# *Effect of the Double Glazing on the performance of an Air Solar Collector*

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**ABSTRACT**

In this paper, the effect of double-glazing on the performance of a solar collector operating under the desetic climatic conditions of Adrar region has been investigated. This study based on a mathematical model obtained from the application of energy conservation laws at different parts of the system and performed in a winter typical day on January 1$^{st}$, 2015 in Adrar. To solve these systems of differential equations, the Finite Differences Method with an implicit scheme is used. The resulting matrix system is solved iteratively using Gauss Seidel algorithm. A program written in FORTRAN is developed for achieving the parametric study. This work aims to investigate the variation of temperature for each component of the system, the temperature difference in terms of time as well as the effect of the gap space between the two glass covers on the collector performance.

## I. Introduction

Solar energy is an excellent source of energy for heating air which can be used for grain drying, space heating and many other applications. The performance of a solar heater depends on the losses from the collector surfaces. One of the simplest and most direct applications of this naturally and freely available energy is to convert solar radiation into heat. This conversion is accomplished using solar panels with the objective of heating water or air for domestic and industrial applications [1]. Glazing is the top cover of a solar collector. It performs three major functions in particular: to minimize convective and radiant heat loss from absorber, to transmit the incident solar radiation inside the solar collector with minimum loss, and to protect the absorber plate from outside environment [2, 3]. Other important characteristics of glazing materials are reflection ($\gamma$), absorption ($\alpha$), and transmission ($\tau$). In order to attain maximum efficiency, reflection and absorption should be as low as possible, while transmission should be as high as possible [4]. In theory, the performance of solar collector depends on climatic conditions and several operating condition such as collector orientation and slope, thickness of cover materials, wind speed, collector length, collector depth, and the material of absorber [5, 6]. The objective of the present study was to investigate the effect of the double-glazing on the performance of an air solar collector in the climatic conditions of Adrar region [7].

## II.Physical domain modelling

The physical domain of a double glazing air thermal solar collector makes us possible to determine the characteristics of each element from which it is composed. The detailed model can be described as follows : The double glazing collector that we will study is divided into seven identical layers as can be seen in **figure (1).**

The upper part of the collector is made from two plates of glass for increasing the greenhouse effect **(1, 3).** These glasses are separated by an air space **(2).** The absorbing plate of the collector **(5).** The heat transfer fluid **(6)** circulates underneath the absorber plate **(5).** Finally, the system is envelopped with an insulation layer to reduce the heat losses to the ambient medium ([8-9]).

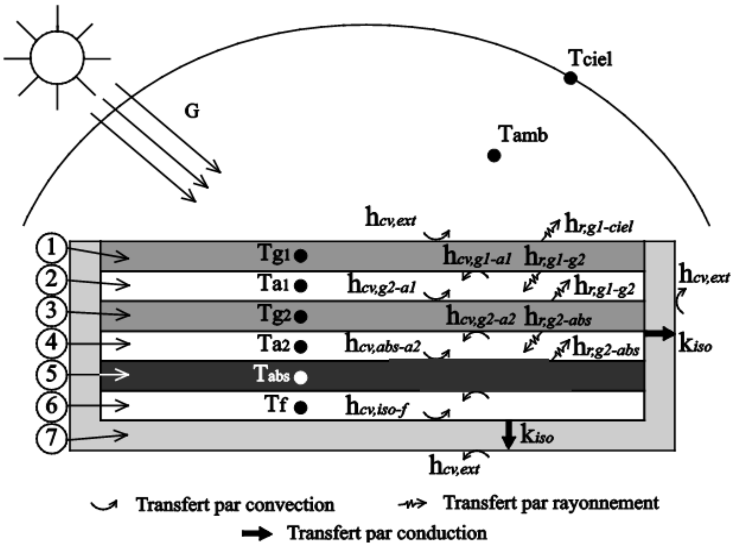



**Figure 1** Representation of the double-glazing flat plate solar collector [7]

**1.** First layer of glass. **2.** First layer of air . **3.** Second layer of glass. **4.** Second layer of air. **5.** Absorbing plate . **6.** Heat transfer fluid . **7.** Insulation and Envelope .

