glass and the surroundings.

2. HEAT BALANCE EQUATIONS
Under steady - state conditions upward heat loss from the absorber plate to the inner glass cover is given by

$$Q^{//}_{p-gi} = (h_{cpg} + h_{rpg})\,(T_p - T_{gi}) \tag{1}$$

and across the glass cover thickness is given by

$$Q^{//}_{gi-go} = k_g\,(T_{gi} - T_{go})/L_g \tag{2}$$

Heat transfer from the outer glass cover to the surroundings is given by

$$Q^{//}_{go-a} = (h_{rga} + h_w)\,(T_{go} - T_a) \tag{3}$$

Heat transfer coefficients
The radiative heat transfer coefficient between the absorber plate and the inner glass cover (h_{rpg}) is:

$$h_{rpg} = \frac{\sigma\left(T_P^{\,2} + T_{gi}^{\,2}\right)\left(T_P + T_{gi}\right)}{\dfrac{1}{\varepsilon_P} + \dfrac{1}{\varepsilon_g} - 1} \tag{4}$$

The radiative heat transfer coefficient between the outer cover and the ambient air (h_{rga}) is:

$$h_{rga} = \sigma\varepsilon_g \left(T_{go}^{\,2} + T_a^{\,2}\right)\left(T_{go} + T_a\right) \quad (5)$$

The radiative heat transfer coefficient (considering sky temperature less than ambient for outer radiation) between the outer glass cover and the atmosphere is given by

$$h_{rgs} = \sigma\varepsilon_g \left(T_{go}^{\,2} + T_s^{\,2}\right)\left(T_{go} + T_s\right) \quad (6)$$

The radiative heat transfer coefficient (considering sky temperature) with reference to ambient temperature is written as

$$h_{rgs-a} = \frac{\sigma\varepsilon_g \left(T_{go}^{\,2} + T_s^{\,2}\right)\left(T_{go} + T_s\right)\left(T_{go} - T_s\right)}{T_{go} - T_a} \quad (7)$$

The convective heat transfer coefficient, h_{cpg}, can be calculated by using the following correlation for Nusselt number proposed by Buchberg et al. [1]:

$Nu=1+1.446\left(1-\frac{1708}{Ra\cos\beta}\right)$ for $1708 < Ra\cos\beta < 5900$

$Nu=0.229(Ra\cos\beta)^{0.252}$ for $5900 < Ra\cos\beta < 9.23\times10^4$

$Nu=0.157(Ra\cos\beta)^{0.285}$ for $9.23\times10^4 < Ra\cos\beta < 10^6$ (8)

3. APPROXIMATE METHODS FOR CALCULATION OF U_t

There are two approaches for the calculation of U_t: (i) by numerical solutions of the above heat balance equations, using computer program and (ii) approximate method available in the literature [2-4].

The empirical relation for calculating U_t proposed by Klein [2]) is as follows:

$$U_t = \left[\frac{N}{\left(\frac{C}{T_P}\right)\left\{\frac{T_P - T_a}{N+f}\right\}^{e}} + \frac{1}{h_w}\right]^{-1} + \left[\frac{\sigma\left(T_P^{\,2} + T_a^{\,2}\right)\left(T_P + T_a\right)}{\left(\varepsilon_P + 0.0591Nh_w\right)^{-1} + \left\{\frac{\left(2N + f - 1 + 0.133\varepsilon_P\right)}{\varepsilon_g}\right\} - N}\right] \quad (9)$$

where

$$f = \left(1 + 0.089\,h_w - 0.116 h_w \varepsilon_p\right)\left(1 + 0.07866\,N\right)$$

$$e = 0.43\left(1 - \frac{100}{T_P}\right)$$

$C = 520\left(1 - 0.000051\,\beta^2\right)$ *for* $0 < \beta < 70^{o}$.

For $70 < \beta < 90$, *use* $\beta = 70°$.

Agarwal and Larsen [3] proposed the following empirical relation for U_t:

$$U_t = \left[\frac{N}{\left(\frac{C}{T_P}\right)\left\{\left(\frac{T_P - T_a}{N+f}\right)\right\}^{0.33}} + \frac{1}{h_w}\right]^{-1} + \left[\frac{\sigma\left(T_P^{\,2} + T_a^{\,2}\right)\left(T_P + T_a\right)}{\left(\varepsilon_P + 0.05N\left(1-\varepsilon_P\right)\right)^{-1} + \left\{\frac{\left(2N + f - 1\right)}{\varepsilon_g}\right\} - N}\right] \quad (10)$$

where

$$f = \left(1 - 0.04\,h_w + 0.0005\,h_w^{\,2}\right)\left(1 + 0.091\,N\right)$$

$$C = 250\left[1 - 0.0044\left(\beta - 90\right)\right]$$

Equations (9) and (10) assume equivalent black body sky temperature, T_s equal to ambient temperature, T_a.

