Modelling of flat plate and V-corrugated solar air heaters operated in natural and forced convection

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Abstract:

A number of analytical and experimental studies have been carried on solar air heaters in order to improve their thermal efficiency. In this paper, various aspects of solar air heaters applied to drying process are investigated. This study presents a mathematical model for simulating the transient processes which occur in solar air collectors with flat plate and V-corrugated absorbers. The proposed model is time-dependent and considers thermo-physical properties and heat transfer coefficients and is based on solving equations which describe the energy conservation in partial differential forms of the components of the system, namely the glass cover, the air gap between the cover and the absorber, the absorber, the working fluid and the insulation.
The differential equations were solved using the implicit finite-difference method and the simulation is carried out using MATLAB program. In order to verify the proposed method, an experiment was conducted in variable ambient conditions and flow rates on a solar air collector with a flat plate galvanized iron absorber.
The comparison between computed and measured results of outlet air temperature shows a satisfactory convergence. The simulation results are also verified with distinguished research results from literature.
The results show that the V-corrugated collector has considerably superior thermal performance than the flat plate collector of about 16.15% for natural convection and 21.64% for forced convection. Furthermore, the model shows that the efficiency of double glazed collector is greater than the single glazed one by 12.11%. A study was made also to optimize the spacing between the absorber and the insulation.

Keywords:

Solar Air Heaters, Flat Plate Collector, V-Corrugated Collector, Thermal Efficiency.

1. Introduction

Solar energy is becoming a valuable renewable energy alternative for the limited fossil fuel resources. Among the simplest and most direct applications of this energy is the conversion of solar radiation into heat, which can be applied in drying process. Hence, the realization of an efficient solar dryer requires a thorough study of its main element, namely the solar air heater that generates thermal energy.
Several designs and improved technologies of solar air heaters have been proposed and discussed in literature. These researches carry out on the type of the air channel (single, double or counter flow channel), the number of glazing, the type of the absorber (flat, corrugated or finned), the operating modes of the solar air heater that are forced or natural convection. For example Karim and Hawlader [1] evaluated, analytically and experimentally, the thermal performance of a V-groove solar air heater. El-Sebaii et al. [2] showed that the V-corrugated is greater than the flat plate absorber by 5% in terms of outlet temperature and by 11% to 14% in terms of efficiency. Singh et al. [3] found that the double glass cover reduces the overall heat loss coefficient by 10-15% compared to the single glass cover. At last but not least AL-KAYIEM et al. [4] investigated theoretically and experimentally a flat plate collector operated in natural convection and optimized the inclination angle of the collector in order to achieve the best collector performance.
The aim of the present paper is to develop a dynamic mathematical model which can be applied to different configurations of solar air heaters, namely flat plate and V-corrugated solar air heaters operated in natural and forced convection in order to assist in the interpretation of the phenomena observed in the collectors, predict trends, and assist in the optimization of suitable devices.

2. Theoretical analysis

This part presents a mathematical model describing the flat plate and V-corrugated solar air collectors (see Fig. 1). In the present model, the analyzed control volume of the collector is divided into different parts which are perpendicular to the air flow direction, namely the glass cover, the air gap between the cover and the absorber, the absorber, the working fluid and the insulation. The governing equations are derived by applying the general energy balance for each element of the collector.

To simplify the analysis, the following assumptions were made:

- One-dimensional heat transfer through the system layers.
- The heat transfer from the collector edges is negligible[1].
- Properties of the glass, the absorber and the insulation are independent of temperature.
- The flowing air temperature is assumed to vary only in the direction of the flow.
- The sky can be considered as a black body for long-wavelength radiation at an equivalent sky temperature.

