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Glass cover temperature and top heat loss coefficient of a single glazed flat plate collector with nearly vertical configuration

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Abstract 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 $\mathrm{W/m^2\,K}$ for wind heat transfer coefficient and −10 °C to 40 °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.

1. Introduction

The integration of solar thermal systems into façades of buildings to meet energy requirements of buildings (for domestic hot water, space heating, air-conditioning and lighting) is supported by many researchers in different parts of world [1–6]. It has been quoted [1] that in most European countries, buildings account for approximately 40% of the total energy use. The concept of solar buildings (solar heated and cooled and PV powered), to meet their energy requirement, is gaining momentum. Design of solar buildings requires integration of solar thermal systems, PV panels into roof or walls [1]. Krauter et al. [2] have mentioned that the application of solar energy technology to buildings often depends on its ability to be integrated into common building structures, such as façade elements. Façade-integrated photovoltaic thermal collector

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Nomenclature

c	empirical factor	Pr	Prandtl number
e	empirical factor	r	ratio of outer to total resistance
e_p	emittance of absorber plate	Ra_L	Rayleigh number $g\beta'(T_p - T_g)L^3\,\mathrm{Pr}/v^2$
f	empirical factor	$\dot{Q}''_t$	top heat loss flux density (W m^{-2})
h_{cpg}	convective heat transfer coefficient between absorber plate and glass cover (W m^{-2} K^{-1})	T_a	ambient temperature (K)
		T_g	average temperature of glass cover (K)
h_{rga}	radiative heat transfer coefficient between glass cover and ambient (W m^{-2} K^{-1})	T_p	average temperature of absorber plate (K)
		U_t	top heat loss coefficient (W m^{-2} K^{-1})
h_w	wind heat transfer coefficient (W m^{-2} K^{-1})	β	collector tilt angle with respect to horizontal (°)
k	thermal conductivity of air (W m^{-1} K^{-1})	β'	volumetric coefficient of expansion (per K)
k_g	thermal conductivity of glass cover (W m^{-1} K^{-1})	σ	Stefan–Boltzman constant (W m^{-2} K^{-4})
L	air gap spacing between absorber plate and glass cover (m)	ε_g	emittance of glass cover
		ε_p	emittance of coating of absorber plate
L_g	thickness of glass cover (m)	v	kinematic viscosity of air (m^2 s^{-1})
Nu	Nusselt number (h_cL/k)		

has the potential to become one of the most desirable methods for electricity generation and water heating [3]. Flat plate collectors with single glazing are mostly used in building-integrated systems.

Matuska and Sourek [4] have proposed the utilization of solar energy for domestic hot water heating by installing façade flat plate solar collectors in many flats (apartments) of Czech Republic (established between 1950s and 1970s, which were ready for major renovations). Tripanagnostopoulos et al. [5] have mentioned that the integration of flat plate solar collectors in buildings should be compatible with the architectural design, while solar collectors with colored absorbers would be aesthetically preferable. The selective colored absorbers could be more effective for improving the thermal performance of flat plate collectors in a wide range of operating temperatures than the absorbers with color paints of high emissivity [5]. Zhai and Wang [6] have quoted that the current use of energy in buildings accounts for approximately 25% of total energy consumption in China, and mainly consists of domestic hot water, heating and air-conditioning systems. The government of China has been planning big in the five year plans encouraging solar energy research for the purpose of developing key technologies involved in the integration of solar thermal systems with buildings [6]. China has been pursuing plans of putting into millions of square meter of solar water collectors and integration of solar collector modules into buildings [6].

Top heat loss coefficient is required for evaluating thermal performance of solar collectors. A correct value of U_t is also important for design, simulation of heat losses or thermal performance evaluation of flat plate collectors with vertical configuration. These are used at high latitudes and are integrated with building walls. Top heat loss coefficient, U_t, has to be computed for various values of different variables like emittance of absorber coating (e_p), absorber plate temperature (T_p), ambient temperature (T_a), wind heat transfer coefficient (h_w), air gap spacing between absorber plate and glass cover (L) and angle of inclination of collector (β). Top heat loss coefficient of a flat plate collector can be computed by numerical solution of heat balance equations or approximately by empirical equations [7–12].

