Experimental and Numerical Study of a New Corrugated and Packing Solar Collector

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Abstract. The thermal performance of a new solar air collector with corrugated packing is studied in this work. A new innovate shape was suggested in this paper to improve the thermal performance of the solar air collector. Mathematical models are developed to investigate the thermal performance of the collector and the results are verified by experiments. Effects of the operating parameters, solar radiation intensity and inlet air velocity are studied to optimize the thermal performance of the collector. Comparisons are conducted between the experimental and numerical results and good agreement was found. The new innovate shape shows better performance than other corrugated shapes and the absorber and air gained higher temperatures. The maximum temperature of air inside duct at the exit point was 69.1 C° at solar heat flow of 725 W/m² and inlet velocity 1.0 m/s. The new shape shows better performance than other corrugated absorber.

Keywords: Solar air collector; Corrugated packing; Thermal performance; Computational fluid dynamics

Nomenclature

| Symbol | Description                      | Unit         |
|--------|----------------------------------|--------------|
| As     | South direction area heat transfer (m²) |             |
| Ae     | East direction area heat transfer (m²) |             |
| At     | Top direction area heat transfer (m²) |             |
| qS     | South direction conduction heat transfer (W) |       |
| qN     | North direction conduction heat transfer (W) |       |
| qE     | East direction conduction heat transfer (W) |       |
| qW     | West direction conduction heat transfer (W) |       |
| qt     | Top direction conduction heat transfer (W) |       |
| qb     | Bottom direction conduction heat transfer (W) |       |
| qSolar | Solar energy (W/m²)               |             |
| u,v,w  | Velocity of x, y and z components (m/sec) |       |
| Cp     | Specific heat at constant pressure (J/kg K) |       |
| kg     | Thermal conductivity of glass cover (W/m K) |       |
| μ      | Dynamic viscosity of fluid (kg/m sec) |             |
| aG     | Absorptivity of the glass |       |
| ρ      | Density of fluid (kg/ m³) |             |
kw Thermal conductivity of absorbing wall (W/m K)
h_i Heat transfer coefficient of air inside duct (W/m^2 K)
h_o Heat transfer coefficient due to wind (W/m^2 K)
h_gab Heat transfer coefficient of air between absorber plate and glass cover (W/m^2 K)

1. Introduction

Solar energy is clean and their applications are increasingly utilized in buildings and in drying crops in the world. One of the main effective methods to enhance the convective heat-transfer rate is to increase the heat-transfer surface area and to increase the turbulence inside the channel by utilizing fins or corrugated surfaces and many studies have been carried out on this topic. For example, Ali and Hanaoka [1] studied experimentally the effects of the operating parameters on laminar flow forced-convection heat transfer for air flowing in a channel having a V-corrugated upper plate heated by radiation heat flux while the other walls are thermally insulated. Karim and Hawlader [2] studied experimentally the thermal performance for v-groove solar air collector for drying applications. In addition, they investigated both experimentally and theoretically in an effort to improve the performance of conventional air heaters. In both studies displayed the results indicate better thermal efficiency for a v-corrugated collector compared to a flat plate collector. Effects of operating variables on the thermal performance have been investigated. The results show that the temperature of the fluid at the exit of the collector decreases with flow rate resulting in an increase of efficiency due to decreased thermal losses to the environment [3]. [4] Investigated numerically the effects of wavy surface and thermal radiation heat transfer over an inclined wavy solar collector. The mathematical model presented is based on the energy transfer phenomenon within the various components of the collector. The transfer equations are discretized using the finite difference method. The proposed approach indicates that the wavy surface increases in general the performance of the solar collector.

