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Energy and exergy analyses of a flat plate solar collector using different nanofluids

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Abstract: *This work theoretically investigates the effect of nanofluids on the energy and exergy efficiencies of a flat plate solar collector. Four different types of nanofluids, Al2O3/water, SiO2/water, TiO2/water and CuO/water, have been taken into consideration to examine the performance of the solar collector. The efficiencies are determined for a range of volume fractions of each nanoparticle and subsequently compared to each other. It is shown that the application of CuO/water nanofluid as a fluidic medium in a conventional solar collector can enhance its energy and exergy efficiencies by 38.46 and 15.52% respectively, when compared to water. The ultimate objective of the present study is to understand the feasibility of each kind of nanofluid prior to the application in solar to thermal energy conversion.*

Keywords: *Energy; Exergy; Nanofluid; Solar collector; Dead state; Building.*

Nomenclature

A_p	=	absorption area, m^2	P	=	mechanical power, W	η_{Ex}	=	exergy efficiency
C_p	=	specific heat, J/kg.K	$\dot{Q}$	=	thermal energy rate, W	η_o	=	optical efficiency
e	=	exergy loss	$\dot{Q}_u$	=	energy gain rate, W	τ	=	transmittance
						α	=	absorptance
$\dot{E}_g$	=	exergy gain rate per unit area, W/m^2	s	=	entropy per unit mass, J/kg.K	φ	=	nanoparticles volume fraction, %
$\dot{E}_{sun}$	=	exergy flow from sun, W/m^2	S	=	absorbed irradiation, W/m^2	ρ	=	density, kg/m^3
F_R	=	heat removal factor	T	=	temperature, K	σ	=	overall entropy production, J/kg.K
F'	=	collector efficiency factor	T_c	=	absorber plate temperature, K	**Subscripts**		
g	=	gravitational acceleration, m/sec^2	U_l	=	overall heat loss, W/m^2.K	a	=	ambient
						bf	=	base fluid
I_T	=	incident solar energy per unit area, W/m^2	V	=	volume flow rate, L/min	$dest$	=	destruction
						f, in	=	inlet fluid
k	=	heat conductivity, W/m.K	z	=	height from reference level, m	f, out	=	outlet fluid
						np	=	nanoparticles
$\dot{m}$	=	mass flow rate, kg/s	η_{En}	=	energy efficiency	nf	=	nanofluid

1. Introduction

Implications of renewable energy sources such as sun, wind, water and geothermal, is one of the most common interests among the scientists and engineers across the world these days. The main advantage of these energy sources is the minimal impact on the environment. However, the challenge is still huge in identifying the means of collecting and converting this energy in forms that are useful. In case of solar energy, the primary devices are solar thermal collectors and photovoltaics (Dziadik, 2012). A typical solar collector is made of a black surface, known as an absorber, and an array of tubes embedded or fused on the surface. The basic principle is that the black surface absorbs thermal energy in the form of heat from the sun and transfers the energy to the fluid flowing inside the attached tubes.

Flat plate solar collectors are considered to be one of the most cost effective devices amongst the existing solar energy conversion appliances (Kostić and Pavlović, 2012; Mahian et al., 2013). It has been documented that the efficiency of this kind of thermal collector is mainly affected by the transfer of heat from the absorber to the working fluid (Otanicar, 2009). Maximizing the heat transfer between these two mediums and minimizing the heat loss to the surrounding environment could result in building a highly efficient solar thermal collector (Otanicar, 2009). However, this study looks at an alternative approach to produce an efficient thermal collector through the replacement of the working fluid, by a highly conductive heat transfer medium such as nanofluid. It has been shown that adding a small amount of nanoparticles in water results in a substantial increase in its thermal conductivity (Yousefi et al., 2012a). A range of studies have reported the properties of particle mixed fluids considering their size in μm or even mm range (Wang and Mujumdar, 2007; Xuan and Li, 2000). The results are promising in some cases, but their disadvantages such as, instability of suspensions, flow resistance, clogging and abrasion, limit their practical applications (Wang and Mujumdar, 2007; Xuan and Li, 2000). However, the recent development in science and technology pave the way for manufacturing solid particles in nanoscale. Nanofluids are a new class of advanced heat-transfer fluids engineered by dispersing nanoparticles smaller than 100 nm (nanometre) in diameter in conventional heat transfer fluids (Choi and Eastman, 1995; Fotukian and Esfahany, 2010).

