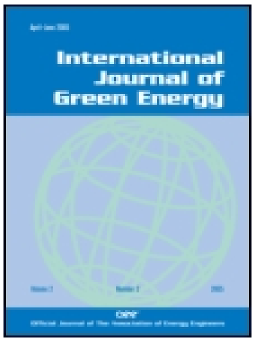

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International Journal of Green Energy

ISSN: 1543-5075 (Print) 1543-5083 (Online) Journal homepage: http://www.tandfonline.com/loi/ljge20

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INTERNATIONAL JOURNAL OF GREEN ENERGY
2016, VOL. 13, NO. 5, 521–529
http://dx.doi.org/10.1080/15435075.2014.968922

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Chawki Mahboub[a,b], Noureddine Moummi[a,b], Abdelhafid Brima[a,b], and Abdelhafid Moummi[a,b,c]

[a]Department of Mechanical Engineering, University of Biskra, Algeria; [b]Laboratory of Mechanical Engineering (LGM), University of Biskra, Algeria; [c]Laboratoire de Génie Civil, Hydraulique, Développement Durable et Environnement (LARGHYDE), Université de Biskra, Algérie

ABSTRACT
The suitable design is the most important key to a cost-effective solar air heater. Although there are many techniques that have been proposed to improve the solar air heaters' performance by means of different turbulence promoters, they cannot ensure a compromise between the cost and the effectiveness. The aim of this study is to find simple and tolerable solution to get rid of the inconvenience resulting from the widely adopted heat-transfer-enhancement techniques by providing an optimized solar air heater design. The proposed design consists of a slightly curved smooth flow channel with an absorber plate of convex shape. A prototype of a curved solar air heater of 1.28 m^2 collector area was built and tested under summer outdoor conditions in Biskra (Algeria). The performance was evaluated in terms of thermal and effective efficiency for mass flow rates of 0.0172, 0.029, and 0.0472 kg/sm^2. It is observed that the overall efficiency of this solar air heater is considerably higher in comparison with the efficiency range of the conventional smooth flat plate heaters reported in the literature for similar operating conditions.

KEYWORDS
Experimental investigation; rectangular curved duct; solar air heater; solar energy; thermo hydraulic performance

Introduction

Nowadays, the dependence on solar energy to meet the increasing demand of thermal energy in residential sector and in some industrial applications has become an urgent requirement. The working efficiency of the devices used for solar to thermal conversion process is one of the important criterions that may affirm whether or not solar energy is advantageous over the other energy sources. This efficiency is defined as absorbing incident solar radiation, converting it to thermal energy and delivering this last to a heat transfer medium with minimum losses (Kalogirou 2004).

Among various configurations of solar thermal collectors, solar air heaters were found suitable for low- and moderate-temperature applications (Garg and Adhikari 1999), such as active building heating, air conditioning, and drying process (Duffie and Beckman 1991; Youcef-Ali et al. 2001; Prakash and Kumar 2013). A major disadvantage that solar air heaters suffer from is that air has low thermal properties compared with other working fluids, which often brings limitations to the overall efficiency of this type of solar collectors, especially the conventional flat plate configuration. To overcome this inconvenience, there are many possible designs for solar air heaters, from the point of view of both materials and configurations that lead to a variety of costs and collection efficiencies (Close 1963).

Since the absorber plate conductivity and resistance to corrosion have insignificant impact on the performance of solar air heaters (Close 1963), the improvement in the characteristics of the materials involved is restricted to the optical properties by either antireflective coatings, surface texture, electrolytic, or chemical treatments (Kalogirou 2004). Solar air heaters performance, however, is very sensitive to other complex phenomena associated with the momentum and heat transfer in the flow channel.

Lewis (1975), Prasad and Saini (1991), Gupta, Solanki, and Saini (1997), and Verma and Prasad (2000) have conducted parametric analyses basing on a unified approach of rough surfaces originally developed for rectangular ribs, for optimizing the thermo-hydraulic performance of surfaces having rough elements of different shapes. It has been found that optimal thermo-hydraulic performance corresponds to a particular value of dimensionless combination of roughness and flow parameters called roughness Reynolds number.

