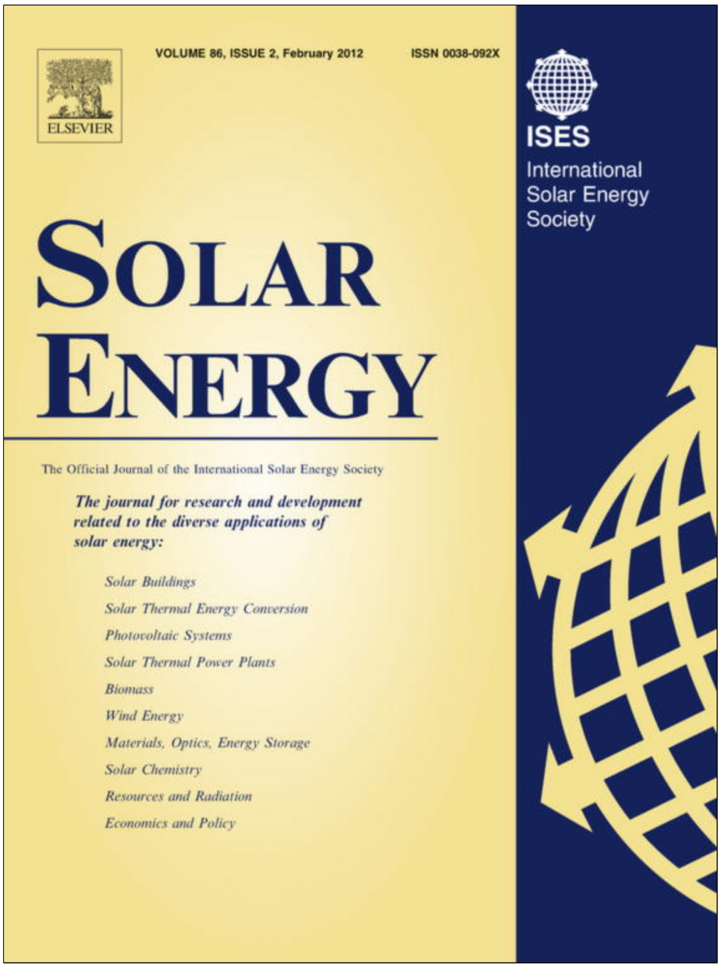
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Brief Note

Calculation of the glass cover temperature and the top heat loss coefficient for 60° vee corrugated solar collectors with single glazing

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Abstract

In this paper, the top heat losses from a 60° vee corrugated solar collector with single glazing have been investigated. An approximate method for computation of glass cover temperature and top heat loss coefficient has been followed. A modified equation from Akhtar and Mullick's relation was proposed. The predicted values of the glass cover temperature and the top heat loss coefficient were compared with the results obtained by iterative solution of the energy balance equations over a wide range of operating conditions. A good accuracy is provided by the proposed equation which is recommended to be used in the energy analysis of the present configuration.

Keywords: Flat plate solar collector; Vee corrugated absorber plate; Glass cover temperature; Top heat loss coefficient

1. Introduction

Because of their advantages over the other collector's types, simple in design, few mechanical parts, low cost and easy installation, the flat plate solar collectors are more appropriate techno-economically for the low temperature solar energy applications than the more complex ones. Although these collectors are classified into the medium to low temperature category which limits their use, they have been popularized after the inclusion of some substantial improvements on their structural design. The design of suitable solar collectors, which leads to increase the useful energy gain and consequently to reduce the heat losses through principally the upper side, is the main subject of many researches which recommend the use of fins or corrugated surfaces as extended heat transfer area to get highest efficiencies.

The energy analysis of such collectors requires the knowledge of the top heat loss coefficient which is necessarily an iterative process to be calculated from the heat balance and thus large time consumption since many thousands of solutions may be required (Duffie and Beckman, 1991). To solve this problem, empirical equations for top heat loss coefficient based on measurement data were first developed by Hottel and Woertz (Duffie and Beckman, 1991; Klein, 1975). Later, in a series of papers, several corrected expressions for the accurate computation of the top heat loss coefficient have been proposed (Duffie and Beckman, 1991; Klein, 1975; Garg and Rani, 1980; Malhotra et al., 1981; Agarwal and Larson, 1981; Garg and Datta, 1984). However, because of the large errors resulting from the regrouping of the convective and radiative terms in the previous relations, improved equation forms for computing the glass cover temperature of flat plate solar collectors with single and double glazing have been developed (Mullick and Samdarshi, 1988; Samdarshi and Mullick, 1991; Akhtar and Mullick, 1999, 2007), which allows to accurately evaluate the top heat loss coefficient directly from the energy balance.