Conventional solar thermal collectors exhibit poor performance considering the amount of thermal energy received from the sun. It has been documented that the low performance might be attributed to the higher specific heat and low thermal conductivity of the absorbing fluid, higher entropy generation, and low absorber temperatures (Natarajan and Sathish, 2009; Tyagi et al., 2009). Recently, some studies have shown that the performance of a solar collector might be improved by changing the heat transfer medium to nanofluids. Natarajan and Sathish (2009) reported the effect of carbon nanotubes (CNT) on the properties of the base fluid (water). The study observed improved thermal properties and ultimately was able to enhance the efficiency of a solar water heater using CNT nanofluid as the working medium. Tyagi et al. (2009) examined the application of aluminium nanofluids in a direct absorption solar collector (DAC) and reported 10 % enhancement in efficiency compared to the conventional DAC (Yousefi et al., 2012b). A number of studies have reported the improved performance of solar thermal collectors using nanofluids as the working medium (Otanicar and Golden, 2009; Otanicar et al., 2010). However, the exergy analysis along with energy needs to be calculated for a better understanding of the application of nanofluids in solar to thermal energy conversions.

The aim of the present work is to determine the effect of nanofluids on the energy and exergy efficiencies of a flat plate solar collector. Several parameters such as, volume flow rate, nanoparticles volume concentration, density, specific heat and output temperatures are taken into consideration in explaining the performance of the solar thermal collector. An attempt is made to compare the results of the present study to the literature reports.

2. Analytical Approach

2.1 The first law of thermodynamics (analysis of energy efficiency)

The first law of thermodynamics is also known as the law of conservation of energy. It states that the total internal energy for a given system is a function of work and heat that are being done on the system. The amount of heat needs to be added or subtracted to the work in determining the internal energy of the system depending on its flow direction (plus and minus for heat flows in and out of the system, respectively). According to the 1st law, the energy balance Equation for a stationary process observed through a control volume can be written as (Cabrera et al., 2013):

$$\dot{Q} - P = \sum_{outlet} \dot{m}_k \left(h + gz + \frac{w^2}{2} \right)_k - \sum_{inlet} \dot{m}_k \left(h + gz + \frac{w^2}{2} \right)_k \tag{1}$$

where w and h represent mass flow velocity $(\dot{m}/\rho A)$ and specific enthalpy (J/kg) assumed at the system inlet and outlet respectively.

The present study considers the flat plate solar thermal collector as the selected system for thermodynamic analysis and nanofluid as its fluidic medium. Therefore, possible heat gain (Q_u) by the absorbing medium (nanofluid) can be written as,

$$\dot{Q}_u = \dot{m}C_p\left(T_{f,out} - T_{f,in}\right) \tag{2}$$

The heat capacity and density of the nanofluids are calculated using Equations (2) & (3) (Xuan and Roetzel, 2000; Zhou and Ni, 2008)

$$C_{p,nf} = \varphi C_{p,np} + C_{p,bf}(1-\varphi) \tag{3}$$

$$\rho_{nf} = (1-\varphi)\rho_{bf} + \varphi\rho_{np} \tag{4}$$

Equation (2) does not consider the parameters related to the heat loss from the solar thermal collector to the surrounding. Hottel–Whillier (Struckmann, 2008) proposed a more convenient solution for the calculation of heat gain involving the missing parameters in Equation (2).

$$\dot{Q}_u = A_p F_R\left[S - U_1\left(T_{f,in} - T_a\right)\right] \tag{5}$$

Absorbed irradiation per unit area of solar collector absorber plate (S) is determined by,

$$S = I_T(\tau\alpha) \tag{6}$$

where *(τα)* is known as optical efficiency *(η_o)* or product of transmittance – absorptance of the solar collector (Sukhatme and Nayak, 2017).

