Exergy analysis for the optimization of flat-plate solar collectors characteristics

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Abstract:

In this paper exergy analysis is used to find the optimal characteristics of flat-plate collectors. These concern the diameter and spacing of riser tubes, the emissivity and absorptivity of the absorber plate and the transmittance of the cover. The analysis is done for a standard collector area of 2m^2 operating under standard flow rate of 0.015 kg/s-m^2 specified in the appropriate ISO standard and inclined at 45^o.

This paper employs exergy analysis to derive a general equation for the exergy efficiency of flat plate collectors. This efficiency depends on the values of collector heat loss coefficient, absorbed solar radiation and collector efficiency factor. The latter depends on the distance between the riser tubes, the diameter of the riser tubes, the collector fin efficiency, and the convection coefficient inside the tube - which depends on the mass flow rate and tube inside diameter, which affect the Reynolds number and that the Nusselt number. A model developed to simulate the collector considers all the modes of heat transfer and the optical analysis.

The results show that the exergy efficiency of a flat-plate solar collector is maximized for good quality collectors. It is shown that the transmittance-absorptance product has a serious effect on the drop of collector performance. The exergy efficiency is insensitive to the size of the riser pipe diameter and distance between riser tubes. Additionally, the performance of the collector is not so much dependent on the collector inlet temperature but depends a lot on the amount of radiation falling on the collector. The effect of collector area depends on the flow rate. When the flow rate is kept constant and the area increases, the collector performance is similar to the lower flow rate case whereas when this increase proportionally to the area, the performance is constant only more useful energy is delivered due to the bigger collector area.

Keywords:

Exergy analysis, flat-plate solar collector, optimization, heat loss coefficient, collector efficiency factor.

1. Introduction

Solar energy collectors are special kind of heat exchangers that transform solar radiation energy to internal energy of a transport fluid. Flat plate collectors are the most popular type of solar devises for low temperature applications. The main use of these collectors is in solar water heating systems operating at maximum temperatures of 80-90°C. A number of researchers have used exergy analysis to design flat plate collectors. Badescu [1] optimized the width and thickness of the fins of a flat-plate collector by minimizing the cost per unit useful heat flux. The proposed procedure allows computation of the necessary collection surface area. A rather involved, but still simple, flat plate solar collector model is used in the calculations. Model implementation requires a specific geographical location with a detailed meteorological database available. Fins of both uniform and variable thickness were considered. The optimum fin cross section is very close to an isosceles triangle. The fin width is shorter and the seasonal influence is weaker at lower operation temperatures. Fin width and thickness at the base depend on the season. The optimum distance between the tubes is increased by increasing the inlet fluid temperature, and it is larger in the cold season than in the warm season.

Badescu [2] also considered the best operation strategies for open loop flat-plate solar collector systems. A direct optimal control method (the TOMP algorithm) is implemented. A detailed collector model and realistic meteorological data from both cold and warm seasons are used. The maximum exergetic efficiency is low (usually less than 3%), in good agreement with experimental measurements reported in literature. The optimum mass-flow rate increases near sunrise and sunset and by increasing the fluid inlet temperature. The optimum mass-flow rate is well correlated with global solar irradiance during the warm season. Also, operation at a properly defined constant mass-flow rate may be close to the optimal operation.

Torres-Reyes et al. [3] presented a procedure to establish the optimal performance parameters for the minimum entropy generation during the collection of solar energy. The Entropy Generation Number, Ns, and the criterion for the optimal thermodynamic operation of a collector under non-isothermally, finite-time conditions, are reviewed. The Mass Flow Number, M, corresponding to the optimum flow of working fluid as a function of the solar collection area, is also considered. A general method for the preliminary solar collector design, based on Ns, M and the “Sun–Air” or stagnation temperature, is developed. This last concept is defined as the maximum temperature that the collector reaches at non-flow conditions for a given geographic location, geometry and construction materials. The thermodynamic optimization procedure was used to determine the optimal performance parameters of an experimental solar collector.

