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Research Article

Heat Transfer Analysis of a Flat-plate Solar Collector Running a Solid Adsorption Refrigerator

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Abstract: Adsorption solar cooling appears to have prospect in the tropical countries. The present study is a theoretical investigation of the performance of a solar adsorption refrigerator using a flat-plate solar collector. The values of glass cover and absorber plate temperatures obtained from numerical solutions of heat balance equations are used to predict the solar coefficient of performance of the solar refrigerator. The simulation technique takes into account the variations of ambient temperature and solar radiation along the day. The effects of optical parameters of the glass cover such as absorption and transmission coefficients on glass cover and absorber plate temperatures and consequently on the coefficient of performance are analyzed. As a result, it is found that the absorber plate temperature is less to the absorption coefficient than the cover glass temperature. Also the thermal radiation exchange has more effect on the cover glass temperature. The higher values of COP are obtained between 11 and 13 h during the morning when the temperatures of the absorber plate and the ambient temperatures increase. Moreover the COP increases with the coefficient of transmission of the glass cover but the main parameter acting on the variations of the COP remains the temperature of the evaporator.

Keywords: Absorber temperature, coefficient of performance, flat-plate solar collector, glass-cover temperature, mathematical model, optical parameters, solar refrigeration, simulation

INTRODUCTION

Solar radiation is known to be the largest and the world most abundant and clean energy source. In recent years, many promising technologies have been developed to extract sun's energy. One of these important technologies is the solar refrigeration systems. Refrigeration is an attractive application of solar energy due to the supply of sunshine and the needs for refrigeration reaches maximum level at the same time. Solar solid adsorption ice maker appears its reasonable application for solar refrigeration during recent decades due to its ability to combat against ozone layer depletion which is caused by the utilization of CFCs and HCFCs in cooling system. The advantage and development of adsorption machine is widely studied by Meunier (1998). Solar ice making with adsorption technology has been also extensively studied by Leite and Daguenet (2000) and Boubakri (2003). Yong and Sumathy (2004) investigated the performance of a solar-powered adsorption air-conditioning system using a lumped parameter model. Khattab (2006) developed a mathematical model to simulate and optimize the performance of a solar-powered adsorption refrigeration system with the solid adsorption pair of charcoal and methanol. Artificial neural network (AAN) has been used by Laidi and Hanini (2013) to predict the solar COP of a solar intermittent refrigerator system for ice production working with activated carbon (AC)/methanol pair. Various researchers presented recently modeling and simulation studies of solar cooling system (Hassan *et al*., 2011; Habib *et al*., 2013; Rouf *et al*., 2013).

A tropical country like Senegal has prospect in utilizing solar energy as a driving source for adsorption refrigeration and air conditioning system. In preservation of medicine and food at the rural and remotes places, solar heat driven adsorption machine could play a major roll for Senegal. From this point of view, Metcalf *et al*. (2011) presented a feasibility study of a solar powered adsorption cycle under climatic condition of Dakar.

From the above perspective, the present study investigates the effect of some important optical parameters on the performance of a solar powered adsorption refrigerating machine using zeolite-water pair. In this study, a realistic theoretical simulation model has been introduced for the main component of

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the solar heat driven adsorption refrigerator. The mathematical model is established taking into account the variation of ambient temperature and solar irradiation along the day.

MATHEMATICAL MODEL

The adsorbent bed of the flat plate solar refrigerator, which is shown in Fig. 1, is the key to solving the model. Through analyzing and calculating the heat transfer in the adsorbent bed, the performance of solar adsorption refrigeration, such COP can be obtained. The adsorbent bed consists of zeolite-water working pair. It is assumed that temperature, pressure and concentration throughout the adsorbent bed are uniform.

Assuming one-dimensional heat transfer through the system layers, neglecting thermal capacity and temperature drop across the glass cover, under steady-state conditions and a solar collecting area of one square meter, the energy balance equations are describe as follows.