Liu et al. (2007a, 2007b) have performed comparative studies of the thermal performances of solar air collectors with flat-plate, v-groove, and transversely positioned wave-like plates, in which the air flows along the groove of the absorbing plates. It is found that for all the operating conditions considered, the cross-corrugated collector has the superior thermal performance, followed by the v-groove collector and then comes the flat plate collector. The results also show that a slender configuration along the air flow direction, a small gap between the absorbing plate and the bottom plate, a selected coating on the absorbing plate and the glass cover that has a very high absorptivity of solar radiation but quite a small emissivity of thermal radiation, an air mass flow rate above 0.1 kg/sm^2, and maintaining the air inlet temperature close to that of the ambient, will lead to improved thermal performance for all these collectors.

An energy and exergy study has been done by Languri et al. (2011) on a double pass flat plate solar air collector

CONTACT Chawki Mahboub chawki1981@yahoo.fr Department of Mechanical Engineering, University of Biskra, B.P. 145 R.P. 07000 Biskra, Algeria.
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with and without porous medium embedded inside the lower channel of the collector. The obtained results show that employing porous medium can increase the thermal efficiency of the collector of more than 30%, while a second law analysis shows that the friction with the porous medium can also increase the pressure drop in the air.

Experimental studies have been carried out by Moummi et al. (2004), Karim and Hawlader (2006), Alta et al. (2010), Akpinar and Koçyiğit (2010), El-Sebaii et al. (2011), Aoues et al. (2011), and Labed et al. (2012) to investigate the effect of fins and v-corrugations on thermal performance of solar air collectors in single and double pass modes. From the reported results, the v-corrugated solar air collector appears to be the most efficient; and providing a double pass configuration can bring further improvement in the efficiency.

Kumar, Saini, and Saini (2012) have reviewed and presented the correlations for heat-transfer coefficient and friction factor developed by various investigators for solar air heater ducts having artificial roughness of different geometries. They came to the conclusion that the heat transfer may be enhanced as more secondary flow cells are produced.

All the previous investigations show that using various choices of artificial roughness could optimize the flow passage geometry leading to a significant enhancement of the heat transfer between the absorber plate and the air, yet it still requires a higher pumping power due to the increasing in pressure drop. Moreover, the inclusion of artificial roughness yields a higher manufacturing cost of the solar air heater. A simple and effective design allows maximizing the heat transfer while keeping the pressure drop as minimum as possible on one hand, and preserving the device as simple as the conventional smooth flat plate configuration which means low manufacturing cost on the other hand, has been proposed, constructed, and investigated experimentally in this paper.

The solar air heater under consideration and the conventional smooth flat plate solar collector are quite similar, with the former having slightly convex shape (Figure 1). The heat-transfer and fluid flow characteristics in such configuration depend on a dimensionless parameter called Dean number which represents the ratio of the square root of the product of the inertial and centrifugal forces to the viscous forces. The curved geometry of the flow passage was found to generate a secondary flow vortices in the channel cross section due to the effect of the centrifugal forces which dislocate the maximum value of the main flow velocity toward the absorber plate leading to a remarkable increase in the heat transfer rate as well as in the pressure drop (Cheng and Akiyama 1970; Silva, Valle, and Ziviani 1999).

Of particular interest to our purpose, Silva, Valle, and Ziviani (1999) have demonstrated that the friction factor and Nusselt number show dissimilar trends as a function of Dean number for rectangular channels. One can observe from their results that at relatively low Dean number, which could correspond to channels of slightly curvature, the pressure drop remains almost constant due to the weak variation in the main flow profile, whereas the heat transfer increases with increasing Dean number due to the presence of a secondary flow pattern. Thus, the curvature radius of the heater should be large compared with the hydraulic diameter of the cross section of the flow channel to ensure that the solar air heater operating within relatively low Dean number which allows to achieve higher effective efficiency as compared to the smooth flat plate heater.

Experiment

In order to provide low-cost solar air heater, an experimental device was designed and constructed in the Mechanical Engineering Laboratory at the University of Biskra (Figure 2) taking into consideration the malleability, the cost, and the availability in the local market of the materials involved. It consists of Plexiglas plate of 3 mm thickness as transparent cover, which has been used instead of glass for the reason of flexibility despite of the degradation of its optical properties through frequent exposure to sun; black painted galvanized iron sheet of 1 mm thickness and 1.28 m^2 collector area, which would be appropriate to be used as absorber plate in solar air heaters since no significant improvement of the collector efficiency can be achieved using selective materials (Hachemi 1999; Moummi et al. 2004); polystyrene board of 40 mm thickness used as back insulation layer; hardboard sheet (also known as masonite or isorel) of 3 mm thickness

