Int. J. Environ. Res., 8(1):39-48,Winter 2014
ISSN: 1735-6865

Exergy Analysis of a Flat Plate Solar Collector in Combination with Heat Pipe

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Received 17 May 2013; Revised 10 July 2013; Accepted 17 July 2013

ABSTRACT: The use of solar collectors in combination with heat pipes is rapidly growing in recent years. Heat pipes, as heat transfer components, have undeniable advantages in comparison with other alternatives. The most important advantage is their high rate of heat transfer at minor temperature differences. Although there have been numerous studies on the heat analysis or first thermodynamic analysis of flat plate solar collectors in combination with heat pipes, the exergy analysis of these collectors is needed to be investigated. In this work, energy and exergy analysis of a flat plate solar collector with a heat pipe is conducted theoretically. Next, the exergy efficiency of pulsating heat pipe flat plate solar collectors (PHPFPSC) is compared with conventional collectors by using the experimental data. The results indicate that the use of heat pipes for heat transfer from the absorption plate to the water reservoir has significantly higher availability and exergy efficiency than the case with conventional collectors with intermediate fluid.

Key words: Solar Collector, Heat Pipe, Pulsating Heat Pipe (PHP), Exergy Analysis, Exergy Efficiency

INTRODUCTION

New sources of energy have been widely considered all around the world during recent years (Amirabedin and McIlveen-Wright, 2013; Buzzigoli and Viviani, 2013; Ataei *et al.,* 2012; Zeinolabedin *et al.,* 2011; Cui *et al.,* 2011; Alipour *et al.,* 2011; Xing *et al.,* 2010; Ismail and Abdul Rahman, 2010; Ataei and Yoo, 2010; Ataei *et al.,* 2009; Karbassi *et al.,* 2008; Mehrdadi *et al.,* 2007; Roshan *et al.,* 2012). There have been studies explaining thermodynamic analysis of solar collectors, using the concepts of exergy and the exergy efficiency (Bejan *et al.,* 1981, Gupta and Kaushik, 2010, Luminosu and Fara, 2005, Saidur *et al.,* 2012, Suzuki, 1988). Many authors (Xiaowu and Ben, 2005, Gunerhan and Hepbasli, 2007, Gupta and Kaushik, 2010) investigated the exergy efficiency of a solar water heater. Their results showed that the collectors have the least exergy efficiency or the most irreversibility. Farahat et al. (Farahat *et al.,* 2009) performed thermal and exergetic computations. Their exergetic optimization was performed under a special operating condition, and the optimum values of various parameters were found. Heat pipes take advantage from high heat transfer due to the latent heat transfer and small temperature discrepancies between the cold and hot region. These properties make them a good choice for increasing the efficiency of solar energy equipments (Abreu and Colle, 2004, Abu-Zour *et al.,* 2006, Azad, 2008, Bong *et al.,* 1993, Hussein, 2007, Mathioulakis and Belessiotis, 2002, Riffat *et al.,* 2005). Pulsating heat pipes (Charoensawan *et al.,* 2003, Charoensawan and Terdtoon, 2008, Rittidech *et al.,* 2007, Song and Xu, 2009, Yang *et al.,* 2008) have a significant potential for increasing the efficiency of solar energy equipments, such as solar water heaters, due to their simplicity, nontwisting structure, requiring equipments with conventional technology, independent from gravity and high efficiency (Arab, 2008, Rittidech *et al.,* 2009, Rittidech and Wannapakne, 2007).

Kargar Sharif Abad et al. (Kargar Sharif Abad *et al.,* 2013) used pulsating heat pipes in a solar water desalinator, which led to a new desalination system of a high efficiency. In another experimental investigation by them (Kargarsharifabad *et al.,* 2013), the performance of pulsating heat pipes in large scales was examined for various evaporator lengths, filling ratios, and inclination angles, the efficient values of which was obtained in a specific climate condition in Yazd, Iran.

