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Exergetic Optimization of Solar Air Heaters and Comparison with Energy Analysis

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Abstract

In this paper, an exergetic optimization of the solar air heater is developed. For this means, an integrated mathematical model of thermal and optical performance of the solar heater is derived. The most geometric parameters and operation conditions are considered as variables in this analysis. Some correlations for exergy efficiency of heater components are obtained. Then, exergy efficiency of the heater is derived by using these correlations. In the process of deriving an equation for the exergy efficiency, while the overall thermal loss coefficient and other heat transfer coefficients of the heater are assumed to be variable, the common error of using the Petela efficiency is corrected to reach the improved equation of solar radiation exergy. Finally, through the MATLAB toolbox the exergy efficiency equation is maximized. Then exergy efficiency is compared with the thermal efficiency of the heater, resulting in an extraordinary increase of the exergy efficiency according to the optimized parameters and benefit of this approach for such systems.

Keywords: Exergy, solar air heater, optimization

1. Introduction

Flat plate solar air heaters are non-adiabatic radiative heat exchangers; they are essentially used at low temperature levels $(T < 375K)$ in air heating and drying systems. *Figure 1* shows a schematic view of a solar air heater.

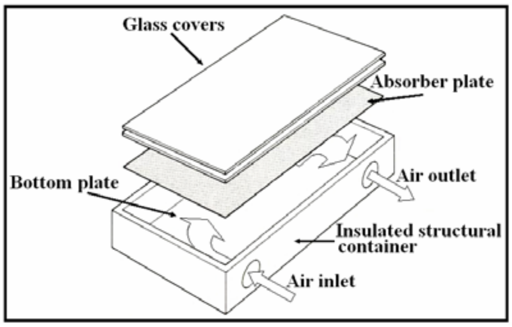

Figure 1. Schematic view of a solar air heater.

In a solar space heating system, the heater is the main part of the system. Hence, the optimal and thermal performance analysis of the heater becomes highly important. It is common knowledge that the solar-thermal energy efficiency increases without extreme points with the flow rate. The absence of maximum points for the function $\eta_{th} = f(\dot{m})$ has created difficulties in the design of solar air heaters. The exergy efficiency of a solar air heater presents points of local maxima and a point of global. In addition, the energy equation alone does not encounter the internal losses; it cannot be a sufficient criterion for the solar air heater efficiency. Therefore, the consideration of this article will be on the design and determination of the operation conditions of this type of collector, rather than absorbed energy maximized, in short, minimizing the existing irreversibility. Altfeld et al. (1988) found the exergy efficiency, optimal fluid inlet temperature, inlet fluid velocity and the optimal flow duct geometry of a solar air heater. However, they assumed that the overall loss coefficient is constant. Dutta Gupta and Saha (1990) made a thermal and exergy analysis, with the assumption of a constant overall loss coefficient and temperature changes of inlet fluid, computing the optimal inlet temperature for several cases. Geng Liu et al. (1995) reached

*Author to whom correspondence should be addressed a general equation, in integral form, for exergy production and disposal. They also presented energy and exergy efficiency for elements of a system containing a flat plate collector. Reyes et al. (2004), by dimensionless correlations for exergy components, found exergy efficiency, optimal fluid inlet temperature, inlet fluid velocity and optimal flow duct length of a solar air heater with respect to definite conditions. They computed the overall loss coefficient of the heater from the empirical equation. However, this equation has specific restrictions.

None of the studies mentioned, concluded a model for the collector optical and thermal performance, other than assuming that the overall loss coefficient or any other heat transfer coefficients are constant, or at minimal effect. None of them has completely performed optimization with respect to the design and performable conditions.

Design conditions include dimensions of heater and performable conditions along with the fluid inlet temperature and the passing mass flow rate. A common error using the Petela efficiency equation obtaining solar radiation exergy can be noted. There do not seem to be any attempts towards the maximization of exergy efficiency in relation to all optimization parameters and performable conditions simultaneously.