As shown in Fig. 1, the glass cover absorbs heat from the solar radiation, thermal radiation emanating from the absorber plate and from the outside environment and natural convection between the absorber plate and the glass cover, while it loses heat due to convection of ambient air and infrared radiation. All heat transfer processes result in the changing of the temperature of the glass cover. The heat balance for the glass cover is given by the following equation:

$$2\varepsilon_g \sigma T_g^4 = \alpha_g E + \alpha'_g \sigma T_p^4 + \sigma T_a^4 - (h_{r,g-a} + h_w)(T_g - T_a) - h_{r,p-g}(T_p - T_g) \quad (1)$$

$$\sigma T_p^4 = \tau_g E + \rho_g \sigma T_p^4 + \varepsilon_g \sigma T_g^4 - h_{r,p-g}(T_p - T_g) - h_{c,p-g}(T_p - T_g) - h_{p-i}(T_p - T_i) \quad (2)$$

$$COP = \frac{Q_{ev}}{Q_s} = \frac{Q_{ev}}{E\ A_s} \quad (3)$$

$$COP_m = \frac{T_{ev}(T_p - T_a)}{T_p(T_a - T_{ev})} \quad (4)$$

$$\eta = \frac{COP}{COP_m} \quad (5)$$

The radiation heat transfer coefficient and are given by Eq. (1 and 2) and the convection heat transfer coefficient by the non-dimensional correlation, Eq. (3 and 4) and the convection heat transfer coefficient for outer wind by Eq. (5).

The absorber plate gets the solar radiation heat, emissive power from the surrounding, while at the same time it transfers heat to the glass cover through natural convection, to the surrounding environment through radiation and to the insulation through conduction. The energy balance for the absorber plate can be expressed as.

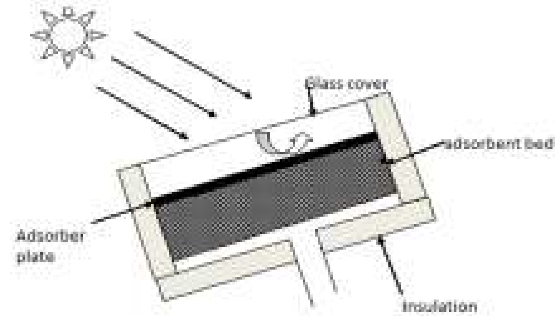


Fig. 1: Sketch structure of the flat-plate collector for the solar refrigeration

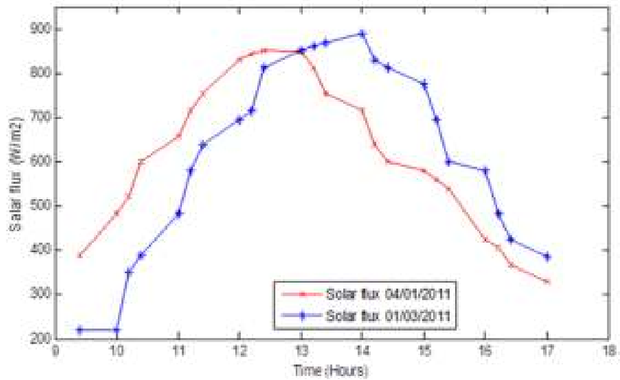
The parameter used in this present work to assess the adsorption refrigerating system performance is the solar Coefficient of Performance (COP). This parameter is defined as the ratio between the total heat extracted by evaporation of the desorbed mass of vapor and the total incident global irradiance on the surface of solar collector during the whole day, it is expressed by the following formula.

The COP of a typical adsorption chiller can be expressed by assuming an ideal refrigeration machine operating in a Carnot cycle. The Carnot coefficient of performance of an ideal heat-driven refrigerating machine is given by the following expression (Kim and Infante Ferreira, 2008).

Which limits the maximum COP achievable with any real heat-driven refrigeration machine working at the same temperatures? Because the heat source temperature varies in different projects, the performance of a real engine is often compared to that of a Carnot cycle working at the same temperatures. The efficiency ratio (η) is defined as the ratio of the coefficient of performance to the Carnot coefficient of performance (COP_m).