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Nomenclature

A	aspect ratio (average gap spacing to vee height)	ε	emissivity
f	ratio of outer to inner thermal resistance	ϕ	half opening angle of the vee corrugation (°)
h_w	wind convection coefficient (W/m^2 K)	θ	collector tilt angle (°)
H	vee height (m)	σ	Stefan–Boltzmann constant (W/m^2 K^4)
k	thermal conductivity (W/m K)		
L	average gap spacing, thickness (m)	*Subscripts*	
Nu	Nusselt number	a	ambient, apparent
Ra	Rayleigh number	g	glass cover
T	temperature (K)	p	plate
U_t	top heat loss coefficient (W/m^2 K)	s	sky

However, all these proposals have been developed primarily for collectors with flat absorber plate, which makes them inappropriate for use to predict the top heat losses from collectors with corrugated absorber plate. For this reason and for the aim of making a better assessment of the top heat losses' dependence on the absorber plate geometry, we investigate through the present study the effect of the geometrical characteristics of a 60° vee corrugated plate on the top heat losses from solar collectors with single glazing over a wide range of operating conditions. A modified version from the equation proposed by Akhtar and Mullick (1999), that provides good predictions for the glass cover temperature of the present configuration, was derived. The accuracy of the results obtained from the proposed model was verified by comparison with those obtained from the iterative solution of the heat balance equations.

2. Analysis

The vee corrugating of the absorber plate surface is regarded as one of the most successful techniques that are used to enhance the thermal performance of solar collectors. This technique is frequently implemented to devise selective surfaces characterized by high solar absorbtance (Duffie and Beckman, 1991; Hollands, 1963) and on the other hand to provide heat transfer enhancement in the flow passage area (El-Sebaii et al., 2011). Unfortunately, due to a cellular base flow occurs within the trough of the vee corrugations permitting better mixture of transfer rate (El-Sherbiny et al., 1978), and also to the increase over the plane emissivity of the absorber's surface (Hollands, 1963), an increasing in top heat losses is expected.

During the calculation of these heat losses, it is common practice to linearize the problem then to convert it to a simple electrical network which allows us to develop the concept of an overall top heat loss coefficient for simplification purpose (Duffie and Beckman, 1991). This coefficient could be determined more preferably by empirical modeling rather than solving a non linear system of heat balance equations. Therefore, in the following, an approximate method for computation of glass cover temperature and top heat loss coefficient of solar collectors with single glazing as that described by Akhtar and Mullick (1999) is adopted and subjected to some modifications corresponding to the present configuration (Fig. 1).

2.1. Glass cover temperature

According to Akhtar and Mullick's analysis, the glass cover temperature could be calculated as follows:

$$T_g = \frac{(fT_p + CT_a)}{(f+1)} \qquad (1)$$

where $C = (T_s/T_a + h_w/3.5)/(1 + h_w/3.5)$ used when the sky temperature lower than ambient is considered for outer radiation using Swinbank's relation, or $C = 1$ when the sky and ambient temperatures are considered equal; and f is the ratio of outer to inner thermal resistance which still has to be determined for the present problem. Hence, basing on the semi-analytical correlation developed by Akhtar and Mullick (1999), an examination of the effect of the vee corrugations on convection and radiation heat transfer processes between the vee corrugated plate and the glass cover yields to the following ratio of outer to inner thermal resistance, which is closely analogous to that proposed in the previous reference:

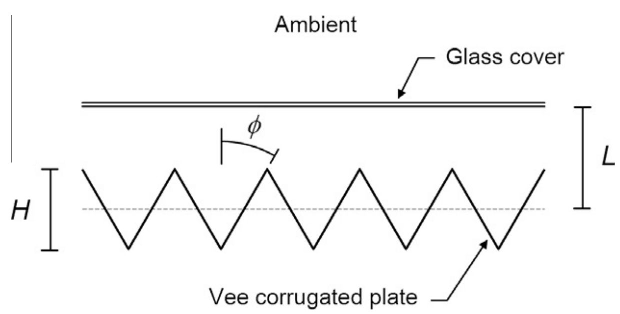

Fig. 1. Cross section of 60° vee corrugated absorber plate and flat glass cover.

$$f = \frac{(12\times 10^{-8}(T_a + 0.2T_p)^3 + h_w)^{-1} + 0.3L_g}{(6\times 10^{-8}(\varepsilon_a + 0.028)(T_p + 0.5T_a)^3 + 0.6L^{-0.2}((T_p - T_a)\cos\theta)^{0.25}\times\Gamma)^{-1}} \tag{2}$$