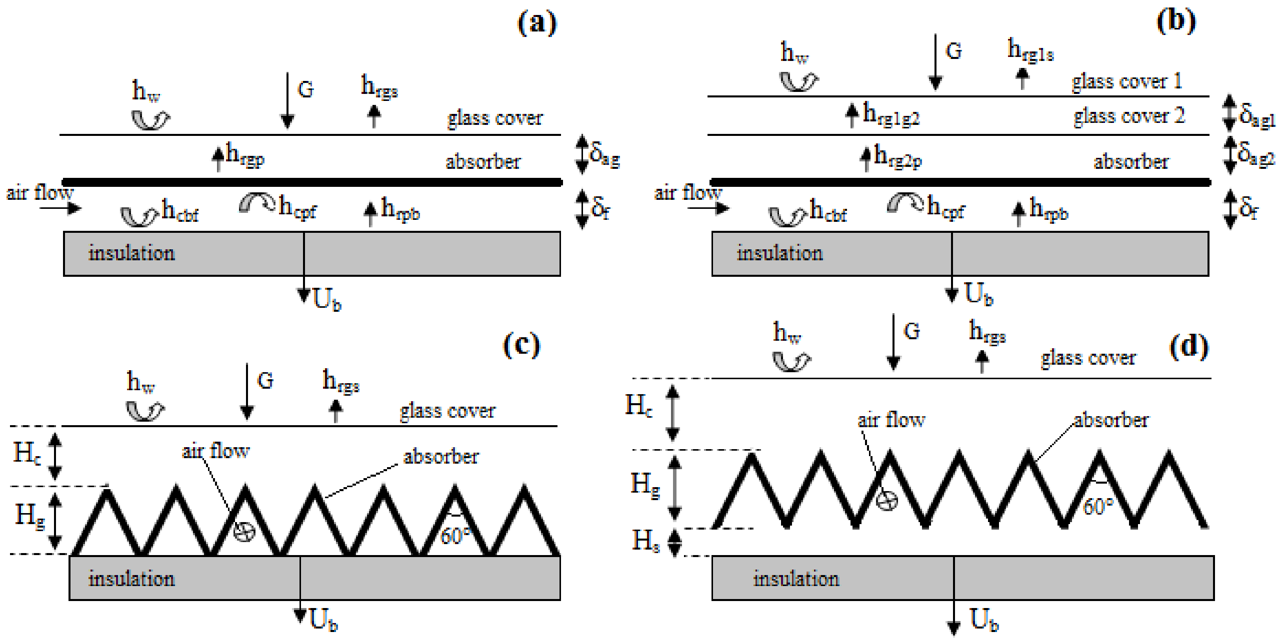

Fig. 1. Solar air heaters: a) Flat plate, b) Double glazed flat plate, c) V-corrugated, d) V-corrugated with spacing between the absorber and the insulation

2.1. Forced convection

2.1.1. Mathematical modelling of flat plate solar air heater

The considered solar air heater configuration is shown in Fig. 1a.

2.1.1.1. Energy balance

The governing equation that describes the glass cover is

$$\rho_g \delta_g C_g \frac{dT_g}{dt} = \alpha_g G + (h_w + h_{rgs})(T_a - T_g) + h_{cg(ag)}(T_{ag} - T_g) + h_{rgp}(T_p - T_g)\,. \qquad (1)$$

By analyzing the air gap between the cover and the absorber, the energy balance equation may be written as follows

$$\rho_{ag}(T_{ag})\delta_{ag} C_{ag}(T_{ag}) \frac{dT_{ag}}{dt} = h_{cg(ag)}(T_g - T_{ag}) + h_{c(ag)p}(T_p - T_{ag})\,. \quad (2)$$

The energy balance equation for the absorber is given by (3)

[1] Duffie and Beckman [12] reported that the edge losses of well-constructed large collector arrays are usually negligible.

$$\rho_p \delta_p C_p \frac{dT_p}{dt} = r\alpha_p \tau_g G + h_{c(ag)p}(T_{ag} - T_p) + h_{rgp}(T_g - T_p) + h_{cpf}(T_f - T_p) + h_{rpb}(T_b - T_p)\,, \qquad (3)$$

where r is the rate of the aperture area of the absorber on which the solar radiation falls, in this case $r=1$.

The energy balance under transient properties for the working fluid can be written as

$$\rho_f(T_f)\delta_f C_f(T_f)\frac{\partial T_f}{\partial t} = h_{cpf}(T_p - T_f) + h_{cbf}(T_b - T_f) - \frac{m_f C_f(T_f)}{W}\frac{\partial T_f}{\partial x}\,. \quad (4)$$

For the insulation, $\rho_b \delta_b C_b \frac{dT_b}{dt} = h_{rpb}(T_p - T_b) + h_{cbf}(T_f - T_b) + U_b(T_a - T_b)\,.$ (5)

2.1.1.2. Heat transfer coefficients

The heat transfer coefficients are calculated using familiar methods and formulas available in the literature.