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MATERIALS & METHODS

The heat pipe flat plate solar collector (HPFPSC) is assumed to be a system with constant properties and in steady state condition. The conservation of energy and exergy equilibrium are assumed to be governed for the system. Since HPFPSC is a closed system, the conservation of mass is not used for analysis. In this study, a heat pipe is assumed to be a closed system for energy and exergy equilibrium analysis, and other complicated phenomena such as two-phase heat conduction, temperature and pressure gradient are not considered. In fact, the heat pipe is a hypothetical solid body with a low conduction resistance through which heat can easily transfer (Q_{HP}). Considering the Considering the whole input and output energy for the control volume shown in Fig. 1, the following equation for conservation of energy is given(Hepbasli 2008):

$$\sum \dot{E}_{in} = \sum \dot{E}_{out} \quad (1)$$

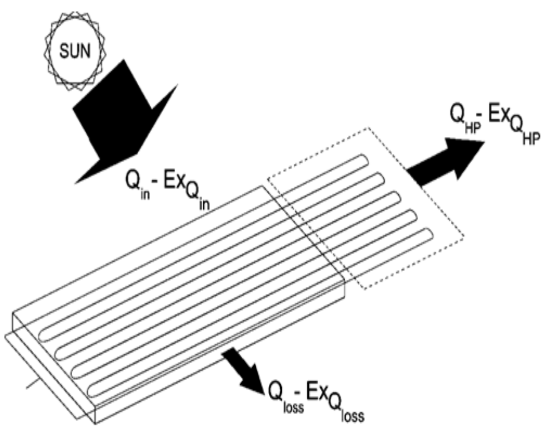

Fig. 1. Schematic of energy and exergy flows

Where the rate of energy is represented by $\dot{E}$ and the input and output energy are denoted by subscripts *in* and *out*, respectively. The input and output parameters are in the form as shown in Eq.(2).

$$\sum \dot{E}_{in} = \dot{Q}_{in} = A_C I_T \quad (2)$$

Where I_T is overall influx radiation received by solar collectors, which includes the direct and diffuse terms of radiation. The collectors receive the solar rays. Due to changes in refractive index of light, partial absorption happens, and a fraction of received rays reflects. The amount of the absorption depends on the absorption coefficient of absorber. Hence, overall influx of the sun can be derived by Eq.(3).

$$\underbrace{A_C I_T}_{\text{irradiance heat}} = \underbrace{\eta_o A_C I_T}_{\text{available heat}} + \underbrace{(1-\eta_o) A_C I_T}_{\text{heat loss}} \quad (3)$$

Where η_o is optical efficiency of the collector, and, for the flat plate collector is equal to the multiplication of τ and α ($\eta_o = \tau\alpha$) which are the collector's effective transmittance and absorption respectively. The left side of Eq. (3) represents the received solar radiation, and the terms of the right side demonstrate the available heat and heat loss respectively. The output parameters are shown in Eq.(4)

$$\sum \dot{E}_{out} = \dot{Q}_{HP} + \dot{Q}_{loss} \quad (4)$$

The heat transferred through a heat pipe is defined as $\dot{Q}_{HP}$ and the heat loss is represented by $\dot{Q}_{loss}$. According to the complexity of thermo-hydrodynamic phenomena occurred in pulsating heat pipes, there have been no explicit analytical equations for calculating the heat transfer in the heat pipes and all the analysis of this field have been performed experimentally (Katpradit *et al.,* 2005, Rittidech *et al.,* 2007, Rittidech *et al.,* 2003, Sakulchangsatjatai *et al.,* 2004, Shafii *et al.,* 2001).

The amount of heat that the heat pipe transfers is called heat pipe efficiency. This efficiency depends on the type of working fluid, inner diameter, filling ratio, the angle from the horizon, operator length, and etc. The following relation defines the dependency of the transferred heat to various parameters:

$$\dot{Q}_{HP} = f\ \{D_i, L_e, h_{fg}, K_l, Cp, \rho_l, \rho_V, ...\} \quad (5)$$

There have been experimental studies on obtaining relations between non-dimensional dependent parameters and non-dimensional transferred heat based on the thermo-physical properties of a problem (Katpradit *et al.,* 2005, Rittidech *et al.,* 2007, Rittidech *et al.,* 2003, Shafii *et al.,* 2001, Shafii *et al.,* 2010).