In which ε_a is the apparent emissivity of the vee corrugated plate. Using the radiation analysis on an enclosure with the fundamental relations of the view factor, the apparent emissivity for a vee corrugated plate of opening angle 2ϕ is given by the following expression which gives a good overall fit with the data of Hollands (1963):

$$\varepsilon_a = \left(1 + \left(\frac{1}{\varepsilon_p} - 1\right)\sin\phi\right)^{-1} \tag{3}$$

And Γ is a dimensionless function developed by regression analysis basing on the results obtained by numerical solution of the heat balance equations of 60° vee corrugated collector, using the convective heat transfer coefficient correlation provided by El-Sherbiny et al. (1978):

$$\Gamma = 1 + \frac{0.653}{(1+A)^{0.38}} + \frac{0.014h_w}{(1+A)^{0.09}} \tag{4}$$

It is worth mentioning that the geometrical parameters A and ϕ of the vee corrugated plate are suitably included in the Eq. (2) in such a way that this latter will be identical to Akhtar and Mullick's relation in the case of flat absorber plate, where $A^{-1} = 0$ and $\phi = 90^\circ$.

2.2. Top heat loss coefficient

Once the glass cover temperature is determined from the previous relations, the top heat loss coefficient could be readily obtained from (Akhtar and Mullick, 1999):

$$U_t = \left[\frac{L_g}{k_g} + \left(h_w + \sigma\varepsilon_g\frac{\left(T_g^4 - T_s^4\right)}{(T_g - T_a)}\right)^{-1} + \left(\frac{k\mathrm{Nu}}{L} + \sigma\frac{\left(T_p^2 + T_g^2\right)(T_p + T_g)}{1/\varepsilon_a + 1/\varepsilon_g - 1}\right)^{-1}\right]^{-1} \tag{5}$$

In which, the Nusselt number of an enclosed gap bounded by one 60° vee corrugated and one flat surface is given by the following correlation (El-Sherbiny et al., 1978):

$$\mathrm{Nu} = \mathrm{Nu}_c + K\left(1 - \frac{\mathrm{Ra}_c(\sin 1.8\theta)^{1.6}}{\mathrm{Ra}\cos\theta}\right)\left[1 - \frac{\mathrm{Ra}_c}{\mathrm{Ra}\cos\theta}\right]^+ + B\left[\left(\frac{\mathrm{Ra}\cos\theta}{\mathrm{Ra}_\theta}\right)^{1/3} - 1\right]^+ \tag{6}$$

where

$$\mathrm{Nu}_c = \left(\frac{1}{1 - 0.3025/A + 0.06825/A^2}\right)A\ln\left(\frac{2A+1}{2A-1}\right) \tag{7}$$

$$\mathrm{Ra}_c = 1708\left(1 + \frac{0.036}{A} + \frac{2.69}{A^2} - \frac{1.70}{A^3}\right) \tag{8}$$

$$K = \frac{2460}{\mathrm{Ra}_c}\left(1 - \frac{0.195}{A} + \frac{5.97}{A^2} - \frac{4.16}{A^3}\right) \tag{9}$$

$$B = 2.23 - 0.0123\theta + 0.34\times 10^{-3}\theta^2 \tag{10}$$

$$\mathrm{Ra}_\theta = 11300(1 + 0.204\sin[4.50(\theta - 37.8^\circ)]) \tag{11}$$

where the operator $[\ ]^+$ means that the value of the quantity is to be taken as zero if the argument inside the brackets is negative but otherwise takes on the value of the argument.

3. Results and discussion

The predicted values of the glass cover temperature obtained from the present model are presented in graphical form along with the results obtained by iterative method based on the convective heat transfer coefficient correlation due to El-Sherbiny et al. (1978) for many influential parameters. The comparison between these results was performed over a range of wind convection coefficient varying from 5 to 50 W/m^2 K, plane emissivity of the absorber plate from 0.05 to 0.95, collector tilt angle from 0° to 60° and absorber plate temperature from 353 to 423 K. While, an average gap spacing of 25 mm, ambient temperature of 293 K, glass cover thickness, emissivity and thermal conductivity of 5 mm, 0.88 and 0.78 W/m^2 K respectively were considered. For outer radiation, the Swinbank's relation for the sky temperature was used.

It can be seen from Figs. 2–5 that all results are in very good agreement over the entire investigated range of variables and for two values of the aspect ratio A.

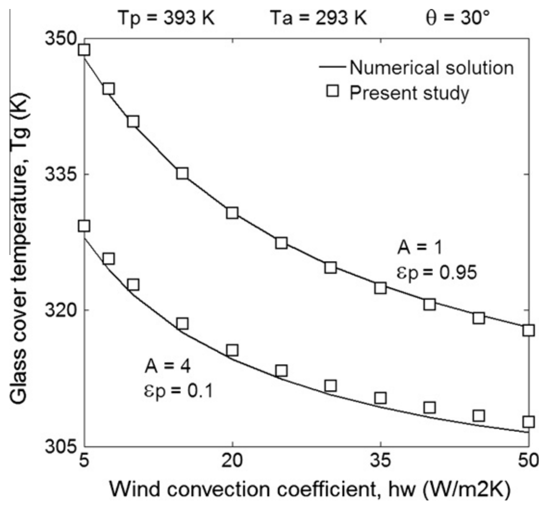

Fig. 2. Variation of the glass cover temperature with the wind convection coefficient.

