International Journal of Sustainable and Green Energy
2022; 11(4): 75-83
http://www.sciencepublishinggroup.com/j/ijsge
doi: 10.11648/j.ijrse.20221104.12
ISSN: 2575-2189 (Print); ISSN: 2575-1549 (Online)

Case Analysis of the Optical Parameters of Thermal Solar Flat-Plate Collector on Heat Fluid Transfer and Pressure

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To cite this article:

Serigne Thiao, Awa Mar, Omar Drame, Diouma Kobor, Issakha Youm. Case Analysis of the Optical Parameters of Thermal Solar Flat-Plate Collector on Heat Fluid Transfer and Pressure. *International Journal of Sustainable and Green Energy*. Vol. 11, No. 4, 2022, pp. 75-83. doi: 10.11648/j.ijrse.20221104.12

Received: November 14, 2022; **Accepted**: November 29, 2022; **Published**: December 15, 2022

Abstract: Solar thermal collectors are the key component of active solar cooling/heating systems. This article presents a theoretical study to evaluate the effects of optical parameters such as the transmission coefficient of the glass on the outlet temperature of heat transfer fluid. We have therefore developed a detailed model for flat plate solar collectors where a detailed analysis of the effects of parameters such as glass cover thickness and transmission coefficient will be analyzed. To simplify our study, we have considered an area of flat solar collector approximately equal to $1m^2$. A numerical procedure is implemented to obtain the solution for the nonlinear set of equations representing the mathematical model. A computational fluid dynamic program with ANSYS is adapted to simulate the outlet temperatures of the heat transfer fluid as function of the glass cover thickness and the optical parameters. The effects of the glass cover thickness and optical parameters such as transmission coefficients on the outlet temperatures of the heat transfer fluid are analyzed. The results obtained show that the optimal thickness and the optimal transmission coefficient of the glass are respectively 6 mm and 0.9. We have also shown that for good resistance to pressure, the optical transmission coefficient must be equal to 0.9. The outlet temperature of the heat transfer fluid can reach 75°C, therefore able to meet our drying and solar heating needs.

Keywords: Fluid Dynamic, Glass Cover Thickness, Optical Parameters, Solar Flat-Plate

1. Introduction

Global energy demand is growing exponentially and is largely covered by fossil energy.

The sources of fossil energy used reach their limits. On the one hand, due to the more expensive exploitation sources of coal, oil and natural gas. On the other hand, due to global climate change related to carbon dioxide releases from the combustion of fossil fuels and refrigerant fluids. The use of fossil energy is becoming stronger, which forces us to explore other energies sources that are available and a potential that could exceed our needs. We are obliged today to privilege the use of renewable energies. These renewable energies differ in their origins. Solar thermal collectors systems are generally considered as an efficient way to recover the energy emitted by the sun. Solar collectors are mainly classified into flat plate solar collectors (FPSC), photovoltaic thermal hybrids (HPV/T), evacuated-tube solar collectors (ETSC) and compound parabolic solar collectors (CPSC) [1]. The solar radiation can be useful for our life to the heating building, heated water in order to produce steam and used it in any way such as in industrial and domestic [2, 3]. A broad spectrum of applications can be derived from this: production of domestic water heating, heat production for industry, desalination of seawater or thermoelectricity production. Solar collectors are the key component of active solar cooling/heating systems. There are three principal parts of flat plate solar collector: absorber plate which absorbs solar radiation and transfers it to the working fluid, transparent cover which allows short wave radiation to pass and prevents them from exiting, and insulation which resists back and rear side heat losses [4]. They gather the sun's energy, transform its radiation into heat, and then transfer that heat to a fluid usually water or air [5, 6, 7]. The solar flat plate perform three functions as absorbing solar radiation, converting it to heat energy, and transferring the energy to a working fluid passing through the collector [4].

Solar flat plate is very important because, the flat plate solar thermal collector is much more cost effective than the evacuated tube solar thermal collector for low temperature applications such solar cooling and heating.

Solar Flat collectors are the most basic design of solar collectors invented in the 1880s for buildings applications [8]. The average monthly energy efficiencies of the solar collector in July and August were 45.3% and 32.9%, respectively, while the average monthly exergy efficiencies reached 2.62% and 2.15%, respectively [9]. Adel A. Ghoneim et al., [10] introduce a detailed nonlinear model for evacuated tube solar collectors where a more comprehensive optical and thermal analysis is proposed.