Malek Mohammadi (2000) concentrated on the optimization of parabolic solar collectors with the use of a general mathematical model of thermal and optical performance similar to other cases corrupted with the common error using the Petela efficiency equation. Najian (2000), assuming constant thermal flux in absorber tube, obtained some equations for the destroyed exergies, also analyzing the flat plate solar collector. Unfortunately, this not only assumed a constant overall loss coefficient and other heat transfer coefficients but also the maximization of exergy efficiency with respect to all optimization parameters. Performable conditions were not reached simultaneously. Farahat et al. (2004a) introduced the method and basis of flat plate collector optimization, without concluding; also in another work, they corrected previously mentioned errors and presented exergy analysis of the linear parabolic solar collectors with the use of a general mathematical model of thermal and optical performance (Farahat et al., 2004b).

2. Heater Exergy Efficiency

It is impossible to analyze exergy of a heater without modeling through its optical and thermal performance. These two subjects are much related, hence there will be a primary attempt at the heater optical and thermal analysis and modeling; thereafter deriving correlations for different terms of the exergy balance equation of the heater. Then the exergy efficiency equation of the heater will be computed with respect to specific parameters.

Finally, through the MATLAB toolbox the exergy efficiency equation of the heater will be maximized.

3. Thermal Analysis

A brief note: proof for governing relations on the heater thermal performance has not been concluded, Sukhatme (1993). By writing the energy balance equation according to inlet and outlet temperatures, Q_u useful heat gain can be obtained:

$$Q_u = \dot{m}C_p\left(T_{out} - T_{in}\right) \quad (1)$$

In the above equation T_{in}, T_{out} are fluid inlet and outlet temperatures from the heater, respectively, and C_p, $\dot{m}$ are heat capacity and the mass flow rate of fluid, respectively.

$$\dot{m} = \rho V_{av} A \quad (2)$$

In this equation, V_{av} and A are average air velocity and cross sectional area of duct at the inlet to the heater, respectively. Furthermore, density, ρ and other properties of fluid are evaluated at the arithmetic mean of the fluid inlet and outlet temperatures. On the other hand, an energy balance by considering heat lost from heater to ambient yields (Sukhatme, 1993),

$$Q_u = A_p F_R\left[S - U_l\left(T_{in} - T_a\right)\right] \quad (3)$$

where T_a is the ambient temperature and F_R is the heat removal factor, defined by:

$$F_R = \dot{m}C_p\left[1 - \exp\left\{- F' A_p U_l / \dot{m}C_p\right\}\right] / A_p U_l \quad (4)$$

That F' is the collector efficiency factor defined by:

$$F' = \left(1 + U_l / h_e\right)^{-1} \quad (5)$$

where h_e is heat transfer coefficient between the absorber plate and air stream given by:

$$h_e = \left[h_{fp} + h_r h_{fb} / \left(h_r + h_{fb}\right)\right] \quad (6)$$

That h_r is equivalent radiative heat transfer coefficient defined in this case by:

$$h_r = \frac{4\sigma T_{av}^3}{\left(1/\varepsilon_p + 1/\varepsilon_b - 1\right)} \quad (7)$$

where

$$T_{av} = \left(T_p + T_b\right)/2 \quad (8)$$

In this equation, T_p and T_b are average absorber plate temperature and average bottom plate temperature, respectively. An energy balance on the absorber plate yields the following equation for steady state:

$$Q_u = A_p S - U_l A_p\left(T_p - T_a\right) \quad (9)$$

In the current equations, h_{fp}, h_{fb}, ε_p, ε_b, σ and A_p are convective heat transfer coefficient between the absorber plate and the air stream, the convective heat transfer coefficient between the bottom plate and the air stream, the emissivity of the absorber plate surface, the emissivity of the bottom plate surface, Stefan-Boltzmann constant and the area of the absorber plate, respectively. h_{fp} and h_{fb} are calculated from empirical equations and the values of these parameters are taken to be equal. U_l is the overall loss coefficient. During previous studies, it was assumed as a constant factor or a variable with little effect, while it is not constant. This parameter is the sum of the loss coefficient from the top, the bottom and the sides:

$$U_l = U_t + U_b + U_e \tag{10}$$

$$U_b = k_i/\delta_b \tag{11}$$

that k_i and δ_b are thermal conductivity of the insulation and the thickness of the back insulation, respectively.

