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Thermal Aspects of Solar Air Collector

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1. Introduction

The amount of solar radiation striking the earth's surface not only depends on the season, but also depends on local weather conditions, location and orientation of the surface. The average value of this radiation is about 1000 w/m^2 when the absorbing surface is perpendicular to the sun's rays and the sky is clear. There are several methods exist to absorb and use this free, clean, renewable and very long lasting source of energy [1]. The solar collectors are one of the devices which can absorb and transfer energy of the sun to a usable and/or storable form in many applications such as drying the agricultural, textile and marine products as well as the heating of building [2]. There is variety of designs for the solar thermal collectors depending on their applications. For example, parabolic trough solar air collector is widely used in solar power plants where solar heat energy is used to generate electricity [3].

Flat plate solar collectors are the most common types of solar collectors used in many applications such as solar hot water panels to provide hot water or as solar air heater for pre-heating the air in building heating or industrial HVAC systems. In the solar hot water panel, a sealed insulated box containing a black metal sheet, called absorber surface, with built-in pipes is located faced to the sun. Solar radiation heats up water inside pipes, causing it to circulate through the system by means of natural convection. Then, water is delivered to a storage tank located above the collector. Such passive solar water heating devices used widely in hotels and home especially in southern Europe.

Several models of thermal solar flat plate collector are available and generally all of them consist of four major parts:

1. A flat-plate absorber, which absorbs the solar energy
2. A transparent cover(s) that allows solar energy to pass through and reduces heat loss from the absorber
3. A heat-transport fluid (air or water) flowing through the collector (water flows through tubes) to remove heat from the absorber
4. And finally, a heat insulating backing.

The exergy of a system defined as the maximum possible useful work during a process that brings the system into equilibrium with a heat reservoir [3]. Exergy can be destroyed by irreversibility of a process. One of the powerful methods of optimizing complex thermo dynamical systems is to do an exergy analysis, which is called the second law analysis as well. In 1956, Rant [4] proposed the term exergy that was previously developed by Gibbs [5] in 1873. One can find the details of this concept in the thermodynamics literature [6-9]. Saravanan et al. [10] used the exergy tool to analyze the performance of cooling tower. Bejan was the first person who developed and published the governing equations of exergy to solar collectors [11, 12]. Later Londono-Hurtado and Rivera-Alvarez [13] developed a model to study the behavior of volumetric absorption solar collectors and its performance. Their model is used to run a thermodynamic optimization of volumetric absorption solar collectors to maximize the energy output of heat extracted from the collector. Luminosu and Fara [14] used the thermodynamically analysis using exergy method to find out the best flow rate of the test fluid in their experiments.

Altfeld et al. [15,16] claimed that the heat transfer characteristics of the absorber are less important in case of considering a highly insulated solar air heater. Torres-Reyes et al. [17] carried out energy and exergy analysis to determine the optimal performance parameters and to design a solar thermal energy conversion system. Kurtbas and Durmuş [18] analyzed several different absorber plates for solar air heater and found that there was a reverse relationship between dimensionless exergy loss and heat transfer, as well as pressure loss.

2. Mathematical modeling

In this section, a review has been done on the theoretical modeling of several different designs of solar air heaters. Theoretical modeling of a single cover solar air collector is derived comprehensively in this section. In this case, a single air flow between absorber and glass plates assumed to convey the heat of the solar radiation, Fig.1.