The most popular approximate method for calculation of U_t, using Klein's equation quoted by Duffie and Beckman [9] is:

$$U_t = \left[\frac{N}{\frac{C}{T_p}\left[\frac{T_p - T_a}{N+f}\right]^e} + \frac{1}{h_w}\right]^{-1} + \frac{\sigma(T_p^2 + T_a^2)(T_p + T_a)}{(\varepsilon_p + 0.00591Nh_w)^{-1} + \left[\frac{2N+f-1+0.133\varepsilon_p}{\varepsilon_g}\right] - N} \quad (1)$$

where f is the $(1 + 0.089h_w - 0.1166h_w\varepsilon_p)(1 + 0.07866N)$, e the $0.43(1 - 100/T_p)$, and C is the $520(1 - 0.0051\beta^2)$ for $0° < \beta < 70°$. For $70° < \beta < 90°$, use $\beta = 70°$.

Approximate method for calculation of U_t of flat plate collector has been widely used since seventies [13]. It has been analyzed [13,14] that empirical equations [9,11] resulted into large errors in U_t because the equations were derived by regrouping and approximating the convective and radiative terms since the glass cover temperature, T_g, is unknown. An improved technique for predicting U_t of flat plate collector with single glazing was proposed [13]. Analytical equation was used for calculation of U_t with an empirical relation for T_g [13]. Mullick and Samdarshi [13] proposed the use of the following analytical equation for U_t for a flat plate collector with single glazing:

$$U_t^{-1} = [h_{c_{pg}} + h_{r_{pg}}]^{-1} + [h_w + h_{r_{ga}}]^{-1} + L_g/k_g \quad (2)$$

The Eq. (2) can be written as

$$U_t^{-1} = \left[h_{c_{pg}} + \frac{\sigma(T_p^2 + T_g^2)(T_p + T_g)}{1/\varepsilon_p + 1/\varepsilon_g - 1}\right]^{-1} + \left[h_w + \frac{\sigma\varepsilon_g(T_g^2 + T_a^2)(T_g + T_a)}{T_g - T_a}\right]^{-1} + L_g/k_g \quad (3)$$

In the above analytical equation sky temperature is taken equal to ambient temperature (as in Eq. (1)). The thermal resistance of glass cover is a small fraction of total thermal resistance to upward heat flow and can be approximated. The wind heat transfer coefficient has been considered as independent variable. For estimation of glass cover temperature following empirical equation has been suggested [13]:

$$T_g = T_a + h_w^{-0.38}[0.567\varepsilon_p - 0.403 + T_p/429](T_p - T_a) \quad (4)$$

It was reported [13] that analytical Eq. (3) gives results (values of U_t) within three percent to numerical solutions for all possible combination of variables whereas the computational level of errors in U_t resulting from use of approximate method; Eq. (1), is very high (of about 30%).

The convective heat transfer coefficient between absorber plate and glass cover, $h_{c_{pg}}$, in Eq. (3) can be estimated from correlations for Nusselt number suggested by Hollands et al. [15] and Buchberg et al. [16]. Hollands et al. [15] summarized that their correlation predicts convective heat transfer coefficient with a maximum error of 5% for angle of inclination of collector between 0° and 60° and with an error of 10% for angle of inclination from 60° to 75°. Buchberg et al. [16] have proposed three region and two region correlations for estimation of $h_{c_{pg}}$ but those correlations are valid for value of β from 0° to 60°. Therefore, analytical equation for computation of U_t [13] can be used for value of β between 0° and 60° and cannot be used for single glazed flat plate collectors with vertical configuration. Badescu [17] has proposed equations for calculation of top heat loss coefficient of a single glazed flat collector for near vertical configuration taking into account weather conditions of Mars. Approximate method; Eq. (1), is still used for estimation of U_t of single glazed flat collector with vertical configuration/performance evaluation of thermal systems as evident from literature [18–20]. Eq. (3) can be employed for computation of U_t of single glazed FPC with vertical configuration if $h_{c_{pg}}$ and T_g are known. The convective heat transfer between absorber plate and glass cover can be calculated from the correlations for Nusselt number suggested by Elsherbiny et al. [21] for enclosed cavities for value of β from 60° to 90°.

In the present work empirical relation for glass cover temperature of a single glazed flat plate collector with vertical configuration has been developed following the procedure given by Mullick and Samdarshi [13]. Top heat loss coefficient is calculated by analytical Eq. (3).