The convection heat transfer between the glass cover and the wind is proposed by Mac Adams [5] $h_w = 5.67 + 3.86V\,.$ (6)

The radiation heat transfer coefficient from the glass cover to the sky can be stated as following $h_{rgs} = \frac{\sigma\varepsilon_g(T_g^4 - T_s^4)}{(T_g - T_a)}$, (7) where the sky temperature T_s is estimated by the formula given by Swinbank [6] $T_s = 0.0552 T_a^{1.5}\,.$ (8)

The radiation heat transfer coefficient between the glass cover and the absorber is proposed by Duffie and Beckman [7] as $h_{rgp} = \frac{\sigma(T_p + T_g)(T_p^2 + T_g^2)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_g} - 1}\,.$ (9)

The convection heat transfer coefficient in the air gap between the cover and the absorber is calculated according to Hollands et al. [8] $h_{cg(ag)} = h_{c(ag)p} = \frac{Nu_{ag}K_{ag}}{\delta_{ag}}$, (10) where the Nusselt number[2] is expressed by $Nu_{ag} = 1 + 1.44\left[1 - \frac{1708}{Ra\cos\theta}\right]^{+}\left(1 - \frac{(\sin 1.8\theta)^{1.6}1708}{Ra\cos\theta}\right) + \left[\left(\frac{Ra\cos\theta}{5830}\right)^{\frac{1}{3}} - 1\right]^{+}$, (11) and the Rayleigh number equals to $Ra = \frac{g|T_g - T_p|\delta_{ag}^3}{T_{ag}\upsilon_{ag}\alpha_d}\,.$ (12)

The convection heat transfer coefficient between the absorber and the air flow is expressed as follows $h_{cpf} = h_{cbf} = \frac{Nu_f K_f}{D_h}$, (13) with $D_h = 2\delta_f$, (14)

- For laminar air flow $Re_f < 2100$, the Nusselt number may be calculated from the equation proposed by the Mercer et al. [9] $Nu_f = 4.9 + \frac{0.0606(\mathrm{Re}_f \Pr_f D_h / L)^{1.2}}{1 + 0.0909(\mathrm{Re}_f \Pr_f D_h / L)^{0.7}\Pr_f^{0.17}}\,.$ (15)
- For turbulent air flow $Re_f \geq 2100$, the Nusselt number may be calculated from the equation below proposed by Kays and Grewford [10] $Nu_f = 0.0158\,\mathrm{Re}_f^{0.8}\,.$ (16)

[2] The "+" symbol in the superscript means that only positive values of the terms in the square brackets are to be used.

The radiation heat transfer coefficient between the absorber and the insulation is proposed by Duffie and Beckman [7] $h_{rpb} = \dfrac{\sigma(T_p + T_b)(T_p^2 + T_b^2)}{\dfrac{1}{\varepsilon_p} + \dfrac{1}{\varepsilon_b} - 1}$. (17)

The conduction heat transfer coefficient across the insulation is estimated by $U_b = \dfrac{k_b}{\delta_b}$. (18)

2.1.2. Mathematical modelling of double glazed flat plate collector

The double glazed flat plate solar air heater is represented in Fig. 1b. The modelling of double glazed collector requires two additional formulas to those used for simple glazed one to be considered.

The energy balance for the glass cover 1 is

$$\rho_{g1}\delta_{g1}C_{g1}\frac{dT_{g1}}{dt} = \alpha_{g1}G + (h_w + h_{rg1s})(T_a - T_{g1}) + h_{cg1(ag1)}(T_{ag1} - T_{g1}) + h_{rg1g2}(T_{g2} - T_{g1})\,. \quad (19)$$

The energy balance for the air gap1 between the two glass covers is

$$\rho_{ag1}(T_{ag1})\delta_{ag1}C_{ag1}(T_{ag1})\frac{dT_{ag1}}{dt} = h_{cg1(ag1)}(T_{g1} - T_{ag1}) + h_{c(ag1)g2}(T_{g2} - T_{ag1})\,. \quad (20)$$

The energy balance for the glass cover 2 is

$$\begin{aligned}\rho_{g2}\delta_{g2}C_{g2}\frac{dT_{g2}}{dt} &= \alpha_{g2}\tau_{g1}G + h_{c(ag1)g2}(T_{ag1} - T_{g2}) + h_{c(ag2)g2}(T_{ag2} - T_{g2}) + h_{rg1g2}(T_{g1} - T_{g2}) \\ &+ h_{rg2p}(T_p - T_{g2})\end{aligned}\,. \quad (21)$$

The energy balance equations of the other components are similar to those given for the simple glazed one.

2.1.3. Mathematical modelling of V-corrugated solar air heater

The V-corrugated collector is represented in Fig. 1c. The angle between V-grooves is 60°.

2.1.3.1. Energy balance

For the air gap $\rho_{ag}(T_{ag})(H_c + \frac{H_g}{2})C_{ag}(T_{ag})\frac{dT_{ag}}{dt} = h_{cg(ag)}(T_g - T_{ag}) + 2h_{c(ag)p}(T_p - T_{ag})$. (22)

For the absorber

$$\begin{aligned}\rho_p\delta_pC_p\frac{dT_p}{dt} &= r\alpha_p\tau_gG + h_{c(ag)p}(T_{ag} - T_p) + h_{rgp}(T_g - T_p) + h_{cpf}(T_f - T_p) \\ &+ h_{rpb}(T_b - T_p)\end{aligned}\,, \quad (23)$$

with r=2/3.

