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Numerical Simulation of Inclined Tubular Collector

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Abstract—*In the calculation of convective and radiative heat losses from tubular collector, glass cover temperature is involved, which is generally not known. Although an empirical correlation exists for glass cover temperature in horizontal tubular collector, no similar relation is available for the glass cover temperature of a tubular absorber with inclined collector. In the present work, numerical solutions of the heat balance equation are found for inclined tubular collector using a computer program. A new correlation for glass cover temperature, including the effect of inclination is obtained by an empirical relation. The useful energy of the collector calculated by using the correlation of glass cover temperature available in the literature and from the proposed correlation are compared to those obtained from the numerical solutions of heat balance equations. The range of variables involved are--absorber temperature 60 to 240 °C, absorber coating emittance 0.1 to 0.95, wind heat transfer coefficient 15 to 45 W/m^{2}K, absorber diameter 25 to 65 mm, ambient temperature 0-40 °C and air gap spacing 10-20 mm. The angle of inclination is taken to be 0°, 30° & 60°, measured from the horizontal. The study shows that using the proposed correlation, there is an improvement in computation of glass cover temperature.*

1. INTRODUCTION

Natural convection heat transfer in the annulus between two concentric cylinders is an important research topic due to its wide application in engineering problems. Many experimental and theoretical investigations have been conducted in recent years as an alternatives to the conventional liquid flat plate collector due to the wide range of applications. Some of these have been commercialized .The objectives in developing these designs have been varied. They include a desire to improve the efficiency, to reduce cost, to increase the operating temperature, or to reduce the weight of the collector.

The thermal performance of a solar energy system under various operating conditions can be predicted with reasonable accuracy if the heat loss factor (U_l) of the collector for that condition is known. The heat loss factor of the collector can be computed analytically or from empirical equations. Numerical solutions involve more computational work and are time-consuming. An empirical equation is preferred by a designer.

The useful energy of a collector is the difference between the absorbed solar radiation and the thermal losses due to convection and radiation between absorber and glass cover followed by forced convection due to wind and radiation losses from glass cover to the surroundings. Many correlations are available in the literature for the calculation of heat loss factor.

SK Nanda and SC Mullick [1] made a first attempt to determine a solution for predicting heat loss factor for a tubular absorber with a concentric glass cover without the requirement of an iterative solution. They made an attempt to solve the heat loss problem for a cylindrical absorber, following the Hottel and Woertz analysis [2] for a flat-plate absorber. However, in the case of the absorber of a concentrator, more than one glass cover is not likely to be used. In their analysis, f is expressed as a function of wind velocity, absorber temperature, emittance of the black coating and the ratio of cover to absorber diameter. The factor f was used to calculate the glass cover temperature approximately. The glass cover temperature was substituted in the actual expression to calculate the heat loss factor with fairly good accuracy.

An approximate value of the glass cover temperature has been obtained by considering the following equation form:

$$\frac{T_2 - T_a}{T_1 - T_a} = f = \frac{R_{2a}}{R_{12}} = \frac{D_1 h_{12}}{D_2 h_{2a}}$$

$$T_2 = (fT_1 + T_a)/(1+f) \quad (1)$$

The ratio f of outside to inside heat transfer resistances was correlated with the independent variables such as the wind velocity, the absorber temperature, the emittance of black paint, etc.

$$f = f(V_w, T_1, \in_1, D_1/D_2, T_a)$$

An empirical equation was found by the least-squares regression analysis, i.e.

$$f = \frac{D_1}{D_2}(0.189 + 0.176\in_1)v_w^{-0.485} \times \exp[0.00371\in_1^{0.388}(T_1 - 273)] \quad (2)$$

The value of *f* for known values of the variables V_w, T_1, ε_l and D_1/D_2 was calculated from equation (2) and substituted into equation (1) to obtain an approximate value of T_2. This approximate value of glass cover temperature will suffice to compute the individual heat transfer coefficients with reasonably good accuracy. Hence, the overall heat loss coefficient, U_l, is computed.

$$U_l = \left[\frac{1}{C\left(T_1 - T_2^{0.25}\right) + \dfrac{\sigma\left(T_1^2 + T_2^2\right)\left(T_1 + T_2\right)}{\dfrac{1}{\in_1} + \left(\dfrac{D_1}{D_2}\right)\left(\dfrac{1}{\in_2} - 1\right)}} + \left(\frac{D_1}{D_2}\right)\frac{1}{h_w + \sigma \in_2 \left(T_2^2 + T_a^2\right)\left(T_2 + T_a\right)} \right]^{-1} \quad (3)$$