## Main technical specification of collector

Distance between the absorber and the second glass: **dp= 3cm.**
Distance dv between the first glass and the second glass is

| | | |
|---|---|---|
| Variable where | : | **dv ∈] 0.1 cm, 2 cm**. |
| The hourly air flow rate | : | **dvdt = $m^3$ / h.** |
| Épaisseur de l’isolant | : | **$e_{pi}$ = 5 cm.** |
| Glass thickness | : | **$e_{pv}$ = 2mm**. |
| Absorber thickness | : | **$e_{pp}$ = 1mm.** |
| Insulation thickness | : | **$e_{pi}$ = 5cm.** |

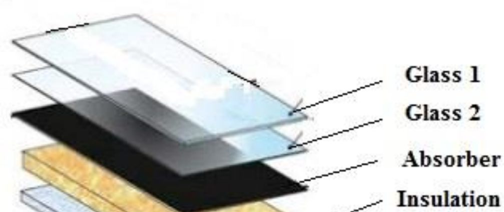


**Figure2:** Double glazing collector components.

## III.Mathematical modeling

A transient mathematical model is developed for describing the thermal behaviour of the flat solar collector. In this proposed model, the Control Volume Finite Differences Method is used, for that the solar collector which is divided into four parts (glass cover, absorber, fluid and insulation) according to the perpendicular direction of the liquid flow ; see the diagram in figures (1and figure2).

### Governing equations for double-glazing flat plate collectors

The energy balance for the double glazing solar collector can be written by applying the principle of energy conservation at each component of the system.

### A) Simplifying assumptions

In order to simplify the model, the thermal analysis is carried out based on the following assumptions:

(a) The sky is considered as a black body of equivalent temperature.

(b) The ground temperature is taken equal to the ambient temperature.

(c) Radiant heat exchange surfaces are assumed to be gray and diffusing bodies.

(d) The physical properties of the materials are assumed to be constant.

(e) The wind speed is supposed to be parallel to the faces of the system.

(f) The air temperature at the inlet of a slice is equal to its outlet from the previous one.

(g) The flow regime is transient.

(h) The temperatures of the interior and exterior faces of the glass, absorber, and insulation are assumed to be uniform.

(i) The temperature of the absorber lower face which is in opposite of the insulation is at the same temperature of its upper face.

### B) Energy balance in the double glazing solar collector [8]

In figure (1), it is shown the different modes of heat transfers occurring in the system:

The first layer of glass (1) is subjected to two convective transfers: The first one between the ambient air and its upper face and the second between the first air layer (2) and its lower face. It is also subject to two radiative transfers: The first is between the sky and its upper face and the second between its lower face and the second layer of glass (3).

The second layer of glass (3) is subjected to two convective transfers: (one between the first layer of air (2) and its upper face and the other between the second layer air (4) and its lower face) but also with two radiative transfers (one with its upper face and the lower face of the first layer of glass (1) and the other between the upper face of the absorber (5) and its underside).

The absorbing plate (5) is subjected to two convective transfers (one between the second layer of air and its upper face and the other between the heat transfer fluid (6) and its lower face) but also to a radiative transfer between its upper face and the second layer of air (3).

The heat transfer fluid (6) is subjected to two convective transfers: the first with the absorber and the second with the insulation.

The heat losses from the bottom of the collector are characterized by a conductive flow passing through the insulation and a convective flow between the insulation and the ambient air. In addition, it should be noted that each layer (1-6) has a lossy side wall characterized by a conductive flow associated with a convective flow.

## C) Thermal balance of the system

To model the solar collector, we apply the first principle of thermodynamics:

$$\varphi_e + \varphi_g = \varphi_s + \varphi_{st} \qquad (1)$$

With:

$\varphi_{st}$ : The flow of stored heat.

$\varphi_g$ : The heat flow generated.

$\varphi_e$ : The incoming heat flow.

$\varphi_s$ : The heat flow output.

**i) For the first layer of glass**

**-The outer face**

$$\frac{C_{pv1}M_{v1}}{2S}\frac{\partial T_{v1e}}{\partial t} = \frac{P_{v1}}{2} + h_{c,\mathrm{v1e-a}}(\mathrm{T}_a - T_{v1e}) + h_{r,v1e-ci}(T_{ci} - T_{v1e}) + h_{cn,v1}(T_{v1i} - T_{v1e}) \qquad (2)$$

**- The inner face**

$$\frac{C_{pv1}M_{v1}}{2S}\frac{\partial T_{v1i}}{\partial t} = \frac{P_{v1}}{2} + h_{c,\mathrm{v2e-}v1i}(\mathrm{T}_{v2e} - T_{v1i}) + h_{r,v2e-v1i}(T_{v2e} - T_{v1i}) + h_{cn,v1i}(T_{v1e} - T_{v1i}) \qquad (3)$$