2. Empirical relation for temperature of glass cover

Values of individual heat transfer coefficients vary relatively small due to variations in glass cover temperature of single glazed FPC. Therefore, Eq. (3) can predict values of U_t of single glazed flat plate collector with vertical configuration with a reasonable degree of accuracy with approximate values of T_g. An approximate value of glass cover temperature has been obtained by empirical relation. Assuming one dimensional heat flow, neglecting internal energy and temperature drop across glass cover, the ratio of outer to total resistance can be written as

$$\frac{T_g - T_a}{T_p - T_a} = \frac{R_{ga}}{R_{pg} + R_{ga}} = r \quad (5)$$

where R_{pg} is the thermal resistance between plate and glass cover and R_{ga} is the thermal resistance between glass cover and surroundings.

Then from Eq. (5), T_g can be calculated as

$$T_g = T_a + r(T_p - T_a) \quad (6)$$

Since the factor 'r' is mainly a function of individual heat transfer coefficients, an empirical relation for factor 'r' can be expressed as a function of basic variables the plate temperature T_p, emittance e_p and wind heat transfer coefficient h_w. The empirical relation for 'r' is obtained through non linear regression analysis noting that 'r' tends to zero as h_w tends to infinity.

The following relation for 'r' is obtained:

$$r = h_w^{-0.42}\left(0.621\varepsilon_p + \frac{T_p}{505} - 0.27\right) \quad (7)$$

Then from Eq. (6), T_g can be calculated as

$$T_g = T_a + h_w^{-0.42}\left(0.621\varepsilon_p + \frac{T_p}{505} - 0.27\right)(T_p - T_a) \quad (8)$$

The range of variables covered in the present work is 20–150 °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 angle, 5–30 W/m^2 K for wind heat transfer coefficient and −10 °C to +40 °C for ambient temperature. Minimum value of temperature difference between absorber and ambient temperature ($T_p - T_a$) is assumed to be equal to 20° for analysis. It has been verified that the maximum absolute error obtained in values of temperature of glass cover calculated from Eq. (8), in comparison to results of numerical solution of heat balance Eqs. (A.1) and (A.2), is within 9° over entire range of variables covered in the present analysis. The values of U_t computed by Eq. (3) with values of T_g obtained from Eq. (8) are compared with values of U_t obtained by Eq. (3) with T_g obtained from numerical solution of heat balance Eqs. (A.1) and (A.2) over entire range of variables covered in the present work. The maximum absolute error in computed values of U_t of single glazed flat-plate collector with vertical configuration has been found to be less than 2% in comparison to values of U_t obtained by iterative solution of heat balance equations.

3. Results and discussions

The semi-empirical Eq. (1) predicts values of U_t with reasonable accuracy, in comparison to numerical solution, for certain ranges of variables for which it has been developed and it is more inaccurate for other ranges of variables. This statement is supported by the results discussed in subsequent paragraphs.

The values of U_t obtained by numerical solution of heat balance equations are taken base values and percentage errors in computation of U_t by use of semi-empirical Eq. (1) and the analytical Eq. (3), with T_g from Eq. (8), are calculated. The results have been shown in Figs. 1–6. Fig. 1 shows the varia-

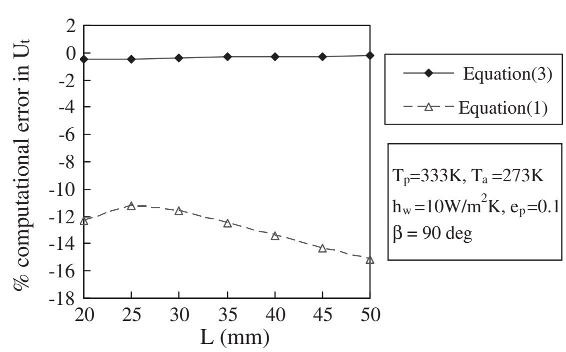

Figure 1 Percentage error in U_t with air gap spacing.

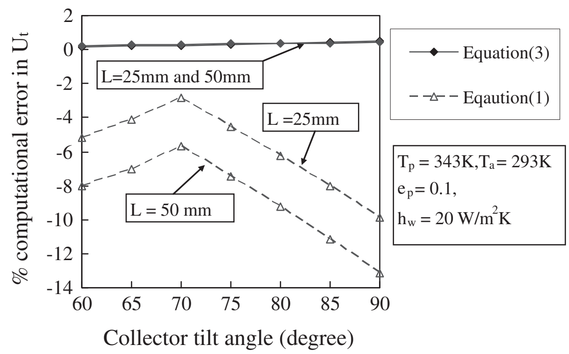

Figure 2 Percentage computational error in U_t for different values of collector tilt angle.