$$U_e = (L_1 + L_2)L_3 k_i / L_1 L_2 \delta_e \tag{12}$$

$$L_3 = \delta_1 + \delta_2 + \delta + \delta_b \tag{13}$$

where L_1, L_2, L_3 are the length, width and height of the absorber plate and δ_1, δ_2, δ are the distance between the absorber plate and the first glass cover, the first glass cover and the second glass cover, the absorber plate and the bottom plate, and δ_e is the thickness of the sides' insulation, respectively. The calculation of the top loss coefficient, U_t, based on convection and re-radiation losses from the absorber plate to ambient.

For purposes of calculation, it is assumed that the transparent covers and the absorber plate constitute a system of infinite parallel surfaces and that the flow of heat is one-dimensional and steady. Further, it is assumed that the temperature drop across the thickness of the covers is negligible, and that the interaction between the incoming solar radiation absorbed by the covers and the outgoing loss may be neglected. For the outgoing re-radiation that is of long wavelengths, the transparent covers can be assumed to be opaque. Making the above assumptions, there is:

$$\frac{Q_t}{A_p} = h_{p-c1}\left(T_p - T_{c1}\right) + \frac{\sigma\left(T_p^4 - T_{c1}^4\right)}{\left(\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1\right)} \tag{14}$$

$$\frac{Q_t}{A_p} = h_{c1-c2}\left(T_{c1} - T_{c2}\right) + \frac{\sigma\left(T_{c1}^4 - T_{c2}^4\right)}{\left(\frac{1}{\varepsilon_c} + \frac{1}{\varepsilon_c} - 1\right)} \tag{15}$$

$$\frac{Q_t}{A_p} = h_a\left(T_{c2} - T_a\right) + \sigma\varepsilon_c\left(T_{c2}^4 - T_{sky}^4\right) \tag{16}$$

$$U_t = \left(Q_t/A_p\right)/\left(T_p - T_a\right) \tag{17}$$

$$T_{sky} = T_a - 6 \tag{18}$$

where h_{p-c1}, h_{c1-c2}, h_a are the convective heat transfer coefficients between the absorber plate and the first glass cover, between the first glass cover and the second glass cover, between the second glass cover and the surrounding air. Also T_{c1}, T_{c2}, T_{sky} are the temperatures attained by the first cover, the second cover, effective temperature of the sky, and Q_t/A_p is heat loss rate per unit area of the absorber plate, and ε_c is the emissivity of the covers for long wavelength radiation where the radiative exchange takes place.

These equations are a set of three nonlinear equations which have to be solved for the unknowns Q_t/A_p, T_{c1} and T_{c2} after substituting the values of h_{p-c1}, h_{c1-c2} and h_a. The current heat transfer coefficients are calculated from empirical equations, (Sukhatme, 1993).

Finally U_l is a function of parameters such as the glass covers' temperatures, the air properties in and out of the glass covers, the environment temperature, the absorber plate temperature, the sky temperature, the wind speed and the reflecting surface properties. Also, the area of the absorber plate is the product of the length of the absorber plate and the width of the absorber plate, and the cross-sectional area of the duct is the product of the width of the absorber plate and the spacing between the absorber plate and the bottom plate.

$$A_p = L_1.L_2 \tag{19}$$

$$A = L_2.\delta \tag{20}$$

Characteristic dimension for calculating Reynolds number is the equivalent diameter given by:

$$D_e = \frac{4 \times \text{Cross} - \text{sectional area of duct}}{\text{Wetted perimeter}} \tag{21}$$

The thermal efficiency of the heater is given by:

$$\eta_{th} = Q_u / I_T A_p \tag{22}$$

4. Optical Analysis

The heater does not absorb all the sunlight shining on the absorber plate.