For the working fluid

$$\rho_f(T_f)\frac{H_g}{2}C_f(T_f)\frac{\partial T_f}{\partial t} = 2h_{cpf}(T_p - T_f) + h_{cbf}(T_b - T_f) - \frac{m_fC_f(T_f)}{W}\frac{\partial T_f}{\partial x}\,. \quad (24)$$

The energy balance for the glass cover and the insulation are given by (1) and (5) respectively.

2.1.3.2. Heat transfer coefficients

The convection heat transfer coefficient in the air gap between the cover and the absorber is calculated as follows $h_{cg(ag)} = h_{c(ag)p} = \dfrac{Nu_{ag}K_{ag}}{\left(H_c + \dfrac{H_g}{2}\right)}$, (25) where the Nusselt number is given by (11) and the Rayleigh number equals to $R_a = \dfrac{g\left|T_g - T_p\right|\left(H_c + \dfrac{H_g}{2}\right)^3}{T_{ag}\upsilon_{ag}\alpha_d}$. (26)

The convection heat transfer coefficient between the absorber and the air flow is proposed by Hollands et al. [8] and expressed by (13), with $D_h = \dfrac{2}{3}H_g$. (27)

- For laminar flow $Re_f < 2800$, $Nu_f = 2.821 + 0.126\,\mathrm{Re}_f \dfrac{H_g}{L}$. (28)
- For transition flow $2800 < Re_f < 10^4$, $Nu_f = 1.9\times10^{-6}\,\mathrm{Re}_f^{1.79} + 225\dfrac{H_g}{L}$. (29)
- For turbulent flow $10^4 < Re_f < 10^5$, $Nu_f = 0.0302\,\mathrm{Re}_f^{0.74} + 0.242\,\mathrm{Re}_f^{0.74}\dfrac{H_g}{L}$. (30)

The other heat transfer coefficients are similar to those given for the flat plate solar air heater.

2.1.4. Mathematical modelling of V-corrugated solar air heater with a spacing between the absorber and the insulation

All energy balance equations and heat transfer coefficients are similar to those expressed for the simple V-corrugated collector except that the spacing between the absorber and the insulation is replaced by $H_s+H_g/2$ instead of $H_g/2$ and D_h is replaced by $H_s+H_g/2$ instead of $2/3Hg$.

2.2. Natural convection

2.2.1. Modelling of air mass flow rate

Air mass flow in natural convection mode is mainly determined by two aspects, which are stack pressure built up in the air channel and pressure losses at the inlet, outlet and along the channel. Consequently, natural convection air mass flow [11] can be calculated as

$$m_f = C_d \rho_f A_{in} \sqrt{\frac{2gL\sin(\theta)(T_{fm} - T_a)}{\left(1 + \left(\dfrac{A_{out}}{A_{in}}\right)^2\right)T_a}}. \qquad (31)$$

The mean air temperature in the air channel is calculated according to Hirunlabh et al. [12] $T_{fm} = 0.75T_{fout} + 0.25T_a$. (32) The coefficient of discharge C_d, according to Flourentzou et al. [13] is estimated equal to 0.6.

2.2.2. Mathematical modelling of flat plate and V-corrugated solar air heaters

The mathematical models of these solar air heaters in natural convection mode are similar to those performed for forced convection except that, the convection heat transfer coefficient between the absorber and the air flow is expressed by (13), the Nusselt number by (11) and the Rayleigh number equals to $Ra = \frac{g\left|T_p - T_b\right|\left(D_f\right)^3}{T_f \upsilon \alpha_d}$, (33) with $Df = \delta_f$ for the flat plat collector and $Df = Hg/2$ for the V-corrugated one.

2.3. Thermal efficiency

Thermal performance of collectors is compared by using the concept of thermal efficiency. It is generally believed that the thermal efficiency is the major requirement for the prediction of thermal performance of the complete solar air heater. It is calculated by using the following formula $\eta = \frac{m_f C_f (T_{out} - T_{in})}{LWG}$. (34)

2.4. Numerical solution of the mathematical models

In numerical solving of the energy balance system, governing equations are nonlinear, and the number of unknown variables is large. Under these conditions implicitly formulated equations are almost always solved using iterative techniques. To do so, Systems of differential equations are solved using the implicit finite difference method.