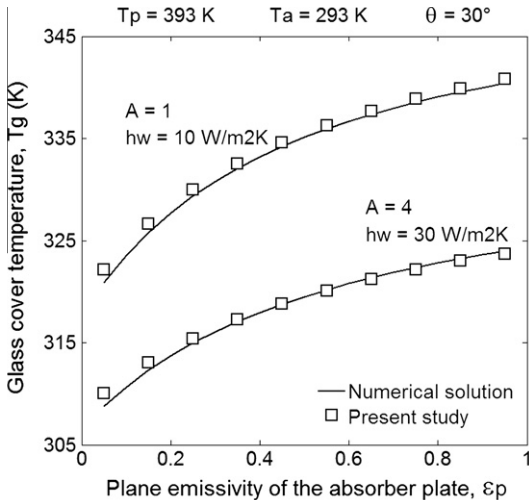

Fig. 3. Variation of the glass cover temperature with the plane emissivity of the absorber plate.

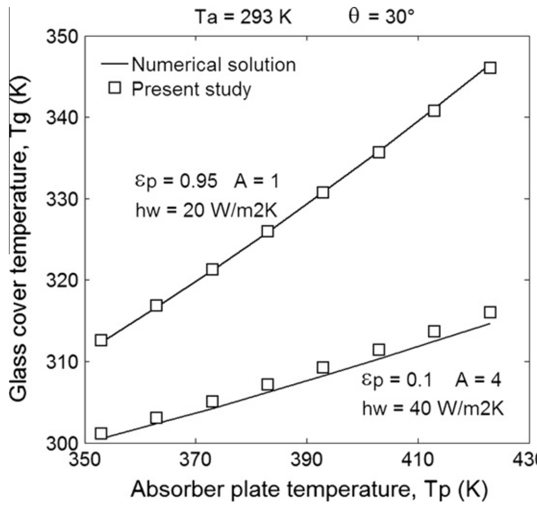

Fig. 5. Variation of the glass cover temperature with the absorber plate temperature.

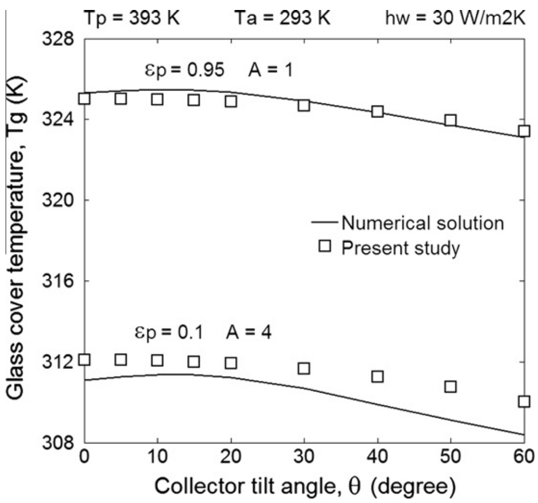

Fig. 4. Variation of the glass cover temperature with the collector tilt angle.

The maximum error between the iterative results of the glass cover temperature and those given by the present model is about 3 °C; and it can be noted that the best predictions are given for non selective absorber plate. Since the natural convection in the enclosed gap depends strongly on Rayleigh number (temperature difference, $T_p - T_g$) which is in turn sensitive to the wind convection coefficient, the effect of this latter has been included in the approximation of the natural heat transfer correlation through the dimensionless function Γ, which significantly reduces the computational error of the glass cover temperature. As it is expected, at the same operating conditions the glass cover temperature as well as the top heat loss coefficient increase with the decreasing aspect ratio.

A comparison between the results of the top heat loss coefficient obtained from the present model and those obtained by iterative method was also carried out (not shown here), by which an excellent agreement was observed. It was found that the relative error from the evaluation of the top heat losses, using the previous empirical relations for estimating the glass cover temperature, is less than 1% over the whole range of variables, which may confirm that the followed approximate method described in (Akhtar and Mullick, 1999) provides more accurate predictions of heat losses than the single equation methods. At some extreme conditions, an increasing in the top heat losses by up 40% relative to the flat plate collectors could be reached. Therefore, the use of the present model in the energy analysis of the 60° vee corrugated solar collectors is justified.

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