Malhotra et al. [4] proposed the following semi empirical correlation for U_t taking into account the variation of air gap spacing, L between two parallel plates and proposed the following empirical relation for U_t:

$$U_t = \left[\frac{N}{\left(\frac{204.43}{T_P}\right)\left\{L^3\cos\beta\left(\frac{\left(T_P - T_a\right)}{\left(N+f\right)}\right)\right\}^{0.252} L^{-1}} + \frac{1}{h_w}\right]^{-1} + \left[\frac{\sigma\left(T_P^{\,2} + T_a^{\,2}\right)\left(T_P + T_a\right)}{\left(\varepsilon_P + 0.0425\left(1-\varepsilon_P\right)\right)^{-1} + \left\{\frac{\left(2N + f - 1\right)}{\varepsilon_g}\right\} - N}\right] \quad (11)$$

where

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

Equation (11) has been formulated assuming sky temperature lower than ambient temperature ($T_s < T_a$) from the relation proposed by Swinbank [5]: $T_s = 0.0552\,T_a^{1.5}$.

Mullick and Samdarshi [6] proposed an analytical equation for predicting U_t of flat plate solar collector with single glazing by introducing empirical relations for the calculation of glass cover temperature (T_g) to predict the convective and radiative heat transfer coefficients. Following analytical equation for U_t was proposed:

$$U_t = \left[\frac{1}{h_{cpg} + \dfrac{\sigma\left(T_P^2 + T_g^2\right)\left(T_P + T_g\right)}{\dfrac{1}{\varepsilon_P} + \dfrac{1}{\varepsilon_g} - 1}} + \frac{1}{h_w + \dfrac{\sigma\varepsilon_g\left(T_P^4 - T_s^4\right)}{(T_P - T_a)}} + \frac{L_g}{k_g} \right]^{-1} \quad (12)$$

For the calculation of glass cover temperature, T_g, following relations were reported.

Case:1 In case the sky temperature is assumed equal to ambient temperature, $T_s = T_a$

$$T_g = T_a + h_w^{-0.38}\,[\,0.567\,\varepsilon_P - 0.403 + \frac{T_P}{429}\,]\left(T_p - T_a\right) \quad (13)$$

Case: 2 In case the sky temperature is assumed lower than ambient temperature, $T_s = 0.0552\ T_a^{1.5}$ (Swinbank's relation)

$$T_g = T_a + h_w^{-0.42}[\,0.6336\varepsilon_P - 0.6547 + \frac{T_P}{346} - 1.16\exp\left\{-0.072\left(T_P - T_a\right)\right\}]\left(T_P - T_a\right) \quad (14)$$

The analytical equation substantially lowered the errors in computation of U_t in comparison to empirical relations. However, the relations proposed for glass temperatures are approximate.

Akhtar and Mullick [7] proposed the following procedure for calculating the value of top heat loss coefficient, U_t for the two cases:

In case the sky temperature is assumed equal to ambient temperature, $T_s = T_a$

$$\frac{1}{U_t} = \left[\frac{\sigma(T_p^{\,2} + T_g^{\,2})\left(T_p + T_g\right)}{\dfrac{1}{\varepsilon_p} + \dfrac{1}{\varepsilon_g} - 1} + \frac{kNu}{L} \right]^{-1} + \left[\sigma\varepsilon_g\left(T_g^{\,2} + T_a^{\,2}\right)\left(T_g + T_a\right) + h_w\right]^{-1} + \frac{L_g}{k_g} \quad (15)$$

The glass cover temperature is calculated as

$$T_g = \frac{\left(fT_p + T_a\right)}{\left(1 + f\right)}$$

where f is the ratio of inner to outer thermal resistance and given by:

$$f = \frac{\left[12\times10^{-8}\left(T_a + 0.2T_P\right)^3 + h_w\right]^{-1} + 0.3L_g}{\left[6\times10^{-8}\left(\varepsilon_P + 0.028\right)\left(T_P + 0.5T_a\right)^3 + 0.6L^{-0.2}\left\{\left(T_P - T_a\right)\cos\beta\right\}^{0.25}\right]^{-1}}$$