The term $\dot{Q}_{loss}$ includes the radiation heat loss of the collector and heat loss due to the overall heat transfer coefficient (U_L) between the collector and environment. The defined heat loss is derived as shown in Eq.(6):

$$\dot{Q}_{loss} = (1-\eta_o) A_C I_T + A_C U_L (T_p - T_a) \quad (6)$$

Using Eq. (6), the energy equilibrium equation for HPFPSC is obtained in the following form:

$$A_C I_T = \dot{Q}_{HP} + (1-\eta_o) A_C I_T + A_C U_L (T_p - T_a) \quad (7)$$

For simplicity, Eq. (7) is rewritten in the form of Eq. (8):

$$\eta_o A_C I_T = \dot{Q}_{HP} + A_C U_L (T_p - T_a) \quad (8)$$

According to Eq. (8), a part of absorbed energy of radiation is lost, and the remaining is transferred by the use of heat pipe.

There is no conservation equation for exergy. The exergy studies take advantage from equilibrium equations which have been presented in different forms (Alta *et al.,* 2010, Farahat *et al.,* 2009, Gunerhan and Hepbasli, 2007, Gupta and Kaushik, 2010, Koca *et al.,* 2008). Using the exergy equation used in (Gunerhan and Hepbasli, 2007), the following equilibrium equation for exergy is derived.

$$\sum \dot{E}x_{in} - \sum \dot{E}x_{out} = \sum \dot{E}x_{des} \quad (9)$$

Where the inflow and outflow of the exergy of system are denoted by subscripts *in* and *out* respectively. The subscript *des* on the exergy parameter illustrates the destructive exergy. Each term used in Eq. (9) is defined based on temperature and parameters of a system.

The exergy flow in for a process of heat transfer depends on the temperature of the boundary of the system (point j) and the amount the transferred heat (Sonntag *et al.,* 1998), which can be calculated by Eq. (10):

$$\dot{E}x_{Q_j} = \dot{Q}_j (1 - \frac{T_a}{T_j}) \quad (10)$$

As shown in Figure 1, the inflow exergy rate for HPFPSC only includes the exergy rate of solar radiation (Gunerhan and Hepbasli, 2007), which is defined in Eq. (11).

$$\sum \dot{E}x_{in} = \dot{E}x_{Q_{in}} = I_T A_C (1 - \frac{T_a}{T_s}) \quad (11)$$

The parameter T_s is the observable temperature of sun, which is 0.75 of black body temperature (Bejan *et al.,* 1981). There are other relations for the exergy rate of solar radiation which are not applicable for solar collectors (Farahat *et al.,* 2009, Petela, 1964). The rate of outflow exergy of HPFPSC includes the rate of exergy loss and the rate of exergy transferred through a heat pipe.

$$\sum \dot{E}x_{out} = \dot{E}x_{Q_{loss}} + \dot{E}x_{Q_{HP}} \quad (12)$$

Exergy loss includes exergy loss due to optical heat loss and exergy loss due to heat transfer from collector to the environment, which are presented in the first and second terms of the Eq.(13), respectively.

$$\dot{E}x_{Q_{loss}} = (1-\eta_o) A_C I_T (1 - \frac{T_a}{T_s}) + A_C U_L (T_p - T_a)(1 - \frac{T_a}{T_p}) \quad (13)$$

The first term in the right side of Eq. (13) defines the exergy loss caused by optical properties of glass and absorber part of the collector. Since the outlet heat transfer in the heat pipe ($\dot{Q}_{HP}$) occurs at T_c, the exergy flow through a heat pipe can be calculated according to Eq. (10), which is given by Eq. (14):

$$\dot{E}x_{Q_{HP}} = \dot{Q}_{HP} (1 - \frac{T_a}{T_c}) \quad (14)$$

The rate of exergy destruction which is generated due to the temperature difference in HPFPSC includes 3 terms:

1-Destructive exergy caused by the temperature difference between absorber and sun: Since the rate of absorbed heat by the absorber is $\eta_o I_T A_c$, one can conclude that:

$$\dot{E}x_{des,\Delta T_{s-p}} = \eta_o I_T A_C \left[(1 - \frac{T_a}{T_s}) - (1 - \frac{T_a}{T_p}) \right] = \eta_o I_T A_C T_a (\frac{1}{T_p} - \frac{1}{T_s}) \quad (15)$$

2-Destructive exergy due to the temperature difference between the absorber and evaporator of a heat pipe:

$$\dot{E}x_{des,\Delta T_{p-e}} = \dot{Q}_{HP} \left[(1 - \frac{T_a}{T_p}) - (1 - \frac{T_a}{T_e}) \right] = \dot{Q}_{HP} T_a (\frac{1}{T_e} - \frac{1}{T_p}) \quad (16)$$