F_R may be determined form Equation (7)

$$F_R = \frac{\dot{m}C_p}{U_1 A_p}\left[1 - \exp\left\{-\frac{F'U_1 A_p}{\dot{m}C_p}\right\}\right] \tag{7}$$

The energy balance equation, represented by Equation (1), may be written in a simplified form for the absorber plate of a solar thermal collector as (Sukhatme and Nayak, 2017):

$$\dot{Q}_u = A_p S - U_1 A_p\left(T_c - T_a\right) \tag{8}$$

The absorber plate temperature (average), plate area, absorbed irradiation flux by the unit area of the absorber plate and overall heat loss are assumed as constant factors or variables with little effect. The efficiency of the flat plate solar collector for a given amount of solar irradiation, absorbed by the collector surface, can be written as follows:

$$\eta_{En} = \frac{\dot{Q}_u}{A_p I_T} = \frac{\dot{m}C_p\left(T_{f,out} - T_{f,in}\right)}{I_T} \text{[without considering heat loss]} \tag{9}$$

$$\eta_{En} = F_R(\tau\alpha) - F_R U_1 \frac{T_{f,in} - T_a}{I_T} \quad \text{[including heat loss]} \tag{10}$$

The present study determines the collector performance for normal incidence conditions, therefore, *$F_R(\tau\alpha)$, F_R*, and *U_1* are assumed to be constant within the range of the tested temperatures (Yousefi et al., 2012a). Considering the above assumptions, one can note that the efficiency definition is only a comparison between quantities, which were metrically homogeneous, but not conceptually equivalent.

2.2 The second law of thermodynamics (analysis for exergy efficiency)

Energy efficiency is not enough to describe a thermodynamic process because the energy equation does not account for the internal losses. The second law of thermodynamics provides more information about the optimal operating stage, inefficiencies, corresponding magnitudes and traces (Farahat et al., 2009; Luminosu and Fara, 2005). It starts with considering that the real processes are not reversible and gains entropy through the processes. Some of the common irreversible processes are molecular diffusion, friction, hysteresis etc. According to Clausiu's statement, the second law can be written as,

$$\sum_{outlet}(\dot{m}.s)_k - \sum_{inlet}(\dot{m}.s)_k = \sum_j\left(\frac{\dot{Q}}{T}\right)_j + \sigma \tag{11}$$

During the first law analysis, there is a term for work, but no consideration for irreversibility. Besides, the second law discusses irreversibility, but avoids the term work. To gather more information, the first and second laws are combined together. By combining Eqs. (1) and (11), one can obtain the Gouy-Stodola equation (Sarhaddi et al., 2010):

$$P = \sum_n \dot{Q}_n\left(1-\frac{T_a}{T_n}\right) + \sum_{Inlet}\dot{m}_k\left(h - T_a s + \frac{w^2}{2} + gz\right)_k - \sum_{Outlet}\dot{m}_k\left(h - T_a s + \frac{w^2}{2} + gz\right)_k - T_a\sigma \tag{12}$$

Exergy can be expressed as the obstruction of any work proportion to its dead state. There is no further work, when the environment comes to an equilibrium with the system. At this state, the system is defined as dead state. So, for a control volume, Eq. (12) may be rewritten in terms of exergy, as follows:

$$\left(\dot{E}_P\right)_{out} - \left(\dot{E}_P\right)_{in} = \left(\dot{E}_Q\right)_{in} - \left(\dot{E}_Q\right)_{out} + \left(\dot{E}_m\right)_{in} - \left(\dot{E}_m\right)_{out} - \sigma T_a$$

$$\left(\dot{E}_P + \dot{E}_Q + \dot{E}_{\dot{m}}\right)_{in} = \left(\dot{E}_P + \dot{E}_Q + \dot{E}_{\dot{m}}\right)_{out} + \sigma T_a$$

Therefore,

$$\sigma T_a = \sum_{in}\dot{E}_j - \sum_{out}\dot{E}_k \tag{13}$$

where exergy of work, $\dot{E}_p$, exergy of heat $\dot{Q}$ available at temperature *T*, $\dot{E}_Q$ and exergy of a mass flow, $\dot{E}_{\dot{m}}$ are defined as follows:

$$\dot{E}_P = P$$

$$\dot{E}_Q = \dot{Q}\left(1 - \frac{T_a}{T}\right) \quad and$$

$$\dot{E}_{\dot{m}} = \dot{m}\left[\left(h - h_o\right) - T_a\left(s - s_o\right) + \frac{w^2 - w_o{}^2}{2} + g\left(z - z_o\right)\right]$$

