Thermal analysis of a flat plate collector with Solidworks and determination of convection heat coefficient between water and absorber

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Abstract:
In this study, a simulation of a flat plate collector is developed in order to determine its performance for different operation conditions. The design and the thermal analysis of the collector are made with the commercial software Solidworks in the flow simulation studio. The heat transfer phenomena inside the collector are complicated combining convection and radiation something that creates obstacles in the simulation. The determination of boundary conditions has a significant role in this simulation because of the nature of the problem. Four different cases are investigated by comparing selective and non-selective collector, with and without cover and the final results show that the selective covered collector performs better on usual operating conditions. For this case, the main flow characteristics are calculated and presented in this analysis. Moreover the convection heat transfer coefficient between water and riser tube was calculated 14.5% greater than the theoretical value for isotheral tube conditions. The total analysis is made for a strip of the collector because the examination of the total system has a huge computational time. Parameters as the transmittance–absorptance product (τα), absorber emissivity and existence of cover are investigated and in every case the efficiency curve is presented. The final results are useful for optimizing the collector and for determining the heat transfer conditions between absorber and water.

Keywords
Solidworks, flat plate collectors, simulation, flow simulation, efficiency

1. Introduction

The role of energy becomes increasingly important to fulfil needs of modern societies and to follow the fast economic and industrial growth worldwide. The reduction on fossil fuels, the cost of them and the environmental problems leads to use alternative and renewable ways to cover the huge energy demand. Solar energy is able to cover an important proportion of the consumed energy with low cost technologies such as flat plate collectors. For example, Greece is a country with annually incident solar energy 5800 MJ/m^2 [1] and in Athens the daily average radiation is approximately to 4.35 kWh/m^2 [2]. So it is obvious that the use of solar energy has an important role in energy map, especially for high irradiation level countries.

The simplest application of solar energy is solar domestic hot water systems with flat plate collectors. This is a low capital cost technology fact that makes it financial sustainable. But, the main goal is to make this system energy efficient too by improving the performance in usual operating conditions. The other important applications for solar collectors are industrial processes where the working fluid temperature varies for different applications. So flat plate collectors are used for heating water under 100^oC and concentrated collectors or evacuated tube collectors are used for vapor production, with the electricity production being the main application.

Solar energy collectors are special kind of heat exchangers that transform solar energy to internal energy of water. The thermal analysis of the collectors is very complicated because all the possible modes of heat transfer and radiation are taken into consideration. The determination of the heat losses coefficient is the main goal of an energetic analysis because this leads to the determination of the useful energy rate from the solar collector.

The analysis of thermal collectors is able to be done experimentally and by simulating the heat transfer phenomena inside them. Sopian et al. [3] conducted an experimental study on the thermal performance of a non-metallic unglazed solar water heater integrated with a storage system. Simultaneously many studies which compare experimentally results with CFD results have been made [4,5]. Many researchers have used exergy analysis in order to improve the efficiency of collectors by decreasing the losses. Farahat et al [6] and Chamoli [7] used matlab to optimazize a flat plate collector with this method. Parametrical analysis is also useful in order to determine the value of crucial parameters which influences on efficiency. Garg and Rani [8] calculated the overall heat loss coefficient and the collector efficiency under different conditions such as the absence of cover, with single and double glazing under different ambient conditions, tilt angles, wind speeds, emissivity of both glass cover and absorber plate. Pillai and Agarwal [9] discussed the influence of various parameters on the efficiency of solar collectors and concluded that at low solar insolation in the range of 200-600 W/m^2 double glazed collectors are superior to single glazed. Anderson et al [10] examined the performance by changing the colors of solar collector. Based on the transmittance-absorptance result of various colored collectors the hypothetical performances of these collectors were calculated using the Hottel-Whillier-Bliss 1-D steady-state model given by Duffie and Beckmann 2006 [11]. By these experiments they concluded that the color of the collector plays a major role in thermal efficiencies of the collectors.

The greater part of the researchers uses mathematical models and simulation programs in order to determine the efficiency of collectors. Augustus and Kumar [12], Janjai et al. [13] create mathematical models developed on empirical relation in their studies on collectors. Lecoeuche and Lalot [14] applied neural network technique to predict the thermal performance of a solar flat plate collector and Luminosu [15] used the temperature criterion aid by a program to simulate solar flat plate collectors. Gorla [16] performed an analysis based upon the two-dimensional finite element method to characterize the performance of solar collectors and Kaplanis and Kaplani [17] made an optimization of glazed and unglazed collector with MATLAB. On the CFD studies, ANSYS is the most common tool [18-20] in the solar collector's efficiency analysis. Manjunath et al [21] used FLUENT to simulate a part of an unglazed solar flat plate collector and Álvarez et al [22] COMSOL. Other tools as COLTST used from Hossain [23] and TRNSYS by Ponshanmugakumar and Vincent [24] and Bunea [25].