In equation (3) the parameter S, the radiation absorbed flux through the absorber plate is as follows (Sukhatme, 1993),

$$S = (\tau\alpha) I_T \quad (23)$$

$$I_T = I_b R_b + I_d R_d + (I_b + I_d) R_r \quad (24)$$

The first term on the right of the last equation is the amount of beam radiation that is absorbed by the absorber plate, and the second term illustrates the radiation absorbed by the absorber plate.

I_T is the incident solar energy per absorber plate area unit, $(\tau\alpha)$ is effective product transmittance-absorptance that is equal to the optical efficiency η_o. Other parameters such as I_b, I_d, R_b, R_d and R_r are hourly beam radiation, hourly diffuse radiation, tilt factor for beam radiation, tilt factor for diffuse radiation and tilt factor for reflected radiation, respectively.

5. Exergy Analysis

Deriving the exergy efficiency or the second law efficiency, the quality of the energy will be obtained. In general, through two methods the exergy exchange with the heater occurs. First through the agent fluid flow and second through heat transfer, (Bejan, 1988; Kenneth,1995).

The exergy accompanying a compressible fluid at temperature T and the pressure difference, ΔP, is given by:

$$\dot{E} = \dot{m} C_p \left(T - T_a - T_a \ln \frac{T}{T_a} \right) + \dot{m} R T_a \ln \left(\frac{P}{P_a} \right) \quad (25)$$

where P_a and R are ambient pressure and ideal gas constant, respectively. The exergy flux by heat transfer rate, $\dot{Q}$ in the range of hot T_h and cold T_c temperatures of air, is given by:

$$\dot{E} = \int_{T_c}^{T_h} \dot{Q} \frac{T_a}{T^2} dT \quad (26)$$

The general form of the exergy balance equation is:

$$\dot{E}_{in} + \dot{E}_s + \dot{E}_{out} + \dot{E}_l + \dot{E}_d = 0 \quad (27)$$

where $\dot{E}_{in}$, $\dot{E}_s$, $\dot{E}_{out}$, $\dot{E}_l$ and $\dot{E}_d$ are the inlet, stored, outlet, leakage and destroyed exergy rates, respectively.

The inlet exergy rate includes two parts. Inlet exergy rate with fluid flow:

$$\dot{m} C_p \left(T_{in} - T_a - T_a \ln \frac{T_{in}}{T_a} \right) + \dot{m} R T_a \ln \left(\frac{P_{in}}{P_a} \right) \quad (28)$$

The absorbed solar radiation exergy rate by the heater: generally, in previous studies, considering the Petela theorem (Bejan, 1988), the exergy rate was calculated with the following equation, which for such systems violates the second law of thermodynamics.

$$\eta_o I_T A_p \left[1 - \frac{4}{3} \frac{T_a}{T_s} + \frac{1}{3} \left(\frac{T_a}{T_s} \right)^4 \right] \quad (29)$$

The term in the bracket is, η_P, the Petela efficiency. Assuming the sun as an infinite thermal source, the equation can be corrected (Dutta Gupta, 1990):

$$\eta_o I_T A_p \left(1 - \frac{T_a}{T_s} \right) \quad (30)$$

The summation of equation (28) and equation (30) will result in the total inlet exergy rate of the heater.

Stored exergy rate is null at steady conditions.

Outlet exergy rate contains only the outlet fluid flow exergy rate:

$$-\dot{m} C_p \left(T_{out} - T_a - T_a \ln \frac{T_{out}}{T_a} \right) + \dot{m} R T_a \ln \left(\frac{P_{out}}{P_a} \right) \quad (31)$$

In equation (28) and equation (31), P_{in} and P_{out} are the agent fluid pressure at entrance and exit from the heater.