The irreversibility can then be quantified as the difference in exergy measured at the inlet and outlet sections of the control volume. The simplest exergy balance equation per unit interception area of a solar collector can be expressed in steady state as shown below (Suzuki, 1988):

$$\dot{E}_g = \eta_o\dot{E}_{sun} - \dot{E}_{loss} \tag{14}$$

The exergy loss due to the fluid pressure drop is assumed to be negligibly small. Eq. (14) can also be written as (Jafarkazemi and Ahmadifard, 2013),

$$\sum\dot{E}_{in} - \sum\dot{E}_{out} = \sum\dot{E}_{dest} \tag{14*}$$

The exergy collection rate in steady state is exergy gained by the heat transfer fluid while the fluid temperature increases from $T_{f,\,in}$ at the inlet to $T_{f,\,out}$ at the outlet. The expression of the exergy collection rate, assuming that the fluid is incompressible, can be obtained by use of the following equation without considering mechanical exergy,

$$\dot{E}_g = \dot{m}C_p\left(T_{f,out} - T_{f,in} - T_a \ln\frac{T_{f,out}}{T_{f,in}}\right) \tag{15}$$

There are two important points that should be noted in considering the exergy available ratio for solar radiation. One is that the solar flux radiating on earth can be assumed as always being in a steady state but never in equilibrium state. The other is that the radiation of the sun is a type of open system which means losses of photons cannot be recovered unlike an equilibrium closed system. From these facts the Carnot's expression of (1 - T_a/T_s) is thought to be appropriate for the solar radiation exergy which has the same form as Jeter's result (Jeter and Stephens, 2012). From the above mentioned, the exergy flux from the sun is defined here as:

$$\dot{E}_{sun} = I_T\left(1-\frac{T_a}{T_s}\right) \tag{16}$$

The heat transfer process from the sun to the collector's working fluid consists of two main parts, absorbing the solar radiation by the absorber plate and heat transfer from the absorber plate to the working fluid. The exergy destructions occur during these two processes including flowing parts (Suzuki, 1988), as can be seen in **Figure 1**.

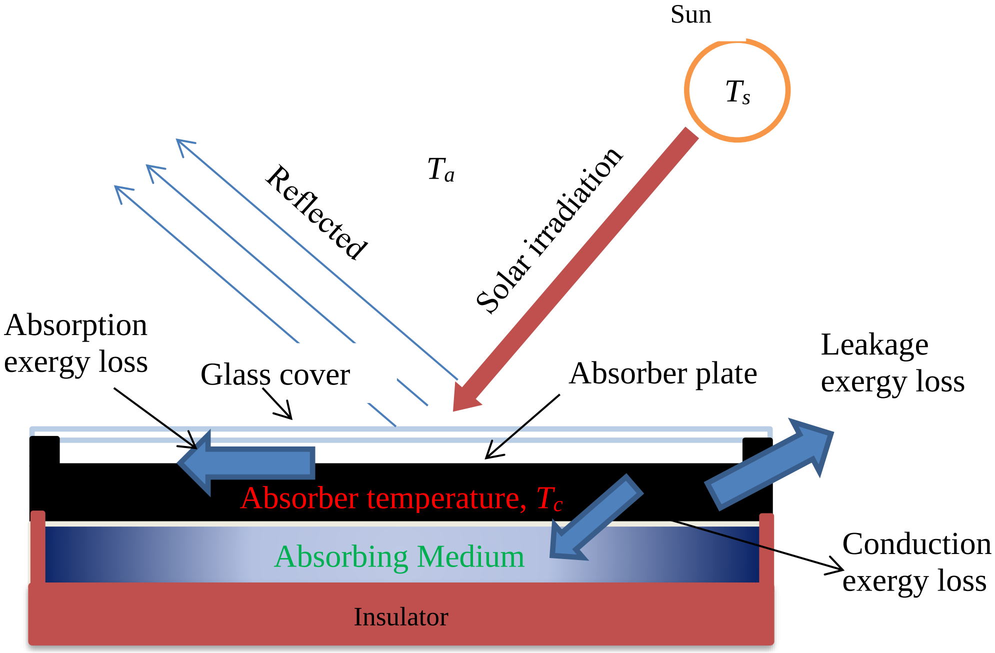

Figure 1. Schematic diagram for different exergy losses

- *Absorption exergy loss (radiation ➡ plate):* an exergy annihilation process when the solar radiation at T_s, is absorbed by the absorber at T_c.
- *Leakage exergy loss (plate ➡ ambient):* an exergy loss process accompanied with heat leakage from the absorber out into its surroundings.
- *Conduction exergy loss (plate ➡ fluid):* an exergy annihilation process caused by heat conduction between the absorber and the heat transfer fluid.