In this study, the computational tool is Solidworks which gives many design opportunities. The design of the collector and the thermal simulation were made in the same environment by adding flow simulation tab. We defined the computational domain to be a strip of the collector in order to reduce the computational time, so a full thermal analysis is succeeded. The convection between the absorber plate and the cover was defined by an innovative way which is explained in the following paragraphs. In the study, covered and uncovered, selective and non-selective collectors are presented in order to compare their efficiency. Moreover, sensitivity analysis for the transmittance – absorptance product (τα) is presented in order to determine the influence of optical losses on the collector's efficiency. Finally, the heat transfer convection coefficient between tube and water is calculated and compared to a theoretical model for isothermal tube.

2. Mathematical Bases

In this paragraph, the equations that describe the heat transfer inside the collector are presented. By studying the involved parameters, it is easier to understand how the performance of the collector is changing in every case. The equations (1) and (2) give three expressions for the efficiency of the collector and equation (4) shows the heat that water receives:

$$\eta = \frac{Q_u}{Q_{solar}} = \frac{Q_{absorbed} - Q_{losses}}{Q_{solar}} = \frac{S - U_L \cdot \left(T_p - T_a\right)}{G_T}, \qquad (1)$$

$$\eta = F_R(\tau\alpha) - F_R U_L \cdot \left(\frac{T_{f,i} - T_a}{G_T}\right) = F_R(\tau\alpha) - F_R U_L \cdot X \,, \tag{2}$$

$$X = \frac{T_{f,i} - T_a}{G_T}\,, \tag{3}$$

$$Q_u = \dot{m} \cdot c_p \cdot \left(T_{f,o} - T_{f,i}\right), \tag{4}$$

Now the losses of the collector are defined, because these have an important role in our methodology:

$$q_L = q_t + q_b + q_e = U_L \cdot \left(T_p - T_a\right), \tag{5}$$

$$U_L = U_t + U_b + U_e\,, \tag{6}$$

$$q_t = U_t \cdot \left(T_p - T_a\right) = h_{pc}^{tot} \cdot \left(T_p - T_c\right), \tag{7}$$

The heat transfer between absorber and cover is presented on next equations with more details. More specifically, the convection and radiation losses are determined [11, 26].

$$q_t = q_{conv} + q_{rad} = h_{pc} \cdot \left(T_p - T_c\right) + \frac{\sigma \cdot \left(T_P^4 - T_c^4\right)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} + \frac{1}{F} - 2}\,, \tag{8}$$

$$q_{conv} = h_{pc} \cdot \left(T_p - T_c\right) = h_{p-air} \cdot \left(T_p - T_{air}\right) = h_{air-c} \cdot \left(T_{air} - T_c\right), \tag{9}$$

$$h_{p-air} = h_{air-c} = 2 \cdot h_{pc} \quad \text{for } T_{air} = \frac{T_p + T_c}{2}\,, \tag{10}$$

From the above equations, equation (8) is of a great interest as far as view factor F, a critical parameter for our simulation, is concerned. Heat transfer between water and riser is presented in equations 11 & 13 in order to determine the convection coefficient. Equation (11) presents the direct way of its calculation by using the simulation results. On the other hand, equation (13) presents a model from heat transfer theory for the calculation of convection coefficient in the case of the uniform temperature in the inner tube surface.

$$h_w = \frac{Q_u}{\left(\pi \cdot \mathrm{D_i} \cdot L\right) \cdot \left(T_s - T_f\right)}\,, \tag{11}$$

$$T_f = \frac{T_{f,i} + T_{f,o}}{2}\,, \tag{12}$$

$$Nu_m = \frac{h_w \cdot D_i}{k} = 3.66 + \frac{0.0668 \cdot \mathrm{Re} \cdot \mathrm{Pr} \cdot D_i / L}{1 + 0.04 \cdot \left(\mathrm{Re} \cdot \mathrm{Pr} \cdot D_i / L\right)^{2/3}}\,, \tag{13}$$

3. The study case & Methodology

For designing and simulating a solar flat plate collector, Solidworks and especially the flow simulation environment are used. In order to reduce the computational time, a small part of a collector is studied (figure 1).