3. Results and discussions

3.1. Relevant parameters employed for numerical calculation

The specifications of the elements that constitute the solar air heaters simulated in MATLAB code are: The glass cover is a low-iron glass that has high transmissivity and small absorption coefficient. The absorber material is cupper with black nickel selective coating. And the chosen material for the insulation is polyurethane foam that has low conductivity which is required to reduce the heat losses from the collector.

The input of the program includes design and operational parameters. Table A.1 and A.2 present respectively the various design and operational relevant parameters used for numerical calculations.

In the proposed numerical method, the thermo-physical properties of the air are temperature-dependent. Based on their values at some temperatures; the cubic spline interpolation is used to determine them at every temperature.

3.2. Convergence study

The theoretical outlet temperature of the solar air heater was evaluated for different numbers of nodes (n = 4, 6, 14, 22, 38 and 80) along the flow direction. The interval of each simulation is 60 min.

Table 1 presents the running time and the mean error of the obtained temperatures compared to the 80 nodes model. Consequently, the results suggest limiting the number of nodes to 22 in order to reduce the running time with acceptable errors from the optimum case of 80 nodes. And so, all simulations in the present paper are made for a node number equal to 22.

Table 1. Range of error and running time for each number of nodes

Number of nodes n	4	6	14	22	38	80
Mean error (%)	2.20	1.48	0.52	0.28	0.12	0
Running time (min)[3]	2.98	3.54	8.2	12.76	26.25	74.20

3.3. Theoretical results of the flat plate solar air heater

The proposed numerical solution is able to predict the instantaneous temperatures of any component at any section of the collector. For example, the predicted temperatures of the collector components

[3] The computer used for simulations has the following properties: Processor Intel Core i5, Installed Memory (RAM) 3.88.

at the middle section are presented in Fig. 2. As it can be recognized from the figure, under constant operating conditions, the response of the system becomes stable after a transitional period.

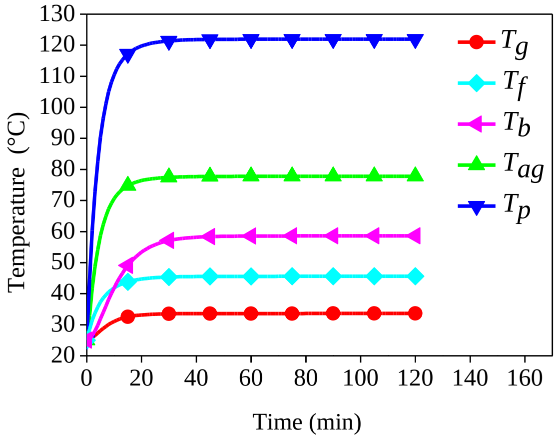

Fig. 2. Temperatures of the flat plat collector's components at the middle cross section.

As it was expected, the temperature of the absorber records the highest value along the running time. Since, the mean function of the absorber is to trap as much as possible of the incident solar radiation transmitted by the upper cover, and to transfer the retained heat by convection to the working air fluid when it flows through the collector.
The aim of the glass cover is to transfer the greater part of the solar radiation received and opposes the heat losses. From the cover temperature presented above, the cover has the lowest variation along time, thus it works efficiently for this aim.

3.4. Comparison between single and double glazed flat plat solar air heaters

Numerical calculations have been performed on tow flat plat solar air heaters, with respectively single and double glass covers, under identical conditions. Fig. 3 shows that, in permanent regime, the outlet temperature of the working fluid and instantaneous efficiency are greater for the double glazed collector than single glazed one by about 8.36°C and 12.11% respectively which is in line with the results of [3]. The reason why the use of the double glazed collector increases the thermal efficiency is that it decreases the thermal losses from the top of the collector.