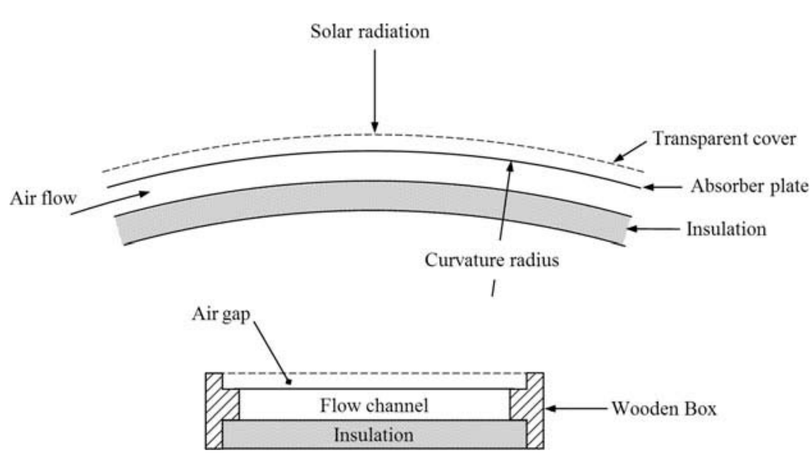

Figure 1. Side and cross section view of the curved solar air heater.

Figure 2. Experimental set-up.

as bottom plate which forms a smooth underside of the curved flow channel, and at the same time protects the polystyrene layer from possible damage due to the direct contact with hot air; plywood panel of 3 mm thickness used as back cover; and a collector box that holds the different parts of the solar air heater together has been fabricated from single and double hardwood plank of 22 mm thickness which has a good thermal insulation. The flow channel between the absorber plate and the bottom plate is of 0.04×0.8 m cross-section and 1.6 m length with sidewalls of 44 mm thickness. The air gap between the transparent cover and the absorber plate is of 20 mm height with sidewalls of 22 mm thickness. The back insulation layer is of 40 mm thickness, and the curvature radius of the absorber plate is 3 m which is considered optimum for the present purpose.

To reduce the size of the dead zones and to assimilate a uniform distribution of the air velocity in the flow channel during operation, the inlet and outlet cross-sections having a row of Ø 20 mm perforations through which the air is driven by an extractor system controlled with voltage regulator. The ambient, air inlet and outlet temperatures were measured using NTC thermistor probes connected to a digital temperature panel meter. The global solar radiation was measured by Frederiksen hand pyranometer calibrated with a Kipp & Zonen CM21 pyranometer, which was placed nearby the transparent cover at the same slope and orientation of the solar air heater. The pressure drop of the air across the solar heater flow channel was measured using a Kimo differential pressure transmitter CP300 through pressure taps mounted at the inlet and outlet cross sections and the pressure drop through the exit converging section was measured using a Phywe inclined tube manometer filled with red oil of specific gravity 0.87 (see Table 1).

The experiments took place in Biskra (34°51'N latitude, 5°44'E longitude and 87 m altitude), eastern of Algeria. The solar air heater was oriented toward the south and tilted from horizontal plane by approximately 18°, which is considered suitable for the geographical location of Biskra in summer season during the period from 24th June to 9th July 2013, where the experiments have been conducted under clear sky and low to moderate wind speed conditions.

Performance analysis

Collector performance

The performance of solar air heaters could be evaluated from different efficiency equations that may provide, according to their complexity and purposes that they serve (Duffie and

Table 1. Technical data of the measuring instruments.

Instrument	Measuring range	Accuracy	Resolution
Hand pyranometer 4890.20	0–1999 W/m^2	±5% of full scale	1 W/m^2
Vane probe Anemometer LV110	0–65000 m^3/h	±3% of reading ±10 m^3/h	1 m^3/h
Pressure transmitter CP300	0–100 Pa	±0,5% of reading ±1 Pa	1 Pa
Inclined tube manometer Phywe	0–200 Pa	–	10 Pa
	0–400 Pa	–	20 Pa
Digital temperature panel TP3	−30 to 110°C	±1°C	0.1°C (< 100°C) 1°C (≥ 100°C)