The semi-empirical equation predicts the heat loss factor to within ±1.2 per cent of the values obtained by the actual solution of the simultaneous equations, in the range of variables—wind velocity 0.5 m/s to 10.0 m/s, absorber temperature 20 °C to 200 °C above ambient air temperature and emittance of the black coating 0.1 to 0.95. However, when the absorber temperature, emittance and ambient air temperature are all high, and the absorber diameter, air gap and wind velocity are all small, the error can be higher.

N C Bhowmik and S C Mullick [3] proposed a relatively simple equation for the factor *f*. A semi-empirical equation for the heat loss factor as a function of the various variables involved was developed.

For a tubular absorber with a concentric glass cover, with diameters D_1 and D_2, respectively, equation (3) for was modified as:

$$U_l = \left[\frac{1}{C\left\{\dfrac{T_1 - T_a}{1+f}\right\}^{0.25}} + \frac{D_1}{D_2}\cdot\frac{1}{h_w} \right]^{-1} + \frac{\sigma\left(T_1^2 + T_a^2\right)\left(T_1 + T_a\right)}{\left\{\left(\in_1 - 0.04\left(1-\in_1\right)\left(\dfrac{T_1}{450}\right)\right)\right\}^{-1} + \dfrac{D_1}{D_2}\left(\dfrac{1}{\in_2} - 1\right) + \dfrac{f}{\in_2}} \quad (4)$$

The value of C in the convection term is obtained from the correlation

$$C = \frac{1.45 + 0.96\left(\in_1 - 0.5\right)^2}{D_1\left(\dfrac{1}{D_1^{0.6}} + \dfrac{1}{D_2^{0.6}}\right)^{1.25}}$$

And factor has been correlated with the wind coefficient in place of wind velocity

$$f = \frac{D_1}{D_2^{1.4}}\{0.61 + 1.3\in_1\}h_w^{-0.9}\exp[0.000325(T_1 - 273)$$

The equation predicts the overall heat loss factor, U_l, to within ±5% of the value obtained by exact solution of the simultaneous equations, in the range of variables-- absorber temperature, 60°C to 220°C and emittance of the black coating, 0.1 to 0.95.

S C Mullick and S K Nanda [5] proposed a different approach to evaluate the heat loss factor of a tubular absorber with a concentric glass cover. The glass cover temperature was estimated by an empirical relation, replacing the empirical relation for the factor *f* of earlier work The number of empirical relationships was thus reduced from two (for the heat loss factor and for *f*) to one (for glass cover temperature). However, in their work no specific dependence of h_w on wind velocity was assumed.

$$T_2 = T_a + 0.04075\left(\tfrac{D_1}{D_2}\right)^{0.4} h_w^{-0.67}\left[2 - 3\in_1 + \tfrac{(6+9\in_1)T_1}{100}\right](T_1 - T_a) \quad (5)$$

In the present work heat balance equations are solved employing an iterative procedure using a computer program. The glass cover temperature is obtained by using numerical solutions of heat balance equations employing empirical correlation given by S A Nada [4].

2. HEAT LOSSES IN TUBULAR COLLECTOR

The heat losses from the tubular collector is evaluated by considering convection and radiation losses from the absorber in upward direction. Fig. 2.1 shows the electric analogue of the heat transfer network, whilst Fig. 2.2 is a schematic diagram of a cylindrical absorber with a concentric glass cover.