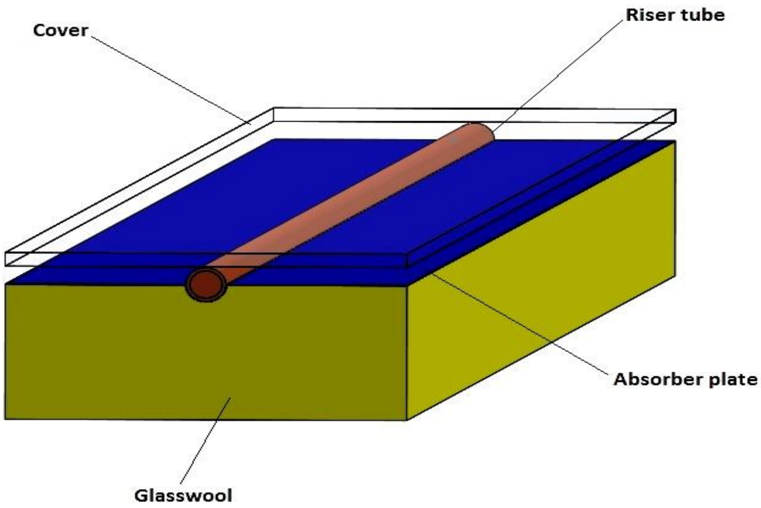

Fig.1. Computational domain in Solidworks

This small computational domain includes a riser tube and a strip consisted of absorber plate, back insulation layer and cover (glass). The performance of this strip is similar with the performance of the total collector. The main characteristics of this computational domain are given below:

Table 1. Main characteristics of computational domain

Characteristics	Values
Domain dimensions	1m x 0.1m x 50mm & Collective Area=0.1m^2
Tube	D_i=8mm & D_o=10mm
Absorber	Copper with 0.2mm thickness
Insulation	40mm Glasswool with thermal conductivity 0.04 W/m K
Cover	Single glass with 4mm thickness

The thermal simulation with Solidworks has many difficulties because the heat transfer between the surfaces is complicated. Thus, it is essential to refer the main assumptions that have been made in our analysis. First of all, the flow in the inlet is selected to be fully developed because this boundary condition simulates better the real situation. The around surfaces are assumed as adiabatic walls which means that the U_e coefficient is zero. This fact does not have a great effect on the results because the collector is insulated and the side area of the total collective area is much lower than the total collective area. Simultaneously, the separation between 2 consecutives tubes is observed in the middle of their distance, so the temperature in the left and in the right side of the strip is the maximum in width direction, which confirms the adiabatic condition. It is important to say that the optical losses ($\tau\alpha$) are defined in the absorptance of the absorber because the glass transmittance is constant and equals to 1. Finally, the distance between absorber plate and cover was reduced from 43.3mm to 5.3mm, so as to increase the radiation view factor between these surfaces, something crucial for the thermal analysis of the strip.

Now, it is important to explain the methodology followed in this analysis. The first obstacle was the way to simulate the air between the absorber and the cover. The default flow simulation of Solidworks let us use only one working fluid (water) in our computational domain. This restriction leads us to use water inside the tube and to simulate the air convection by determining boundary heat transfer condition with an innovative way. So, the convection between absorber and air is determined by defining the last one to have a constant temperature and convection coefficient to have a constant value. The same thing is done for the convection between air and the internal side of the cover. If we assume that these heat coefficients have the same value and the air temperature is the mean of the absorber and cover temperatures then the coefficient from absorber to air is double compared to the coefficient between absorber and cover.

The other problem is the radiation heat flux between absorber and cover. The equation (8) has a general form by including the view factor F for these surfaces. In the case that the whole collector is studied, this factor tends to 1 and there is not a problem, but in our case this has a lower value. In the initial design, the distance between absorber and glass is 43.3mm which mean that F is equal to 0.6, a very low value. So this distance reduced to 5.3mm and the view factor increased approximately to 0.95, closer to 1. The reduction in this distance has not any other effect in this specific analysis.

More specifically, the view factor is lower to 1 which means that the absorber radiates mainly to cover but also to ambient. This leads the absorber radiation losses to be greater, fact that influences on results. In order to improve this situation, we change the convection losses between the absorber and the air (between the two plates). By this way, we reduce the temperature difference between the air and the absorber so as the convection heat flux to be lower and the sum of convection and radiation losses to have the right value. Under normal condition, air temperature should be the mean value of cover and absorber temperature but in our case, this takes a higher value (about 0.1-2.0 ^oC) which means a lower convection heat flux from absorber plate to air. In order to define the air temperature value, we use the equation (1) as a converge criterion. In other words, different values of air temperature are tested in every case and the value that gives a right energy balance is selected. Simultaneously, by selecting this suitable air temperature the total heat flux that reaches to cover is equal to the top heat flux that leaves the absorber. The next table shows the main parameters of the simulation which are important for this analysis.

Table 2. Parameters of strip for thermal analysis

Parameter	**Value**	**Parameter**	**Value**
ε_p	0.10	U_e	0 W/m^{2}K
ε_c	0.88	U_b	0.875 W/m^{2}K
$(\tau\alpha)$	0.80	h_{pc}	4 W/m^{2}K
G_t	800 W/m^2	h_{ca}	8 W/m^{2}K
m	0.004 kg/sec	h_{back}	7 W/m^{2}K
t_a	10 ^oC	A_c	0.1 m^2

Moreover, it is essential to mention some extra definitions for the computational model. The water flow was selected to be laminar and turbulent in order to cover each case. The computational mesh consists of 150.000 cells with refinement in the fluid domain. An extra refinement has been made in partial cells (solid-fluid interface) in order to simulate properly the heat transfer by taking into consideration the boundary layer distribution.