Double-pass counter flow v-grove collector is considered one of the most efficient solar air-collectors. A mathematical model is developed for this type of collector by Karim, Perez and Amin [5] The simulation results proven the difference between the predicted and experimental results is, at maximum, approximately 7% which is within the acceptable limit considering some uncertainties in the input parameter values to allow comparison. To make a better use of the jet impingement, a GTC (glazed transpired solar collector) with perforating corrugated plate is developed by Zheng and his co-workers [6, 7]. The results indicate that the GTC with perforating corrugated plate is applicative enough for its advantages in economy and thermal performance in rural areas of cold regions. Also they studied the thermal performance of a novel solar air collector with metal corrugated packing in the buildings of cold regions. Mathematical models are developed to investigate the thermal performance of the collector and the results are verified by experiments. The hydraulic analysis is conducted experimentally to study the pressure drops of the air flows in the corrugated packing. The results indicate that the metal corrugated packing solar air collector is more appropriate to be used in the rural buildings of cold regions for its advantages of large heat transfer area, high heat transfer coefficient and good economic performance. The thermal performance of three types of solar air-heaters under several configurations and operating conditions are analyzed, measured and compared by [8]. All the analytical and experimental results show that, although the thermal performance of the type 2 heater is just slightly superior to that of the type 1 heater, both of these cross-corrugated solar air-heaters have a much superior thermal performances to that of the flat-plate one. An investigation was conducted on the thermal performance of a solar air collector with a v-groove absorber [9]. The results show that the v-groove collector has considerably superior thermal performance to the flat-plate collector.

By analyzing the finite element method the nonlinear behavior of steel plate shear walls with corrugated plates is investigated under lateral pushover loading conditions [10]. The results of this
study have demonstrated that in the wall with constant dimensions, the trapezoidal plates have higher energy dissipation, ductility and ultimate bearing than sinusoidal waves, while decreasing the steel material consumption. An experimental investigation on the absorber surface of the collector whose shape was made up to provide better heat transfer surfaces was presented by [11].

This paper introduced mathematical and experimental work to investigate the solar collector performance with new design of absorber plate and different operation parameter cases of the city of Kut in Iraq.

2. Methodology and divided as a sub-suction as follows

2.1. Mathematical analysis

Figure 1 displays the model for the solar collector. Air enters the collector from the opening that is in first end for any duct in this collector with an inlet temperature which is assumed equal to the uniform ambient air temperature. Hot air exits from the other side (second end) of the duct at outlet temperature. The collector has an eleven simple rectangular shape (ducts) connected together in special method and glass cover on one side with collector walls are isolated on the other sides. The mathematical formulation of the airflow problem is ruled by basic conservation of mass, momentum and energy equations. In the present study, the flow is laminar according to Reynolds number values (Re = 2138 for inlet velocity 1 m/s and Re = 2875 for inlet velocity 1.3 m/s). Reynolds number was calculated according to equation (1) and for the inlet air velocity range from 1.0 to 1.3 m/s and hydraulic diameter 0.038 m calculated according to equation (2), noting that all other properties were extracted from the standard air tables at the average temperature.

\[
Re = \frac{\rho V D_h}{\mu}
\]

(1)

Where \(D_h\) is the hydraulic diameter of duct,
Dh= 4 A/p = (4 W*H)/2(W+H)    (2)

Where A and P, are the area and perimeter of the duct

The following assumptions have been made:
1- Three-dimensional of conservation equations.
2- Steady state incompressible flow.
3- Conduction three-dimensional heat transfer through glass cover and absorber plate.
4- Air flow in the duct has been considered laminar according to the value of Reynolds number.
5- All properties are evaluated at an average temperature.

2.2. Governing Equations (Laminar Flow)

For solar collector, laminar 3-D Cartesian Coordinates Navier Stokes equations (continuity, momentum and energy) were solved by finite volume (SIMPLE algorithm) using build in program (Fortran 90), and the graphs were plotted using Tecplot software. The 3-D laminar Navier Stokes equations with forced convection are Awbi [12]:

I-Continuity equation
\[
\frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial y}(\rho v) + \frac{\partial}{\partial z}(\rho w) = 0
\]  
(3)

II-Momentum equation
u-Momentum (x-direction)
\[
\frac{\partial}{\partial x}(uu) + \frac{\partial}{\partial y}(uv) + \frac{\partial}{\partial z}(uw) = -\frac{1}{\rho} \frac{\partial P}{\partial x} + \nu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right)
\]  
(4)

v-Momentum (y-direction)
\[
\frac{\partial}{\partial x}(vu) + \frac{\partial}{\partial y}(vv) + \frac{\partial}{\partial z}(vw) = -\frac{1}{\rho} \frac{\partial P}{\partial y} + \nu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right)
\]  
(5)

w-Momentum (z-direction)
\[
\frac{\partial}{\partial x}(uw) + \frac{\partial}{\partial y}(wv) + \frac{\partial}{\partial z}(ww) = -\frac{1}{\rho} \frac{\partial P}{\partial z} + \nu \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right)
\]  
(6)