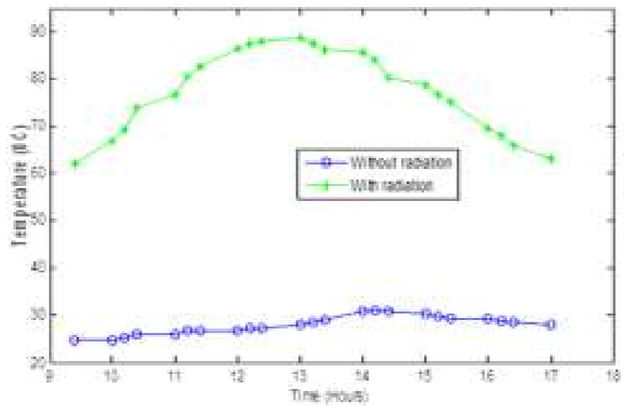
RESULTS AND DISCUSSION

The values of the glass cover temperature and the absorber plate temperature have been obtained by iterative technique of Newton-Raphson (Douglas, 2001) to solve the heat balance Eq. (1) and (2). The tolerance for all the convergence criteria is 10^{-4}. A numerical program, written in FORTRAN, has been developed in order to simulate the behavior of the solar adsorption cooling system. In our simulation, we have used the climatic data measured for two clear type days from 09:00 to 17:00 on January 04, 2011 and March 03, 2011. The main data input to the developed numerical program and used in this simulation are depicted in Table 1 and the range of variables is given in Table 2. Figure 2 shows the variation of the solar incident radiation and the ambient atmospheric temperature with the day hours respectively. Both solar radiation and ambient temperature changed with time and solar radiation reached its highest value at noon in January and around 14:00 in March for the solar radiation,



(a) (b)

Fig. 2: Variation of climatic data with the day time; (a) solar radiation and (b) ambient temperature



(a) (b)

Fig. 3: Temperature with and without radiation; (a) the glass cover and (b) the absorber plate (04/01/2011)

Table 1: Main parameters used in this study

Symbol parameter	Value	Unit
Glass cover		
Glass cover	0.050	[-]
Glass cover emittance	0.880	[-]
Absorber plate		
Absorber plate emittance	0.950	[-]
Absorber plate thermal conductivity	1.750	[W/mK]
Absorber plate thickness	0.050	[m]
Insulation		
Insulating material thermal conductivity	0.003	[W/mK]
Insulation thickness	0.040	[m]
Air gap		
Air gap spacing	0.025	[m]
Air thermal conductivity	0.023	[W/mK]

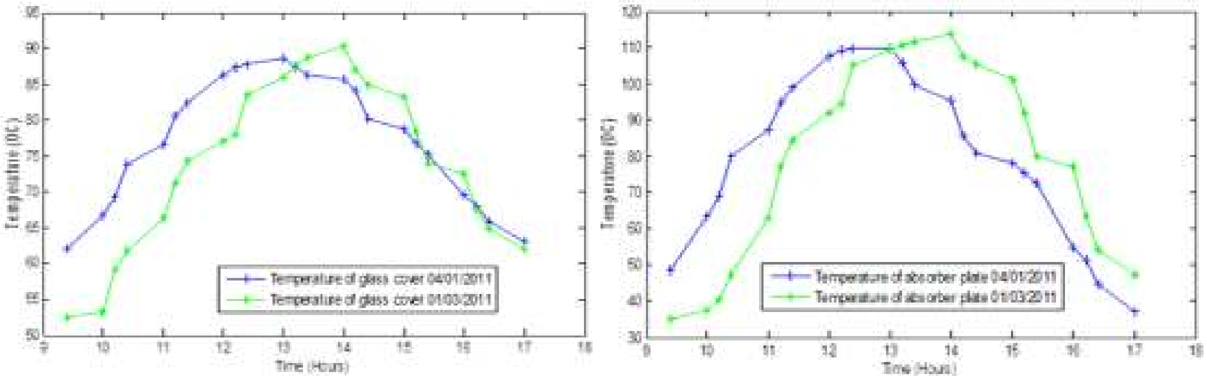
Table 2: Range of variables

Variables	Range
Glass cover transmissivity	0.90-1.0
Glass cover absorptivity for radiation emitted by absorber plate (wavelength more than 2.5)	0.90-1.0
Glass cover reflectivity	0.05-0.30

however, the ambient temperature reached its highest value around 14:00 for both months.