In our study, the performance of this strip is analyzed in various cases. Table 3 shows the parameters which change during the analysis.

Table 3. Parameters that change in the analysis

Parameter	**Value**
$(\tau\alpha)$	0.7-0.8-0.9-1.0
ε_p	0.1 & 0.9 (selective & non selective)
Glass cover	YES -NO

In every case, table 2 shows the default values for the variables that do not change. For instance, when the $(\tau\alpha)$ parameter changes the emissivity of absorber (ε_p) is kept constant at 0.1 according to the table 2.

4. Results

In this paragraph the results of the simulation with Solidworks are presented. By comparing the efficiency in every case, it is possible to determine how each parameter influences on the system performance.

4.1 Influence of transmittance–absorptance product (τα) in efficiency

The (τα) parameter shows the proportion of solar energy which is absorbed in the absorber plate. The following diagram shows how this parameter effects in the collector efficiency.

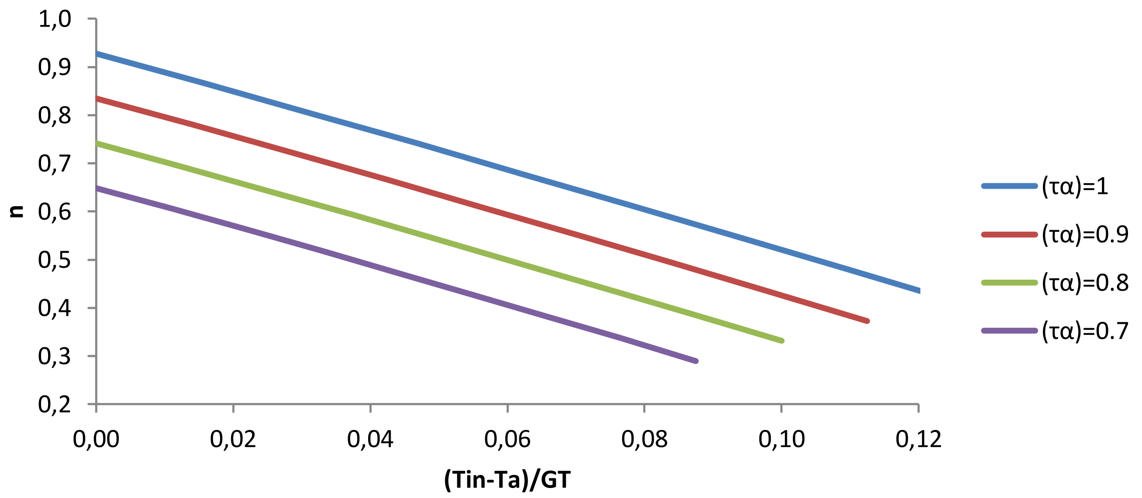

Fig. 2. Collector efficiency for various optical losses

From the figure 2 it is obvious that when the optical coefficient is lower, then the total efficiency is decreased. Moreover, another interesting result is that the change in this coefficient makes the efficiency curve to have a parallel displacement.

4.2 Efficiency for selective, non-selective, glazed and unglazed collector

The construction of the collector plays a very important role in the efficiency. The existence of cover and a selective absorber increase the efficiency according the next figure (3).

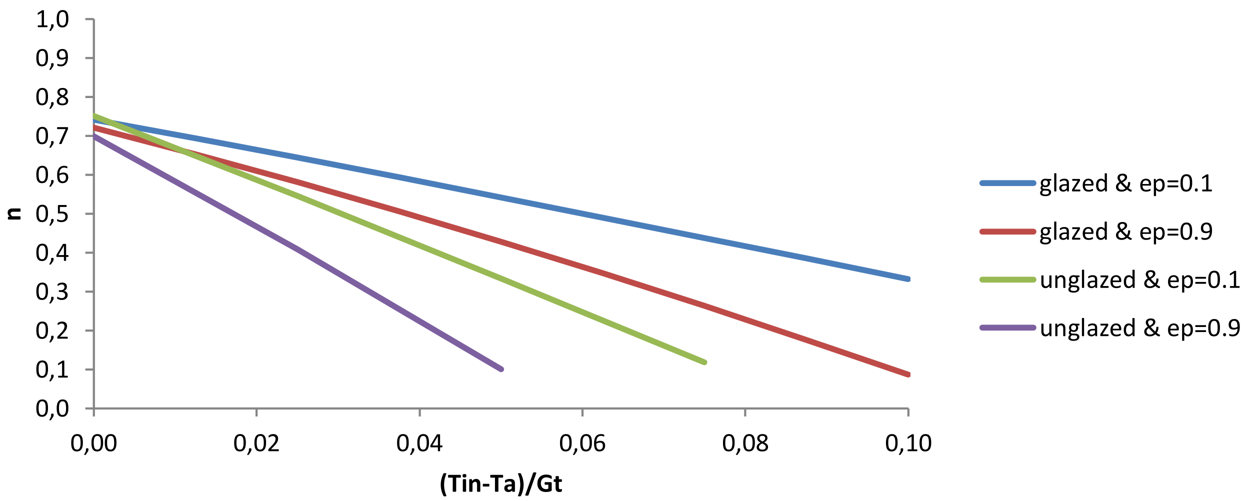

Fig. 3. Performance of different collectors

It is obvious that the cases with cover and with selective absorber are more efficient, especially when the inlet temperature is great. A higher inlet temperature leads to greater absorber temperature making the absorber to radiate more. Therefore, the non-selective collector efficiency is influenced a lot by the increase in the temperature and in high temperatures does not perform well.