Leakage exergy rate is caused by the heat leakage rate from the absorber plate to environment (Altfeld et al., 1988):

$$-U_l A_p (T_p - T_a)(1 - T_a / T_p) \quad (32)$$

Destroyed exergy rate: It consists of three terms. The first one is caused by the absorber plate surface and sun temperature difference (Dutta Gupta, 1990):

$$-\eta_o I_T A_p T_a (1/T_p - 1/T_s) \quad (33)$$

Keep in mind $(T_s = 4350\,\text{K})$ the effective sun temperature, being 3/4 temperature of black body of sun (Bejan et al., 1981). The second one is caused by the duct pressure drop (Altfeld et al., 1988):

$$-\dot{m} \Delta P T_a / \rho T_m \quad (34)$$

where T_m is given by:

$$T_m = (T_{in} + T_{out})/2 \quad (35)$$

And the third one is caused by heat transfer rate from absorber plate to the agent fluid across finite temperature difference:

$$-\eta_{th} I_T A_p T_a (1/T_m - 1/T_p) \quad (36)$$

Implementing correlations, equation (28) through equation (36) in the exergy balance equation, equation (27), and simplifying, yields:

$$\dot{m}C_p\left(T_{out}-T_{in}-T_a\ln(\frac{T_{out}}{T_{in}})\right)-$$
$$\dot{m}R\ln(\frac{P_{out}}{P_{in}})=-\frac{\dot{m}\Delta P}{\rho}\frac{T_a}{T_m}+$$
$$\eta_o I_T A_p(1-\frac{T_a}{T_s})-U_l A_p(T_p-T_a)(1-\frac{T_a}{T_p})- \tag{37}$$
$$\eta_o I_T A_p T_a\left(\frac{1}{T_p}-\frac{1}{T_s}\right)-\eta_{th} I_T A_p T_a\left(\frac{1}{T_m}-\frac{1}{T_p}\right)$$

Looking at the definition of the heater exergy efficiency, increase of fluid flow exergy upon the primary radiation exergy by the radiation source, and equation (37), a new equation is derived for the exergy efficiency (Dutta Gupta , 1990):

$$\eta_E=\frac{\dot{m}C_p\left(T_{out}-T_{in}-T_a\ln\left(T_{out}/T_{in}\right)\right)}{I_T A_p\left(1-T_a/T_s\right)}-\frac{\dot{m}RT_a\ln\left(P_{out}/P_{in}\right)}{I_T A_p\left(1-T_a/T_s\right)} \tag{38}$$

or:

$$\eta_E=1-\left\{\left(1-\eta_o\right)+\frac{\eta_o T_a\left(1/T_p-1/T_s\right)}{\left(1-T_a/T_s\right)}+\frac{U_l(T_p-T_a)\left(1-T_a/T_p\right)}{I_T\left(1-T_a/T_s\right)}+\frac{\dot{m}\Delta P T_a}{\rho T_m I_T A_p\left(1-T_a/T_s\right)}+\frac{\eta_{th} T_a\left(1/T_m-1/T_p\right)}{\left(1-T_a/T_s\right)}\right\} \tag{39}$$

The terms in the right bracket of equation (39) illustrate exergy losses. Pressure drop in the duct obtained from

$$\Delta P=2f\rho L_1 V_{av}^2/D_e \tag{40}$$

where f is the friction factor and can be calculated from the well-known Blasius equation which is valid for smooth surfaces (Sukhatme, 1993):

$$f=0.079\,Re^{-0.25} \tag{41}$$

6. Seeking Optimized Value For Exergy Efficiency Effective Parameters

Using a numerical method, optimizing the exergy efficiency can be started. The objective function is equation (39), and equations (1) to (24) can be constraints for this special case. The basic variables regarding optimization conditions or the heater's designing conditions can be such as the following parameters:

T_{in} , T_{out} , T_p , V_{av} , $\dot{m}$, F_R , F' , Q_u , U_l , S , A_p , A , L_1 , L_2 , L_3 , δ , D_e and etc.