Solar collectors are the main components of a solar heating system. The collectors collect the sun's energy, transform this radiation into heat, and then transfer this heat into a fluid, water or air, which has many household or industrial applications [11]. Performance and efficiency of solar collectors depends on various factors like collector & receiver material, solar radiation intensity, nature of working fluid and on selective coatings etc. have a very crucial role in through the effective utilization of solar thermal energy [12]. They investigate the performance characteristics of solar flat plate collector with different selective surface coatings & by using varying concentrations of heat transfer fluid mixture. The selective coating helps us to increase the incident radiation and heat resistance of the material which would increase the operating temperature. [13].

B. Kalidasan and Srinivas T. [14] showed in their study that among the three different selective coatings tested; the thermal efficiency for black chrome coating is higher. For the black chrome coating the absorptivity was 0.93 and its emissivity was 0.10. The cover must be selected for the highest transmittance of the air mass, solar spectrum of radiation and the lowest transmittance of the infrared reradiated from absorber.

Federico G. et al. [15] analyses the performance of newly developed, highly transmitting and spectrally selective glass coatings based on transparent conductive oxides (TCO) for the use in flat plate collectors: Uncovered, single-glazed and double. Glazed designs are taken into consideration. For them, glass represents an ideal material for the use in solar energy applications thanks to its high solar transmittance, long-term stability and low cost.

Also, according to Vettrivel, H. and Mathiazhagan, P., [16] the efficiency of the solar collector is depend with many parameters such as number of glass cover, wind velocity, space between absorber plate to the glass cover and overall top loss heat transfer coefficient out of which top loss heat transfer coefficient top loss plays an important role for design of solar collector.

X. Zhao, Z. Wang, Q. Tang., [17] investigated the impact of absorber glazing covers. For them, under certain system operating conditions, i.e., weather and water parameters, the efficiency of the solar system would depend upon its own structure, particularly type of the top glazing cover. Ideally the glazing should allow maximum solar irradiance to be transmitted and also be able to minimize heat losses from the absorber to ambient. Double skinned polycarbonate and borosilicate are potential materials for this use due to their higher solar transmittances and lower values. Adsten, M. [18] do studies from the material properties of the collector components to performance analyses of the whole collector. It is important to have knowledge about the components to be able to optimize the system. Solar collectors are the major component in solar thermal systems. Based on the literature the present work focuses toward the effects of optical parameter such as thickness and coefficient transmission of glass cover on the outlet temperature of air. The pressure in the optimal solar plat plate collector are also considered.

2. Theoretical Modeling

As knows in the introduction part, there are three principal parts of solar flat plate collector: absorber plate, transparent cover and insulation. Figure 1 shows a cross section of a conventional solar flat plate collector. Each of the components of the solar flat plate collector must be analyzed. In our study, we are interested to the glass cover and their optical parameters as absorptance, emittance, reflectance and transmittance.

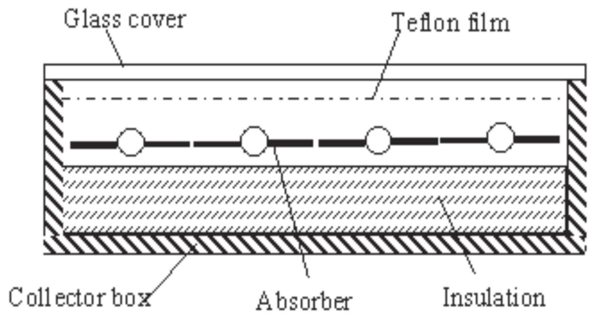

Figure 1. *Conventional solar flat plate collector.*

2.1. Energy Balance

Advantage of flat plate collectors is that they have a simple design which favours ease of manufacture [7].

The thermal balance of solar converters depends on numerous parameters, as: absorber optical properties, working temperature, radiation concentration, convection transfer coefficient.

Absorber plate

The absorbent plate exchanges heat with the other components of the solar flat plate and with the ambient atmosphere by convection, radiation and conduction. In addition, it absorbs part of the solar radiation transmitted by the glass cover.