The existence of glass reduces the top losses fact that leads to a more efficient system. It is important to say that the absorptance of absorber in the unglazed collectors was selected to be 0.9 when the parameter (τα) in glazed is about 0.8. Table 4 presents the main characteristics of these collectors which determine the efficiency in every case.

Table 4. Efficiency characteristics of collectors

Collector	$F_R(\tau\alpha)$	F_RU_L	n(%) for x=0.05
Selective glazed	0.7453	4.111	0.5415
Non-selective glazed	0.7285	6.104	0.4280
Selective unglazed	0.7536	8.456	0.3330
Non-selective unglazed	0.7013	11.96	0.1005

The greater difference is observed in F_RUL which is the main parameter for the thermal losses. The F_R (τα) parameter is not changed a lot between glazed and unglazed because the factor (τα) is lower in the glazed and F_R factor is greater. More specifically, the existence of cover creates reflectivity losses which reduce the factor (τα) whereas the factor F_R increases due to lower heat coefficient U_L. Also the last column of this table presents the efficiency when the inlet temperature is 50°C (a typical value) in order to understand the difference in efficiency more directly. So, we can say that the selective glazed collector gives about 5 time greater efficiency than the non-selective unglazed collector, which is a huge different.

4.3 Comparison selective and non-selective collector

In the previous analysis it was proved that the existence of a cover improves the collector efficiency and leads to a better performance. So, in this section the glazed collector is studied and an analytical comparison between collector with selective and non-selective absorber plate takes place. The next figure presents the useful energy which is gained from the analyzed strip, which is respectively to efficiency diagram.

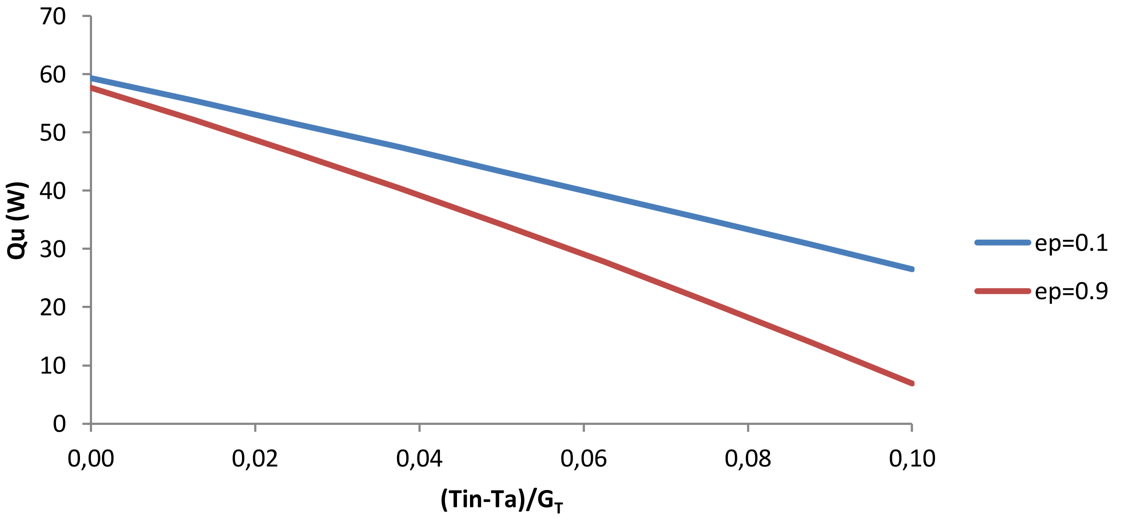

Fig. 4. Useful energy for selective (e_p=0.1) and non-selective (e_p=0.9) absorber plate

Figure 4 shows that the selective collector is more efficient in all cases and especially when the inlet temperature takes greater values the difference between these curves is greater, something that explained according to the next diagram.