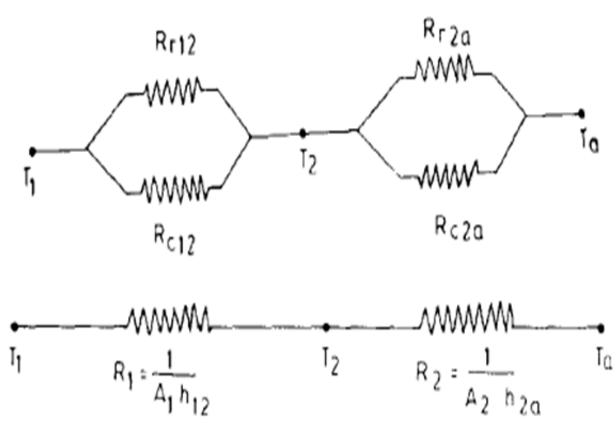

Fig. 2.1: Electrical analogue of heat transfer network

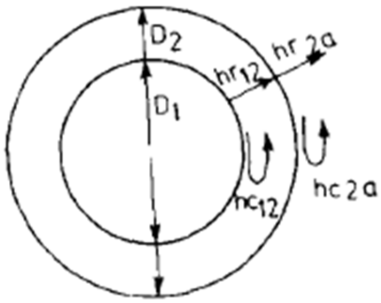

Fig. 2.2: Cylindrical absorber with concentric glass cover

Under steady-state conditions the total radiative and convective heat loss from the absorber at temperature T_1 to the transparent glass cover at T_2 equals that from the glass cover to the ambient air at T_a.

The rate of heat loss from the absorber to the glass cover is given by

$$Q_l = A_1 h_{c12}\left(T_1 - T_2\right) + \frac{A_1\sigma\left(T_1^4 - T_2^4\right)}{\frac{1}{\in_1} + \frac{D_1}{D_2}\left(\frac{1}{\in_2} - 1\right)} \quad (6)$$

And from the glass cover to- the ambient by

$$Q_l = A_2 h_w (T_2 - T_a) + A_2\sigma \in_2 (T_2^4 - T_a^4) \quad (7)$$

Equations (6) and (7) have to be solved simultaneously to find the value of T_2. Substituting the value of T_2, we can obtain Q_l.

The overall heat loss factor, U_l, of the absorber, based on unit target area, is given by

$$U_l = \frac{1}{\frac{1}{h_{12}} + \frac{D_1}{D_2}\cdot\frac{1}{h_{2a}}}$$

$$U_l = \left[\frac{1}{\left(h_{c12} + h_{r12}\right)} + \frac{D_1}{D_2}\left(\frac{1}{h_w + h_{r2a}}\right)\right]^{-1} \quad (8)$$

Where h_{12} and h_{2a} are inside and outside heat transfer coefficients, respectively. The convective and radiative coefficients are nonlinear functions of T_2 Therefore, T_2 has to be found by iterative/numerical solution of the simultaneous equations. However, an approximate value of glass cover temperature will suffice to compute the individual heat transfer coefficients with reasonably good accuracy.

A number of correlations are available to find coefficient of heat transfer by natural convection, $h_{c12,}$ in horizontal tubular absorber with concentric glass cover. But no previous work was found in the literature that studied the effect of the inclination angle on glass cover temperature in cylindrical annulus.

For the coefficient of heat transfer by natural convection, h_{c12} between inclined concentric cylinders the following correlation proposed by S A Nada [4], has been used in the present work:

$$\kappa_{eff}/\kappa = 1.21 Ra_{cc}^{0.33}(L/D_2)^{0.99}(1 - 0.34\cos\alpha(1 - 1.06\cos\alpha)) \quad (9)$$

Where $\boldsymbol{\alpha}$ is the angle of inclination, k_{eff} is effective thermal conductivity, which is defined as the thermal conductivity that a stationary fluid should have to transfer the same amount of heat as the buoyancy driven moving flow and Ra_{cc} is corrected Rayleigh number given by

$$Ra_{cc} = \frac{\ln\left(\frac{D_2}{D_1}\right)^{0.4}}{D_1^3\left[\frac{1}{D_1^{0.6}} + \frac{1}{D_2^{0.6}}\right]^5} Ra_{D1}$$

With Ra_{D_1} is the Rayleigh number based on the absorber tube diameter D_1

$$Ra_{D1} = Gr_{D1} \times Pr$$

$$Gr_{D_1} = \frac{g\beta(T_2 - T1)D_1^3}{\nu^2} \qquad Ra_{D_1} = \frac{g\beta(T_2 - T_1)D_1^3}{\nu\alpha'}$$

Thus

$$h_{c12} = \frac{2k_{eff}}{D_1 \times \ln\left(\frac{D_2}{D_1}\right)} \quad (10)$$

