CFD Simulation of the Thermal Performances of a Double Pass Solar Collector with Vacuum

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Abstract

A numerical study of multi-pass solar collector is proposed in this paper. We developed a three-dimensional numerical model. It was first used to study the efficiency of the solar collector and secondly to evaluate the validity of the developed CFD model by comparison these numerical results with those experimental performed by Fudholi et al. [1]. For the numerical simulations, turbulence and radiation were respectively modeled using k-ε model and discrete ordinates (DO) model. This numerical model was used to carry out a parametrical study and to discuss the effect of vacuum between the two glass covers and of the vacuum layer. Numerical results show that the collector efficiency and the outlet temperature were improved with the creation of the vacuum between the two glass covers. The variation of the distance between this two covers shows that we have an optimal distance for the most improvement in the collector efficiency.

Keywords: CFD simulation; Thermal performance; Solar air heater; Vacuum; Double pass

Introduction

Development of renewable energy sources received important intensities, in view of world’s depleting fossil fuel reserves and environmental threats. Of many alternatives, and as a renewable energy source, solar energy stands out a noticeable energy source to meet the demand, due to its enormous potential. The freely available solar radiation furnishes an absolute and non-polluting reservoir of fuel. The easiest method to utilize solar energy for heating applications is to convert it into thermal energy with the use of the solar collectors. Solar air heaters and solar water heaters are FP collectors which used typically for heating of air and water respectively. In comparison to solar water heater, solar air heater is compact, less complicated and easy to use. Solar air heater is easy to construct with cheaper materials. By using the freely available solar energy and without using any fuels conservative, solar air heater generates warm air for any farmer or industrial level drying applications. Due to the low heat transfer potential between absorber plate and fluid flowing in the duct, thermal efficiency of SAH is reduced, this thermal efficiency wants to be improved by enhancing the heat transfer rate. In literature various designs of solar air heater have been proposed and discussed. Sabaii et al. [2] analyzed experimentally and theoretically the thermal performance of the double glass double pass solar air heater with a packed bed. They performed a comparison among the thermal performances of the system without and with the packed bed, also above or under the absorber plate. Several experiments were as well performed using iron scraps as packed bed material. It was illustrated that it is advisable to achieve the system with a low porosity packed bed above the absorber plate. The best performance was accomplished with gravel packing material raised the absorber plate when the mass flow rate of the air is equal to 0.05 kg/s or lower to supply a lower pressure drop across the system.
and, then, a higher thermo hydraulic efficiency. They found that the values of the thermo hydraulic efficiency with gravel were 22-27% higher than that without a packed bed. The annual averages of the outlet temperature of air and the thermo hydraulic efficiency were, respectively, 16.5% and 28.5% higher than those for the collector without the packed bed, illustrating an enhancement of the heater performance on using a packed bed material, above or under the heater absorber, all year round. Ozgen [3] investigated experimentally a double pass FP SAH with an absorber made with an aluminum cans into the double pass channel. Three singular absorber plates design had been tested experimentally. A first type (Type I) with cans staggered as zigzag on the absorber plate, though, in a second type (Type II), they were arranged in order and the Type III is a simple FP without cans. For an air mass flow rates of 0.03kg/s and 0.05kg/s, experiments were performed, the highest efficiency was reached for Type I at 0.05kg/s. Aldabbagh [4] investigated experimentally the thermal performances of a single and a double pass solar air heaters. Instead of a flat absorber plate, they used a steel wire mesh. They studied the effects of mass flow rate of air on the outlet temperature and the thermal efficiency. The results show the increase of the efficiency with the increase of the mass flow rate. In this work, the flow rate varied between 0.012 and 0.038kg/s. For a same flow rate, the efficiency of the double pass is higher than the single pass by 34-45%. Furthermore, the maximum efficiencies acquired respectively for the single and the double pass air collectors, are 45.93 and 83.65% for the same mass flow rate of 0.038kg/s. The comparison of a packed bed collector results with those of a conventional collector results shows a considerable enhancement in the thermal efficiency. Omojaro [5] investigated experimentally the thermal performance of a single and double pass SAH with fins attached and they used, as absorber plate, a steel wire mesh. The result shows that the collector efficiency increase with the increase of the air mass flow rate. They also found that the efficiency of the double pass was higher than the single pass by 7-19.4%, for the same mass flow rate. The highest efficiency was obtained for, respectively the single and double pass air heater, 59.62% and 63.74% for air mass flow rate of 0.038kg/s. Furthermore, the additional thermal efficiency decreases by the increase of the height of the first channel of the double pass solar air heater. As the air mass flow rate increase, the temperature difference between the outlet flow and the ambient temperature reduces. When compared with a conventional solar air heater, the result of a double or single SAH using steel wire mesh placed in layers as an absorber plate and packing material, shows a much more considerable enhancement in the collector thermal efficiency. Kumar [6] investigated a study of a photovoltaic/thermal solar air heater with double pass design. A vertical fin was added in the lower channel of the system to enhance heat transfer rate and collector efficiency. They found that the extended fin area reduces cell temperature from 82 °C to 66 °C. They reported the impact estimations on thermal and electrical outputs of the packing factor, they recognized that the higher packing factors was beneficial to the production of more electrical output per unit collector area and help in controlling the cell temperature. Sopian et al. [7] investigated an experimental study of the thermal performance of a double pass solar collector with and without porous media in the inferior channel of the system. They concluded the increases of outlet temperature, then the thermal efficiency of the collector, with the existence of the porous media.