Torres-Reyes et al. [4] also presented the thermodynamic optimization of flat plate collectors based on the first and the second law, developed to determine the optimal performance parameters and to design a solar to thermal energy conversion system. An exergy analysis is presented to determine the optimum outlet temperature of the working fluid and the optimum path flow length of solar collectors with various configurations. The collectors used to heat the air flow during solar-to-thermal energy conversion are internally arranged in different ways with respect to the absorber plates and heat transfer elements. The exergy balance and the dimensionless exergy relationships are derived by taking into account the irreversibilities produced by the pressure drop in the flow of the working fluid through the collector. Design formulas for different air duct and absorber plate arrangements are obtained.

Farahat et al. [5] performed an exergetic optimization of flat plate solar collectors. A detailed energy and exergy analysis is carried out for evaluating the thermal and optical performance, exergy flows and losses as well as exergetic efficiency for a typical flat plate solar collector under given operating conditions. In this analysis, the authors considered as variables the absorber plate area, dimensions of solar collector, pipe diameters, mass flow rate, fluid inlet, outlet temperature, the overall loss coefficient. For this purpose a simulation program is developed and used for the thermal and exergetic calculations. The exergetic optimization has been carried out under given design and operating conditions and the optimum values of the mass flow rate, absorber plate area and maximum exergy efficiency were estimated.

The objective of the present work is to find the optimum collector characteristics maximize the exergy efficiency of flat-plate collectors. For this purpose an analytic exergy efficiency equation is derived which considers all parameters affecting the performance of the collector. Analytic equations are also used for the determination of the collector transmittance-absorptance product and the collector heat loss coefficient, which is estimated according to the collector plate temperature, wind convection loss, number of glass covers, collector inclination, ambient temperature and emittance of collector plate and glass cover.

2. Energy Analysis

In this section various relations that are required in order to determine the useful energy collected and the interaction of the various constructional parameters on the performance of a collector are presented.

The useful energy collected from a collector can be obtained from the following formula [6]:

$$Q_u = A_c F_R \left[G_t(\tau\alpha) - U_L \left(T_{f,i} - T_a \right) \right] = \dot{m} c_p \left(T_{f,o} - T_{f,i} \right) \quad (1)$$

where F_R is the heat removal factor given by [6]:

$$F_R = \frac{\dot{m} c_p}{A_c U_L} \left(1 - \mathrm{Exp}\left[\frac{U_L F' A_c}{\dot{m} c_p} \right] \right) \quad (2)$$

In (2) F′ is the collector efficiency factor which is calculated by considering the temperature distribution between two pipes of the collector absorber and by assuming that the temperature gradient in the flow direction is negligible [6]. This analysis can be performed by considering the sheet-tube configuration shown in Fig. 1, where the distance between the tubes is W, the tube diameter is D, and the sheet thickness is δ.

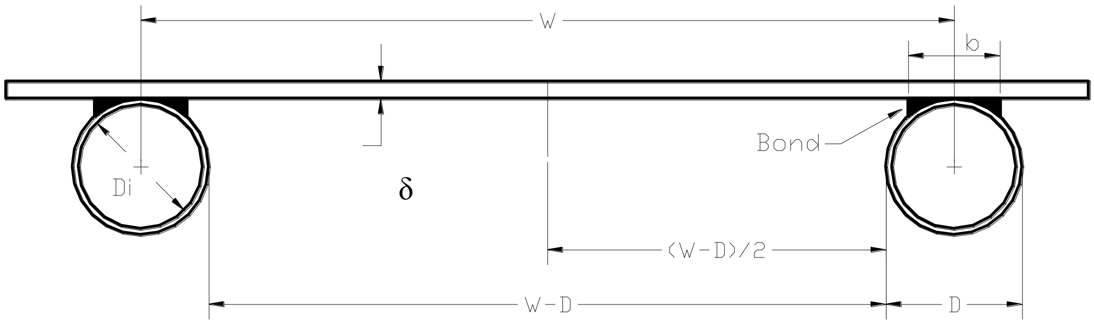

Fig. 1. Schematic diagram of a flat-plate sheet and tube configuration

As the sheet metal is usually made from copper or aluminum which are good conductors of heat, the temperature gradient through the sheet is negligible, therefore the region between the centerline separating the tubes and the tube base can be considered as a classical fin problem. By following this analysis the equation to estimate F′ is given by [6]:

$$F' = \frac{\frac{1}{U_L}}{W\left[\frac{1}{U_L\left[D + (W-D)F\right]} + \frac{1}{C_b} + \frac{1}{\pi D_i h_{fi}} \right]} \quad (3)$$

In (3), C_b is the bond conductance which can be estimated from knowledge of the bond thermal conductivity, the average bond thickness, and the bond width. The bond conductance can be very important in accurately describing the collector performance and generally it is necessary to have good metal-to-metal contact so that the bond conductance value to be high. The value used in the work is 195 W/m-K which applies to copper welding (brazing) of the riser pipes to the fin.