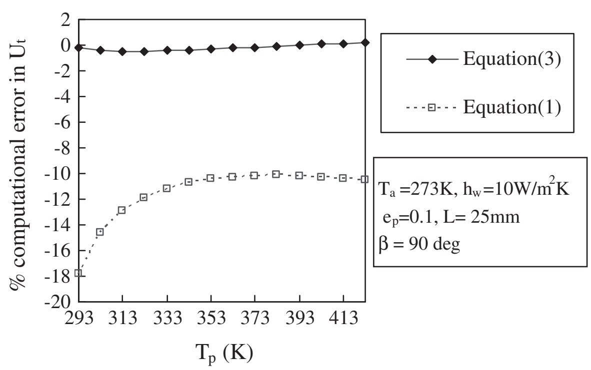

Figure 3 Percentage computational error in U_t at different absorber plate temperatures.

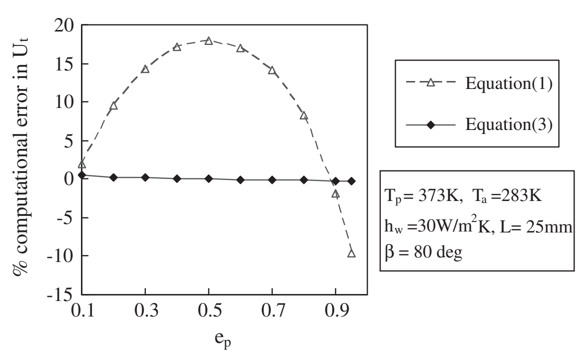

Figure 4 Percentage computational error in U_t with emittance of absorber plate.

tion of percentage computational error in U_t with air gap spacing. It can be seen from Fig. 1 that using the analytical Eq. (3) the error is less than 0.5%, whereas, by using the empirical Eq. (1) the minimum error is 11% for $L = 25$ mm and the error increases for values of L greater than 25 mm. The reason for the increase in computational error in U_t can be stated that the approximate method does not provide/estimate T_g. So the values of h_{cpg} cannot be estimated for any given combination of variables. The analytical equation includes T_g explicitly and h_{cpg} is calculated for any value of L by using appropriate correlation for h_{cpg}. Fig. 1 illustrates that by using the analytical equation the error does not increase even though the value of L increases from 25 mm to 50 mm. The variations of percentage computational error in U_t with collector tilt angle

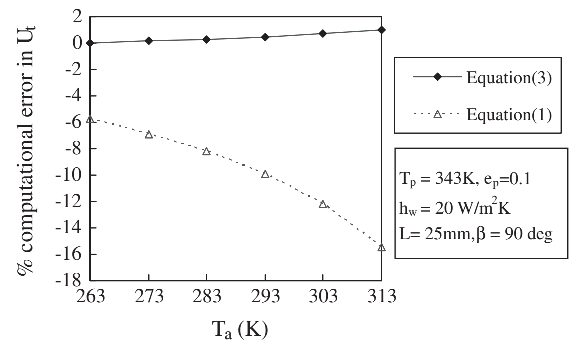

Figure 5 Percentage error in U_t at different ambient temperatures.

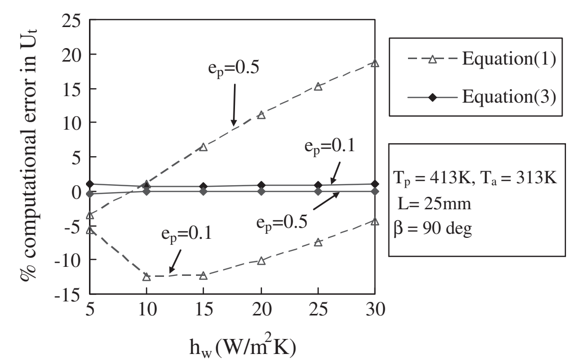

Figure 6 Percentage computational error in U_t for different values of wind heat transfer coefficient.

are shown in Fig. 2. It is noted that computational error in U_t by the use of Eq. (3) is negligible (within 0.5%) in comparison to errors obtained from using empirical Eq. (1). The Eq. (1) does not consider the effect of angle of tilt on U_t for values of β between 70° and 90°. The error by use of Eq. (1) increases for value of tilt angle greater than 70° and the error is maximum at $\beta = 90°$.