Solar radiation exchanged by convection-conduction between the absorber plate and the glass cover are given by equation 1 [19, 20].

$$\varphi_{c,p-c} = h_{c,p-c}\left(T_{pm} - T_v\right)S \quad (1)$$

Heat exchanged by radiation between the absorber plate and the glass cover are given by equation 2.

$$\varphi_{r,p-c} = \sigma \frac{T_{pm}^4 - T_v^4}{\frac{1}{\varepsilon_{pi}} + \frac{1}{\varepsilon_{ci}} - 1} S \quad (2)$$

Heat exchanged by convection-conduction between the absorber plate and the insulation are given by equation 3.

$$\varphi_{c,p-i} = \frac{T_{pm} - T_i}{\frac{1}{h_{bp}}} S \quad (3)$$

If we assume that:

$\rho_1 \sigma T_{pm}^4$ the density flow rate emitted by the absorber plate;

$\tau_1 E$ the fraction of the solar radiation E of the glass cover transmitted by it and supposed to be entirely absorbed by the absorber plate;

$\rho_2 \sigma T_{pm}^4$ the fraction of the radiation emitted by the absorber plate and reflected by the glass cover;

$\varepsilon_v \sigma T_v^4$ the radiation emitted by the glass cover towards the absorber plate;

$h_{pv}\left(T_{pm} - T_v\right)$ and $h_{pi}\left(T_{pm} - T_i\right)$are respectively the heat losses between the absorber plate and the glass, and between the absorber plate and the insulation.

Then in view of the above the thermal balance of the absorber plate can be written:

$$\rho_1 \sigma T_{pm}^4 = \tau_1 X E - \rho_2 \sigma T_{pm}^4 + \varepsilon_v \sigma T_v^4 - h_{pv}\left(T_{pm} - T_v\right) \\ - h_{pi}\left(T_{pm} - T_i\right) \quad (4)$$

X is the concentration factor

$$X = \frac{S_r}{S_{pm}}$$

With:

S_r the area of reflector;

S_{pm} the area of absorber.

Glass cover

The glass covers exchange by convection with the ambient environment under the action of the wind and by radiation with the celestial vault and with the absorber plate. On the other hand, it absorbs part of the solar radiation IR and exchanges with the air confined in the solar flat plate.

Heat exchanged by convection between the glass cover and the ambient environment. It depends on the wind speed and is given by:

$$\varphi_{c,c-a} = h_{wind}\left(T_v - T_a\right)S \quad (5)$$

Flux exchanged by radiation between the glass cover and the ambient environment. It is given by equation 6.

$$\varphi_{r,c-a} = \sigma \alpha_{ci}\left(T_v^4 - \varepsilon_a T_a^4\right)S = \sigma \alpha_{va}\left(T_v^4 - T_c^4\right)S \quad (6)$$

If we assume that:

$2\varepsilon_v \sigma T_v^4$ the power emitted by the two faces of the glass cover;

$\alpha_1 E$ the fraction of the solar radiation absorbed;

$\alpha_2 \sigma T_{pm}^4$ the fraction absorbed of the radiation emitted by the absorber plate;

$\alpha_3 \sigma T_a^4$ the radiation of the environment absorbed;

$h_v\left(T_v - T_a\right)$ and $h_{pv}\left(T_{pm} - T_v\right)$ are respectively the thermal losses between the glass cover and the ambient environment and between the glass cover and the absorber plate.

Then in view of the above the thermal balance of the glass cover can be written:

$$2\varepsilon_v \sigma T_v^4 = \alpha_1 E + \alpha_2 \sigma T_{pm}^4 + \alpha_3 \sigma T_a^4 \\ - h_v\left(T_v - T_a\right) - h_{pv}\left(T_{pm} - T_v\right) \quad (7)$$

With

$$h_v = h_{vent} + \sigma \varepsilon_v \left(T_v + T_a\right)\left(T_v^2 + T_a^2\right)$$

System of global balance equations

The resolution of the system of equation below makes it possible to determine the temperatures of the absorber plate and the glass cover [19, 20].