Factor F in (3) is the standard fin efficiency for straight fins with rectangular profile, obtained from:

$$F = \frac{\tanh[n(W-D)/2]}{n(W-D)/2} \qquad (4)$$

where n is given by:

$$n = \sqrt{\frac{U_L}{k\delta}} \qquad (5)$$

The collector efficiency factor is essentially a constant factor for any collector design and fluid flow rate. The ratio of U_L to C_b, the ratio of U_L to h_{fi}, and the fin efficiency F are the only variables appearing in (3) that may be functions of temperature. For most collector designs F is the most important of these variables in determining F′. The factor F′ is a function of U_L and h_{fi}, each of which has some temperature dependence, but it is not a strong function of temperature. Additionally, the collector efficiency factor decreases with increased tube center-to-center distances and increases with increases in both material thicknesses and thermal conductivity. Increasing the overall loss coefficient decreases F′ while increasing the fluid-tube heat transfer coefficient increases F′.

Therefore it is obvious from the above analysis that by increasing F′ more energy can be intercepted by the collector. By keeping all other factors constant increase of F′ can be obtained by decreasing W. However, decrease in W means increased number of tubes and therefore extra cost would be required for the construction of the collector.

The collector efficiency is found by dividing Q_u given in (1), by the incident radiation A_cG_t. By doing so the following Equation is obtained:

$$\eta = F_R(\tau\alpha) - F_R U_L \left[\frac{T_{f,i} - T_a}{G_t}\right] = \frac{\dot{m}c_p\left(T_{f,o} - T_{f,i}\right)}{A_c G_t} \qquad (6)$$

By plotting η against $\Delta T/G_t$ a straight line is obtained with the slope equal to $F_R U_L$, called the loss coefficient and the intercept on the y-axis equal to $F_R(\tau\alpha)$, called optical efficiency.

Generally the efficiency of a solar thermal system increases by increasing the flow rate. This increase is asymptotic, i.e., the increase is rapid at small flow rates and becomes almost asymptotic to a certain maximum value at higher values of flow rate. In a similar way the collector outlet temperature increases with the collector area and decreases with increasing flow rate and vice versa.

In a real system the variation of the fluid inlet temperature depends to a great extent on the storage tank configuration, thermal load demand and the consequent make-up water to the storage tank which affects the fluid inlet temperature to the collector.

3. Exergy Analysis

According to Kalogirou [6] the temperature at any position y at a fluid inlet temperature $T_{f,i}$ is given by:

$$T_f - T_a = \frac{S}{U_L} + \left(T_{f,i} - T_a - \frac{S}{U_L}\right)\left(\mathrm{Exp}\left[-\frac{U_L F' N W y}{\dot{m} c_p}\right]\right) \tag{7}$$

For y=L the collector area is given by A_c = NWL. If $T_{f,i} = T_a$ the fluid temperature variation at the exit from the solar collector is:

$$\Delta T = T_{f,o} - T_a = \left(\frac{S}{U_L}\right)\left(1 - \mathrm{Exp}\left[-\frac{U_L F'}{c_p}\frac{A_c}{\dot{m}}\right]\right) \tag{8}$$

The flow rate of exergy transferred from the sun to the fluid that is heated while crossing the riser pipe is:

$$\dot{E}_f = \dot{m} e_f = \dot{m}\left[\left(h_{f,o} - h_{f,i}\right) - T_a\left(s_{f,o} - s_{f,i}\right)\right] \tag{9}$$

Where $h_{f,o}$ - $h_{f,i}$ = $c_p(T_{f,o}$ - $T_{f,i})$ is the variation of specific enthalpy. The variation of the specific entropy is:

$$s_{f,o} - s_{f,i} = c_p \ln\left(\frac{T_{f,o}}{T_{f,i}}\right) \tag{10}$$

Therefore:

$$\dot{E}_f = \dot{m} c_p\left(\Delta T - T_a \ln\left(\frac{T_{f,o}}{T_{f,i}}\right)\right) \tag{11}$$