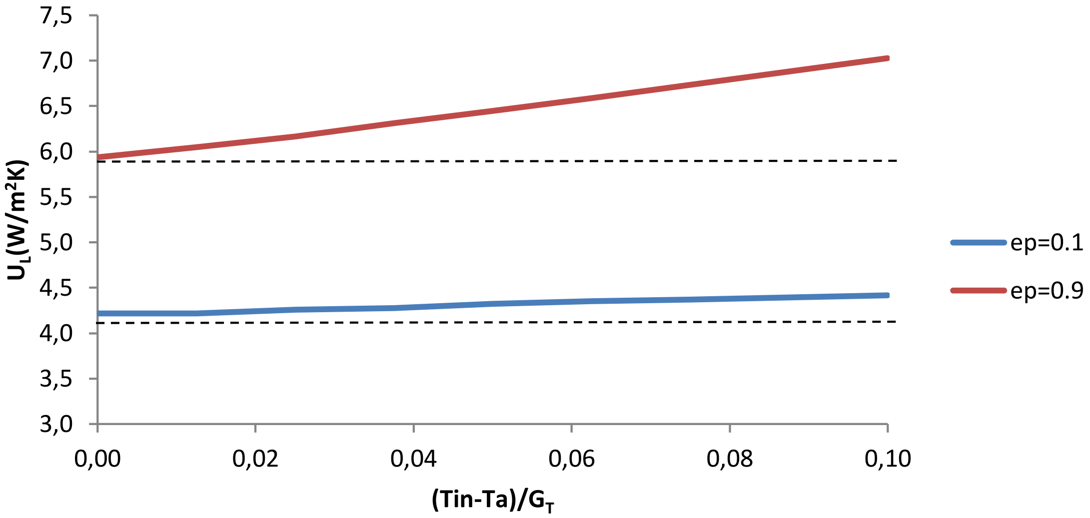

Fig. 5. Collector's losses coefficient for different operating conditions

The collector's losses coefficient is depended on the inlet temperature in the two cases, but in the case of non-selective absorber, the slope of the curve is greater. The different slope in these curves is the reason for the increasing difference in the efficiency (figure 4).

More specifically, a greater temperature in the inlet of the system makes the whole collector and especially the absorber warmer. This makes the absorber to radiate more and the radiation coefficient to increase. The increase in radiation coefficient is greater in the non-selective collector because of its emissivity value. As a conclusion in this explanation, it can be said that the losses coefficient in the selective collector is approximately constant, but in the non-selective collector there is a great dependence on the inlet temperature.

4.4 Main study case: the glazed selective collector

From the previous analysis it is obvious that the optimum solution is the covered selective collector. In this section, a more detailed analysis of this collector takes place in order to present more characteristics of it. Absorber temperature, cover temperature and water outlet temperature are the necessary Solidworks outputs in order to calculate the parameters of table 5.

Table 5. Parameters of covered selective collector

Parameter	**Value**
F_R	0.927
F'	0.938
F''	0.988
U_L	4.310 W/m^2K
v	0.081 m/s

The following diagram gives the efficiency of the collector by an optical way. The optical losses are constant in all cases, but the thermal losses are increasing with a greater water inlet temperature. This increase in thermal losses leads the collector efficiency to have the form in the figure 6.

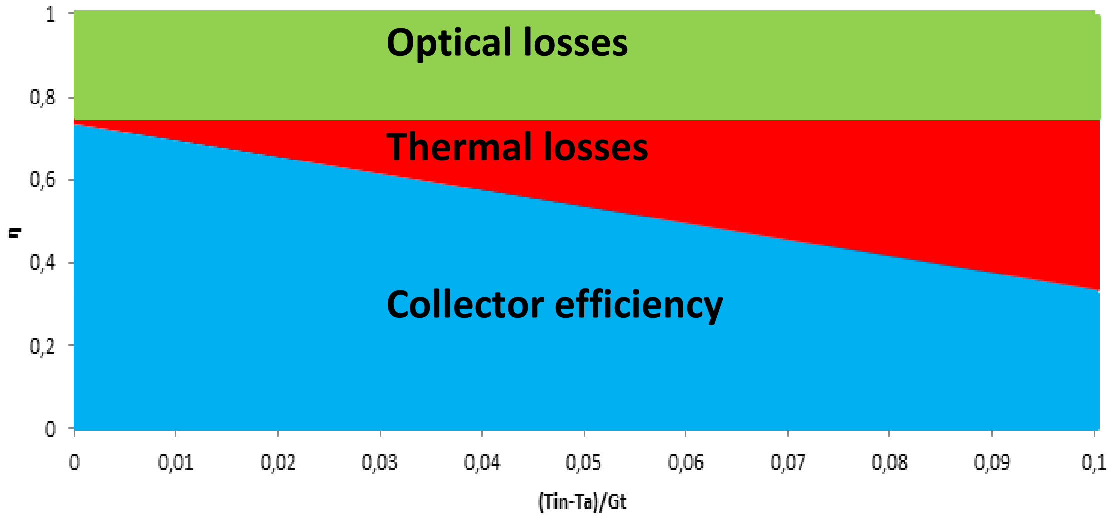

Fig. 6. Efficiency of selective collector and losses

As it is referred, the simulation tool of this analysis calculates the temperature of absorber of cover and of fluid. The next diagram compares these temperatures for different operation conditions.