With respect to the amount of used variables and amount of noted equations, the system degree of freedom will be obtained. Since constituted equations are nonlinear, the problem can be solved numerically. Therefore, the optimized value of some effective parameters that maximize exergy efficiency are obtained. For this path, the MATLAB optimizing functions toolbox was used.

Environmental and designing conditions are as follows:

wind speed $V_a=2.5$ m/s , heater tilt $\beta=20^\circ$ and other parameters defined in this paper are:
$I_T=800\ \mathrm{W/m^2}$, $T_a=300\,\mathrm{K}$, $T_s=4350\,\mathrm{K}$,
$\eta_o=0.85$, $A_p=2\ \mathrm{m^2}$, $L_1=2$ m , $L_2=1$ m ,
$\delta=0.015$ m , $\delta_1=\delta_2=0.025$ m , $\varepsilon_c=0.88$,
$\varepsilon_p=\varepsilon_b=0.95$, $k_i=0.05\ \mathrm{W/m.K}$,
$\delta_b=\delta_s=0.05$ m

Fluid inside the duct is air. Also pressure inside the glass covers is assumed to be equal to atmospheric pressure. According to these assumptions, T_{in} and V_{av} are found to maximize exergy efficiency:

$T_{in}=355.509\,\mathrm{K}$, $V_{av}=5\,\mathrm{m/s}$

where maximum exergy efficiency is equal to $\eta_E=7.4171\%$

After solving the constraint equations, other parameters were obtained as follows:

$T_{out}=364.4060\,\mathrm{K}$, $T_p=372.8992$ K
$U_l=4.7654\ \mathrm{W/m^2.K}$, $F'=0.8426$
$h_{fp}=h_{fb}=19.2806\ \mathrm{W/m^2.K}$, $F_R=0.7983$
$\eta_{th}=41.28\%$, $Q_u=660.4411\,\mathrm{W/m^2}$
$\dot{m}=0.0736\,\mathrm{kg/s}$

Figure 2 demonstrates the exergy efficiency contour according to average air velocity and fluid inlet temperature, in two dimensions.

As can be seen in this figure, exergy efficiency has its maximum value in a particular point. The coordinate of this point is the same value of optimized parameters. This allows having an optimized design regarding other

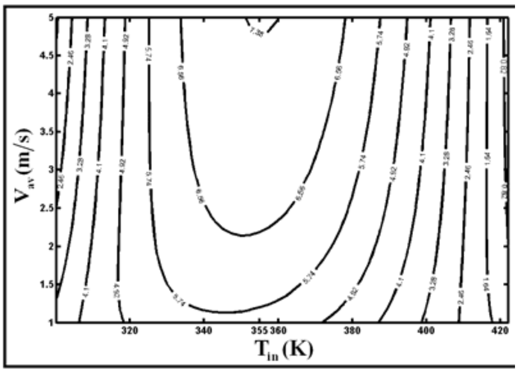

Figure 2. Exergy efficiency contour according to average air velocity and fluid inlet temperature.

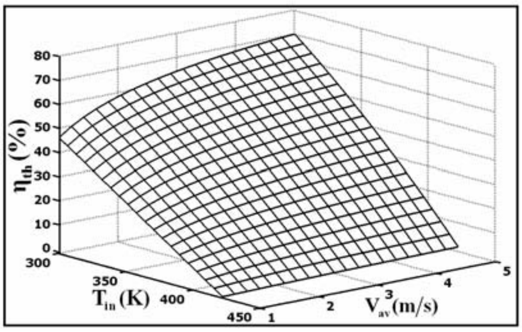

Figure 3. Variation of thermal efficiency according to average air velocity and fluid inlet temperature.

conditions such as design limitations and thermal applications. Although, out of this point, we have sensible exergy efficiency loss and this shows the danger limits more clearly.