And the exergy efficiency is given by dividing $\dot{E}_f$ by the available solar radiation $Q_{s,in}$ which is equal to $A_c G_t$:

$$\eta_{ex} = \frac{\dot{E}_f}{\dot{Q}_{S,in}} = \dot{m} c_p\left(\frac{\Delta T - T_a \ln\left(\frac{T_{f,o}}{T_{f,i}}\right)}{A_c G_t}\right) \tag{12}$$

Replacing ΔT given by (8) in (12), the following relation for the exergy efficiency can be obtained:

$$\eta_{ex} = \frac{\dot{m} c_p}{A_c G_t}\left\{\frac{S}{U_L}\left(1 - \mathrm{Exp}\left[-\frac{U_L F' A_c}{\dot{m} c_p}\right]\right) - T_a \ln\left(1 + \frac{S}{T_a U_L}\left(1 - \mathrm{Exp}\left[-\frac{U_L F' A_c}{\dot{m} c_p}\right]\right)\right)\right\} \tag{13}$$

As can be seen from the exergy analysis, a general equation for the exergy efficiency of flat plate collectors is derived. This efficiency depends on the values of flow rate, collector area and collector efficiency factor. The latter depends on the distance between consequent riser tubes, the diameter of the riser tubes, the collector fin efficiency, and the convection coefficient inside the tube - which depends on the mass flow rate and tube inside diameter (depends on outside tube diameter), which affect the Reynolds number and thus the Nusselt number.

4. Method Description

The objective of this work is to find the parameters that maximize the exergy efficiency. In order to do this a numbers of parameters need to be considered as constants. These are shown in Table 1.

Table 1. Constant parameters used in exergy optimization

Parameter	Value
Fluid specific heat, c_p	4185 J/kg-K
Ambient temperature, T_a	25°C
Bond conductance, C_b	195 W/m^2-K
Absorbing plate thickness, δ	0.5 mm
Thermal conductivity of absorbing plate, k	385 W/m-K (copper)
Collector slope, β	45°

From these constant parameters a number of other parameters are evaluated. These are:

1. Heat loss coefficient, U_L, estimated from [6]. This is done using the analytic procedure based on heat transfer from absorbing plate to ambient. For single cover collectors the top heat loss from plate to cover is given by:

$$Q_{t,p-c} = A_c h_{c,p-g}(T_p - T_a) + \frac{A_c \sigma (T_p^4 - T_g^4)}{(1/\varepsilon_p) + (1/\varepsilon_g) - 1} \tag{14}$$

For tilt angles up to 60° the convective heat transfer coefficient, $h_{c,p-g}$ is given by Hollands et al. [7] for collector inclination (θ) in degrees as:

$$Nu = \frac{h_{c,p-g}L}{k} = 1 + 1.446\left(1 - \frac{1708}{Ra.\cos(\beta)}\right)^{+}\left(1 - \frac{1708[\sin(1.8\beta)]^{1.6}}{Ra.\cos(\beta)}\right) + \left[\left(\frac{Ra.\cos(\beta)}{5830}\right)^{0.333} - 1\right]^{+} \tag{15}$$

where + represents positive values only, and Ra is the Rayleigh number.

Similarly, the heat loss from glass cover to ambient is given by:

$$Q_{t,\,g-a} = A_c(h_{c,g-a} + h_{r,g-a})(T_g - T_a) = \frac{T_g - T_a}{R_{g-a}} \tag{16}$$

The radiation heat transfer coefficient is now given by:

$$h_{r,g-a} = \varepsilon_g \sigma (T_g + T_{sky})(T_g^2 + T_{sky}^2) \tag{17}$$

From (16): $$R_{g-a} = \frac{1}{A_c(h_{c,g-a} + h_{r,g-a})} \tag{18}$$

As resistances $R_{p,g}$ and R_{g-a} are in series their resultant is given by:

$$R_t = R_{p-g} + R_{g-a} = \frac{1}{U_t A_c} \qquad (19)$$

Therefore: $$Q_t = \frac{T_p - T_a}{R_t} = U_t A_c (T_p - T_a) \qquad (20)$$

The energy loss from the bottom of the collector is first conducted through the insulation and then by a combined convection and infrared radiation transfer to the surrounding ambient air. As the temperature of the bottom part of the casing is low, the radiation term ($h_{r,b-a}$) can be neglected, thus the energy loss is given by:

$$U_b = \frac{1}{\frac{t_b}{k_b} + \frac{1}{h_{c,b-a}}} \qquad (21)$$

Finally U_L is the summation of U_t, U_b and U_e.