**ii) For the second layer of glass**

**-The outside**

$$\frac{C_{pv2}M_{v2}}{2S}\frac{\partial T_{v2e}}{\partial t} = \frac{P_{v2}}{2} + h_{c,\mathrm{v2-}v1i}(\mathrm{T}_{v1i} - T_{v2e}) + h_{r,v2e-v1i}(T_{v1i} - T_{v2e}) + h_{cn,v2}(T_{v2i} - T_{v2e}) \qquad (4)$$

**-The inside face**

$$\frac{C_{pv2}M_{v2}}{2S}\frac{\partial T_{v2i}}{\partial t}=\frac{P_{v2}}{2}+h_{c,p-v2i}(T_p-T_{v2i})+h_{r,p-v2i}(T_p-T_{v2i})+h_{cn,v2}(T_{v2e}-T_{v2i}) \quad (5)$$

**iii) For absorber plate**

$$\frac{C_{PP}M_P}{S_P}\frac{\partial T_p}{\partial t}=\frac{P_{v2}}{2}+P_p+h_{c,p-f}(T_f-T_p)+h_{c,p-v2i}(T_{v2i}-T_P)+h_{r,p-v2i}(T_{v2i}-T_P) \quad (6)$$

**iv) For the heat transfer fluid**

$$\frac{C_{pf}M_f}{S}\frac{\partial T_f}{\partial t}=h_{c,p-f}(T_p-T_f)+h_{c,f-I_i}(T_{I_i}-T_f)+\frac{\dot{m}C_{pf}}{S}\frac{\partial T_f}{\partial t} \quad (7)$$

**v) For the insulation material**

**- The inside face**

$$\frac{C_{pI}M_I}{S}\frac{\partial T_{I_i}}{\partial t}=h_{c,f-I_i}(T_f-T_{I_i})+h_{cn,I}(T_{I_e}-T_{I_i}) \quad (8)$$

**-The outer face**

$$\frac{C_{pI}M_I}{S}\frac{\partial T_{I_e}}{\partial t}=h_{vent}(T_a-T_{I_e})+h_{cn,I}(T_{I_i}-T_{I_e}) \quad (9)$$

## IV. Results and discussion

### 1-The results that come from the variation over time

The following results shown in **figures (3, 4 and 5)** indicate the temperature variations according to the legal time obtained on **January 1st, 2015** with a fixed distance between the two panels **dv=2cm**.

**Temperature variation**

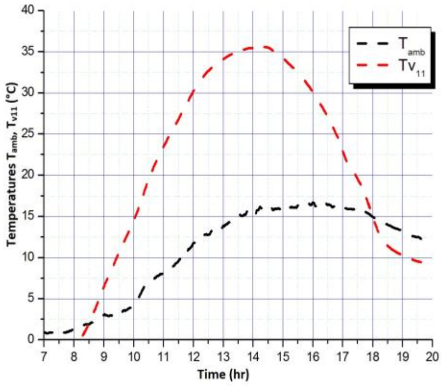



**Figure3:** Comparison between the variation of temperature of $\mathbf{T_{amb}}$ and $\mathbf{Tv_{11}}$ as a function of time

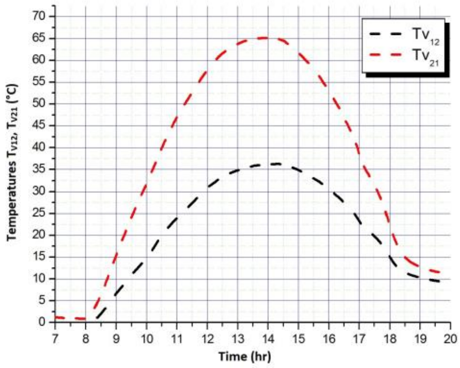



**Figure4:** Temperature variation of protecting glass $\mathbf{Tv_{12}}$ and the temperature of the intermediate glass $\mathbf{Tv_{21}}$

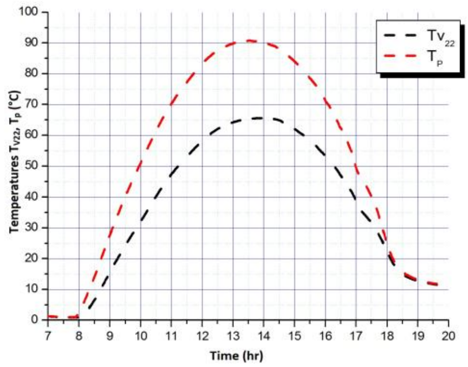



**Figure5:** Comparison between the variation of temperature of **$T_{v22}$** and **$T_p$** as a function of time.

In **figures (3,4,5),** it is shown that the different temperatures of the components of the solar collector vary in proportion to the power absorbed by each component and reach their maximum values in the period from 13 p.m. to 15p.m. The interior temperature of the intermediate glass is slightly higher than the temperature of the interior face of the protective glass. This is mainly due to the increase of greenhouse effect.