In case the sky temperature is assumed lower than ambient temperature, $T_s = 0.0552\ T_a^{1.5}$ (Swinbank's relation)

$$\frac{1}{U_t} = \left[\frac{\sigma(T_p^{\,2} + T_g^{\,2})\left(T_p + T_g\right)}{\dfrac{1}{\varepsilon_p} + \dfrac{1}{\varepsilon_g} - 1} + \frac{kNu}{L} \right]^{-1} + \left[\frac{\sigma\varepsilon_g\left(T_g^{\,4} - T_s^{\,4}\right)}{T_g - T_a} + h_w \right]^{-1} + \frac{L_g}{k_g} \quad (16)$$

The glass cover temperature is obtained as follows:

$$T_g = \frac{\left(fT_p + cT_a\right)}{\left(1 + f\right)}$$

where

$$c = \frac{\left(\dfrac{T_s}{T_a} + \dfrac{h_w}{3.5}\right)}{\left(1 + \dfrac{h_w}{3.5}\right)}$$

4. COMPARISON OF RESULTS WITH NUMERICAL SOLUTIONS

The values of top heat loss coefficient of a single glazed flat plate solar collector obtained from the numerical solution are compared with those obtained from approximate relations discussed above for the two cases i.e. sky temperature equal to ambient temperature for the outer radiation (i.e. $T_s = T_a$) and sky temperature less than ambient temperature i.e. sky temperature obtained from the Swinbank's relation [5]. The range of variables considered in the analysis is 323-423 K in absorber plate temperature, 273-318 K for ambient temperature, 10-50 mm in air gap spacing, 5-45 W

m^{-2} K^{-1} for wind heat transfer coefficient, 0-70° collector tilt angle and absorber plate emittance 0.1-0.95. Although the wind heat transfer coefficient, h_w, is a function of wind velocity, in the present analysis h_w has been taken as a variable and the range is considered as reported in the literature [2, 4]. A comparison of the values of U_t is made in Figs. 1 and 2, respectively, for the cases when $T_s = T_a$ and when $T_s < T_a$.

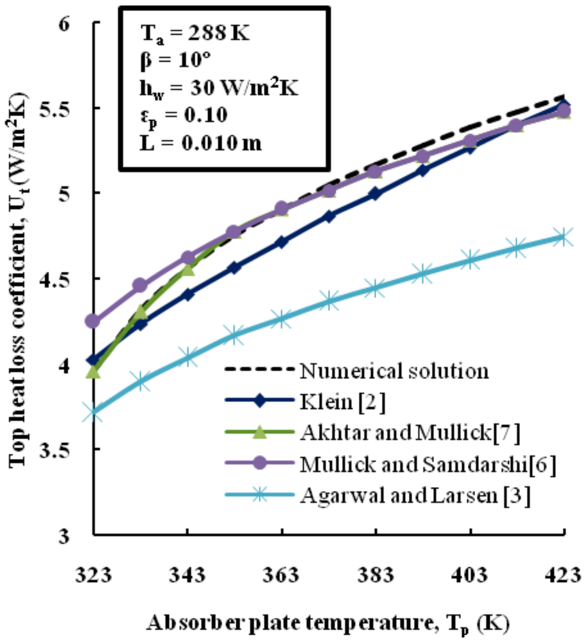

Fig. 1 Variation in U_t with absorber plate temperature (when T_s=T_a)

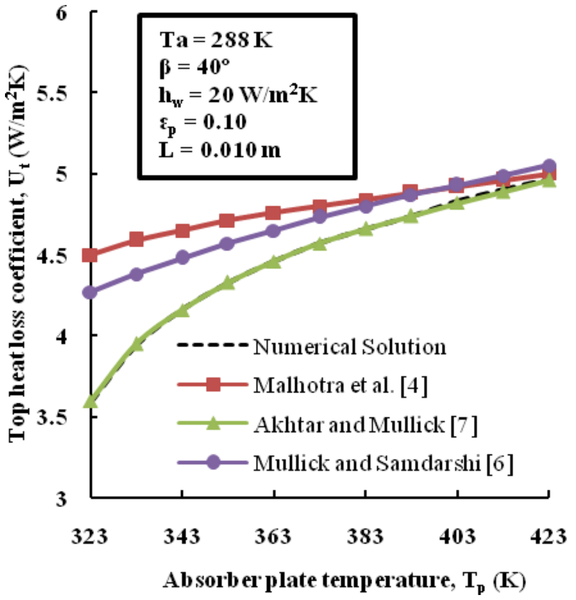

Fig. 2 Variation in U_t with absorber plate temperature (when T_s<T_a)

It can be seen from Fig. 1 that at lower values of absorber plate temperatures, Eqs. (10) and (12) proposed by Agarwal and Larsen [3] and Mullick and Samdarshi [6] show a large deviation from the numerical solutions. At higher absorber plate temperatures the values of top heat loss coefficient obtained from Eq. (12) are close to those obtained from the numerical solution. The values obtained by using Eq. (10) show a large deviation for the entire range of absorber plate temperature. The values obtained from Eq. (9) proposed by Klein [2] are close to the values obtained from numerical solution. The values of top heat loss coefficient obtained from Eq. (15) proposed by Akhtar and Mullick [7] are so close to the values obtained from numerical solutions that they are over lapping.