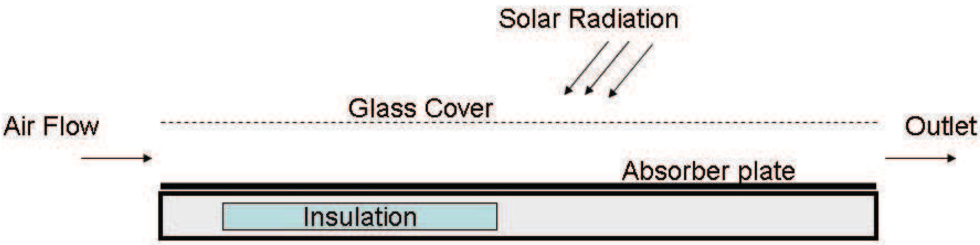

Fig. 1. Schematic view of single air flow solar air heater

The energy balance equations for cover glass, absorber plate and air flow respectively, are as following:

$$\alpha_g S = M_g C_g \frac{\partial T_g}{\partial t} + k_g \delta_g \frac{\partial^2 T_g}{\partial x^2} + h_w (T_g - T_a) + h_{rgs} (T_g - T_s) + h_{cgf} (T_g - T_f) + h_{rpg} (T_g - T_p) \quad (1)$$

$$\alpha_p \tau_p S = M_p C_p \frac{\partial T_p}{\partial t} + k_p \delta_p \frac{\partial^2 T_p}{\partial x^2} + h_{cpf} (T_p - T_f) + h_{rpg} (T_p - T_g) + U_r (T_p - T_a) \quad (2)$$

$$M_f C_f \frac{\partial T_f}{\partial t} + \frac{G_f C_f}{W} \frac{\partial^2 T_f}{\partial x^2} = h_{cgf} (T_f - T_g) + h_{cpf} (T_p - T_f) \quad (3)$$

One may assume following boundary and initial conditions for the given geometry:

$$T_f(t=0)=T_p(t=0)=T_g(t=0)=T_a(t=0) \tag{4}$$

$$\frac{\partial T_g}{\partial x}\Big|_{x=0}=0,\quad \frac{\partial T_g}{\partial x}\Big|_{x=L}=0,\quad \frac{\partial T_p}{\partial x}\Big|_{x=0}=0,\quad \frac{\partial T_p}{\partial x}\Big|_{x=L}=0 \tag{5}$$

3. Back pass solar air heater

Choudhury and Garg [19], Ong[20], Hegazy[21], Al-Kamil and Al-Ghareeb[22] investigated the heat transfer in the back pass solar air collectors. In such solar collectors, the absorber plate is placed behind the glass cover with a gap filled with static air from the glass cover. The flow of air is happens between the bottom surface of the absorber and the top surface of insulated surface, Figure 2.

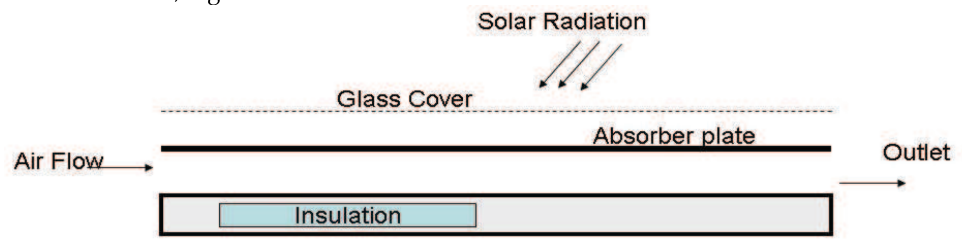

Fig. 2. Schematic view of back pass solar air heater

Garg et al. [23] developed the following mathematical modeling for the glass cover, absorber plate, air flow and the bottom plate, respectively as:

$$h_{gs}(T_s-T_g)+h_{rpg}(T_p-T_g)=U_t(T_g-T_a) \tag{6}$$

$$(\alpha\tau)S=h_{cpf}(T_p-T_{fm})+h_{rps}(T_p-T_a)+h_{rpb}(T_p-T_b)+h_{rpg}(T_p-T_g) \tag{7}$$

$$h_{cpf}(T_p-T_{fm})=h_{cbf}(T_{fm}-T_b)+mC_a(T_{fo}-T_a) \tag{8}$$

$$h_{rpb}(T_p-T_b)=h_{cbf}(T_b-T_{fm})+U_b(T_b-T_a) \tag{9}$$

4. Parallel pass solar air heater

The design of the parallel pass solar air heater is to optimize the heat transfer transportation in the solar air heaters. This design consists of a glass cover, absorber plate and bottom insulated plate. There are two air flow channels which one is located between the glass cover and absorber and another one is located between the absorber and the bottom insulated plate, Figure 3.