In the present work no specific dependence of h_w on wind velocity is assumed and h_w is taken as an independent variable. The radiative heat transfer coefficient, h_{r12}, between the absorber and the glass cover is given by:

$$h_{r_{12}} = \frac{\sigma(T_1^2 + T_2^2)(T_1 + T_2)}{\frac{1}{\in_1} + \frac{D_1}{D_2}\left(\frac{1}{\in_2} - 1\right)} \quad (11)$$

And the radiative heat transfer coefficient from the glass cover at T_2 to the ambient air at T_a is given by:

$$h_{r_{2a}} = \sigma \times \in_2 (T_2^2 + T_a^2)(T_2 + T_a) \quad (12)$$

As the order of variation of the radiative and convective heat transfer coefficients with temperatures is lower than the order of variation of temperatures, an approximate value of glass cover temperature, T_2 will suffice for the calculation of the heat loss factor from equation (8). An approximate value of glass cover temperature can be obtained by considering the following relation:

$$\frac{T_2 - T_a}{T_1 - T_a} = \frac{R_{2a}}{R_{12} + R_{2a}} = f \quad (13)$$

The factor f is a function of the individual heat transfer coefficients and thus a relatively weak function of the temperatures. An empirical expression for the factor f may therefore, be obtained as a function of the basic variables- the absorber temperature T_1, emittance ε_1, wind heat transfer coefficient h_w, absorber diameter D_1, glass cover diameter D_2, ambient temperature T_a and inclination angle $\boldsymbol{\alpha}$, through regression analysis. From equation (13) the value of T_2 is in the form

$$T_2 = T_a + f\,(T_1 - T_a)$$

The following empirical relation for the factor f is obtained by regression analysis,

$$f = 0.0412(D_1/D_2)^{0.4} h_w^{-0.67}[2 - 10.2 \in_1 + 5.52\cos\alpha \\ + (5.24 + 13.21 \in_1)T_1/100]$$

And hence

$$T_2 = T_a + \{0.0412(D_1/D_2)^{0.4} h_w^{-0.67}[2 - 10.2 \in_1 + 5.52\cos\alpha \\ + (5.24 + 13.21 \in_1)T_1/10\}(T_1 - T_a) \quad (14)$$

The simple empirical relationship (14) can estimate the glass envelope temperature T_2 to within ±10 ° (maximum error) of the value obtained by iterative solution of heat balance equations (6) and (7) on a computer over the range of variables-absorber temperature 60 to 240°C, absorber coating emittance 0. 1 to 0.95, absorber diameter 12.5 to 65 mm, wind heat transfer coefficient 15 to 45 W/m^2 ^{0}C, and ambient temperature 0 to 40°C.

The useful energy of the collector is defined as the actual amount of energy that is available after meeting the heat losses.

If I is the intensity of solar radiation, in W/m^2, incident on the aperture plane of the solar collector having a collector surface area of A, m^2, then the amount of solar radiation received by the collector is:

$$Q_i = I \cdot A$$

However, a part of this radiation is reflected back to the sky, another component is absorbed by the glazing and the rest is transmitted through the glazing and reaches the absorber as short wave radiation. Therefore the conversion factor indicates the percentage of the solar rays penetrating the transparent cover of the collector (transmission) and the percentage being absorbed. Basically, it is the product of the rate of transmission of the cover and the absorption rate of the absorber.

Thus,

$$Q_i = I(\tau\alpha)\cdot A$$

As the collector absorbs heat its temperature is getting higher than that of the surrounding and heat is lost to the atmosphere by convection and radiation. The rate of heat loss (Q_l) depends on the collector overall heat transfer coefficient (U_l) and the collector temperature.

$$Q_l = U_l A\,(T_1 - T_a)$$

Thus, the rate of useful energy extracted by the collector (Q_u), expressed as a rate of extraction under steady state conditions, is proportional to the rate of useful energy absorbed by the collector, less the amount lost by the collector to its surroundings. This is expressed as follows:

$$Q_u = Q_i - Q_l \\ = I\,(\tau\alpha)\,A - U_l A\,(T_1 - T_a) \quad (15)$$

3. RESULTS AND DISCUSSION

The numerical solution of heat balance equations, Eqns. (6) and (7), iteratively using a computer program gives the value of glass cover temperature T_2. The values of glass cover temperature are also obtained by using approximate correlation/empirical correlation proposed by S C Mullick and S K Nanda [5].