It can be seen from Fig. 2 that at low absorber plate temperatures the values of top heat loss obtained from Eqs. (11) and (12) proposed by Malhotra et al. [4] and Mullick and Samdarshi [6], respectively, show a significant deviation as compared to the numerical solution. At higher absorber plate temperature the values are in good agreement with the numerical solution. The values of top heat loss coefficient obtained from Eq. (16) proposed by Akhtar and Mullick [7] are so close to the values obtained from numerical solutions that they are over lapping.

A comparison of the percentage error in the values of U_t obtained by using the approximate relations taking the values obtained by numerical solutions of the heat balance equations as the basis is made in Fig. 3.

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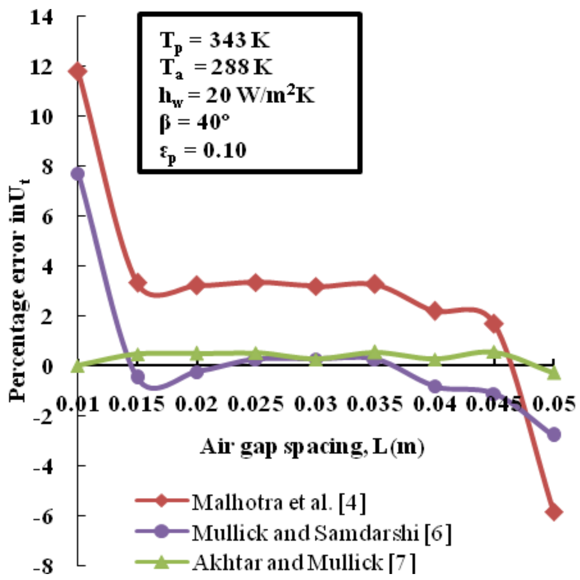

Fig. 3 Variation of percentage error in U_t with air gap spacing when (T_s= 0.0552$T_a^{1.5}$)

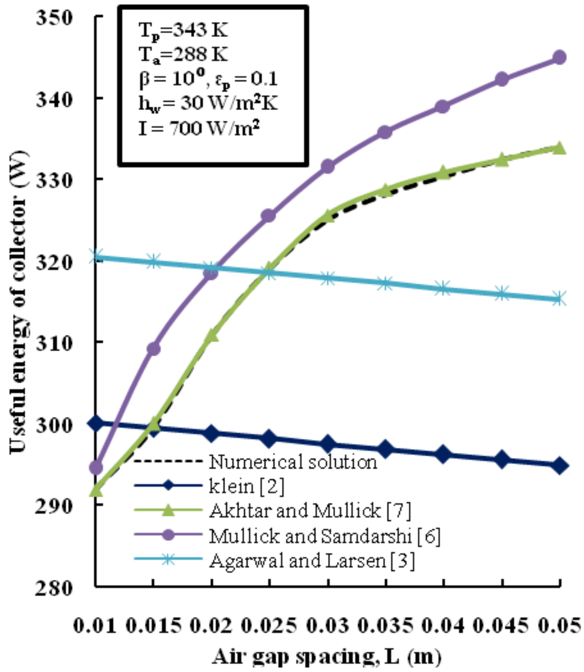

Fig. 4 Variation of useful energy of collector with air gap spacing (when T_s= T_a)

It can be seen from this figure that at low air gap spacing the percentage error obtained from Eqs. (11) and (12) proposed by Malhotra et al. [4] and Mullick and Samdarshi [6] are about 12 percent and 8 percent, respectively, whereas at large air gap spacing the percentage error reduces to -6 percent and -2 percent, respectively. The percentage error obtained from Eq. (16) proposed by Akhtar and Mullick [7] is within ± 0.2 percent.

The useful energy of the collector is the actual amount of energy that is available from the collector, after meeting the heat losses. For the evaluation of useful energy the size of the solar collector is considered 2.1 m × 1.0 m, the thickness of glass cover as 4 mm, thickness of the side and bottom insulation as 50 mm and 25 mm, respectively. The thermal conductivity of the insulation (glass wool) is taken as 0.04 W/m-K, the intensity of solar radiation is kept 700 W/m^2 and the optical efficiency of collector is as 0.85. The variation of useful energy calculated by using the different relations of U_t is shown in the Fig. 4.