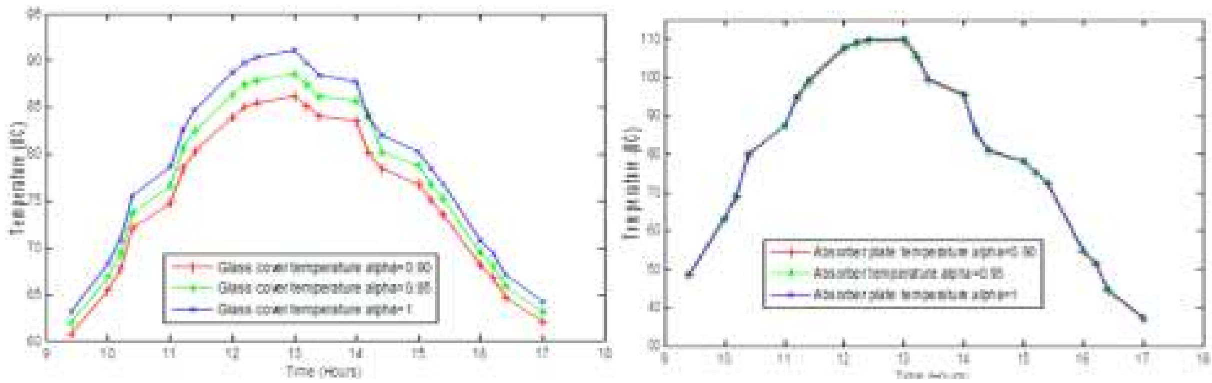
The effect of the heat transfer by radiation is depicted in Fig. 3. It can be seen that both temperature of glass cover and absorber plate decrease when thermal radiation exchange is neglected. It can be observed that the effect of radiation heat transfer on the glass cover temperature is bigger than that of the absorber plate temperature. This means that the radiation is the dominant heat exchange compared with the convective term. In fact the convective heat transfer between the glass and absorber is driven natural convection of air between two parallel planes. The outside convective heat transfer exchange on the glass cover is dependent on wind speed. Consequently, the higher the wind speed on the glass cover is the higher the convective term is and therefore the lower the glass temperature is Khoukhi and Maruyama (2006). In this way by neglecting the radiation we deprive the glass of an important heat input provided by infrared radiation re-emitted by the absorber plate which is at higher thermal potential. This means that the net radiative exchange between the cover glass and the surroundings is high compared with the convective exchange.

Figure 4 represents the cover glass temperature and the absorber plate temperature changes during the day respectively. It is clear that the temperature change follows that of the corresponding ambient temperature. It can be seen that the maximum temperature of the



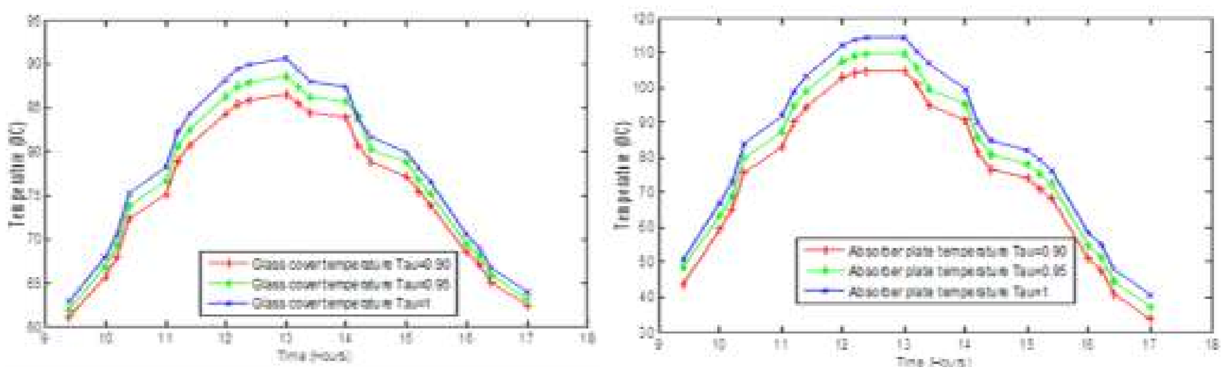
(a) (b)