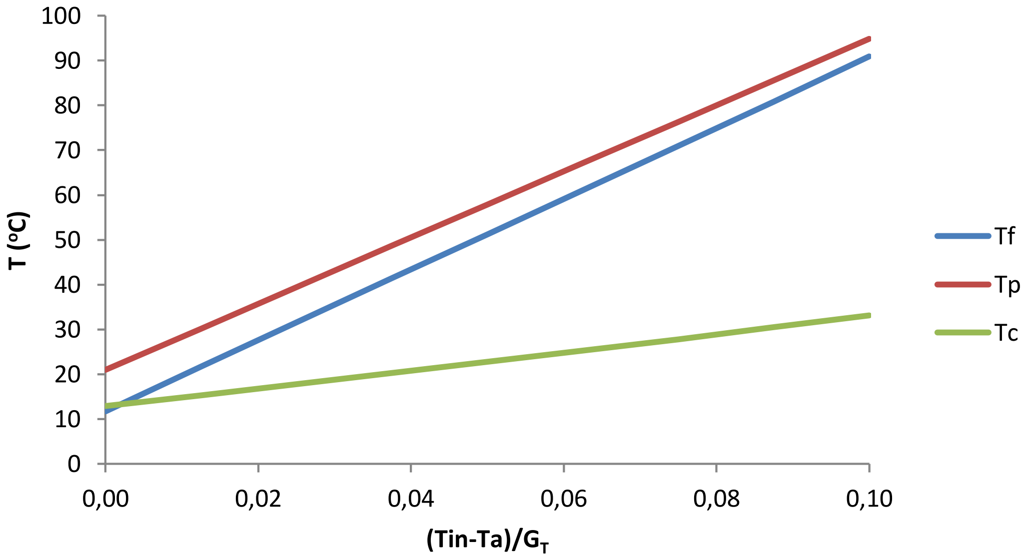

Fig. 7. Absorber temperature (red line), cover temperature (green line) and fluid mean temperature (blue line)

Figure 7 presents the main temperature of this study. It is obvious that all these curves tend to be linear by having an increasing rate, but the slope of every line is different. Cover temperature has the lowest values, because cover comes in touch with the environment. The absorber and the fluid have similar temperature profiles. More specifically, the difference between absorber and fluid temperature is lower while the inlet temperature increases fact that explained by the simultaneous reduce in efficiency and by the change in the heat convection coefficient between the tube and the water. This parameter is very important and figure 8 presents it.

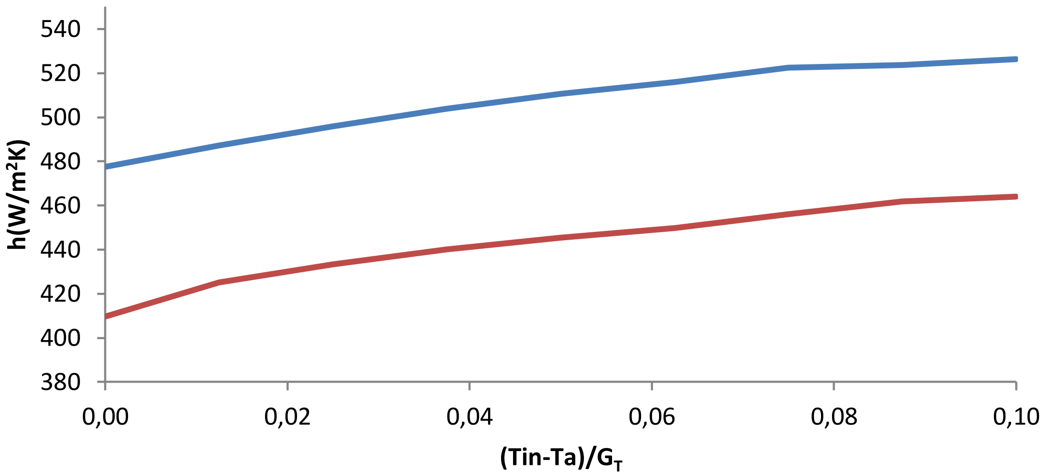

Fig. 8. Comparison of calculated (blue) and theoretical (red) convection coefficient for constant tube temperature

In figure 8 a comparison between the calculated coefficient (blue line) and a theoretical value (red line) according to the equation (2.12) is presented. This comparison leads us to conclude that the convection coefficient in a solar flat plate collector is 14.5% greater than the theoretical value, which is an important result. By knowing this value, the calculation of all the other parameters is easier.

In the end of the paragraph, results taken from Solidworks are presented in figure 9 in order to show the temperature in the absorber and in the back insulation. The results are referred in the case of covered selective collector for inlet water temperature at 50^{o}C.