$$\begin{cases} \rho \sigma T_{pm}^4 = \tau_1 E - \rho_2 \sigma T_{pm}^4 + \varepsilon_v \sigma T_v^4 - h_{pv} \\ \quad \left(T_{pm} - T_v\right) - h_{pi}\left(T_{pm} - T_i\right) \\ \quad . \\ \quad . \\ 2\varepsilon_v \sigma T_v^4 = \alpha_1 E + \alpha_2 \sigma T_{pm}^4 + \\ \alpha_3 \sigma T_a^4 - h_v\left(T_v - T_a\right) - h_{pv}\left(T_{pm} - T_v\right) \end{cases} \quad (8)$$

2.2. Solar Flat Plate Collector Efficiency

Thus, the thermal collector efficiency is based on the standard second-order collector performance given by [21]:

$$\eta = \frac{\rho_1 \sigma T_{pm}^4}{XE} = \alpha \left[1 - \frac{-\rho_2 \sigma T_{pm}^4 + \varepsilon_v \sigma T_v^4 - h_{pv}\left(T_{pm} - T_v\right) - h_{pi}\left(T_{pm} - T_i\right)}{XE}\right]$$

Assuming that ρ_2 is the emissivity factor of the glass cover then the efficiency becomes:

$$\eta = \alpha \left[1 - \frac{\varepsilon}{\alpha} \frac{\sigma\left(T_{pm}^4 - T_v^4\right)}{XE}\right] - \frac{h_{pv}\left(T_{pm} - T_v\right) + h_{pi}\left(T_{pm} - T_i\right)}{\alpha XE}$$

So: $\eta = \eta_R - \eta_C$

With

$$\eta_C = \frac{h_{pv}\left(T_{pm} - T_v\right) + h_{pi}\left(T_{pm} - T_i\right)}{\alpha XE}$$

If we neglect the convective term then the efficiency will be equal to the radiative efficiency

$$\eta = \eta_R = \alpha \left[1 - \frac{\varepsilon}{\alpha} \frac{\sigma\left(T_{pm}^4 - T_v^4\right)}{XE}\right]$$

3. Numerical Modeling

3.1. CFD Modeling

Generally, the Computational fluid dynamic (CFD) tool was related to solving the problems for non-linear partial differential equations that were described as the behaviors and phenomena of the fluid dynamics [22].

Many researchers have reported the application of NFs in thermal systems. In many studies, computational fluid dynamics (CFD) is employed to determine the aerodynamic coefficients of parabolic trough collectors [23, 24]. In order to better perform the CFD analysis of our problem, we first proceeded to create the geometric model of the problem domain according to the design specifications. The domain of the problem considered here is a closed enclosure with an area of $1m^2$ with a 4 cm of air gap thickness between the absorber and the glass cover with thickness of 6 mm. A 3D geometry of the conventional solar collector was created by ANSYS Workbench using Ansys Spaceclaim, which is a design tool for developing the geometric models of the physical problem domain [25, 26]. Figure 2 shows the geometric model of the solar collector with the same dimensions as the designed experimental model.