Fig. 4: Daily temperature changes of; (a) the glass cover and (b) the absorber plate



(a) (b)

Fig. 5: Temperature of; (a) the glass cover and (b) the absorber plate for different values of the absorption coefficient for long-wave radiations of the glass cover (04/01/2011)

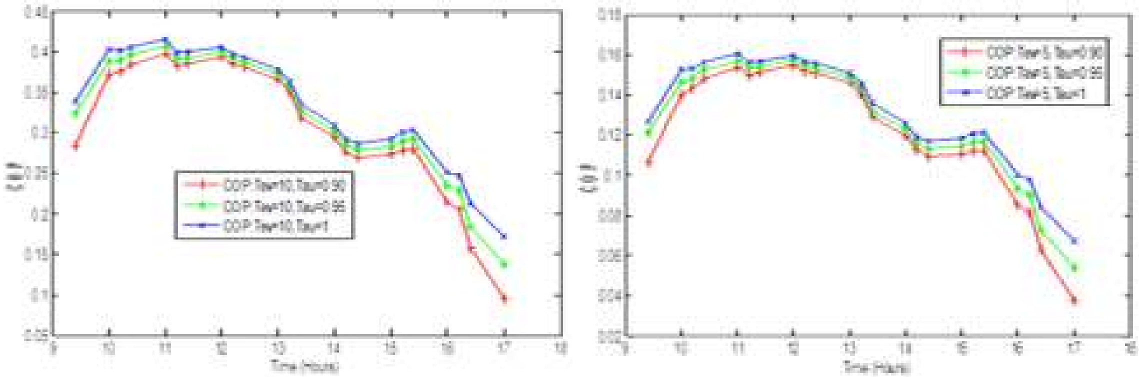


(a) (b)

Fig. 6: Temperature of; (a) the glass cover and (b) the absorber plate for different values of the coefficient of transmission of the glass (04/01/2011)

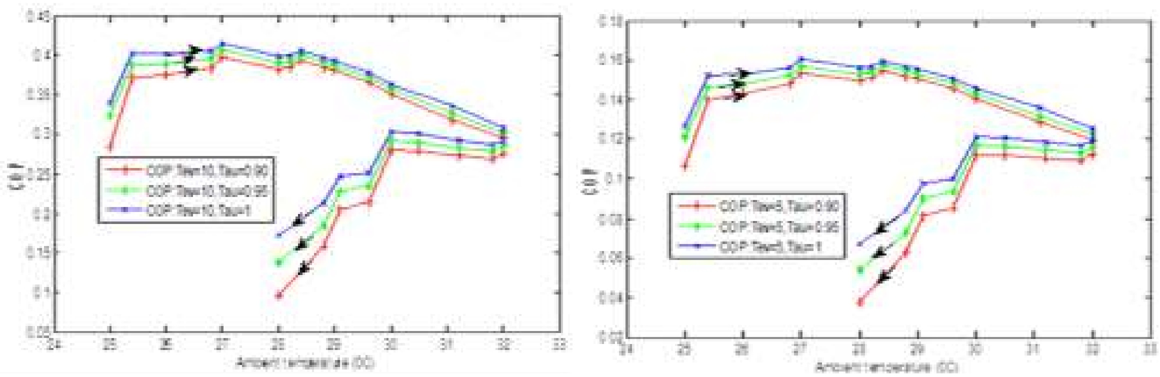
absorber plate is achieved from 12 to 14 and the temperature is around 110°C.