Beckman 1991), qualitative information on how these solar systems do work. Knowing that the thermal efficiency is defined as the ratio of the useful energy gain to the incoming solar energy received by the collector area of the heater, it is usually expressed either under real transient or quasi-steady state operating conditions by the following equations:

$$\eta_{\text{th}} = \frac{\dot{m}C_p(T_{\text{out}} - T_{\text{in}})}{A_c\ I} \tag{1}$$

$$\eta_{\text{th}} = F_o(\tau\alpha) - F_oU_L\frac{(T_{\text{out}} - T_a)}{I} \tag{2}$$

When the collector works on an open cycle where the ambient air is sucked in, it is convenient to calculate the useful energy gain as a function of the air outlet temperature, as in Eq. (2), rather than using air inlet temperature since this latter coincides with the ambient, i.e., $T_{\text{in}} \approx T_a$, and consequently, the resulting expression of the instantaneous thermal efficiency will not be practically useful (Ahmad, Saini, and Varma 1996).

As the solar air heaters are active systems, they require an external energy input (electric) to force the air throughout the flow channel. According to Cortés and Piacentini (1990), a considerable part of this energy which is produced in many countries mainly from thermoelectric sources is lost in conversions and transmission. Thus, and for economic considerations, one should take into account these factors to evaluate the real economic performance of the collector as follows:

$$\eta_{\text{eff}} = \frac{\dot{m}C_p(T_{\text{out}} - T_{\text{in}}) - P_{\text{mec}}/C_f}{A_c\ I} \tag{3}$$

in which the mechanical power is given by:

$$P_{\text{mec}} = \frac{\dot{m}\Delta p}{\rho} \tag{4}$$

and 0.18 is suggested to be taken as typical value of the conversion factor C_f (Cortés and Piacentini 1990; Ahmad, Saini, and Varma 1996; Mittal and Varshney 2006).

Global solar radiation

Since it was not possible practically to measure the exact rate of solar radiation received by the present curved solar heater, we should mention that the calculations performed previously were based on the assumption that as the heater has a small curvature, it collects the global solar radiation as much as a flat plate collector of the same collector area does. This assumption may be briefly demonstrated by simply considering that the collector area consists of N surface elements and the global solar radiation incident on each surface element dA tilted from horizontal by an angle β_i, could be written according to Liu and Jordan's isotropic model as (Eicker 2003):

$$\begin{aligned} I_{\beta_i} &= I_{hB}\left(\cos\beta_i + \tan\theta_z \sin\beta_i \cos\left(\gamma_s - \gamma\right)\right) \\ &\quad + I_{hD}\frac{1+\cos\beta_i}{2} + I_h\ \rho_g\frac{1-\cos\beta_i}{2} \end{aligned} \tag{5}$$

Thus, the average incident solar radiation on the solar collector surface could be evaluated by integrating the previous equation over the entire collector area, where $\beta - \varphi \leq \beta_i \leq \beta + \varphi$, which yields:

$$\begin{aligned} I_\beta &= I_{hB}r\left(\cos\beta + \tan\theta_z \sin\beta \cos\left(\gamma_s - \gamma\right)\right) \\ &\quad + I_{hD}\frac{1+r\cos\beta}{2} + I_h\ \rho_g\frac{1-r\cos\beta}{2} \end{aligned} \tag{6}$$

in which:

$$r = \frac{\sin\varphi}{\varphi} \tag{7}$$

Where, $\varphi = \frac{L}{2R_C}$ is the half curvature opening angle and β is the inclination angle at the mid-length point of the surface. When φ is relatively small (≈ 0.267 rad in our case) the ratio r tends to unity and the Eq. (6) becomes identical to that of incident solar radiation on a plane surface tilted with an inclination angle β.

Results and discussion

A series of experimental tests have been conducted in typical summer days. The experiments have been performed for mass flow rates of 0.0172, 0.029, and 0.0472 kg/sm^2, which are in the typical range of flow rates for solar air heaters. The data of ambient, air inlet and outlet temperatures, solar radiation, and pressure drop were recorded every half an hour between 10:00 and 15:00 local time in Biskra. The overall thermal and effective efficiencies of the collector have been calculated by integrating the experimental data via Eqs. (1) and (3) over the test period. The results are compared with those of smooth flat plate solar air heater reported in the literature at similar (or nearly) operating conditions.