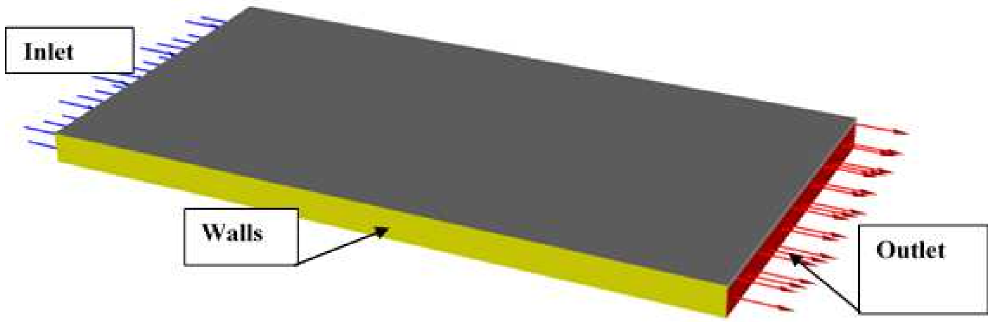

Figure 2. *Geometry of solar flat plate.*

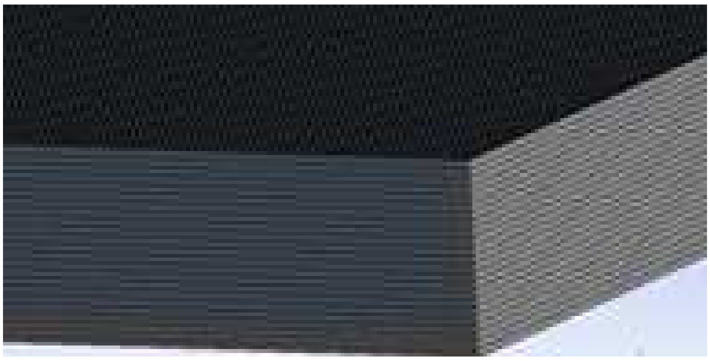

Figure 3. *Mesh.*

Since the geometry of solar collector does not involve any curved surfaces, the quadratic mesh method is the most suitable for the considered problem and could provide accurate results with moderate computation time, as shown in Figure 3. The total number of nodes and elements in the meshed domain is 3,054,096 and 2,909,500, respectively, which is sufficient from the point of view of the problem's complexity to be solved.

For an adequate setting, we carried out the simulation with the solar load model provided by ANSYS FLUENT to determine the solar radiation which enters the calculation domain. Two options were available for the model: Tracing of solar rays and irradiation in discrete ordinates. The ray tracing approach is an extremely efficient and practical way to apply solar loads as heat sources in energy equations. For optical thicknesses greater than 3 mm like our case, the Rosseland model is more efficient. In this case, the Rosseland radiation model with solar charge and solar ray tracing was used [27]. This model allowed us to calculate the intensity of incident solar radiation on a surface, as well as the ambient temperature when the latitude and altitude of the application site were given.

3.2. Conditions and Types of Limit for the Model

To define the boundary conditions and the appropriate types was essential for the accurate solution of a fluid flow problem. Most of the boundary conditions have been established by physical phenomena. However, some have been established by the ANSYS simulation software. Table 1 displays the types of conditions for the different parts of the domain studied. Real or physical boundary conditions are idealized and simplified to put them in the simulation. For example, in this study, the side walls of the solar collector, which were insulated, were considered to be adiabatic.

Table 1. *The types of conditions for the different parts of the domain studied.*

Name	Type	Thermal Conditions	Description	Wall Thickness (mm)
Glass Wall	Wall	Convection losses (2 w/m^2.k)	Semi-transparent	6
Absorber Wall	Wall	Adiabatic wall (Heat flux = 0)	Opaque	2
Bottom Wall	Wall	Adiabatic wall (Heat flux = 0)	Opaque	3
Side Walls	Wall	Adiabatic wall (Heat flux = 0)	Opaque	3

Table 2. *Models and input parameters used for the simulation of the solar collector.*

Function	Specification	
Solver setting	Space	3D
	Time	Unsteady, first order implicit
	Viscous model	k-omega SST
	Inlet velocity	0.02m/s
	Radiation	Rosseland radiation model, solar loading, and solar ray tracing Longitude: -17.26°O Latitude: 14.41° N Timezone (+GMT): 0 Day: 21 Time: 13H AM Month: June Solar calculator
Material Properties	Glass; cooper; wood; Air	Thermo-physical properties including density, thermal conductivity, and specific heat capacity of the materials

3.3. Models Selection and Constants for Simulation

The models and the operating parameters used for the simulation of the solar collector are given in Table 2.

After giving the input parameters, the solution was initialized. The time step for the iterations was set at 0.001–1.0 s based on the ease of convergence and the time needed to complete the simulation.

4. Results and Discussion

We carried out the simulation on June 21 at 1 p.m. with a first simulation which allowed us to make a comparative study of the glass cover thickness effect on the outlet temperature of the heat transfer in air.

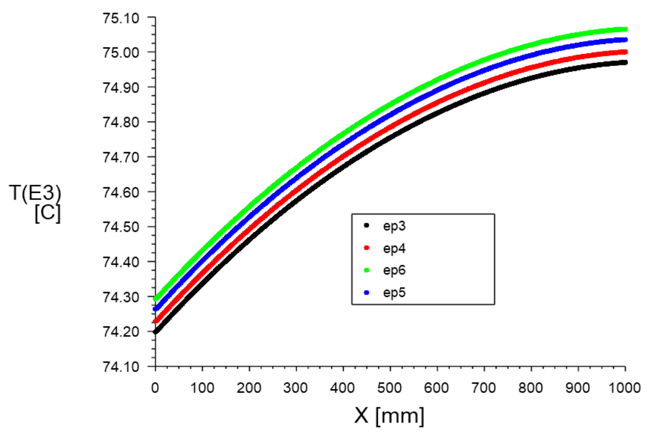

Figure 4. *Evolution of the temperature on the transverse median axis of the air gap for different thicknesses of the glass cover.*

Figure 4 shows that the temperature of the heat transfer fluid increases throughout the collector to reach temperatures of 74°C to 75°C depending on the thickness of the glass cover.