Jha et al. [24] obtained the energy transport equations for the glass cover, channel 1, absorber plate, channel 2 and finally, bottom insulated plate as following

$$M_gC_g\frac{\partial T_g}{\partial t}=\alpha_gS+h_{rpg}(T_p-T_g)+h_{cf1g}(T_{f1}-T_g)-h_{cgw}(T_g-T_w)-h_{rga}(T_g-T_a) \tag{10}$$

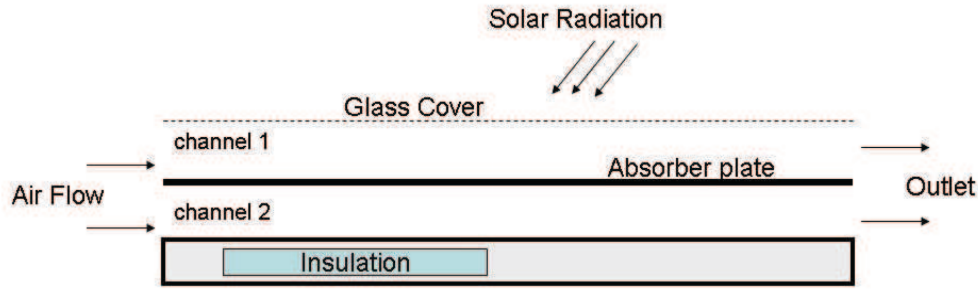

Fig. 3. Schematic view of parallel pass solar air heater

$$M_{f1}C_f \frac{\partial T_{f1}}{\partial t} = \frac{-G_1 C_f}{W}\frac{\partial T_{f1}}{\partial x} + h_{cpf1}(T_p - T_{f1}) - h_{cf1g}(T_{f1} - T_g) \tag{11}$$

$$M_1 C_p \frac{\partial T_p}{\partial t} = \alpha_p \tau_g S - k_p \delta_p \frac{\partial^2 T_p}{\partial x^2} - h_{rpg}(T_p - T_g) - h_{cpf2}(T_p - T_{f2}) - h_{rpb}(T_p - T_b) - h_{cpf1}(T_p - T_{f1}) \tag{12}$$

$$M_{f2}C_f \frac{\partial T_{f2}}{\partial t} = \frac{-G_2 C_f}{W}\frac{\partial T_{f2}}{\partial x} + h_{cpf2}(T_p - T_{f2}) - h_{cbf2}(T_b - T_{f2}) \tag{13}$$

$$M_b C_b \frac{\partial T_b}{\partial t} = -k_b \delta_b \frac{\partial^2 T_b}{\partial t} + h_{rpb}(T_p - T_b) - h_{cbf2}(T_b - T_{f2}) - h_b(T_b - T_r) \tag{14}$$

Along with following boundary and initial conditions

$$\frac{\partial T_p}{\partial x}|_{x=0} = 0\ ,\ \frac{\partial T_p}{\partial x}|_{x=L} = 0,\ \frac{\partial T_b}{\partial x}|_{x=L} = 0 \tag{15}$$

$$T_{f1}(x=0) = T_{fi},\ \ T_{f2}(x=0) = T_{fi} \tag{16}$$

5. Double-pass solar air heater

Ho et al. [25] developed the mathematical modeling for the double pass solar air heater. In such solar collectors, the air is allowed to move faster than the simple collectors. In his mathematical modeling, following assumptions are considered:

- Temperatures of absorbing plate, bottom insulated plate and the fluid are only functions of air flow direction
- Glass cover and absorber plate do not absorb radiant energy
- The radiant energy absorbed by the outlet cover is negligible