The variation of the global solar radiation, air inlet and outlet temperatures during the test period on 7th, 8th, and 9th July 2013 are presented in Figure 3–5 respectively. The measurements show an increase in the global solar radiation to a maximum of over 1050 W/m^2 between 12:30 and 13:00 during which it remained almost steady before decreased thereafter. Similar variations in the air outlet temperature were observed since it depends strongly on the incident solar radiation, however it reached the maximum value between 13:00 and 14:00 due to the transient behavior of the solar collector which requires a time response to the variation in solar radiation intensity; whereas the air inlet temperature varied almost linearly with time during the test period. Consequently, a linear increasing trend of the air temperature rise with increasing solar radiation has been observed for each mass flow rate. For the investigated mass flow rates 0.0172, 0.029, and 0.0472 kg/sm^2, the maximum rises in air temperature obtained from measurements are respectively 31.4°C, 24.8°C, and 19.1°C with corresponding overall thermal efficiencies of 49.22%, 66.35%, and 82.91%, respectively.

Figure 6 shows the variation of the pressure drop across the solar air heater as a function of the mass flow rate. For low mass flow rates, the centrifugal forces are less intense which does not affect the main velocity profile even with the formation of secondary flow vortices, and the pressure drop through the curved channel is equal to that through

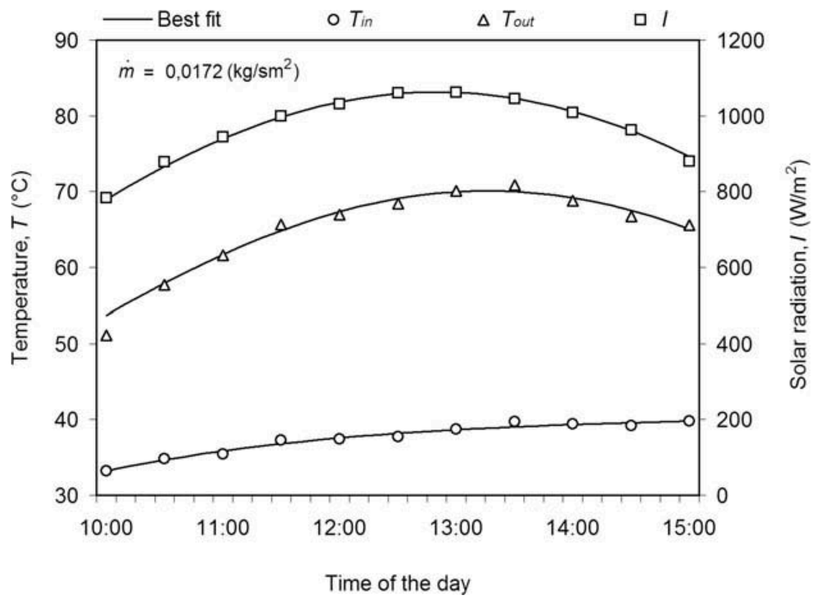

Figure 3. Measured solar radiation, air inlet and outlet temperatures versus time for 0.0172 kg/sm^2.

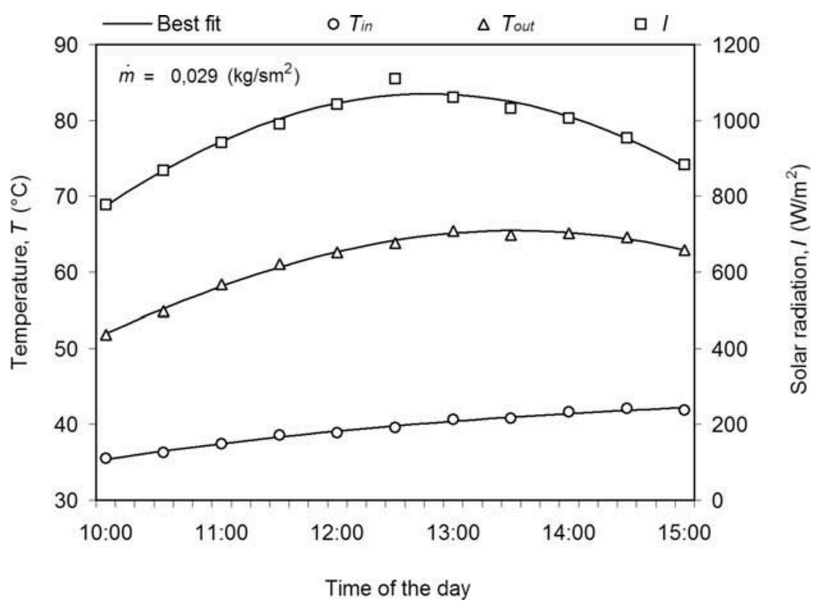

Figure 4. Measured solar radiation, air inlet and outlet temperatures versus time for 0.029 kg/sm^2.