It also shows that below a thickness of 6 mm, the temperature of the heat transfer fluid decreases. Thus we can say that the optimal thickness of our solar thermal collector is 6 mm.

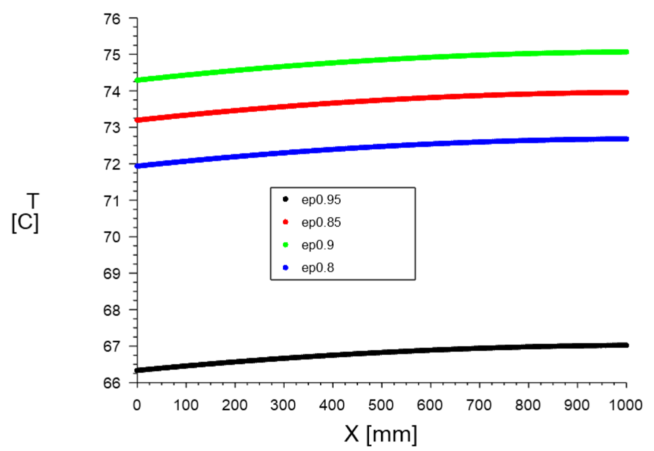

Figure 5. *Evolution of the temperature of the heat transfer fluid along the solar thermal collector for different values of transmission coefficient.*

The evolution of the temperature of heat transfer fluid along the solar thermal collector for different values of the transmission coefficient of the glass cover is given by figures 5, 6, 7, 8 and 9.

Figures 6, 7, 8 and 9 showed that the outlet temperature of the heat transfer fluid varies with the transmission coefficient. Figure 4 shows that the maximum outlet temperature is reached with a coefficient of transmission of 0.90. Thus with our solar thermal collector the optimal transmission coefficient is 0.9.

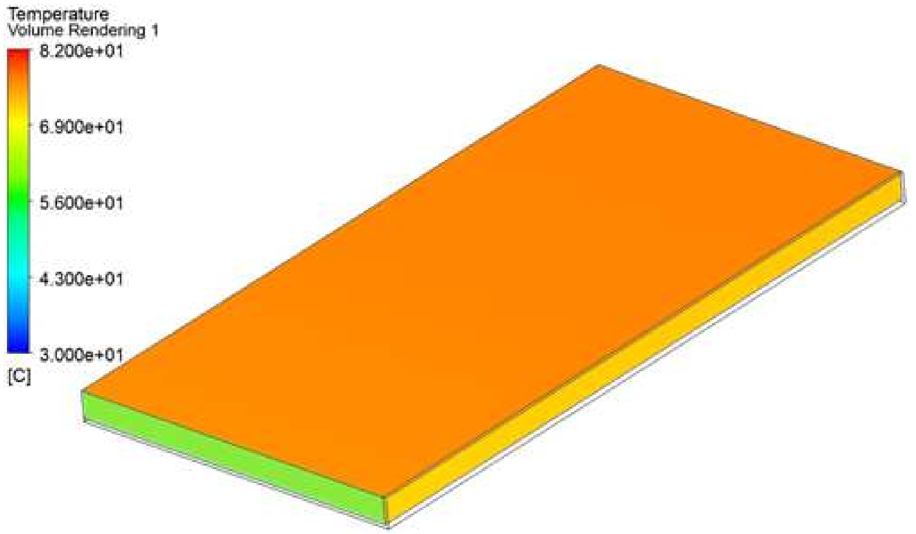

Figure 6. *Evolution of the temperature of the heat transfer fluid along the solar thermal collector for a transmission coefficient epsilon=0.8.*