Fig. 3 shows the variation of percentage computational error in U_t with absorber plate temperature. It can be seen from this figure that using the analytical Eq. (3) the error is less than 0.5%, whereas, by using the empirical Eq. (1) the error decreases with increase in plate temperature. At low plate temperatures the error increases up to 18%. The variation of error in computation of U_t with plate emittance is shown in Fig. 4. It is evident from this figure that using Eq. (3) the error in computation of U_t is negligible (less than 0.6%) while using Eq. (1) the error is substantial. Eq. (1) gives negligible error when $e_p = 0.1$ and $e_p = 0.9$ whereas in the middle range of plate emittance ($0.4 \leqslant e_p \leqslant 0.6$) the error is about 17%. Fig. 5 shows the variation of percentage error in U_t with ambient temperature. It can be observed from this figure that by using the Eq. (3) the error is 1.1% whereas by using the Eq. (1) the error increases with increase in ambient temperature. At higher ambient temperature ($T_a = 313$ K), Eq. (1) results in error of about 15%. The variation of computational error in U_t as a function of wind heat transfer coefficient is shown in Fig. 6. It is observed that using the Eq. (3) the computational error in U_t is within 1.1% for $e_p = 0.1$ and $e_p = 0.5$, whereas, by using the empirical Eq. (1) the error is negligible at $h_w = 10\ \mathrm{W/m^2\,K}$ and the error increases at higher values of h_w when e_p is equal to 0.5. It is also noted that using the Eq. (1) the error is about 12% at $h_w = 10\ \mathrm{W/m^2\,K}$ and the error decreases at higher values of h_w for $e_p = 0.1$.

The results shown in Figs. 1–6 illustrate the effect of one variable at a time. The errors resulting from use of semi-empirical equations would be larger if all the possible combinations of variables covering the entire range of variables are considered. Accordingly in the present work the results have been analyzed considering all the possible combinations of variables (by varying the variables in small steps). It has been found that the maximum absolute error in values of U_t computed from the analytical Eq. (3) is 1.8% in comparison to those obtained from numerical solution of heat balance equations. The maximum absolute errors in computation of U_t by using the empirical Eq. (1), as compared to those obtained by numerical solution of heat balance equations, is about 23%. It has also been verified that the Eq. (3) can be used with reasonable accuracy for air gap spacing between plate and glass cover less than 20 mm. For example, the analytical Eq. (3) with T_g from Eq. (8) result in absolute computational error in U_t equal to 0.1% when $L = 10$ mm, $T_p = 333$ K, $T_a = 273$ K, $e_p = 0.1$, $h_w = 10\ \mathrm{W/m^2\,K}$ and $\beta = 90°$. The maximum absolute error in computation of U_t by the use of analytical Eq. (3) with T_g from Eq. (8) for values of L between 10 and 20 mm and for entire range of other variables T_p, e_p, h_w, T_a, and β covered in present work is found to be less than 2% in comparison to results of numerical solution.

The reasons for the error in Eq. (1) can be briefly analyzed as follows: T_g has not been explicitly employed in Eq. (1) but only indirectly contained in it. In obtaining Eq. (1) the convection and radiation terms have been regrouped rather than as these would appear as per thermal network for heat losses [7,9]. Eq. (1) has been obtained through empirical modifications of the equation of Hottel and Woretz [7] which assumed emittance of absorber plate equal to that of glass cover, in order to permit regrouping of the convection and radiation terms. This was possible as selective coatings were not employed at that time. In the analytical Eq. (3), T_g has been used explicitly and hence it is possible to obtain the convective and radiative heat transfer coefficients and use them as such in Eq. (3) without having to regroup the convection and radiation coefficients as would be required for eliminating glass cover temperature. As a result the analytical Eq. (3) correctly predicts the variation of U_t with collector parameters such as emittance of absorber plate, air gap spacing and climatic variables such as wind heat transfer coefficient over entire range considered in the present work.

4. Conclusions

The method proposed in the present work for computation of U_t of single glazed flat plate collector with vertical configuration predicts values of U_t with greater accuracy for entire ranges of variables covered in the present work. The maximum absolute error in computation of U_t is within two percent in comparison to results of numerical solution.

The advantage of using proposed method over approximate method is that the glass cover temperature can be found by empirical equation hence appropriate correlation for h_{cpg} can be used and other radiative heat transfer coefficients can also be computed.