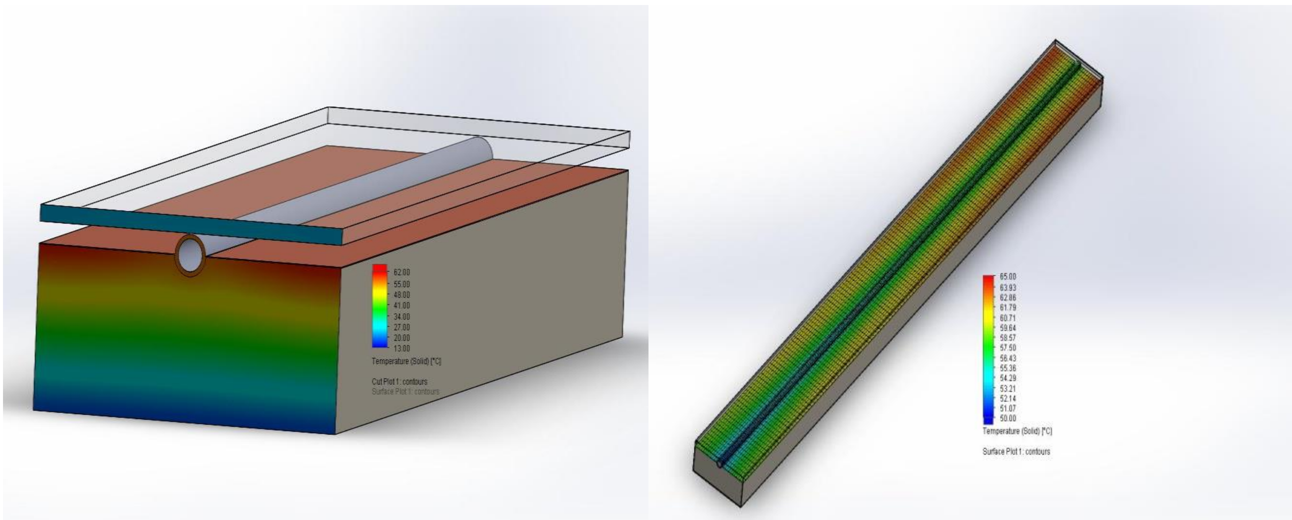

Fig. 9. Temperature distribution in the middle cross section (temperature distribution: 13-62^{o}C) and in the absorber surface (temperature distribution: 50-65 ^{o}C)

The left image shows the insulation temperature in the middle cross section. More specifically, the riser length is 1m and this situation is at 0.5m from the beginning. It is obvious that temperature near the absorber is higher than in the back of the collector and in the region near the tube, the temperature is lower because the water absorbs solar energy.

The other image shows the absorber temperature in the flow direction. The absorber temperature in the flow direction increases because the water carries heat from the start of the riser to the end. Another remarkable point is that the temperature in the sides is greater than in the center of the strip. The reason for this is that in the center there is the tube with water which absorbs useful energy.

5. Conclusions

In this study, a part of a solar flat plate collector is analyzed in order to predict its performance. The following points present the main conclusion of this study:

●The transmittance–absorptance (τα) product influences the efficiency by a direct way (figure 2).
●The covered collector performs better than uncovered (figure 3), especially for greater values of inlet temperature.

●The selective collector appears a higher efficiency rate due to a lower heat losses coefficient (figure 5). This parameter is influenced more by temperature fact that leads to a more curvy efficiency line. So, the case with a selective absorber has a linear efficiency curve, but the other with the non-selective is adjusted better to 2nd polynomial efficiency equation.

●According the figure 7, the absorber temperature is a linear function of parameter X while the cover has a respective behavior but with a lower slope. The important point in this diagram is that the difference between absorber temperature and mean fluid temperature is decreasing with X.

●The heat convection coefficient between tube and water is about 14.5% greater than the theoretical model of equation (12).

●The temperature in the absorber plate is increasing from the inlet to the outlet something which explained by the increase in the fluid temperature (figure 9). Also temperature in the side of the strip has the maximum value, because this side line is the center line between two consecutive water risers. This means that this line is far from the tubes, so it does not heat the water and keeps a great temperature.

Nomenclature

cp	Specific heat capacity, kJ/kgK
D	Diameter, m
F_R	heat removal factor
F′	collector efficiency factor
F′′	Flow efficiency factor
G_T	Solar titled radiation, W/m^2
h	Convection coefficient, W/m^{2}K
k	Thermal conductivity, W/mK
L	Tube length, m
m	Mass flow rate, kg/s
Nu	Nusselt number
Pr	Prandtl number
Q	Heat flux, W
q	Specific heat flux, W/m^2
Re	Reynolds number
S	Absorbed solar energy, W
T	Temperature,K
U	Losses coefficient, W/m^{2}K
v	Velocity, m/s
X	Collector efficiency parameter, m^{2}K/W
Greek symbols	
η	efficiency

($\tau\alpha$)	transmittance–absorptance product
Subscripts and superscripts	
a	Ambient
b	back
c	cover
e	edge
f	fluid
i	inlet
L	losses
o	Outlet
p	Absorber plate
s	Tube surface
t	Top
tot	Total
u	Useful

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