Figure 5 illustrates the effect of absorption for long-wave radiations of glass cover on the temperatures of the cover glass and the absorber plate. It was observed the glass cover temperature increased with the coefficient of absorption and its influence is very important between 12:00 and 14:00. If we notice Eq. (1), it is seen that the temperature of the cover glass is strongly influenced by the absorption coefficient. At the same time, no obvious variation could be observed for the absorber plate temperature, meaning that the



(a) (b)

Fig. 7: Variation of COP according to the time during the day; (a) 10°C and (b) 5°C (04/01/2011)



(a) (b)

Fig. 8: Variation of COP according to the ambient temperature during the day; (a) 10°C and (b) 5°C (04/01/2011)

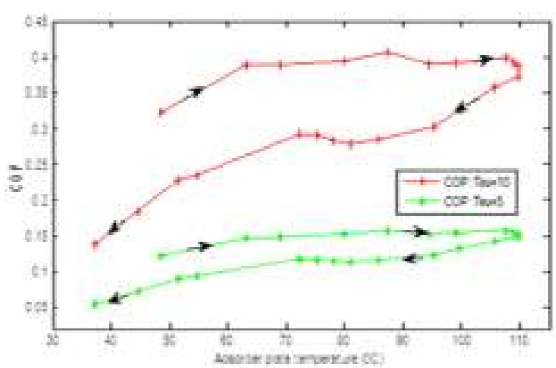


Fig. 9: Variation of COP according to the temperature of the absorbing plate during the day (04/11/2011)

absorber plate is less sensible to the absorption coefficient of the glass cover.

The effect of the transmissivity of the cover glass on the temperatures of the cover glass and absorber plate is indicated on Fig. 6. It is noticed that cover glass temperature and absorber plate temperature increased with the increase of the coefficient of transmission of the glass. This effect is more pronounced between 12:00 and 14:00.

Figure 7 displays the effects of the transmissivity on the coefficient of performance. It is seen from this figure that the simulated COP increases slightly with the transmissivity. It was observed a rapid increase of the COP and its maximum value is reached in a time period of about 2 h. It stays practically constant for the same amount of time because of the sufficient solar radiation and suddenly starts to decline due to the reduction of the solar radiation.

The variations of COP with ambient temperature and absorber plate temperature are given in Fig. 8 and 9. It was observed that the COP starts to decrease in the afternoon when the ambient temperature and the solar radiation are decreasing. It can also be observed from these figures that the values of COP increase with evaporator temperature. As seen from the Eq. (4), the performance of the Carnot cycle gets better with increasing evaporator temperature. Solar energy is a typical variable heat source; furthermore, the adsorption refrigeration is characteristic of period and variable behavior (Zhai and Wang, 2009). The COP varied widely during the day due to the refrigerator operating in transient regime. Similar result was obtained experimentally by Syed *et al*. (2005). The low COP at the beginning and end of the day can be explained from the fact that the cooling capacity at this time was extremely low compared with the generator heat load due to the transient behavior exhibited by the refrigerator and its heat losses. In the morning, the irradiation from the sunrise to 9:00 is necessary to overcome thermal inertia in the system. From then until 15:30, the flat-plate collector supplied heat to the generator. From 15:30 to sunset, the insulation was insufficient to add heat to the generator. Therefore, a brutal drop of COP was observed when the ambient temperature and irradiation start to decrease.