Destructive exergy caused by the temperature difference between the evaporator and condenser of a heat pipe:

$$\dot{E}x_{des,\Delta T_{e-c}} = \dot{Q}_{HP} \left[(1 - \frac{T_a}{T_e}) - (1 - \frac{T_a}{T_c}) \right] = \dot{Q}_{HP} T_a (\frac{1}{T_c} - \frac{1}{Te}) \quad (17)$$

According to Eq. (9), we have:

$$\dot{E}x_{Q_{HP}} = \dot{E}x_{Q_{in}} - \left[\dot{E}x_{Q_{loss}} + \dot{E}x_{des,\Delta T_{s-p}} + \dot{E}x_{des,\Delta T_{p-e}} + \dot{E}x_{des,\Delta T_{e-c}} \right] \tag{18}$$

Replacing Eqs. (11)-(17) in Eq.(18):

$$\dot{Q}_{HP}(1-\frac{T_a}{T_c}) = I_T A_C (1-\frac{T_a}{T_s}) - \begin{bmatrix} (1-\eta_o)A_C I_T (1-\frac{T_a}{T_s}) + A_C U_L (T_p - T_a)(1-\frac{T_a}{T_p}) \\ +\eta_o I_T A_C T_a (\frac{1}{T_p} - \frac{1}{T_s}) + \dot{Q}_{HP} T_a (\frac{1}{T_c} - \frac{1}{T_p}) \end{bmatrix} \tag{19}$$

The thermal efficiency of a solar collector is defined as the amount of output useful energy divided by total solar heat gain, as follows:

$$\eta_{en} = \frac{\dot{Q}_{usefull}}{\dot{Q}_{in}} = \frac{\dot{Q}_{HP}}{A_C I_T} \tag{20}$$

According to Eq. (8), we have:

$$\eta_{en} = \eta_o - \frac{U_L (T_p - T_a)}{A_C I_T} \tag{21}$$

Various explanations have been used in order to investigate the exergy efficiency (Hepbasli, 2008). The most popular one is the rate of net exergy output to the total exergy input:

$$\eta_{ex} = \frac{\dot{E}x_{usefull}}{\dot{E}x_{in}} = 1 - \frac{\dot{E}x_{loss} + \dot{E}x_{dest}}{\dot{E}x_{in}} \tag{22}$$

Using Eq. (19), exergy efficiency for HPFPSC can be derived as:

$$\eta_{ex} = \frac{\dot{E}x_{usefull}}{\dot{E}x_{in}} = \frac{\dot{Q}_{HP}(1-\frac{T_a}{T_c})}{I_T A_C (1-\frac{T_a}{T_s})} = 1 - \begin{bmatrix} (1-\eta_o) + \frac{U_L (T_p - T_a)}{I_T} \frac{(1-\frac{T_a}{T_p})}{(1-\frac{T_a}{T_s})} \\ +\eta_o T_a \frac{(\frac{1}{T_p} - \frac{1}{T_s})}{(1-\frac{T_a}{T_s})} + \frac{\dot{Q}_{HP} T_a (\frac{1}{T_c} - \frac{1}{T_p})}{I_T A_C (1-\frac{T_a}{T_s})} \end{bmatrix} \tag{23}$$

In order to investigate the exergy efficiency of HPFPSCs, several experiments have been carried out in June-July 2011, in Tehran, Iran. The experimental setup mainly included heat pipes, water tank, and measurement equipment. All the heat pipes, according to a study performed by Arab et al. (Arab *et al.*, 2012), were made of copper. Twenty one U-shaped tubes were made by forty two tubes which were bent and had an outer diameter of 4mm, an inner diameter of 2 mm, and the radius of curvature of about 20 mm. The tubes were evacuated to 13 Pa and filled up partially, about 30% to 60%, with distilled water (Charoensawan *et al.*, 2003, Charoensawan and Terdtoon, 2008). pulsating heat pipe was developed with the condenser length (50 cm), and evaporator length of 100 cm. The width of the collectors, which had 42 tubes and a gap of 20 mm, was 1 m.