III-Energy equation
\[
\frac{\partial}{\partial x}(uT) + \frac{\partial}{\partial y}(vT) + \frac{\partial}{\partial z}(wT) = \frac{k}{\rho c_p} \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right)
\]  
(7)

2.3. Boundary Conditions

The mesh used was 51*21*21, and the number of iteration was 5000. The program was already tested for consistency in previous study.

Table 1: boundary conditions

| Variable  | Wall            | Inlet            | Outlet            |
| --------- | --------------- | ---------------- | ----------------- |
| Velocity | u, v, w = 0     | u = u_in v = w = 0 | [\frac{\partial u, v, w}{\partial n} = 0] |
2.4. Heat Conduction

The energy is exchanged through solid wall. For steady state, plane, the heat flow in the x or y or z direction may be calculated by Fourier equation Holman [13]:

\[ q = -kA \frac{\partial T}{\partial x} \]  \hspace{1cm} (8)

The total heat flow at any point in the material is the result of the (q) at that point. In this section, an analytical approach can be used for the solar collector walls. For glass cover shown in Figure 2.

\[ \sum q_{\text{in all face}} = q_{\text{storage}} \]  \hspace{1cm} (9)

Energy balance for node (1) on the glass cover.

\[ \sum q_{\text{in all faces}} = q_s + q_n + q_e + q_w + q_t + q_b + q_{\text{solar1/4}} \]  \hspace{1cm} (10)

\[ q_{\text{solar1}} = \text{the heat absorbed by glass cover from the solar radiation incident} = q_{\text{solar}} \times \alpha_g \]

\[ h_{\text{gb}} A_s (T_{i,j,k} - T_{\text{gab}}) + \frac{k_{gb}}{\Delta y} (T_{i,j,k} - T_{i,j+1,k}) + \frac{k_{gb}}{\Delta z} (T_{i,j,k} - T_{i,j,k+1}) + \frac{k_{gb}}{\Delta x} (T_{i,j,k} - T_{i+1,j,k}) + \frac{k_{gb}}{\Delta y} (T_{i,j,k} - T_{i+1,j,k}) + \frac{k_{gb}}{\Delta z} (T_{i,j,k} - T_{i,j,k+1}) + q_{\text{solar1/4}} A_n = 0 \]  \hspace{1cm} (11)

Where

\[ A_s = \Delta x \Delta z \quad , \quad A_n = \Delta x \Delta z \]
\[ A_e = \left( \frac{\Delta y}{2} \right) \Delta x \quad , \quad A_w = \left( \frac{\Delta y}{2} \right) \Delta x \]  
\[ A_t = \left( \frac{\Delta y}{2} \right) \Delta z \quad , \quad A_b = \left( \frac{\Delta y}{2} \right) \Delta z \]  

Energy balance for node (2) on the glass cover.
\[ \sum q_{in\ all\ faces} = q_s + q_n + q_e + q_w + q_t + q_b + q_{solar\ 1/2} \]  
\[ \sum \frac{k_g A_s}{\Delta y} \left( T_{i,j,k} - T_{i,j-1,k} \right) + \frac{k_g A_n}{\Delta y} \left( T_{i,j,k} - T_{i,j,k} \right) + \frac{k_g A_e}{\Delta y} \left( T_{i,j,k} - T_{i,j,k+1} \right) + \frac{k_g A_w}{\Delta z} \left( T_{i,j,k} - T_{i,j,k-1} \right) + \frac{k_g A_t}{\Delta y} \left( T_{i,j,k} - T_{i+1,j,k} \right) + \frac{k_g A_e}{\Delta y} \left( T_{i,j,k} - T_{i,j,k+1} \right) + \frac{k_g A_w}{\Delta y} \left( T_{i,j,k} - T_{i,j,k-1} \right) + q_{solar\ 1} A_n / 2 = 0 \]  