A similar equation applies for the edges of the collector. In all cases the wind heat loss coefficient is given by:

$$h_w = \frac{8.6V^{0.6}}{L^{0.4}} \qquad (22)$$

2. Factor n estimated from (5)
3. Fin efficiency F, estimated from (4)
4. The convection heat transfer coefficient inside the pipe, $h_{f,i}$, estimated from the principles of heat transfer and according to the type of flow, turbulent or laminar according to the mass flow rate and the riser tube diameter.
5. The collector efficiency factor F′, estimated from (3).

For the above parameters the inputs required are the collector area, A_c, the distance between successive riser tubes, W, the riser tube diameter, D, and the collector mass flow rate. For the optical analysis of the collector and the determination of the transmittance-absorptance product ($\tau\alpha$) the analysis presented in [6] is followed and may not be repeated here.

5. Results

The collector characteristics for good and bad quality solar collectors are shown in Table 2. Initially a standard collector is considered which has a distance between the riser tubes (W) equal 10 cm and a riser pipe diameter of 15mm. This collector is modelled for a solar radiation of 1000 W/m^2 and collector inlet temperature of T_a+15°C. The results are shown in Table 3.

Table 2. Characteristics of good and bad quality collectors

Parameter	Good quality (TiNOx)	Bad quality (Black paint)
Absorbing plate emittance (ε_p)	0.05	0.98
Absorbing plate absorptance (α_p)	0.95	0.85
Glass extinction coefficient (K)	4 m^{-1}	32 m^{-1}

As can be seen from Table 3 there is a considerable difference in the performance of the two collectors. To check the effect the various parameters have on the performance of the collector the good collector is simulated again changing each time one of the parameters. The results are shown in Table 4.

Table 3. Results for the analysis for good and bad quality collectors

Parameter	Good quality (TiNOx)	Bad quality (Black paint)
Useful energy (Q_u)	1531.2 J	1082.5 J
Thermal efficiency (η)	76.56%	54.14%
Exergetic efficiency (η_{ex})	1.759%	1.058%
Transmittance-absorptance produce $(\tau\alpha)_n$	0.866	0.692
Heat loss coefficient (U_L)	4.063 W/m^2-°C	6.908 W/m^2-°C
Heat Removal factor (F_R)	0.9513	0.9196
Collector efficiency factor (F')	0.9818	0.9694

Table 4. Results for the analysis for the good quality collector changing each time one parameter

Parameter	Good quality	K=32 m^{-1}	α_p=0.85	ε_p=0.98
Useful energy (Q_u)	1531.2 J	1356.8 J	1358.6 J	1399.6 J
Thermal efficiency (η)	76.56%	67.84%	67.93%	69.98%
Exergetic efficiency (η_{ex})	1.759%	1.409% (-19.9%)	1.413% (-19.7%)	1.643% (-6.6%)
Transmittance-absorptance produce $(\tau\alpha)_n$	0.866	0.774	0.775	0.866
Heat loss coefficient (U_L-W/m^2-°C)	4.063	4.048	4.047	6.956
Heat Removal factor (F_R)	0.9513	0.9515	0.9515	0.9192
Collector efficiency factor (F')	0.9818	0.9819	0.9819	0.9693

It should be noted that the percentage difference in exergetic efficiency is with respect to the good quality collector. As can be seen the effect of increasing the glass extinction coefficient from 4 m^{-1} to 32 m^{-1} is very similar to the decrease of the absorbing plate absorptance from 0.95 to 0.85 and have a serious effect on the drop of collector performance, whereas the effect of the increase of the absorbing plate emittance from 0.05 to 0.98 is less serious.

Subsequently the effect of changing the receiver diameter (D) and distance between riser tubes (W) is investigated. The results are presented in Table 5 with respect to the good quality (GQ) one.