Ho-Ming [8] investigated theoretically the effect of external recycle on the collector efficiency in solar air heaters. They founded, as a desirable effect, that the considerable improvement in collector efficiency was available if the process was carried out with an external recycle. The enhancement increases with increasing reflux ratio, particularly for operating at inferior air flow rate with elevated inlet air temperature. In this paper, we present a numerical study of the thermal performance of the double-pass solar collector using the package FLUENT 6.3.

Treating this system with a stationary analysis, we will first validate our numerical results by comparing them with experimental results found by Fudholi et al. [1]. In a second time, we study a certain parameter affecting the performance of the double-pass solar collector.

**Modeling and Simulation**

**Geometry description**

A 3D numerical model was developed using the CFD package FLUENT 6.3, in order to validate the numerical approach. This model interprets the geometry of the double-pass solar collector experimentally studied by Fudholi et al. [1]. As shown in Figure 1 the system consists of a glass cover, an aluminum absorber painted in black and an insulated container. The size of the collector is 1.2m wide and 2.4m long. In this type of collector, the air initially enters through the upper channel, and then passes through the lower channel. The channels were separated by the absorber plate.

**Figure 1:** The schematic of a double-pass solar collector.
The physical properties of air varies linearly with temperature (°C):

A. **Specific heat:**
\[ C = 1.0057 + 0.000066(T - 27) \]  
(1)

B. **Density:**
\[ \rho = 1.1774 - 0.00359(T - 27) \]  
(2)

C. **Thermal conductivity:**
\[ k = 0.02624 + 0.0000758(T - 27) \]  
(3)

D. **Viscosity:**
\[ \mu = [1.983 + 0.00184(T - 27)]10^{-5} \]  
(4)

E. **The convection heat transfer coefficient due to wind:**
\[ h_w = 2.8 + 3.3V \]  
(5)

Where \( V \) is the wind velocity and in our case \( V = 1 \text{m/s} \).

**Mesh**

The steps of the mesh generation are indisputable, in numerical simulation. The mesh was created using Gambit (a preprocessor bundled with FLUENT). This preprocessor is a single interface for geometry creation and meshing [9]. The mesh quality affects the accuracy of the results. The mesh generated for our solar system consists of hexahedral cells in all the configuration as shown in Figure 2. This generated mesh consists of 697920 cells.

**Figure 2:** The generated mesh.

### The governing differential equations

For the numerical simulation of the problem, the resolution of the different conservation laws is essential. We should take the following assumption into account:

A. The fluid is incompressible.

B. The flow is stationary.

The equations are writing as follows:

A. **The continuity equation:**
\[ \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0 \]  
(6)

B. **The momentum conservation equation**
\[ \frac{\partial \rho \mathbf{u}}{\partial t} + \nabla \cdot (\rho \mathbf{u} \otimes \mathbf{u}) = -\nabla p + \nabla \cdot \mathbf{T} + \mathbf{F} \]  
(7)

C. **The energy conservation equation**
\[ \frac{\partial (\rho E)}{\partial t} + \nabla \cdot (\rho \mathbf{u} E) = -\nabla \cdot \mathbf{q} + \mathbf{S}_{\text{conv}} + \mathbf{S}_{\text{rad}} \]  
(8)

The system is treated stationary in this simulation. The turbulence is modeled using the standard k-ε model. The working fluid in the system is the air. We calculate the Reynolds number according to the relation (9).

\[ R_e = \frac{\bar{u}D}{v} \]  
(9)

Where
\[ D_c = \frac{4Wd}{2(W+d)} \]  
(10)

\( D_c \) is the equivalent diameter, \( d \) is the channel depth and \( W \) is the width of the collector.