Considering above assumptions, energy equations of cover 1, absorber plate, bottom plate, channels *a* and *b* are as following, respectively:

$$h_{rpg2}(T_p - T_{g1}) + h_1'(T_b(z) - T_{g1}) = U_{g1s}(T_{g1} - T_s) \tag{17}$$

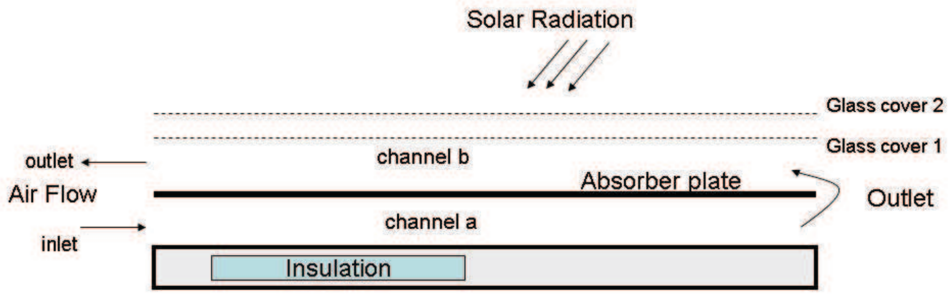

Fig. 4. Schematic view of double- pass solar air heater

$$\alpha_p \tau_{g1} \tau_{g2} S - h_T (T_p - T_s) + h_B (T_p - T_s) + h_1 (T_p - T_b(x)) + h_2 (T_p - T_a(x)) \tag{18}$$

$$h_{rpR}(T_p - T_R) + h_2'(T_a(x) - T_R) = U_{B1s}(T_p - T_s) \tag{19}$$

$$h_2(T_p - T_a(x)) - h_2'(T_a(x) - T_R) = \left[\frac{(R+1)mC_p}{W}\right]\frac{dT_a(x)}{dx} \tag{20}$$

$$h_1(T_p - T_b(x)) - h_1'(T_b(x) - T_{g1}) = \left[-\frac{(R+1)mC_p}{W}\right]\frac{dT_b(x)}{dx} \tag{21}$$

6. Energy analysis

According to Duffie and Beckman [26], the rate of heat received by the air from the collector is

$$\dot{Q}_f = \dot{m}_f C_p (T_o - T_i) \tag{22}$$

One can find the specific collector power by dividing of the heat by the area of the collector [26]

$$\dot{q}_f = \dot{Q}_f / A_{col} = \dot{m}_f C_p (T_o - T_i) / A_{col} \tag{23}$$

The average daily useful heat per unit area of the collector can be obtained by adding up all radiation times as

$$q_{a,d} = 3600 \sum_{i=1}^{8} \dot{q}_{a,hourly} \tag{24}$$

in which $\dot{q}_{a,hourly}$ is the useful heat per unit area of the collector for the hour i of the test day. The energy efficiency of solar energy can be obtained by the ratio of absorbed energy to the total energy of the sun [14].

$$\eta_{en} = \dot{q}_f / G_{col} \tag{25}$$

where G_c is the solar radiation captured by the collector in w / m^2

7. Exergy analysis

According to Bejan et al. [11] and Bejan [12], there are two main sources of entropy generation in a solar air collector, one due to the friction of passing fluid, and the other one due to the thermal heat transfer or temperature change of air. Esens's [22] considered following assumptions to derive the exergy balance equations:

1. The process is steady state and steady flow.
2. The potential and kinetic energies are negligible.
3. Air is an ideal gas, so its specific heat is constant.
4. The humidity of air is negligible.