Table 5. Results of the effect of changing the receiver diameter (D) and distance (W)

Parameter	GQ	D=12mm	D=10mm	W=0.07m	W=0.03m
Useful energy (Q_u)	1531.2 J	1531.1 J	1530.9 J	1545.3 J	1556.8 J
Exergetic efficiency (η_{ex})	1.759%	1.758%	1.758%	1.789%	1.814%
Heat Removal factor (F_R)	0.9513	0.9512	0.9597	0.9515	0.9664
Collector efficiency factor (F')	0.9818	0.9817	0.9817	0.9606	0.9975

As can be seen from Table 5 both parameters have very little effect on the performance of the collector, although a smaller distance between the riser tubes increases the exergetic efficiency but not to a level that will be economically viable, as smaller distance would mean more riser tubes. As for the smaller diameter pipe the performance is almost the same but the cost is lower, so a smaller diameter riser pipe is preferred.

The effect of different inlet temperature and solar radiation are shown in Table 6.

As can be seen from Table 6 the performance of the collector is not so much dependent on the collector inlet temperature but depends a lot on the amount of radiation falling on the collector. Concerning the collector inlet temperature however, the efficiency drops as this increases due to higher thermal losses.

Table 6. Results of estimations with different collector inlet temperature and solar radiation

Parameter	Radiation 1000 W/m^2			Radiation 500 W/m^2		
	Collector inlet temperature			Collector inlet temperature		
	T_a	T_a+5°C	T_a+25°C	T_a	T_a+5°C	T_a+25°C
Useful energy (Q_u)	1651.8 J	1612.6 J	1445.8 J	827.7 J	790.2 J	626.9 J
Thermal efficiency (η)	82.59%	80.63%	72.29%	82.77%	79.02%	62.69%
Exergetic efficiency (η_{ex})	1.768%	1.764%	1.754%	0.901%	0.897%	0.891%

The flow rate through the collector also affects its performance. The flow rate considered in all the above results was 0.015 kg/s-m^2 or for the Area of 2m^2 considered was 0.03 kg/s. The effect of smaller flow rate is shown in Table 7.

Table 7. Effect of flow rate on the performance of the collector

Parameter	Standard (0.03 kg/s)	0.02 kg/s	0.01 kg/s
Useful energy (Q_u)	1531.2 J	1502.2 J	1373.0 J
Thermal efficiency (η)	76.56%	75.11%	68.69%
Exergetic efficiency (η_{ex})	1.759%	2.510%	4.086%
Heat loss coefficient (U_L-W/m^2°C)	4.063	4.110	4.227
Heat Removal factor (F_R)	0.9513	0.9340	0.8570
Collector efficiency factor (F′)	0.9818	0.9797	0.9420
Collector outlet temperature (°C)	52.18	57.92	72.77

As can be seen the smaller the flow rate the higher the collector outlet temperature and the exergetic efficiency as thermodynamically the higher temperature fluid can perform more useful work. The useful energy collected however is lowered as the heat losses are bigger. The heat removal factor is also affected by the lower flow rate.

Finally the effect of collector area is investigated. This is done in two respects; in the first the flow rate is kept constant at 0.03 kg/s and in the second this is increasing according to the area using the rate of 0.015 kg/s-m^2. The results are shown in Tables 8 and 9 respectively.

Table 8. Results of estimations with different collector area at constant flow rate of 0.03 kg/s

Parameter	A_c=2m^2	A_c=4m^2	A_c=6m^2
Useful energy (Q_u)	1531.2 J	2940.8 J	4220.9 J
Thermal efficiency (η)	76.56%	73.52%	73.56%
Exergetic efficiency (η_{ex})	1.759%	3.190%	4.323%
Heat loss coefficient (U_L-W/m^2-°C)	4.063	4.315	4.570
Heat Removal factor (F_R)	0.9513	0.9178	0.8824
Collector efficiency factor (F′)	0.9818	0.9810	0.9801
Collector outlet temperature (°C)	52.18	63.38	70.35

Table 9. Results of estimations with different collector area at a flow rate of 0.015 kg/s-m^2

Parameter	A_c=2m^2	A_c=4m^2	A_c=6m^2	A_c=8m^2	A_c=10m^2
Useful energy (Q_u)	1531.2 J	3058.6 J	4578.5 J	6092.5 J	7601.1 J
Thermal efficiency (η)	76.56%	76.46%	76.31%	76.16%	76.01%
Exergetic efficiency (η_{ex})	1.759%	1.762%	1.761%	1.760%	1.758%
Heat loss coefficient (U_L-W/m^2-°C)	4.063	4.182	4.279	4.362	4.436
Heat Removal factor (F_R)	0.9513	0.9522	0.9520	0.9516	0.9511
Collector efficiency factor (F′)	0.9818	0.9837	0.9843	0.9845	0.9845
Collector outlet temperature (°C)	52.18	52.16	52.14	52.11	52.09