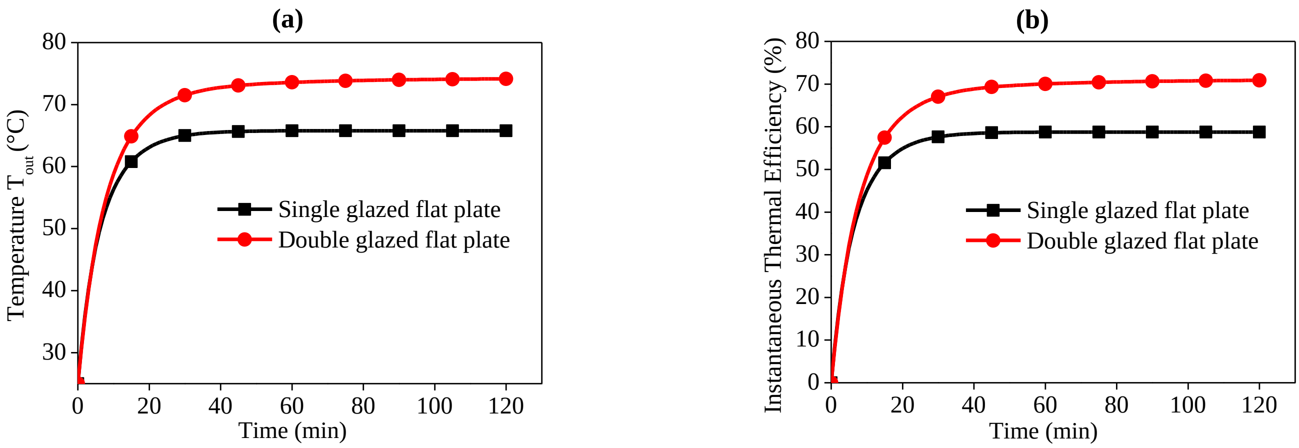

Fig. 3. Single and double glazed flat plate solar air heater: a) Outlet air temperature, b) Instantaneous thermal efficiency.

3.5. Comparison between flat plate and V-corrugated solar air heaters

In order to make the comparison possible between the flat plate and the V-corrugated collectors, the cross sectional areas of air gaps and air channels must be equal for the two types of collector.

The results under identical operating conditions are shown in Fig. 4. It is found that the V-corrugated collector is considerably superior to the flat-plate collector by about 21.64% higher efficiency for forced convection and about 16.15% higher efficiency for natural convection, indicating that the use of the V-corrugated absorber improves significantly the thermal performance of a solar air heater, which is in line with the results of [1,2,14].

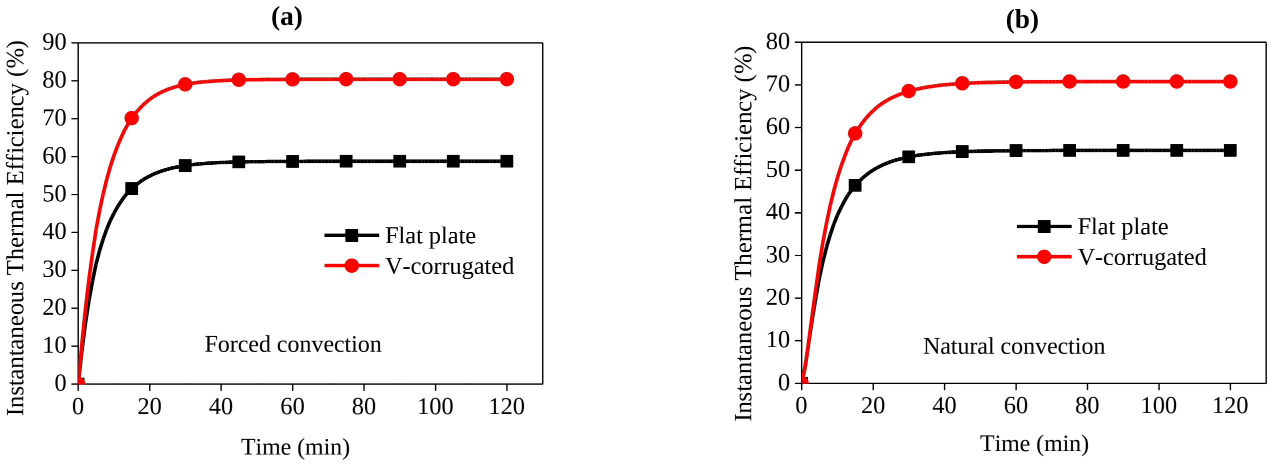

Fig. 4. Instantaneous thermal efficiency of flat plate and V-corrugated solar air heaters: a) Forced convection, b) Natural convection.

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This improvement of thermal efficiency is due to the enhanced convection heat transfer coefficient between the air working fluid and the V-corrugated absorber. In addition to that, the use of the V-corrugated absorber increases the contact surface between it and the working fluid resulting in larger heat gains for the fluid.

3.6. Comparison between different structures of V-corrugated solar air heaters

In this part, the effect of performing a millimetric spacing between the V-corrugated absorber and the insulation is studied (Fig. 1d). To do so, four configurations of V–corrugated solar air heaters with a spacing H_s equal respectively to 0 mm, 2 mm, 4 mm and 6 mm are compared.

And as it was expected, by reducing the spacing H_s see even eliminating it, we can increase the outlet temperature (Fig. 5) and enhance the thermal performance of the collector.