Where
\[ A_s = \Delta x \Delta z \quad , \quad A_n = \Delta x \Delta z \]  
\[ A_e = \Delta y \Delta x \quad , \quad A_w = \Delta y \Delta x \]  
\[ A_t = \Delta y \Delta z \quad , \quad A_b = \Delta y \Delta z \]  

Energy balance for node (3) on the glass cover.
\[ \sum q_{in\ all\ faces} = q_s + q_n + q_e + q_w + q_t + q_b + q_{solar\ 1/4} \]  
\[ \sum \frac{k_g A_s}{\Delta y} \left( T_{i,j,k} - T_{i,j-1,k} \right) + h_o A_n \left( T_{i,j,k} - T_o \right) + \frac{k_g A_e}{\Delta y} \left( T_{i,j,k} - T_{i,j,k+1} \right) + \frac{k_g A_w}{\Delta z} \left( T_{i,j,k} - T_{i,j,k-1} \right) + \frac{k_g A_t}{\Delta y} \left( T_{i,j,k} - T_{i+1,j,k} \right) + \frac{k_g A_e}{\Delta y} \left( T_{i,j,k} - T_{i,j,k+1} \right) + \frac{k_g A_w}{\Delta y} \left( T_{i,j,k} - T_{i,j,k-1} \right) + A_n A_{solar} 1 / 4 = 0 \]  

Where \( h_o \) the convection heat transfer due to wind, is given by McAdams [14] as:
\[ h_o = 5.7 + 3.8 \nu \]  
\[ A_s = \Delta x \Delta z \quad , \quad A_n = \Delta x \Delta z \]  
\[ A_e = \left( \frac{\Delta y}{2} \right) \Delta x \quad , \quad A_w = \left( \frac{\Delta y}{2} \right) \Delta x \]  
\[ A_t = \left( \frac{\Delta y}{2} \right) \Delta z \quad , \quad A_b = \left( \frac{\Delta y}{2} \right) \Delta z \]  

Similar approach was used for absorber and the number of mesh was 51*3*21. The conduction in glass and absorber was performed in subroutines within main program, which solve the governing equations for duct flow simultaneously.  

2.5 Experimental Works

The experimental rig was built to study air flow and heat transfer in corrugated packing solar collector shown Figure. 3. The aim was to investigate the impacts of the change inlet air velocity and input power on air flow and heat transfer in the system. It consists of eleven ducts, glass cover and wood box all these as solar collector. The solar collector dimensions are 126 cm length, 65 cm width and 13 cm height. The one duct is manufactured of Aluminium plate with length 126 cm, width 4 cm and 4 cm height. All these ducts are assembled in special technique. The glass cover consists of glass windows with dimensions of 65 cm width and length of 126 cm and a thickness of 0.4 cm. A gap was created between the glass cover and the absorbing surface with thickness of 1.5 cm from the top of the wave and
this gap was constant throughout the test. The absorbent surface formed from the assembly of aluminum ducts, so that it gives a crispy surface at an angle 90 ° (V-grooved).

![Image](image_url)

**Figure: 3 Corrugated packing solar collector**

2.6. Experimental Apparatus

Figure 4 shows a schematic diagram of the experimental apparatus, and a photograph of experimental apparatus is shown in Figure 5, which consists of:

1. Blower.    2. Pipe.  3. Control Valve.  4. Transition Piece.  5. Test Main Section  6. Solar Simulator.  7. Voltage Regulator.  8. Measuring Units.  a- Sensors, Arduino and lap top (temperature measurement).  b- Solar Power Meter.  c- Hot Wire Anemometer.  d- Voltmeter.  9. Electrical Control.

![Image](image_url)

**Figure: 4 Schematic diagram of the experimental apparatus**
3. Result and discussions

The results have been calculated numerically and experimentally. The numerical results represent three-dimensional, laminar flow forced convection inside solar collector. Different parameters were discussed and studied numerically to exhibit the performance of a solar collector such as air velocity (1.0, 1.1, 1.2 and 1.3 m/s) and solar heat flux (400, 540, 640 and 725 W/m²). The experimental part also, exhibited the performance of a solar collector under the same conditions that were utilized in numerical part. The temperature change was compared through a channel for numerical and experimental testing. Figure. 6 shows the comparison between experimental and numerical results. The comparison seems acceptable. The numerical results are higher than experimental results because of the losses through walls.