An adjustable stand was used for each collector to obtain a desired inclination angle . Fig. 2 (a) illustrates the placement of the collectors facing the south. ψ is the angle between the direct radiation and the normal vector of the plate, and θ_z is the solar zenith angle. Fig. 2 (a) clearly illustrates the orientation of the collector. The detail of the experimental setup is illustrated in Fig. 2 (b).

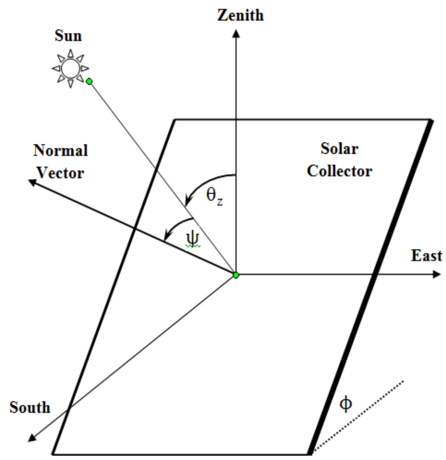

Fig. 2. Detail of the experimental setup. (a) orientation of the collectors, (b) experimental setup

A schematic of the pulsating heat pipes and their connection to the charging system is shown in Fig. 3. First, the inside of the pulsating heat pipes were cleaned with compressed air to prepare the pipes to be filled with degassed water. The amount of the working fluid was measured with a graduated cylinder with an accuracy of 0.1 cc. The pipes were filled with a specific amount of water as the T-type valve opened. The pressure of the system was measured again with a vacuum gauge (SUPCO-VG64) with an accuracy of 1 Pa.

The pulsating heat pipe flat plate solar collector (PHPFPSC) has an optical efficiency of 0.87. Apparent temperature of the sun is considered as 4350K.

A rotameter, k-type thermocouples, and a data logger with an accuracy of $0.1\,^0C$ were used to measure and obtain the experimental data. The schematic of the pulsating heat pipe with U-shaped tubes and measurement equipment were shown in Fig. 4. The data were recorded with the data logger at the sampling rate of 1 Hz. The amount of hourly solar radiation was measured with a pyranometer (CMP11, KIPP & ZONEN Inc.), and the ambient temperature was acquired from Iran Meteorological Organization's data.

The experiments were performed in Tehran between 9 A.M. to 5 P.M.. Because the solar radiation varies daily, the experiments were performed in limited periods and consecutive days in order to ensure that the variation of the solar radiation was negligible.

RESULTS & DISCUSSION

In PHPFPSC, the heat transfer occurs at a higher temperature and without significant pressure loss. Also, due to the nature of PHP, which can transfer the heat in a constant temperature, the temperature of the evaporator and condenser sections of PHP remain almost constant, which leads to less exergy destruction. Using the concluded the relations of the previous sections, the exergy analysis of a solar plate collector is given in Table 1 and is compared with the results of Luminosu and Fara (Luminosu and Fara, 2005). The highest value for the exergy efficiency of PHPFPSC is 9.91% which is more than three times of the same value for a simple solar collector, introduced in it.

An exergy efficiency relation for simple solar collectors which takes into account the pressure gradient, caused by the natural convection, is introduced in (Farahat *et al.*, 2009). The forth term in Eqs. (23) is the exergy destruction due to the heat transfer of the collectors. Although the amount of heat transfer increases in a PHP, low temperature difference between the evaporator and condenser decreases this term in Eq. (23). According to the nature of PHPs, pressure losses are negligible in comparison with other losses. Hence, there is no exergy destruction due to the pressure drop.

The variations of exergy efficiency and energy efficiency of PHPFPSCs with the overall heat loss coefficient are given in Fig. 5.

Increasing the overall heat loss coefficient leads to more heat loss in the collector and according to 2nd term at the right side of Eq. (21), energy efficiency of the collector decreases. Moreover, this can result in higher exergy destruction and lower exergy efficiency (2nd term in the bracket of Eq. (23)). Thus, one can conclude that the use of better insulation materials, glass with higher light transmission factor and thermal break frames can directly improve the energy and exergy performance of PHPFPSC.