straight smooth channel as it is expected. With the increasing mass flow rate, the centrifugal forces become more intense, causing larger main velocity gradients near the absorber plate that results in pressure drop greater than that through straight smooth channel. It was found that the pressure drop across the curved solar air heater is higher by about 25% at mass flow rate of 0.06 kg/sm^2 compared with that of the smooth flat plate collector reported by Karwa and Chitoshiya (2013). It was also found that the increase in the pressure drop in the curved collector is mainly attributed to the contractions at the channel entrance and exit.

Figure 7 shows the variation of the overall daily thermal and effective efficiencies of the curved solar air heater versus mass flow rate along with the thermal efficiency of the smooth flat plate collector reported in Ahmad, Saini, and Varma (1996) and Karim and Hawlader (2006). The thermal efficiency follows an increasing trend up to the limit of the investigated range of flow rates beyond which it is expected to stabilize close to a maximum value because the overall heat losses had nearly reached their minimum value (El-Sebaii et al. 2011). The results of the curved configuration show that, whatever the value of the flow rate, an enhancement in thermal efficiency of 20–30% as compared to the smooth flat plate collector could be achieved. It can be also seen from the results that a maximum effective efficiency of about 75% could be obtained at mass flow rate around 0.045 kg/sm^2.

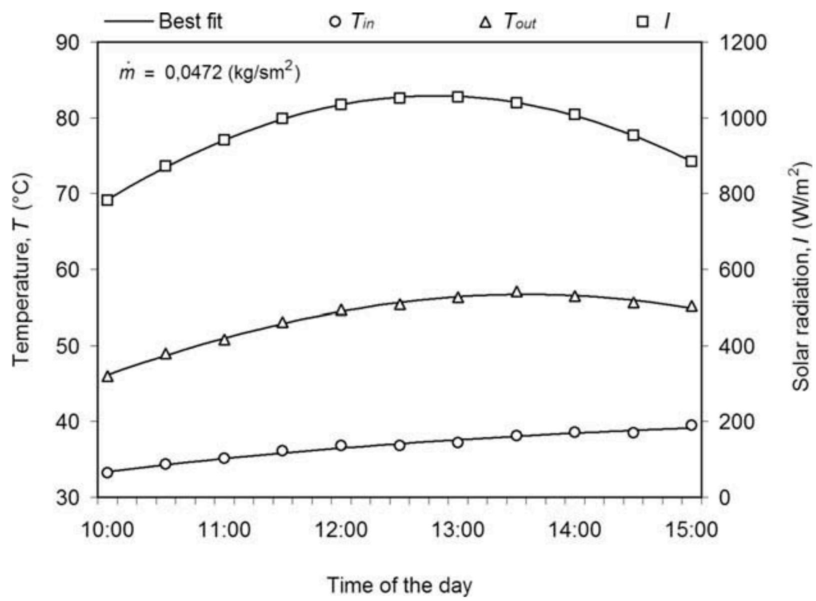

Figure 5. Measured solar radiation, air inlet and outlet temperatures versus time for 0.0472 kg/sm^2.