![Comparison between experimental and numerical results](image)

Figure: 6 Comparison between experimental and numerical results

Figure. 7 shows the comparison with Ali and Hanaoka (2002). Ali and Hanaoka (2002) used upper V-corrugated plate with no fins. Our new shape shows higher temperatures for absorber and air because both surfaces (top and bottom) transfer heat to the air. The fins transfer the heat quickly between upper and bottom surfaces.
Figure: 7 Comparison with Ali and Hanaoka (2002) results at velocity = 0.65 m/s

Figure. 8 shows the variation in air temperature measured through the channel section in the various solar heat flows (400, 540, 640 and 725 W / m²) and air velocity is a constant input. All of these curves in the figure shows an increase in air temperature through the channel. Improved heat exchange zone and enhanced heat absorption contributed to the enhancement of thermal gains of the V-corrugated collector. V-corrugated collector benefited from its absorption of greater capacity of solar heat flux in increasing temperature.

Figure: 8 .Change the air temperature through the channel at different solar heat fluxes and inlet velocity = 1.0 m/s.

Figure. 9 shows the variation of measured air temperature along the duct at different inlet air velocities (1.0, 1.1, 1.2 and 1.3) m/s and solar heat flux was constant at 725 W/m². Normally, all these curves in figure demonstrates an increase in the air temperature through the duct. However, the temperature decreases with an increase in inlet velocity from 1.0 to 1.3 m/s. For sample at 725 W/m², the maximum temperature of air inside the duct was in the exit point about 69.1 C° at 1.0 m/s and the minimum was about 61.8 C° at 1.3 m/s.
Figure 9. Air temperature change through the channel at different inlet velocities and solar heat flow = 725 W/m²

Figure 10 shows the variation of absorber plate temperature with solar heat flux at different inlet air velocities (1.0, 1.1, 1.2 and 1.3 m/s). The absorber plate temperature increases with an increase in solar heat flux. Figure 11 shows the variation of absorber plate temperature with inlet velocity at different heat fluxes (400, 540, 640 and 725 W/m²). The absorber plate temperature decreases with an increase in inlet air velocity.

Figure 10. Temperature variation of absorption plate with heat flow at different inlet velocities.
Figure: 11. Variation of absorption plate temperature with inlet velocity at different heat flux.

The effect of solar heat flux and inlet air velocity on glass cover temperature were similar to its effects on absorber plate temperature as shown in Figures 12 and 13.

Figure: 12. Variation of glass cover temperature with heat flux at different inlet air velocities.
Figure: 13. Variation of glass cover temperature with inlet velocity at different heat flux.

Figure 14 shows the isothermal contours inside duct in solar collector for 400 W/m² solar heat fluxes and 1.0 m/s inlet air velocities. It was observed that temperature begins to increases along duct, while in the radial direction, the high temperature at the channel walls begins to decline towards the center of that channel. Figure 15 shows the velocity vector inside duct for 400 W/m² and inlet velocity = 1.0 m/s. The velocity was decreases towards the duct walls until it becomes zero at the walls, while at the center of the channel at its maximum value (boundary layer development).

Figure 14. Isothermal contours through the duct at solar flux = 400 W/m² and inlet velocity = 1.0 m/s

Figure 15. Flow field of air through the duct for solar heat flow = 400 W/m² and inlet velocity=1.0 m/s
4. CONCLUSIONS

1- The maximum temperature of air inside duct at the exit point of the duct was 69.1°C at solar heat flow of 725 W/m² and inlet velocity 1.0 m/s, and the minimum was about 52.6°C at solar heat flux of 400 W/m² at inlet velocity 1.3 m/s.

2- Good agreement was found when compared between numerical and experimental results with deviation of about 4-11% for air temperature (at exit), 1.2-8% for absorber plate temperature (at center) and 1-6.7% for glass cover temperature (at center).

3- The new shape shows better performance than other corrugated absorber. The new shape separates the flow into many parallel duct flows where walls of the ducts work as fins.

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