Another important parameter the efficiency of solar collectors is the intensity of solar radiation. Increasing solar radiation, while there are no significant variations

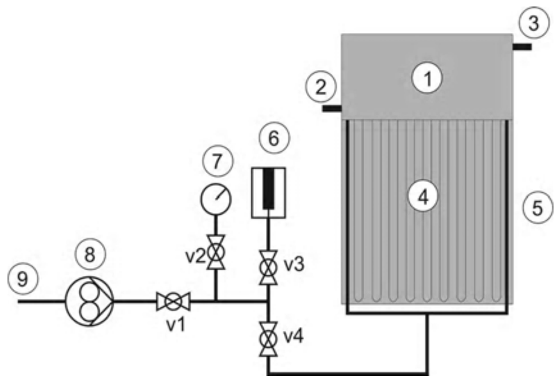

Fig. 3. The schematic of the charging system: (1) water tank. (2) inlet water, (3) outlet water, (4) heat pipes, (5) absorber plate, (6) graduated cylinder, (7) vacuum gauge, (8) vacuum pump, (9) atmosphere

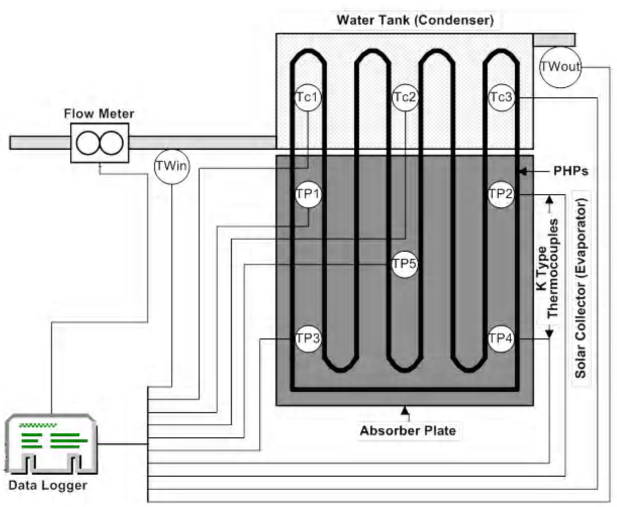

Fig. 4. The schematic of the pulsating heat pipe with U-shaped tubes and measurement equipment

Table 1. Exergy analysis for a solar collector: A comparison between current study and (Luminosu and Fara, 2005)

Researchers	Parameters	Hourly interval					
		8-10	10-12	12-14	14-16	16-18	18-20
Luminosu and Fara (Luminosu and Fara, 2005)	$\langle T_{in} \approx T_{out} \rangle_h\ (K)$	301.15	303.15	305.15	307.15	306.15	303.15
	$\langle I_T \rangle_h\ (W/m^2)$	503	795	788	489	397	293
	$\langle T_{out,\exp} \rangle_h\ (K)$	328.15	346.15	351.15	335.15	328.15	320.15
	$\langle \eta_{en,\exp} \rangle_h\ (\%)$	38.00	39.00	42.00	43.00	42.00	42.00
	$\langle \eta_{ex,\exp} \rangle_h\ (\%)$	1.60	2.50	2.90	1.80	1.40	1.20
Present work	$\langle T_{in} \approx T_{out} \rangle_h\ (K)$	301.55	302.25	303.85	304.25	303.85	306.25
	$\langle I_T \rangle_h\ (W/m^2)$	682.1	800.0	876.8	836.0	726.0	536.8
	$\langle T_{c,\exp} \rangle_h\ (K)$	355.95	365.85	368.35	368.05	363.55	351.35
	$\langle \eta_{en,\exp} \rangle_h\ (\%)$	47.41	47.69	52.66	51.86	49.87	48.61
	$\langle \eta_{ex,\exp} \rangle_h\ (\%)$	7.79	8.91	9.91	9.67	8.80	6.71

in optical properties of glass and coefficient of heat loss, results in higher energy and exergy efficiency. This is illustrated in Fig. 6.

Any improvement in the optical properties of the cover glass will pass more fractional ratio of the received sun ray and leads to a higher η_o value (at a constant surface area) and higher energy absorption by water. Hence, an increment in the energy and exergy efficiencies is observed. Fig. 7 shows the changes of energy and exergy efficiencies with the variation of the optical efficiency of the glass.