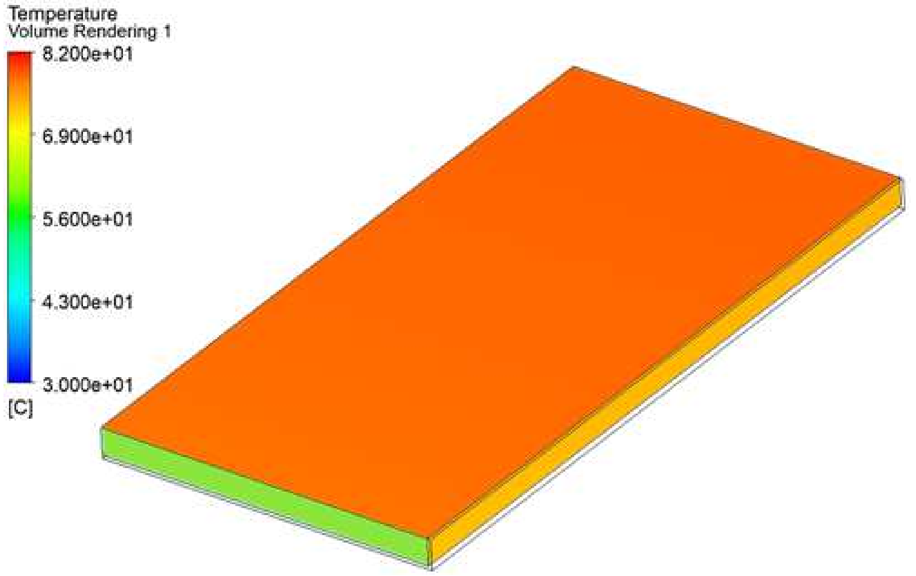

Figure 7. *Evolution of the temperature of the heat transfer fluid along the solar thermal collector for a transmission coefficient epsilon=0.85.*

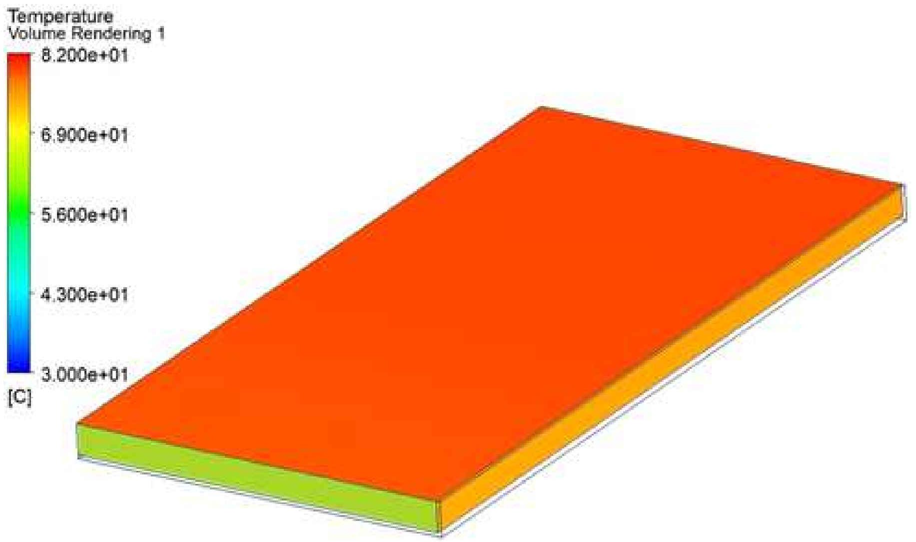

Figure 8. *Evolution of the temperature of the heat transfer fluid along the solar thermal collector for a transmission coefficient epsilon=0.9.*

Figure 10 gives the evolution of the pressure in the solar thermal collector for the optimal coefficient of transmission.

To check the pressure in the solar thermal collector, we displayed the value for the transmission coefficient equal to 0.9. We note that there is no risk of damage to the glass caused by the excess pressure because the critical pressure value has not been reached for 6 mm of thickness of glass cover. The particles have not expanded enough to create an overpressure in the solar thermal collector.