Table 3.1: Range of variables

Variables	Range
Absorber emittance,ε_1	0.1-0.95
Absorber temperature, T_1	333-513 K
Absorber diameter, D_1	25-65 mm
Wind heat transfer coefficient, h_w	15-45 Wm^2/K
Glass cover diameter, D_2	35-75 mm
Air gap Spacing, L	10-20 mm
Ambient temperature, T_a	273-313 K
Collector tilt angle, $\boldsymbol{\alpha}$	0-60 (deg.)

3.1 Variation of glass cover temperatures for the heat transfer analysis of tubular solar collector

The estimation of glass cover temperature is required for the estimation of radiative and convective heat transfer coefficients. The glass cover temperature are calculated by the numerical solution of heat balance equations, Eqns (6)-(12) iteratively using a computer program. The glass cover temperature is also calculated by correlation given by Mullick and Nanda [5], and proposed correlation, Eqn (14). These values are compared with glass cover temperature obtained from the numerical solutions of heat balance equations. The variation of glass cover temperatures with wind heat transfer coefficient is shown in Figs 3.3-3.4.

The glass cover temperatures decreases with the increase in the wind heat transfer coefficient as shown in Figs 3.3-3.4, which may be attributed to the fact that as the wind heat transfer coefficient increases, the forced convection from the glass cover also increases, it cools the surface of the glass cover, hence decrease the glass cover temperature. It can also be seen from the Figs that the glass cover temperature obtained from Eqn (14) are well acceptable with the numerical solution.

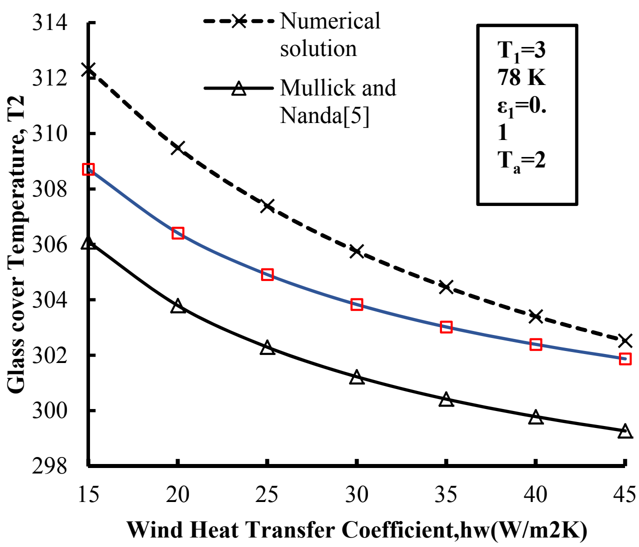

Fig. 3.3:Variation of glass cover temperature with wind heat transfer coefficient at ε_1=0.1

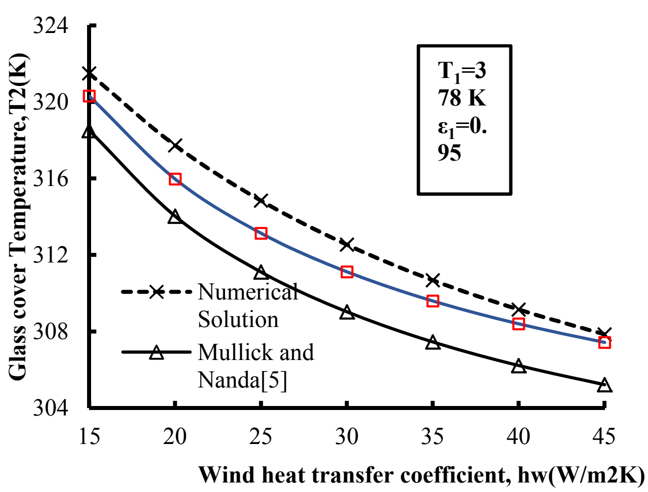

Fig. 3.4: Variation of glass cover temperature with wind heat transfer coefficient at ε_1=0.95

3.2 Variation of useful energy of the collector

The useful energy of the collector is the actual amount of energy that is available from the collector. For the evaluation of useful energy, the intensity of solar radiation is kept 700 W/m^2 and the optical efficiency of collector is taken as 0.85.