It can be seen from this figure that at small air gap spacing the useful energy obtained from Eqs. (9) and (12) proposed by Klein [2] and Mullick and Samdarshi [6] show a very small deviation from the numerical solution, whereas, Eq.(10) proposed by Agarwal and Larsen [3] shows a large deviation as compared to the numerical solution. At L = 15 mm the value of useful energy obtained from Eq. (9) is same as that obtained from the numerical solution, and at L = 25 mm the value of useful energy obtained from Eq. (10) do not show any deviation from the numerical solution. The values of useful energy obtained by using Eq. (15) proposed by Akhtar and Mullick [7] are so close to the values obtained from numerical solutions that they are over lapping for the entire range.

The percentage error in the calculation of useful energy of the collector with air gap spacing is shown in Fig. 5. The figure shows that the percentage error obtained in useful energy by using the approximate relations are large except by the relation proposed by Akhtar and Mullick [7].

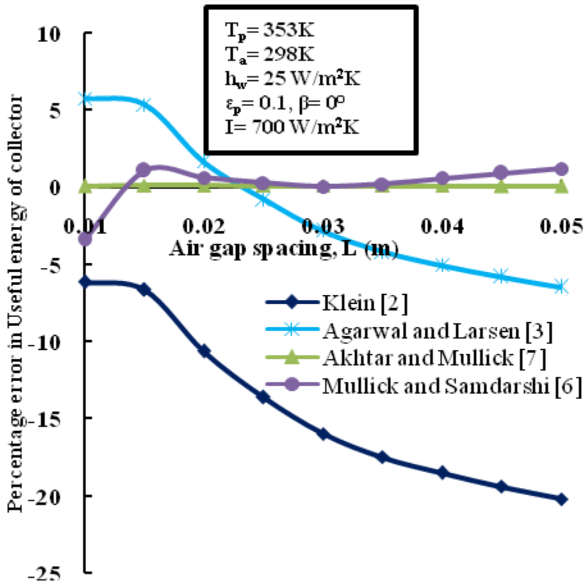

Fig. 5 Percentage error in useful energy with air gap spacing at T_s = T_a

5. CONCLUSIONS

1. The empirical relations [2-4] cause large errors in the calculation of U_t due to incorrect regrouping of convective and radiative terms.
2. The empirical relation causes large error in the calculation of useful energy available from the solar collectors. The study shows that the empirical relations of U_t predict the values correctly only for certain assumed conditions and cause large errors in the calculation of U_t and useful energy for other range of variables.
3. The analytical method proposed by Mullick and Samdarshi [6] substantially lowered the errors in computation of U_t in comparison to the empirical relations. However, the relations used for glass cover temperatures were approximate. The maximum error obtained in U_t is 8 percent. The values of glass cover temperature obtained were found within 10 degree as compared to the numerical solutions of heat balance equations.
4. The improved equation form for computing the glass cover temperature of flat plate collector proposed by Akhtar and Mullick [6] computes the glass cover temperature accurately which is within 2 degrees as compared to the values obtained from numerical solutions of heat balance equations. The computational error in U_t as compared to the numerical solution is within 1 percent for the entire range of variables. The useful energy available from the collector can also be evaluated correctly by using the relation of U_t proposed by Akhtar and Mullick [6].

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NOMENCLATURE

f	ratio of outer to inner thermal resistance
h_c	convective heat transfer coefficient, W/m² °C
h_r	radiative heat transfer coefficient, W/m² °C
h_w	wind heat transfer coefficient, W/m² °C
k_g	thermal conductivity of the glass plate, W/m-K
L	air gap spacing, m
L_g	thickness of the glass plate, m
N	number of glass covers
Nu	Nusselt number
Q''_{gi-go}	heat transfer in the glass cover, W/m²
Q''_{go-ga}	heat transfer from the glass cover to the surroundings, W/m²
Q''_{p-gi}	heat transfer between the absorber plate and glass cover, W/m²
Ra	Raleigh number
T	temperature, K
U_t	top heat loss coefficient, W/m²K

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Subscripts

a ambient air
g glass cover
i inner surface of glass cover
o outer-surface of glass cover
p absorber plate
pg absorber plate to glass cover
ga glass cover to ambient air

Greek Letters

ε_g emissivity of the glass plate
ε_p emissivity of the absorber plate
β collector tilt angle from horizontal, (deg)
β' coefficient of volume expansion, K^{-1}
ν kinematic viscosity, m^2/s
σ Stefan-Boltzman constant, (W/m^2K^4)

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