**Boundary condition**

The boundary conditions imposed on Gambit at different element of our configuration are cited below:

1) **The condition "mass flow inlet" for the air entry:** we consider this type of condition for an incoming flow, in our study in every case we have a value of mass flow rate.

2) **The condition "outflow" for the air exit.**

The boundary condition mixed for the glass cover wall: this condition allows the simulation of the convective and radiative heat transfer in the glass cover.

**Numerical Results**

**Validation of the CFD model**

For the CFD model validation, the numerical thermal efficiency variation is calculated according to the following relation:
\[ \dot{m} C \frac{T_o - T_i}{A_c I} \]  

(11)

Where \( \dot{m} \) the mass flow rate, \( C \) the specific heat of the fluid, \( T_o \) the outlet temperature, \( T_i \) the inlet temperature, \( A_c \) the area of the collector, and \( I \) is the solar intensity.

The ambient temperature considered to calculate the convective and conductive losses is these considered experimentally by Fudholi et al. [1], \( T_a = 25 \, ^\circ C \). The sky temperature \( T_{sky} \) necessary to estimate radiative exchange is calculated according to the relation (12):

\[ T_{sky} = 0.0552 T_a^{1.5} \]  

(12)

To validate our system, we present the collector efficiency as a function of the mass flow rate. The numerical efficiency is compared to the experimental result of Fudholi et al. [1], as shown in Figure 3. The difference between the experimental results and the numerical data does not exceed 1.3% so that’s a satisfactory agreement between the two results. Figure 4 shows also a satisfactory agreement when we present the experimental and numerical outlet temperature.

Figure 3: Efficiency variation

Figure 4: The outlet temperature.

Fudholi et al. [1] explore a theoretical study of the double pass solar collector; Figure 5 shows a comparison between experimental, theoretical and numerical results. Figure 6 shows the theoretical and numerical temperature distribution along the absorber plane, the bottom plane and the glass cover of the solar collector. From this variation, we note that the increase of the mass flow rate dropped the temperature of the different component of the system. Figure 7 & 8 show the temperature contours respectively in the glass cover and the absorber plate. The stream lines presented in Figure 9 shows the air circulation between the upper channel and the lower one.

Optimization of operating parameters of the solar collector

The performance of the double pass solar collector depends on several parameters. Therefore, in this part we treat the effect of the creation of the vacuum between the first glass cover and a second added glass cover as shown in Figure 10.
Figure 5: Results comparison.

Figure 6: Temperature distribution in different system components

Figure 7: Temperature contours in the glass cover.
Numerical results show an important enhancement in the thermal performance of the system as proved in Figure 11-13 where we present respectively the outlet temperature distribution, the system efficiency and the glass cover temperature with and without vacuum. The flat plate collectors lose heat by convection, conduction and radiation, this heat losses do not occur in vacuum. To suppress radiative heat loss, vacuum result in both glass internal surfaces, a low emissivity coatings. The flat plate collector operates at low temperature only, but with vacuum, it is possible to operate at high temperature due to the thermal insulating properties of the vacuum.
Figure 11: Outlet temperature with and without vacuum.

Figure 12: Efficiency with and without vacuum.

Figure 13: Glass cover temperature with and without vacuum.
In another part, the glass covers are subject to large continued stresses from firstly atmospheric pressure loading above their surfaces and secondly the differential thermal expansion when one glass sheet is heater than the other. The vacuum layer influences the mechanical stresses distribution over the glass surfaces. Therefore, when varying the distance between the glass sheets, we note the existence of an optimal distance as shown in Figure 14. For a chosen mass flow value (0.06816kg/s), we present in Figure 15 the outlet temperature for the different values of the vacuum layer.

![Figure 14: Outlet temperature variation with vacuum layer.](image1)

![Figure 15: Optimal vacuum layer.](image2)

**Conclusion**

This paper is dedicated to a numerical simulation of a double pass solar collector. To create the geometry and meshes we used Gambit and then FLUENT for the numerical simulation. The standard k-ε was chosen as the turbulent model and the DO model for the radiation. The numerical simulations were approved for different operating parameters. In a first time, to enhance the thermal properties of the system we have create a vacuum layer between two glass cover. Indeed, for a mass flow rate of 0.045kg/s the outlet temperature and the efficiency are improved respectively from 316 and 0.44 without vacuum to 325 and 0.68 with vacuum. In a second time, the vacuum layer affects the thermal properties of the system and approved the existence of an optimal layer for the most enhancement. To attain and maintain an enclosure low pressure value for a tolerable product lifetime presents a considerable engineering challenge, particularly when the vacuum layer is very small.

**References**