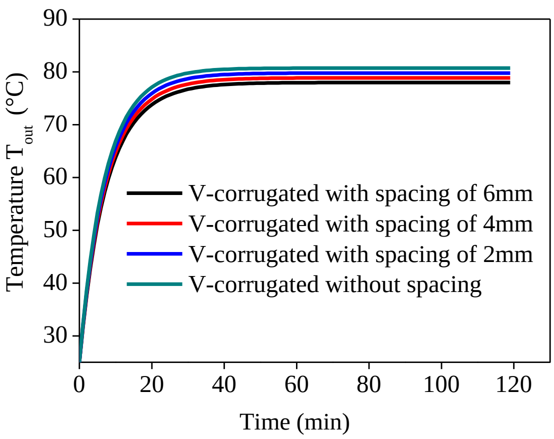

Fig. 5. Outlet air temperature of different structures of V–corrugated solar air heater.

This result can be explained by the fact that making a small spacing of some millimeters between the absorber and the insulation reduces the turbulence of the working fluid.

3.7. Experimental verification

All the experiments that were done to verify the code took place in Marrakech (Morocco), using an existing flat plate solar collector (see Fig. 6). They were conducted on three days of December 2014 with clear sky condition. The ambient temperature and the solar radiation were measured and recorded every 30 minutes from 9h to 17h and were fitted by the following gauss function $f(y) = a + \left(\frac{b}{c\sqrt{\pi/2}} \right) \exp\left(-2 * \left(\frac{(y-d)}{c} \right)^2 \right)$, (35) in which y represents ambient temperature or solar radiation and a,b,c and d are the fitting coefficients. The average wind velocity was 9km/h and the average humidity was 65%.

The experimental apparatus used for this experimental study are:

- Single glazed flat plate collector of 2.5 m² area (2.5 m x 1 m) orientated southward under the angle of 30◦ versus the horizontal. The top cover is an ordinary glass of 3 mm (τ_g=0.9, ε_g=0.85). The absorber is a 0.5 mm galvanized iron painted in black (α_p=0.7, ε_p=0.23, C_p= 444 J/(kg K), ρp=7860 kg/m^3). The insulation in the bottom and the edges of the collector is a 5 mm polyurethane foam (k_b=0.025 W/(m K), ε_b=0.6). The thickness of the air gap between the top glass cover and the absorber is 5 cm and the thickness of the air channel between the absorber and the insulation is 2.5 cm.
- A centrifugal fan (300 m^3/h, 80 mm CE, 220V), with an upstream restrictor which allows to vary the air flow rate from 50 to 300 m^3/h.
- PT 100 platinum thermocouples.
- A pyranometer Kipp & Zonen CM11 for measuring the solar radiation on 30° inclined surface.

Fig. 6. Front view of the utilized flat plat solar air heater.

The solar air heater was operated at flow rates of 300 m^3/h (0.0939 kg/s), 150 m^3/h (0.0469 kg/s) and at natural convection during respectively the first, the second and the third day of experience.

Fig. 7 compares the theoretical and experimental outlet temperatures of the air working fluid for a flow rate of 300, 150 m^3/h and for natural convection. The comparison shows good agreement between measured and predicted results, although, the mean error equals to 7.05%.