The variation of useful energy is shown in Figs 3.5-3.6. The Fig shows the variation of useful energy of the collector with collector tilt angle at emissivities, ε_1 = 0.1 and ε_1 = 0.95, respectively. The useful energy of the collector increases with the increase in the collector tilt angle. This is due to the fact that, as tilt angle is increased, heat loss factor decreases resulting in decrease in heat losses. Thus from Eqn (15), the useful energy of the collector increase. It can be seen from the Fig 3.5 that at emissivity ε_1 = 0.1, the useful energy obtained from correlation given Mullick and Nanda [5] shows a larger deviation from the numerical solution whereas the results obtained from proposed correlation are well acceptable within limits with the numerical solution for the whole range of collector tilt angle, the error being minimised at higher emissivity, ε_1 = 0.95 in Fig 3.6.

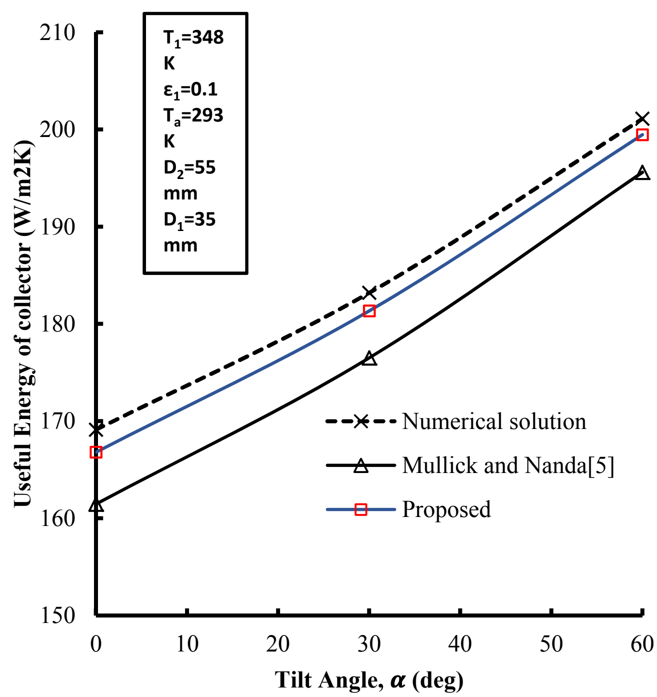

Fig. 3.5: Variation of useful energy of collector with collector tilt angle at ε_1 = 0.1

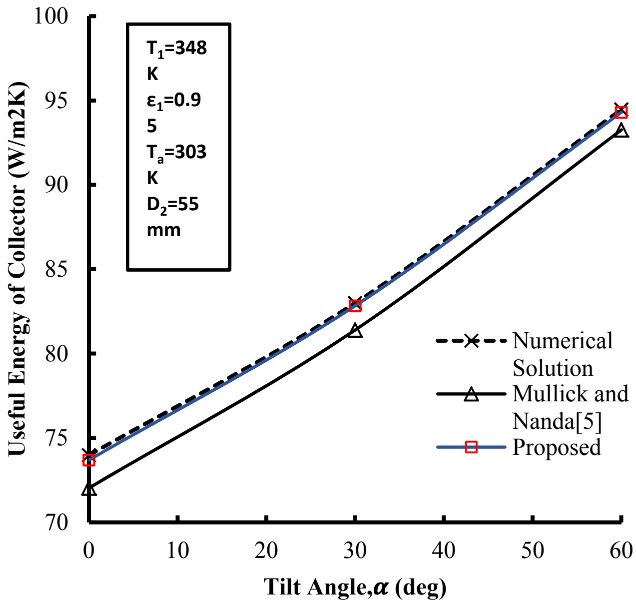

Fig. 3.6 Variation of useful energy of collector with collector tilt angle at $\varepsilon_1 = 0.95$

4. CONCLUSIONS

1. For the same flux, heat-transfer coefficients decreases with the angle of inclination of the tubular collector. The minimum value occurs in the vertical position and the maximum value in the horizontal position.
2. The empirical correlations available in the literature have large errors, since they have been developed assuming the horizontal annulus.
3. The analytical method proposed by Mullick and Nanda [5] used for calculation of glass cover temperature, results in an maximum error of ±15 °C, whereas proposed correlation have a maximum error of ±9 °C as compared to the numerical solution of heat balance equations.
4. The empirical correlation proposed by Mullick and Nanda [5] causes large error in the calculation of useful energy available from the solar collectors.

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