Figure 3 demonstrates the variation of thermal efficiency according to average air velocity and fluid inlet temperature, in three dimensions for environmental and design conditions that were mentioned previously.

In this figure, by decreasing the fluid inlet temperature and increasing the average air velocity simultaneously, the thermal efficiency increases to its maximum value.

Figure 4 shows the thermal and exergy efficiency changes regarding the area of the total heater surface.

Exergy efficiency is nearly independent of the heater area. However, by increasing the area of the heater surface, there is little decrease in thermal efficiency.

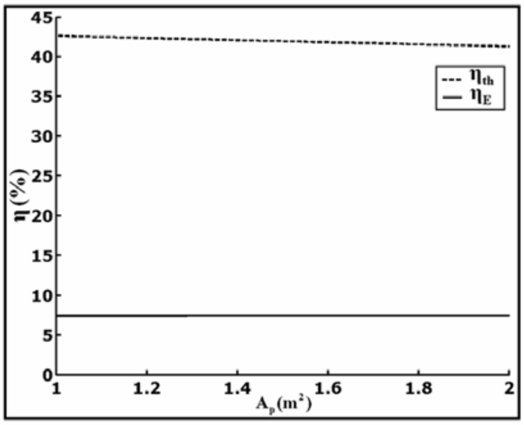

Figure 4. Variation of thermal and exergy efficiency versus area of heater surface.

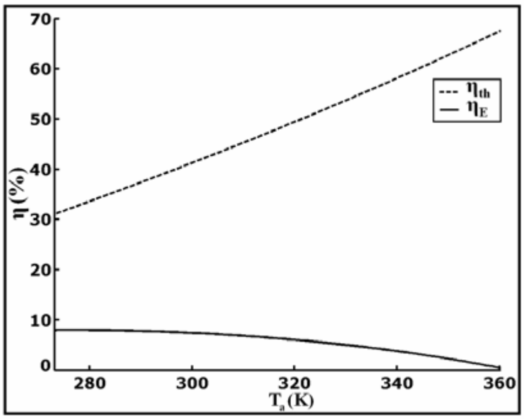

Figure 5. Variation of thermal and exergy efficiency versus ambient temperature.

Figure 5 shows the ambient temperature effect on thermal and exergy efficiency.

A decrease in exergy efficiency can be seen by increasing the ambient temperature. Due to the temperature changes during the day to have maximum exergy efficiency, other parameters and heater performance conditions should change during the day. On the other hand, by increasing the ambient temperature, the thermal efficiency will increase.

In *Figure 6*, the wind speed influence on thermal and exergy efficiency can be seen.

Increasing the wind speed, there is little decrease in exergy efficiency but great decrease in thermal efficiency.

The effect of optical efficiency on thermal and exergy efficiency can be seen in *Figure 7*.

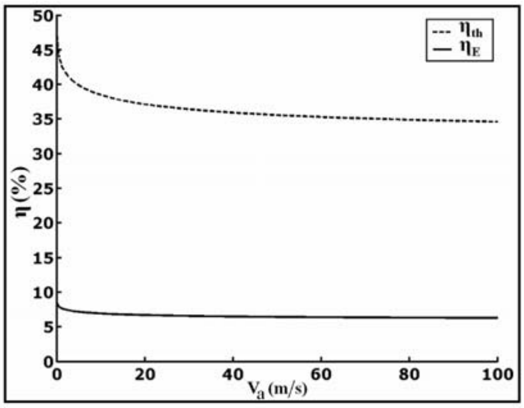

Figure 6. Variation of thermal and exergy efficiency versus wind speed.