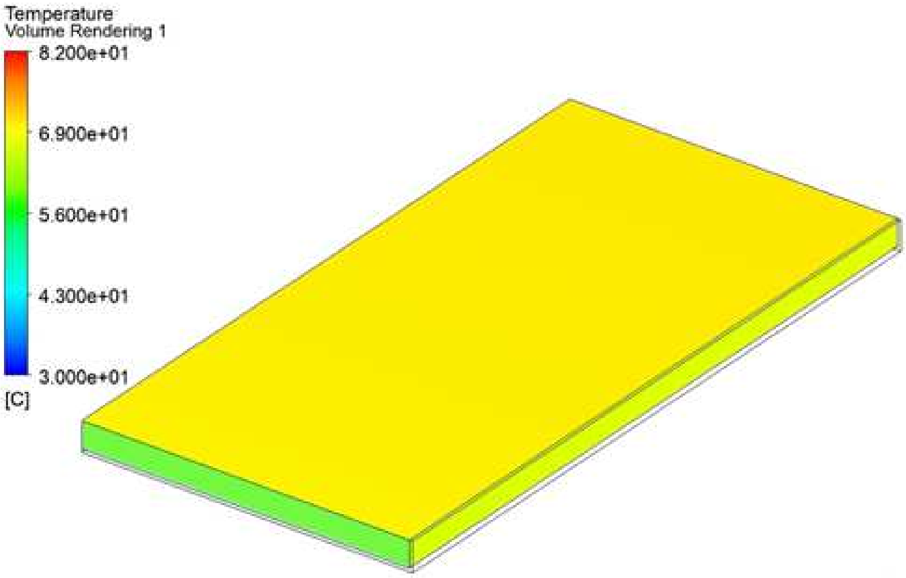

Figure 9. *Evolution of the temperature of the heat transfer fluid along the solar thermal collector for a transmission coefficient epsilon=0.95.*

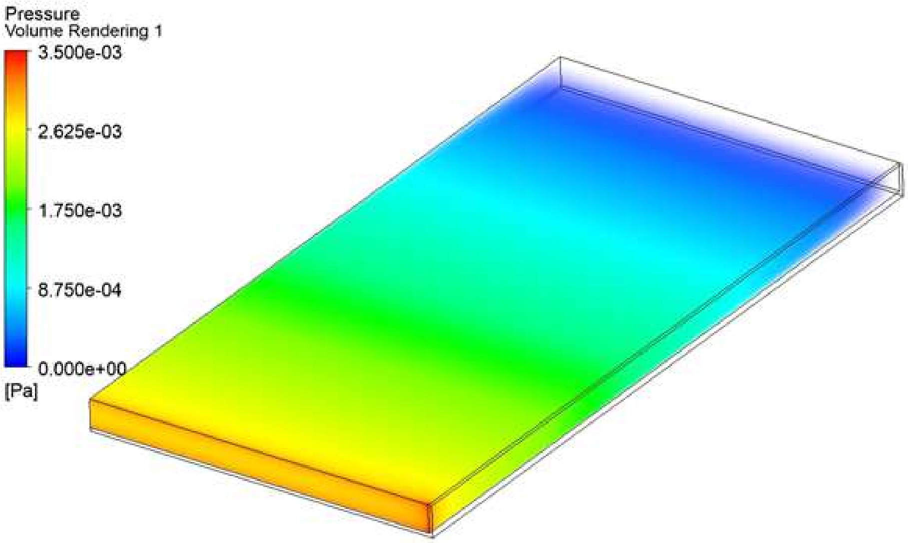

Figure 10. *Evolution of the pressure in the solar thermal collector with a transmission coefficient equal to 0.9.*

5. Conclusion

The solar thermal collector is an essential element for heat production systems for the drying of agro-food products and the production of solar thermal cold. Thus, to achieve acceptable solar thermal output temperatures, it is necessary to properly optimize the system. In this study we have determined the optimal thickness and the optimal transmission coefficient of the glass cover of 6 mm and 0.9 respectively. With these values, the outlet temperature of the heat transfer fluid can reach 75°C, therefore able to meet our drying and heating needs.

Nomenclature

S: area Absorber area (m^2)

E: Solar radiation intensity perpendicular to collector surface (W/m^2)

h: Heat transfer coefficient (Wm^2/K)

T: Temperature (K)

V: Wind speed velocity (m/s)

X: concentration factor
x: Solar collector lengh (mm)

Greek Symbols

α, α': Absorptivity
ε: Emissivity
η: Efficiency

Acknowledgements

This work was supported by the laboratory of chemistry and physics of materials of the Assanse Seck University of Ziguinchor, Senegal.

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