The general exergy balance for a steady state and steady flow process is

$$\dot{I} = \dot{E}x_{heat} - \dot{E}x_{work} + \dot{E}x_i - \dot{E}x_o \tag{26}$$

Considering above assumptions, following relations are defined for the mentioned terms:

$$\dot{E}x_{heat} = \sum (1 - \frac{T_0}{T_s})\dot{Q}_s \tag{27}$$

$$\dot{E}x_{work} = 0 \tag{28}$$

$$\dot{E}x_i = \sum \dot{m}_{in}[(h_{in} - h_0) - T_0(s_{in} - s_0)] \tag{29}$$

$$\dot{E}x_o = \sum \dot{m}_{out}[(h_{out} - h_0) - T_0(s_{out} - s_0)] \tag{30}$$

Eq (28) comes from the fact that there is no work has done during the process. From mass balance we have

$$\sum \dot{m}_i = \sum \dot{m}_o = \dot{m}_a \tag{31}$$

Upon substitution of Eqs (27) to (31) in Eq (26), one can find the rate of irreversibility as

$$\dot{I} = (1 - \frac{T_a}{T_s})\dot{Q}_s - \dot{m}_f[(h_o - h_i) - T_a(s_o - s_i)] \tag{32}$$

$\dot{Q}_s$ is the total rate of the exergy received by the collector absorber area from the solar radiation and is evaluated by this relation

$$\dot{Q}_s = G_c.A_{col}.\tau\alpha \tag{33}$$

where $\tau\alpha$ is absorbance-transmittance product of the covering glass and the absorber plate. The changes in enthalpy and entropy of test liquid, air, in the collector can be obtained using following two expressions:

$$h_o - h_i = C_p(T_o - T_i) \tag{34}$$

$$s_o - s_i = C_p \ln(\frac{T_o}{T_i}) - R\ln(\frac{P_o}{P_i}) \tag{35}$$

One can find the final form of irreversibility expression by substituting Eqs (33) to (35) in Eq (32) as

$$\dot{I} = (1 - \frac{T_a}{T_s})G_{col}.A_{col}.\tau\alpha - \dot{m}_f C_p (T_o - T_i) + \dot{m}_f T_a C_p \ln(\frac{T_o}{T_i}) - \dot{m}_f T_a R \ln(\frac{P_o}{P_i}) \qquad (36)$$

T_s is apparent temperature of sun surface, which considered to be 6000 K. In this equation the first term comes from the entropy generated due to heat transfer; second and third terms are related to the temperature change of air and last term is related to the entropy generated due to the friction of fluid. By definition, irreversibility is the total entropy generated times the ambient temperature [8]

$$\dot{I} = \dot{S}_{gen} T_a \qquad (37)$$

Therefore, the total entropy generated during the process will be

$$\dot{S}_{gen} = \frac{1}{T_a}[(1 - \frac{T_a}{T_s})G_{col}.A_{col}.\tau\alpha - \dot{m}_f C_p (T_o - T_i)] + \dot{m}_f C_p \ln(\frac{T_o}{T_i}) - \dot{m}_f R \ln(\frac{P_o}{P_i}) \qquad (38)$$

According the second law of thermodynamics, the exergy efficiency defined as [17]

$$\eta_{ex} = 1 - \frac{\dot{I}}{\dot{E}x_{heat}} = 1 - \frac{\dot{S}_{gen} T_a}{(1 - T_a / T_s)\dot{Q}_s} \qquad (39)$$

8. Error analysis

There are two types of errors in doing the energy and exergy analysis;
One group comes from direct measurement, such as ΔG_c , ΔT , ΔP and $\Delta \dot{m}$

The second group of errors comes from indirect measurement, which are $\Delta \eta_{en}$ and $\Delta \eta_{ex}$. Luminosu and Fara [14] proposed following relations for error analysis:

$$\Delta \eta_{ex} = \Delta \dot{I} / \dot{E}x_{heat} + \dot{I} \Delta \dot{E}x_{heat} / \dot{E}x_{heat}^2 \qquad (40)$$

$$\Delta \eta_{en} = \Delta \dot{q}_a / G_c + \dot{q}_f \Delta G_{col} / G_{col}^2 \qquad (41)$$

where each error term may be computed as following:

$$\Delta \dot{E}x_{heat} = (\Delta T / T_s + T_a \Delta T / T_s^2) A_{col} (\tau\alpha) G_{col} + (1 - T_a / T_s) A_{col} (\tau\alpha) \Delta G_{col} \qquad (42)$$

$$\Delta \dot{I} = T_a \Delta \dot{S}_{gen} + \dot{S}_{gen} \Delta T \qquad (43)$$

$$\begin{aligned} \Delta \dot{S}_{gen} = & [R \ln(P_o / P_i) + C_p \ln(T_i / T_o) + C_p (T_o + T_i) / T_a] \Delta \dot{m} + G_{col} A_{col} (\tau\alpha) \Delta T / T_a^2 \\ & + \dot{m} C_p [1 / T_o + 1 / T_i + 2 / T_a + (T_o + T_i) / T_a^2] \Delta T + \dot{m} R [1 / P_o + 1 / P_i] \Delta P \\ & + A_{col} (\tau\alpha) [1 / T_s + 1 / T_a] \Delta G_{col} \end{aligned} \qquad (44)$$

$$\Delta \dot{q}_a = C_p[\Delta \dot{m}(T_o + T_i) + 2\dot{m}\Delta T] / A_{col} \quad (45)$$

9. Nomenclature

A	Absorbing area [m^2]
C	Specific heat [kJ/kg.K]
$\dot{E}x$	Rate of exergy [kJ/s]
G	Solar radiation flux [kw / m^2]
h	Enthalpy [kJ/kg]
$\dot{I}$	Irreversibility [kw]
$\dot{m}$	Mass flow rate [kg/s]
p	pressure [Pa]
$\dot{Q}$	Rate of heat energy received [kw]
$\dot{q}$	Ratio of heat energy received by the unit area [kw / m^2]
R	Gas constant for carrier fluid [kJ/kg.K]
$\dot{S}$	Rate of entropy generated [kw/K]
T	Temperature [K]

Greek Letters

η	Efficiency (dimensionless)
$\tau\alpha$	Absorbance-transmittance product (dimensionless)
Δ	Error in measuring or calculation

Subscripts

a	Ambient
f	Carrier fluid (air)
c	convection
col	Collector
g	Glass cover
d	Daily
en	Energy
ex	Exergy
gen	Generated
heat	Heat energy
hourly	Hourly
i	Inlet flow of carrier fluid
o	Outlet flow of carrier fluid
p	Absorber plate
s	Sun
work	Work

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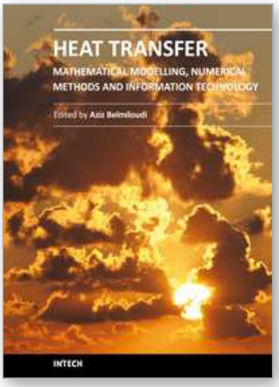

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Over the past few decades there has been a prolific increase in research and development in area of heat transfer, heat exchangers and their associated technologies. This book is a collection of current research in the above mentioned areas and describes modelling, numerical methods, simulation and information technology with modern ideas and methods to analyse and enhance heat transfer for single and multiphase systems. The topics considered include various basic concepts of heat transfer, the fundamental modes of heat transfer (namely conduction, convection and radiation), thermophysical properties, computational methodologies, control, stabilization and optimization problems, condensation, boiling and freezing, with many real-world problems and important modern applications. The book is divided in four sections : "Inverse, Stabilization and Optimization Problems", "Numerical Methods and Calculations", "Heat Transfer in Mini/Micro Systems", "Energy Transfer and Solid Materials", and each section discusses various issues, methods and applications in accordance with the subjects. The combination of fundamental approach with many important practical applications of current interest will make this book of interest to researchers, scientists, engineers and graduate students in many disciplines, who make use of mathematical modelling, inverse problems, implementation of recently developed numerical methods in this multidisciplinary field as well as to experimental and theoretical researchers in the field of heat and mass transfer.