Appendix A

A.1. Heat balance of single glazed flat plate collector

Under steady state conditions, the top heat loss flux density from absorber plate at an average temperature T_p to the glass cover at an average temperature T_g equals to that from glass cover to surroundings.

Top heat loss flux density from absorber plate to glass cover is given by

$$\dot{Q}_t'' = (h_{cpg} + h_{rpg})(T_p - T_g) \tag{A.1}$$

And from glass cover to surroundings by

$$\dot{Q}_t'' = (h_w + h_{rga})(T_g - T_a) \tag{A.2}$$

Solar radiation absorbed in the glass cover has not been taken into account. For calculation of U_t, the standard practice is to account for this energy influx in the overall energy balance by increasing the energy absorption term (through an artificial enhancement in the optical efficiency, since magnitude of this energy influx is proportional to the insolation) rather than by decreasing the heat loss term [9]. It is desirable to have an expression for U_t for any absorber temperature, independent of insolation level [9]. Eqs. (A.1) and (A.2) have to be solved iteratively to find value of T_g since convective and radiative heat transfer coefficients are non linear functions of T_g.

A.2. Radiative and convective heat transfer coefficients

The radiative heat transfer coefficient between absorber plate and glass cover, h_{rpg} is given by

$$h_{rpg} = \frac{\sigma(T_p^2 + T_g^2)(T_p + T_g)}{1/\varepsilon_p + 1/\varepsilon_g - 1} \tag{A.3}$$

The convective heat transfer coefficient between absorber plate and glass cover is an important parameter in heat balance of flat plate collector especially when absorber has selective coating. Elsherbiny et al. [21], based on their extensive experimental data, proposed following correlations for Nusselt number for enclosed cavities for angle of tilts between 60° and 90°:

(i) For $\beta = 90°$, for Rayleigh number range, $10^3 \leqslant \mathrm{Ra}_L \leqslant 10^7$, $5 \leqslant AR \leqslant 110$

$$\begin{aligned}
\mathrm{Nu}_1 &= 0.0605\mathrm{Ra}_L^{1/3} \\
\mathrm{Nu}_2 &= \left[1 + \left\{\frac{0.104\mathrm{Ra}_L^{0.293}}{1 + \left(\frac{6310}{\mathrm{Ra}_L}\right)^{1.36}}\right\}^3\right]^{1/3} \\
\mathrm{Nu}_3 &= 0.242\left(\frac{\mathrm{Ra}}{\mathrm{AR}}\right)^{0.272} \\
\mathrm{Nu}_{90} &= [\mathrm{Nu}_1, \mathrm{Nu}_2, \mathrm{Nu}_{3,}]_{\max}.
\end{aligned} \tag{A.4}$$

(ii) For inclined layers ($\beta = 60°$)

$$\mathrm{Nu}_1 = \left[1 + \left\{\frac{0.0936\mathrm{Ra}_L^{0.313}}{1+G}\right\}^7\right]^{1/7}$$

where

$$G = \frac{0.5}{\left[1 + \left(\frac{\mathrm{Ra}_L}{3160}\right)^{20.6}\right]^{0.1}}$$

$$\mathrm{Nu}_2 = \left[0.104 + \frac{0.175}{AR}\right]\mathrm{Ra}_L^{0.283}$$
$$\mathrm{Nu}_{60} = (\mathrm{Nu}_1, \mathrm{Nu}_2)_{\mathrm{max.}} \quad \text{(A.5)}$$

(iii) For angle of inclination between 60° and 90°

$$\mathrm{Nu}_\beta = \left[\frac{(90\,\mathrm{deg} - \beta)\mathrm{Nu}_{60} + (\beta - 60\,\mathrm{deg})\mathrm{Nu}_{90}}{30\,\mathrm{deg}}\right] \quad \text{(A.6)}$$

The radiative heat transfer coefficient between glass cover and surroundings of flat plate collector is given as

$$h_{rga} = \sigma\varepsilon_g(T_g^2 + T_a^4)(T_g + T_a) \quad \text{(A.7)}$$

The variations in air properties: thermal conductivity, kinematic viscosity, coefficient of volumetric expansion and Prandtl number with mean temperature have been taken into account.

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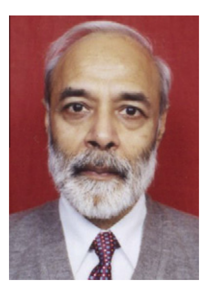

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