As can be seen from Tables 8 and 9, as expected, as the area increases more useful energy is collected. When the flow rate is kept constant (Table 8) this leads to higher collector outlet temperature and higher exergetic efficiency as thermodynamically the higher temperature fluid can perform more useful work. The normal thermal efficiency drops due to the lower heat removal factor obtained at the lower flow rate compared to the collector area. When the flow rate increases in proportion to the collector area (Table 9), all performance parameters remain approximately constant irrespective of the collector area.

6. Conclusions

It is shown in this paper how the exergy analysis can be used to select the optimum configuration of a flat-plate collector. The results show that the exergy efficiency of a flat-plate solar collector is maximized for good quality collectors. By checking which parameter affects the collector performance more it was shown that the effect of increasing the glass extinction coefficient from $4m^{-1}$ to $32m^{-1}$ is very similar to the decrease of the absorbing plate absorptance from 0.95 to 0.85 and both have a serious effect on the drop of collector performance, whereas the effect of the increase of the absorbing plate emittance from 0.05 to 0.98 is less serious. As is shown the exergy efficiency is insensitive to the size of the riser pipe diameter and distance between riser tubes, although a smaller distance between the riser tubes increases the exergetic efficiency but not to a level that will be economically viable, as smaller distance would mean more riser tubes. As for the smaller diameter pipe the performance is almost the same but the cost is lower, so a smaller diameter riser pipe is preferred. Additionally, the performance of the collector is not so much dependent on the collector inlet temperature but depends a lot on the amount of radiation falling on the collector. Concerning the collector inlet temperature however, the efficiency drops as this increases due to higher thermal losses. Finally the effects of flow rate and collector area are investigated. It is shown that the smaller the flow rate the higher the collector outlet temperature and the exergetic efficiency as thermodynamically the higher temperature fluid can perform more useful work. The useful energy collected however is lowered as the heat losses are bigger. The heat removal factor is also affected by the lower flow rate. The effect of collector area depends on the flow rate. When the flow rate is kept constant and the area increases, the collector performance is similar to the lower flow rate case whereas when this increase proportionally to the area, the performance is constant only more useful energy is delivered due to the bigger collector area. The values of exergetic efficiency obtained are in agreement with the values cited in the relevant literature.

Nomenclature

A_c	collector area, m^2
B	bond width , m
C_b	bond conductance, W/m-K
c_p	specific heat capacity, J/kg-K
D	riser tube outside diameter, m
D_i	riser tube inside diameter, m
$\dot{E}_f$	exergy flow rate, W
e	specific exergy, J/kg
F'	collector efficiency factor
F	fin efficiency
F_R	heat removal factor
G_t	solar radiation, W/m^2

h specific enthalpy, J/kg; convection heat transfer coefficient, W/m^2-K

h_{fi} heat transfer coefficient inside absorber tube, W/m^2-K

h_w wind loss coefficient, W/m^2-K

I solar radiation, W/m^2

k absorber plate thermal conductivity, W/m-K

L collector length, m

$\dot{m}$ mass flow rate, kg/s

n factor given by (5)

N number of riser tubes

$Q_{s,in}$ available solar radiation, W/m^2

Q_u rate of useful energy collected, W

Ra Rayleigh number

s specific entropy, J/kg-K

S power absorbed per unit area of collector, W/m^2

T_a ambient temperature, K

T_f fluid temperature, K

$T_{f,i}$ collector inlet temperature, K

$T_{f,o}$ collector outlet temperature, K

T_p plate temperature, K

U_L overall heat loss coefficient, W/m^2-K

V wind speed, m/s

W distance between riser tubes, m

Greek symbols

β collector slope, degrees

δ absorber (fin) thickness, m

ε_p plate emittance

ε_g glass cover emittance

ΔT temperature difference, K

$\tau\alpha$ transmittance-absorptance product

Subscripts

a ambient

b back

c cover, convection

e edges

f fluid

g glass

p plate

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