In fact, the temperature increasing of the protective and intermediate glasses compared to the temperature of the outside surface of the protective glass is due to the heat exchange with the absorber.

The inside face temperature of the intermediate glazing is lower because the glass is transparent and has not a large absorption coefficient since it transmits a great part of solar energy to the absorber.

In addition, it should be noted that the temperature of the absorbing plate is the highest compared to the other components.

**2-The results dealing with the variation of the air space beween the double glazing**

**-Variation of the convective heat transfer coeffcient $h_{c-v2v\,1}$, -Variation of the fluid outlet temperature $T_{fs}$**

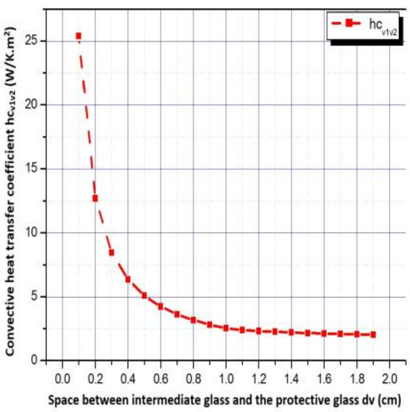



**Figure 6** : Variation of the **$h_{c-v2v1}$** convection exchange coefficient as a function of the air space $esp_{v2v1}$.

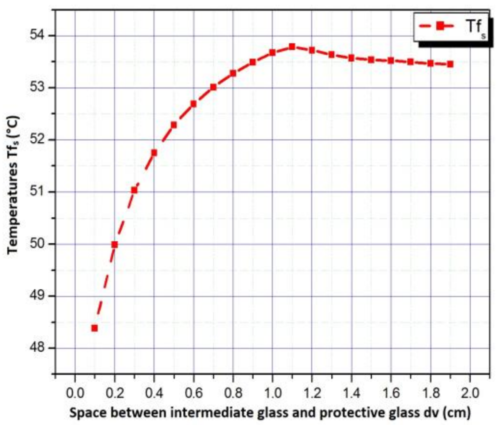
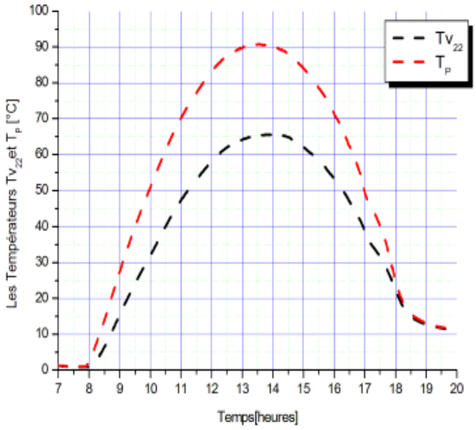
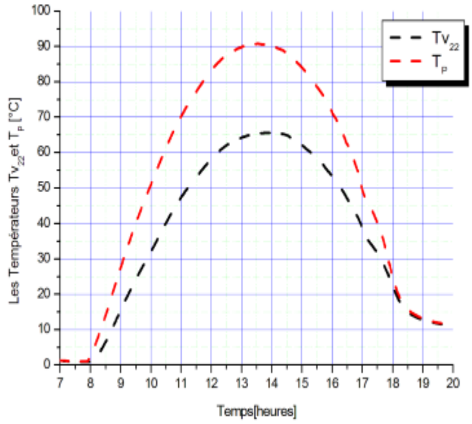