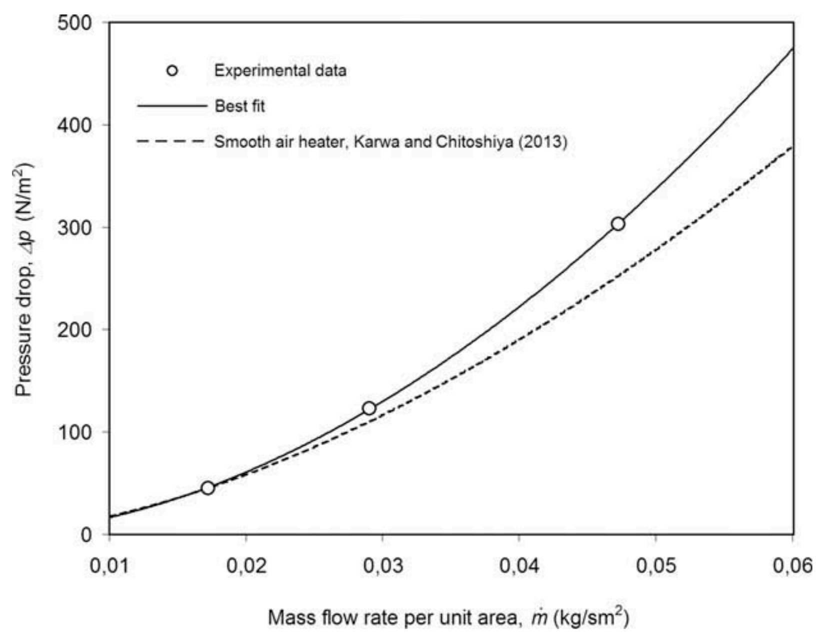

Figure 6. Pressure drop versus mass flow rate.

In comparison with the results of the previously cited studies, one can notice that the impact of the centrifugal forces on the heat transfer in the solar air heater seems to be analogous to the effect of the artificial roughness from the standpoint of providing a good mixing air due to the secondary flow vortices which would be more effective as the airflow rate increases. However, unlike channels having artificial roughness inserted perpendicular to the main flow which involves recirculation and separated flow regions at the front and rear of each roughness element (Lewis 1975), the vortices in the curved channels are formed in the cross-section plane and the pressure gradient in the direction of the flow is favorable everywhere which is indeed the reason why the pressure drop is relatively low.

The variation of the instantaneous thermal efficiency against the temperature parameter $\frac{T_{\rm out}-T_{\rm in}}{I}$ was plotted in Figure 8. A linear regression analysis has been used to fit the experimental data points which resulted in straight line with an intercept equal to 1.25 and a negative slope equal to 22.43 W/m^2 K, since for the case without air recycling ($T_{\rm in} \approx T_a$) all the efficiency curves for the different mass flow rates merge and only one performance curve is obtained (Ahmad, Saini, and Varma 1996). The characteristic curves of both configurations show that the present collector is more efficient than the smooth flat plate collector over the considered range of operating conditions. However, the present collector seems to be more sensitive to the outdoor conditions as its performance varies from the intercept efficiency to the stagnation

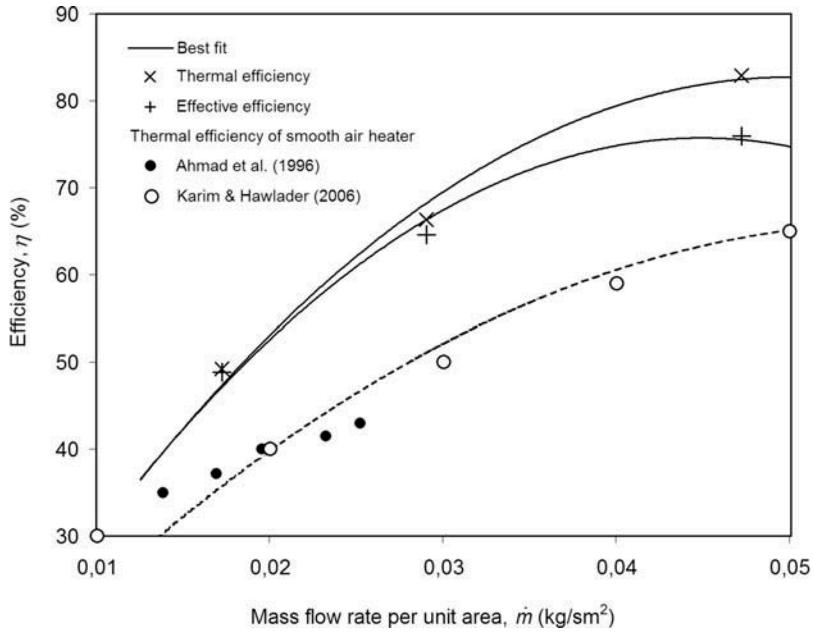

Figure 7. Thermal and effective efficiency versus mass flow rate.