More temperature differences between the glass plate and ambient results in higher heat loss in the collector, which decrease the energy efficiency of system, which was predictable from Eq. (6). Decreasing the ambient temperature leads to higher $(T_p - T_a)$ and higher heat loss in the collector. Hence the exergy efficiency decreases. However, higher ambient temperature and so lower $(T_p - T_a)$ value result in higher exergy efficiency. But after a critical point, according to Eqs. (15) and (16), the exergy destruction increases, due to higher temperature differences between the absorber and the sun, and between the absorber and evaporator. Hence the exergy efficiency decreases after the critical point. One can conclude that there should be a specific temperature difference between the collector and the ambient to have the best performance and optimized values for energy and exergy efficiencies. Fig. 8 illustrates the changes of energy and exergy efficiencies versus temperature differences between the glass plate and ambient.

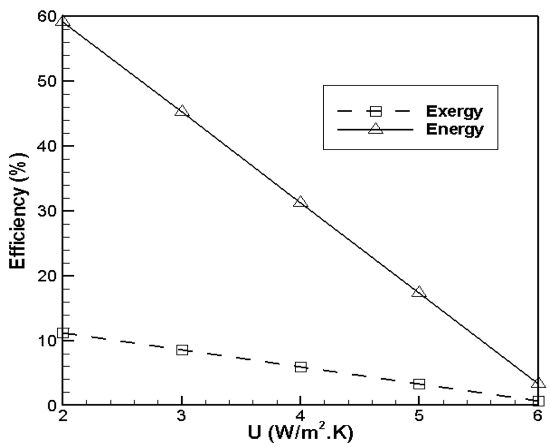

Fig. 5. Variations of exergy efficiency and energy efficiency of PHPFPSC with overall heat loss coefficient : $I_T = 500 W / m^2, \eta_o = 0.87$

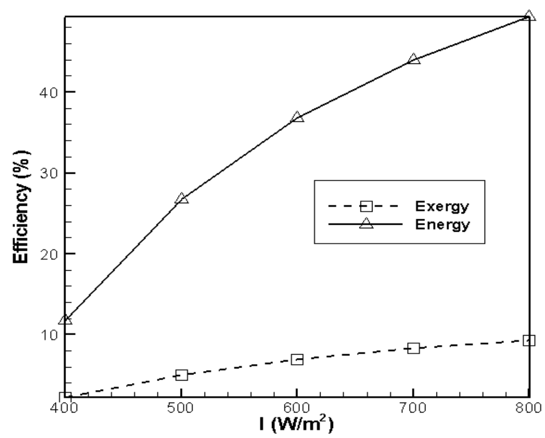

Fig. 6. Variations of exergy efficiency and energy efficiency of PHPFPSC with changes of the solar radiation : $U = 4.32 W/m^2.K$, $\eta_o = 0.87$

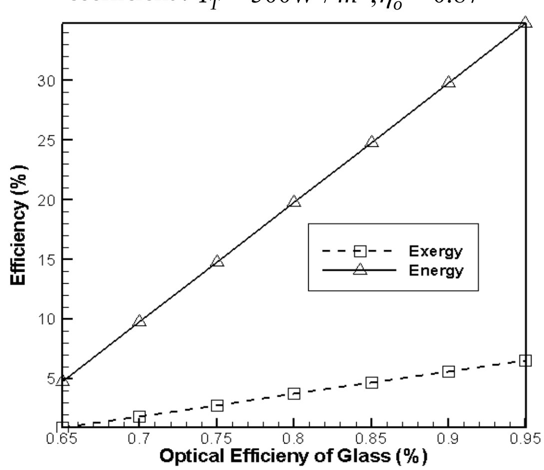

Fig. 7. Variations of exergy efficiency and energy efficiency of PHPFPSC with changes of the optical efficiency of glass :

$$U = 4.32 W/m^2.K, I = 500 W/m^2$$

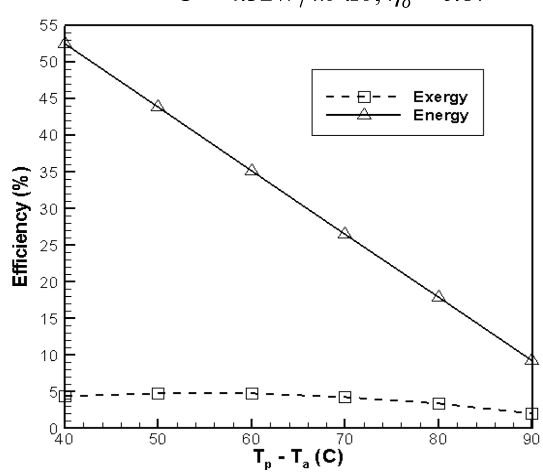

Fig. 8. Variations of exergy efficiency and energy efficiency of PHPFPSC with changes of the optical temperature difference between glass plate and ambient :