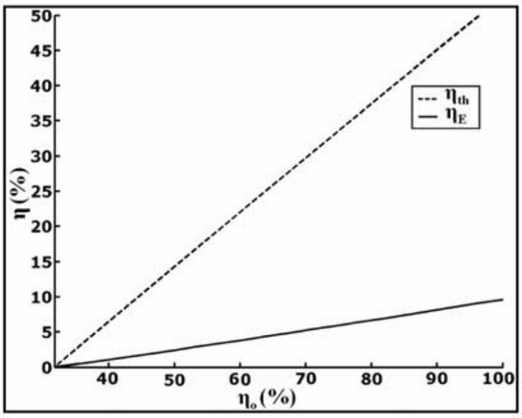

Figure 7. Variation of thermal and exergy efficiency versus optical efficiency.

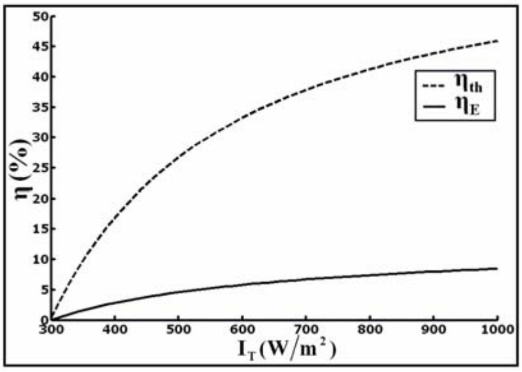

Figure 8. Variation of thermal and exergy efficiency versus incident solar energy per absorber area unit.

By increasing optical efficiency, thermal and exergy efficiency increases.

Figure 8 shows thermal and exergy efficiency changes with respect to incident solar energy per absorber area unit.

By increasing this parameter, thermal and exergy efficiency have increased.

7. Conclusion

In this paper, an integrated mathematical model of thermal and optical performance of a heater was derived. This simulation was followed with the achievement of the heater exergy efficiency. In the process of deriving the heater exergy efficiency, while the overall thermal loss coefficient and other heat transfer coefficients of the heater were assumed to be variables, the common error of using the Petela efficiency was corrected. Finally, through the MATLAB toolbox the exergy efficiency equation was maximized. Figures show that the behavior of exergy and thermal efficiency are always not the same. Results show that exergy analysis is a better method for optimizing and designing solar air heaters because exergy efficiency is a proportion to common quantities in solar engineering such as thermal efficiency, temperature, pressure drop, mass flow rate of fluid and others. On the other hand, the exergy analysis developed for the selected model allows the establishment of the optimal values for the characteristic quantities of the solar air heater. Unlike other optimization methods, this method decreases internal irreversibility, which is very important.

Nomenclature

A	area [m^2]
C_p	heat capacity of the fluid [kJ/kg K]
D	diameter [m]
$\dot{E}$	exergy rate [J/s]
f	friction factor
F'	collector efficiency factor
F_R	heat removal factor
h	heat transfer coefficient [$W/m^2.K$]
I	solar radiation [W/m^2]
k	conductivity [W/m.K]
L	dimensions of heater [m]
$\dot{m}$	mass flow rate [kg/s]
P	fluid pressure [Pa]
Pr	Prandtl number
Q	heat transfer rate [W]
R	ideal gas constant [kJ/kg K] and tilt factor
Re	Reynolds number
S	radiation absorbed flux [W/m^2]
T	temperature [K]
U	heater loss coefficient [W/m^2 K]
V	velocity [m/s]

Greek Symbols

β	heater tilt [degree]
Δ	difference in pressure or temperature
δ	distance or thickness [m]
ε	emissivity
η	efficiency
ρ	density [kg/m^3]
σ	Stefan-Boltzmann constant
$(\tau\alpha)$	effective product transmittance-absorptance

Subscripts

a	ambient
av	average
b	back, beam, bottom
c	cold, cover
d	destroyed, diffuse
e	equivalent, side
E	exergy
f	air stream
h	hot
i	insulation
in	inlet
l	leakage, overall
m	average

o optical
out outlet
p absorber plate
r radiative, reflected
s stored, sun
sky sky
t top
T incident
th thermal
u useful

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