**Figure 7** : Varation of the fluid temperature as a function of the air space **$esp_{v\,2v1}$**.




**-Thermal efficiency variation $\eta$gl;**

**- Optimal distance between the double glass**

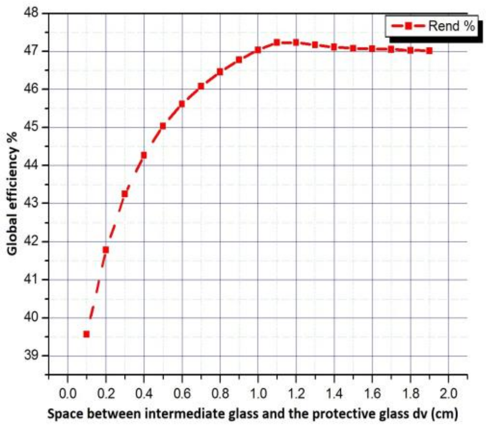



**Figure 8**: Variation of thermal efficiency $\eta$**gl** as a function of the air space $ep_{\mathbf{v2v1}}$

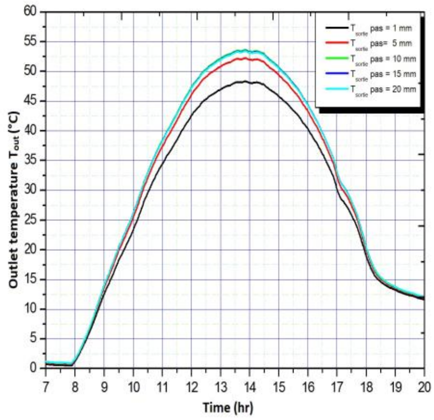



**Figure 9**: Effect of air space on the fluid oulet temperature

According to **figure 6**, when increasing the space between the intermediate and protective glass covers, there are two areas. First zone $0.1cm \leq ep_{v2v1} \leq 1cm$ there is a significant decrease in the exchange coefficient by convection $h_{c-v2v1}$ between the intermediate and protective glasses. Second zone $1cm \leq ep_{v2v1} \leq 1.9cm$ the exchange coefficient by convection $h_{c-v2v1}$ is almost constant.

According to **figure 7**, when increasing the space between the intermediate and protective glasses, there are two areas. First zone $0.1cm \leq ep_{\mathbf{v2v1}} \leq 1.2cm$ there is a significant increase in temperature of the heat transfer fluid $T_{\mathbf{fs}}$. Second zone $1.2cm \leq ep_{\mathbf{v2v1}} \leq 2cm$ the temperature of the heat transfer fluid $T_{\mathbf{fs}}$ is almost constant.

According to **figure 8,** when increasing the space between the intermediate and protective glasses, there are two areas. First zone $0.1cm \leq ep_{\mathbf{v2v1}} \leq 1.2cm$, we note a significant increase in overall performance $\eta$glo. Second zone $1.2cm \leq ep_{\mathbf{v2v1}} \leq 2cm$ the overall yield is almost constant $\eta_{\mathbf{glo}}$.

The **figure 9** illustrates the evolution of the maximum temperature of the fluid as a function of time for different steps between the two panels, it appears clearly that the increase in the distance between the two panels of the collector leads to an increase in the temperature of the leaving fluid, the latter is therefore proportional to the increase in pitch. We also note that the optimal temperature of fluid is found to be maximal at $ep_{v2v1}$ = 1cm.

## V.Conclusion

This study deals with the effect of the double-glazig on the performance of an air solar collector. The obtained results allow us to draw the following conclusions :

Generally, the thermal efficiency is maximal between 13p.m and 15 p.m. When the space between the intermediate and protective glasses in creased, the overall efficiency increases slightly up to a maximum value for epv1v2=1.2cm. The optimum space between the intermediate and protection covers is epv2v1=1.2cm. To have maximum efficiency from our solar collector, the space between the two panels should be taken as 1.2cm. The developed model can be used by designers for predicting the performance of the prototype in predefined contexts. The mode of heat transfert remains conductive in the range of 1.2-2cm.

Finally, it should be noticed that the authors decided to continue their work by developing an experimental setup to compare the theoretical results with those obtained experimentally and also to study in situ its thermal behaviour.

## Nomenclature

$\alpha$ : Absorptivity

$\tau$ : Transmissivity

$\eta$ **:** Collector efficiency

$S$ : The collector surface area [m$^2$]

$T_p$ : The temperature of the absorber [K]

$T_{amb}$ : Ambient temperature [K]

$T_{v1e}$ : The temperature of the outer side of the outer pane [K].

$T_{v1i}$ : The temperature of the inner side of the outer pane [K]

$T_{v2e}$ : The temperature of the outer side of the inner pane [K]

$T_{v2e}$ : The temperature of the inner side of the inner pane [K]

$T_{Ie}$ : The temperature of the outer face of the insulation [K]

$h_{c,p-v2i}$ : Convective heat transfer coefficient between absorber plate and glass [W/m$^2$K.]

$h_{c,p-f}$ : Convective heat transfer coefficient between absorber plate and fluid [W/m$^2$K].

$h_{r,p-v}$ : Radiative heat transfer coefficient between absorber plate and glass [W/m$^2$K].

$h_{r,v1-v2}$ : Radiative heat transfer coefficient between tow glasses [W/m$^2$K].

$h_{cn,v1}$ : Conductivity heat transfer coefficient in the glass 1[W/m$^2$K].

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