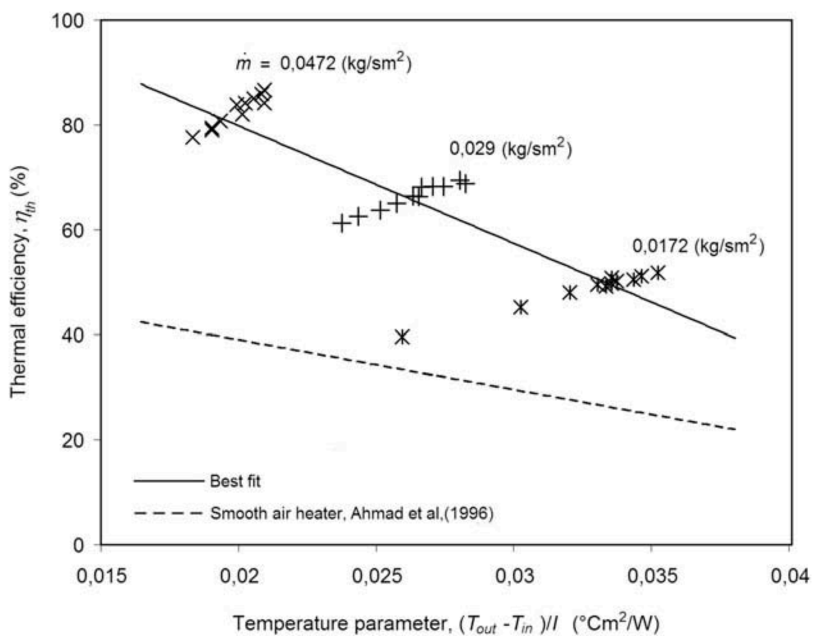

Figure 8. Thermal efficiency versus temperature parameter (T_{out}-T_{in})/I.

condition over a restricted range of the temperature parameter. Referring to the results of the work by Gill, Singh, and Singh (2012), it is obvious that the sensibility of solar collectors to changes in outdoor conditions depends partly on the season during which the experiments are conducted.

Conclusion

An experimental study was conducted to evaluate the performance of a new solar air heater design under the climatic conditions of Biskra (Algeria). According to the results presented above, the conclusions can be summarized as follows:

(1) In terms of effectiveness, the present design can improve the performance of the smooth solar air heater through taking advantage of the centrifugal forces effect on the airflow structure inside the curved duct of the heater. The flow vortices formed in the cross-sectional plane of the curved duct allow enhancing the heat transfer between the absorber plate and air, and at the same time avoiding the important increase in the

pressure drop associated with separated flows which usually occurs when using artificial roughness.

(2) In terms of cost, the present collector can be constructed with the same amount of materials used to fabricate the conventional smooth flat plate collectors; thus, there will be no additional manufacturing cost or construction weight.

Nomenclature

A_c	Collector area of the heater, (m^2)
C_f	Conversion factor
C_p	Specific heat of air, (J/kg K)
De	Dean number, $\mathrm{De} = \mathrm{Re}\sqrt{D_H/R_C}$
D_H	Hydraulic diameter, (m)
F_o	Heat removal factor based on air outlet temperature
I	Solar radiation, (W/m^2)
I_{hB}	Direct beam solar radiation on a horizontal surface, (W/m^2)
I_{hD}	Sky diffuse solar radiation on a horizontal surface, (W/m^2)
I_h	Global solar radiation on a horizontal surface, (W/m^2)
L	Solar air heater length, (m)
$\dot{m}$	Mass flow rate, (kg/s)
Δp	Pressure drop, (Pa)
P_{mec}	Mechanical power, (W)
R_C	Curvature radius, (m)
Re	Reynolds number
T_a	Ambient temperature, (°C)
T_{in}	Air inlet temperature, (°C)
T_{out}	Air outlet temperature, (°C)
U_L	Overall loss coefficient, (W/m^2 K)

Greek Letters

$\tau\alpha$	Transmittance absorptance product
γ	Surface azimuth angle, (°)
γ_s	Solar azimuth angle, (°)
η_{th}	Thermal efficiency
η_{eff}	Effective (thermo-hydraulic) efficiency
θ_z	Solar zenith angle, (°)
ρ	Density of air, (kg/m^3)
ρ_g	Ground reflectance

Acknowledgments

The authors would like to express their gratitude to Pr. Mahmoud Omari, Dr. Azeddine Belakehal, Mr. Abdelkrim Baissi, the engineers and technicians in the Department of Mechanical Engineering. The authors also wish to express their thanks to the reviewers for the helpful comments and the suggestions for improving the manuscript.

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