CONCLUSION

In this study a mathematical modeling of thermal solar flat-plate collector based on its physical model was established and then used to compute the temperatures of the glass cover and the absorber plate. Also a theoretical investigation of the performance of a solar adsorption refrigerator using this flat-plate solar collector has been conducted. Based on the investigation the following conclusions can be drawn:

- Thermal radiation exchanges play an important role to achieve high temperature of the glass cover.
- Absorption of thermal radiation in glass cover leads to an increase in its temperature, but does not affect the absorber plate temperature. However both temperatures increase with transmission coefficient of glass cover.
- High values of the solar coefficient of performance COP are obtained during the morning when absorber plate temperature and ambient temperature and solar radiation intensity increase. COP also increases with the coefficient of transmission of the glass cover, but the temperature of the evaporator has a more distinct influence on COP values.

NOMENCLATURE

A : Absorber area (m^2)
E : Solar radiation intensity perpendicular to collector surface (W/m^2)
h : Heat transfer coefficient ($W\ m^2/K$)
L : Thickness (m)
Nu : Nusselt number
Pr : Prandlt number
Q : Heat quantity (W)
Re : Reynolds number
T : Temperature (K)
V : Wind speed velocity (m/s)

Greek symbols:

α, α' : Absorptivity
ε : Emissivity
λ : Thermal conductivity (W/mK)
η : Efficiency
ρ : Reflectivity
τ : Transmissivity
σ : Stefan-Boltzmann constant (W/m^2K^4)

Subscripts:

a : Ambient air
c : Convection
ev : Evaporator
g : Glass cover
gap : Air gap between absorber plate and glass cover
i : Insulation
p : Absorbing plate
r : Radiation
s : Solar collector
w : Wind

Abbreviations:

COP : Coefficient of performance
$(COP)_m$: Carnot coefficient of performance

APPENDIX

Individual heat transfer coefficients: The radiation heat transfer coefficient between absorber plate and the glass cover is:

The radiation heat transfer coefficient between the glass cover and the ambient is:

The convective heat transfer coefficient between absorber plate and the glass cover is determined using the equation expressed by Duffie and Beckman (1974) assuming the natural convection of the air between two parallel planes:

Where and are the thermal conductivity and the thickness of the air layer between the glass and the absorber, respectively. The Nusselt number is given by the correlation provided by Bekkouche (2009).

With the Prandl number, is between 0.6 and 100 and Reynolds number the outside convective heat transfer of the glass cover exposed to outside wind is calculated using the empirical equation proposed by McAdams and reported by Duffie and Beckman (1974):

The overall heat transfer coefficient between the absorber plate and the insulation is given by:

The radiation heat transfer coefficient between absorber plate and the glass cover, $h_{r,p-g}$ is:

$$h_{r,p-g} = \frac{\sigma(T_p+T_g)(T_p^2+T_g^2)}{\frac{1}{\varepsilon_p}+\frac{1}{\varepsilon_g}-1} \tag{A1}$$

The radiation heat transfer coefficient between the glass cover and the ambient $h_{r,g-a}$ is:

$$h_{r,g-a} = \sigma\varepsilon_g(T_g+T_a)(T_g^2+T_a^2) \tag{A2}$$

The convective heat transfer coefficient between absorber plate and the glass cover $h_{c,p-g}$ is determined using the equation expressed by Duffie and Beckman (1974) assuming the natural convection of the air between two parallel planes:

$$h_{c,p-g} = N_U \frac{\lambda_{gap}}{L_{gap}} \tag{A3}$$

where λ_{gap} and L_{gap} are the thermal conductivity and the thickness of the air layer between the glass and the absorber, respectively. The Nusselt number N_u is given by the correlation provided by Bekkouche (2009):

$$N_u = 0.023 R_e^{0.8} P_r^{0.4} \quad (A4)$$

With the Prandl number, P_r, is between 0.6 and 100 and Reynolds number $R_e > 5000$. The outside convective heat transfer of the glass cover exposed to outside wind is calculated using the empirical equation proposed by McAdams and reported by Duffie and Beckman (1974):

$$h_w = 5.7 + 3.8\,V \quad (A5)$$

The overall heat transfer coefficient between the absorber plate and the insulation is given by:

$$h_{p-i} = \frac{1}{\frac{L_i}{\lambda_i} + \frac{L_p}{\lambda_p}} \quad (A6)$$

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