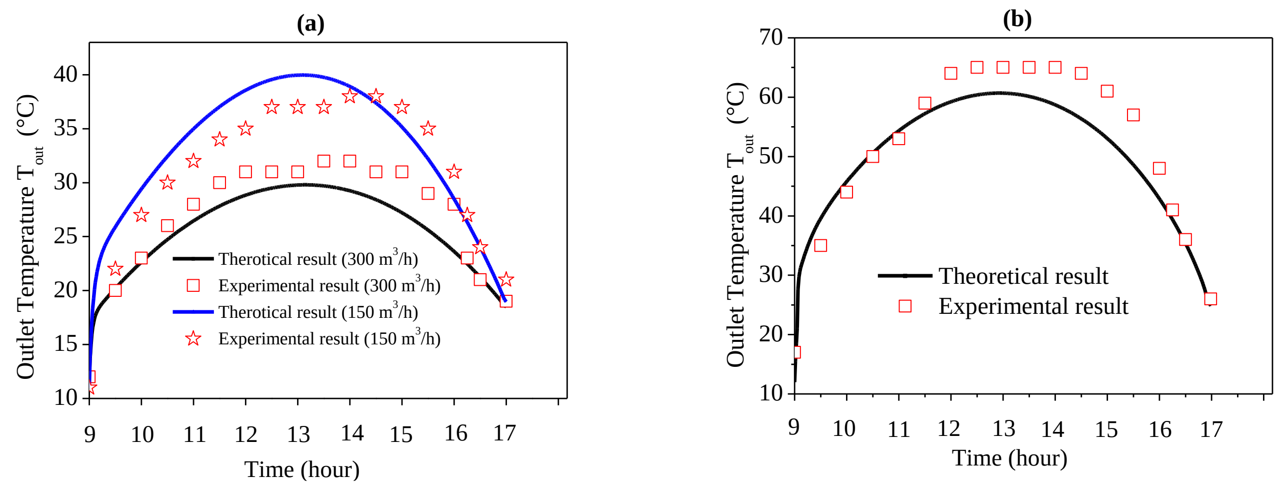

Fig. 7. Predicted and experimental outlet air temperature of the existing flat plate collector: a) The collector is operated at 300 and 150 m^3*/h flow rates, b) The collector is operated at natural convection.*

4. Conclusion

The mathematical models developed for the flat plat and V-corrugated solar air heaters and the simulation codes created in MATLAB are able to correctly predict the instantaneous temperatures of any component at any section of the air collector, the outlet air temperature, and the instantaneous thermal efficiency.

The simulation results are verified by experimental and theoretical results found in the literature. Also, an experimental study at an existing flat plate collector was made in order to verify the proposed model, the analysis shows very good agreement between the measured and the numerically predicted values for different running conditions and flow rates. And it was found that the proposed model has the ability to predict the performance of the collector's different configurations.

Acknowledgments

The authors acknowledge the support of LESPAM (ENS-Marrakech) for allowing us to experiment on a solar air heater installed in their laboratory.

Appendix A

Table A1. Relevant design parameters employed for numerical calculations

Relevant parameter	value	Relevant parameter	value	Relevant parameter	value
τ_g	0.94	C_b (J/(kg K))	1400	δ_{ag2} (cm)	5
α_g	0.04	ρ_g (kg/m^3)	2500	δ_p (mm)	0.5
ε_g	0.1	ρ_p (kg/m^3)	8940	δ_f (cm)	2.5
α_p	0.96	ρ_b (kg/m^3)	32	δ_b (cm)	5
ε_p	0.12	L (m)	2.5	H_c (cm)	2.5
k_b (W/(m K))	0.025	W (m)	1	H_g (cm)	5
ε_b	0.6	δ_g, δ_{g1}, δ_{g2} (mm)	3	H_s (mm)	0,2,4,6
C_g (J/(kg K))	720	δ_{ag} (cm)	5		
C_p (J/(kg K))	385	δ_{ag1} (cm)	1		

Table A2. Relevant operational parameters employed for numerical calculations

Relevant parameter	value	Relevant parameter	value	Relevant parameter	value
G (W/m^2)	700	m_f (kg/ s)	0.025	θ (degree)	30
T_a (°C)	25	V (m/s)	5		

Nomenclature

L length of the collector, m
W width of the collector, m
D_h hydraulic diameter, m
A_{in}, A_{out} inlet and outlet air flow channel cross sectional areas, m^2
G solar radiation on titled surface, W/m^2
T temperatures, K
V wind velocity, m/s
C specific heat, J/(kg K)
h heat transfer coefficient, W/(m^{2}K)
k thermal conductivity, W/(m K)
m_f air mass flow rate, kg/ s
H_c gap between V-corrugated absorber and glass cover, m
H_g height of V-corrugated absorber, m
H_s Spacing between the V-corrugated absorber and the insulation, m
U_b bottom heat loss coefficient, W/(m^2 K)
Re, Pr, Nu, Ra Reynolds, Prandtl, Nusselt and Rayleigh numbers
x Spatial co-ordinate, m
n number of nodes in flow direction
r rate of the aperture area
C_d coefficient of discharge of air channel, 0.6
g gravitational constant, 9.81 m^2 /s

Greek symbols

η instantaneous thermal efficiency of the collector, %
δ Thickness, m
α absorptivity
ε emissivity
τ transmissivity
ρ density, kg/m^3
θ tilt angle of the collector, degrees
α_d Thermal diffusivity of air, m^2/s
v kinematic viscosity of air, m^2/s
σ Stephen-Boltzman constant, 5.67x10^{-8} W /(m^2 K^4)

Subscripts and superscripts

a ambiant
s sky
w wind
g, $g1$, $g2$ glass cover, glass cover 1, glass cover 2
ag air gap between glass cover and absorber
$ag1$ air gap between the glass cover 1 and the glass cover 2
$ag2$ air gap between the glass cover 2 and the absorber
p absorber
f air working fluid
b insulation in the bottom
in inlet air

out outlet air
c convection
r radiation

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