$$U = 4.32 W/m^2.K, I = 500 W/m^2$$

CONCLUSION

The energy efficiency (1st thermodynamic law) is not sufficient to analyze a thermal system. Especially, in solar energy systems, in which the input energy is free, and an investigation on the first law of thermodynamics cannot clearly illustrate the performance of a system. Consequently, it is necessary to perform an exergy analysis on these systems because an exergy analysis is the basis of an economic analysis for a system. Unfortunately, one of the main disadvantages of simple solar collectors is the very low exergy efficiency. Hence, heat pipes as a high performance heat transfer device are recently employed to increase the energy and exergy efficiency of solar collectors. In this work, to make a comparison between the heat performance of thermosyphon solar collectors, which use natural convection for heat transfer between the absorber plate and tank, and heat pipe solar collectors, in which heat pipes serve an effective role in heat transfer between the absorber and tank, and, to examine the effect of using pulsating heat pipes on the heat transfer process of solar applications, an analytical investigation on the exergy efficiency of such systems has been done, and relations for calculation of exergy losses in HPFPSC have been derived. It is concluded that the use of a PHP, due to the elimination of losses in natural convection of fluid and heat transfer in almost constant temperature, can increase the exergy efficiency of system. The effects of various parameters have been studied, and the following statements can be made:

· The higher overall coefficient of the heat transfer of a collector leads to a linear decrement in the energy and exergy efficiencies of the collector, the higher energy loss to ambient and lower availability of the system. An increase in the total heat coefficient causes a high heat transfer between the collector and ambience for a minor temperature difference, which leads to a decrease in the efficiency of the collector, and the exergy loss increases, which results in decreases in the exergy efficiency.

• A high intensity of solar radiation has a negligible effect on the energy loss, and also, the exergy destruction of PHPSPC, energy efficiency, and exergy efficiency increase.

• An increment in the optical efficiency of the collector results in lower heat losses, and both the energy and exergy efficiencies increase. A higher optical efficiency of the glass leads to a lower absorption of the solar radiation, and as a result, the temperature of the glass decreases, which causes a decrease in the energy loss on the surface of the glass, and, the irreversibility of this mechanism.

• The temperature difference between the collector and the ambient has an optimal point for exergy efficiency, and a larger difference can lead to lower exergy efficiency. However, an increase in the temperature difference decreases the energy efficiency due to the possibility of more heat losses to ambient. Finally, it is concluded that a high-performance PHP can improve the energy and exergy efficiencies due to a higher heat transfer coefficient and less heat losses to the ambient.

ACKNOWLEDGEMENT

The authors would like to express their gratitude and acknowledgements for supports provided by Sharif Energy Research Institute (SERI).

NOMENCLATURE

A	**area (m^2)**
C_P	heat capacity of the fuid (kJ/kg K)
D	diameter(m)
$\dot{E}$	energy flow rate(W)
$\dot{E}x$	exergy flow rate(W)
h	specific enthalpy (J/kg)
I	solar radiation intensity (W/m^2)
K	conductivity(W/m.k)
L	length(m)
$\dot{m}$	mass flow rate
P	pressure
$\dot{Q}$	heat flow rate(W)
T	temperature
U	global coeffcient of heat losses (W/m^2K)
Greek symbols	
α	absorbance
Δ	difference in temperature or pressure
η	effciency
τ	transmittance
ρ	density (kg/m^3)
$\tau\alpha$	effective product transmittance–absorbance

Subscripts	
a	ambient
C	collector
c	condenser
des	destructive
e	evaporator
en	energy
ex	exergy
fg	fluid-gas
HP	heat pipe
i	inner
in	inlet
j	local
L	overall
l	liquid
o	optic
out	outlet
p	plate
s	solar
T	total
v	vapor
Abbreviations	
FPSC	Flat plate solar collector
HPFPSC	heat pipe flat plate solar collector
PCM	Phase change material
PHP	pulsating heat pipe
PHPFPSC	pulsating heat pipe flat plate Solar Collectors (PHPFPSC)

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