The above three kinds of exergy loss processes are closely related with the corresponding entropy generation rates through Gouy-Stodola's theorem (Bejan, 1982). These three entropy generation rates can be stated from the thermodynamic considerations as follows:

$$\Delta\dot{s}_{rp} = \int_0^1 kI_T\left(\frac{1}{T_1}-\frac{1}{T_s}\right)d\sigma \tag{17}$$

$$\Delta\dot{s}_{pa} = \int_0^1 U_1(T_1-T_a)\left(\frac{1}{T_a}-\frac{1}{T_1}\right)d\sigma, and \tag{18}$$

$$\Delta \dot{s}_{pf} = \int_0^1 k(T_1 - T_f)\left(\frac{1}{T_f} - \frac{1}{T_1}\right) d\sigma \tag{19}$$

Although Eqs. (17) - (19) cannot be integrated unless a distribution of the local absorber temperature *(T_1)* and the heat transfer coefficient are known, these equations still can be approximated using the mean absorber temperature as follows:

$$\Delta \dot{s}_{rp} = kI_T\left(\frac{1}{T_c} - \frac{1}{T_s}\right) \tag{17´}$$

$$\Delta \dot{s}_{pa} = U_1(T_c - T_a)\left(\frac{1}{T_a} - \frac{1}{T_c}\right), and \tag{18´}$$

$$\Delta \dot{s}_{pf} = \int_{T_{f,in}}^{T_{f,out}} \frac{\dot{m}C_p dT}{T} - \frac{\dot{m}C_p(T_{f,out} - T_{f,in})}{T_{f,out}} \tag{19´}$$

In Eq. (19'), the first term on the right-hand side is an entropy flow received by the fluid from the absorber and the second term represents entropy of the collected energy as it has been in the absorber. The difference of both terms becomes the entropy generation rate while heat transfers from the absorber to the fluid. The exergy loss term in Eq. (14) can be seen from Eqs. (17') - (19') using Gouy-Stodola's theorem as,

$$\dot{E}_{loss} = T_a(\Delta \dot{s}_{rp} + \Delta \dot{s}_{pa} + \Delta \dot{s}_{pf}) \tag{20}$$

Hence, the exergy-balance-equation of a solar collector in steady state can be derived by substituting Eqs. (15), (16), and (20) into Eq. (14). After a few arrangements, it becomes:

$$\begin{aligned} &\dot{m}C_p\left(T_{f,out} - T_{f,in} - T_a \ln\frac{T_{f,out}}{T_{f,in}}\right) = I_T\left(1 - \frac{T_a}{T_s}\right) \\ &-\left[(1-\eta_o)I_T\left(1 - \frac{T_a}{T_c}\right) + I_T T_a\left(\frac{1}{T_c} - \frac{1}{T_s}\right) + U_1(T_c - T_a)\left(1 - \frac{T_a}{T_c}\right) + \dot{m}C_p T_a\left(\ln\frac{T_{f,out}}{T_{f,in}} - \frac{T_{f,out} - T_{f,in}}{T_c}\right)\right] \end{aligned} \tag{21}$$

By rearranging this equation, the following energy balance equation of a solar collector can be easily obtained:

$$\dot{m}C_p(T_{f,out} - T_{f,in}) = \eta_o I_T - U_1(T_c - T_a) \tag{22}$$

The exergetic efficiency is defined here and is expressed using Eq. (21) as follows:

$$\eta_{Ex} = \frac{\dot{E}_g}{\dot{E}_{sun}}$$

$$= 1 - \left[(1-\eta_o)\frac{1 - T_a/T_c}{1 - T_a/T_s} + \frac{1/T_c - 1/T_s}{1/T_a - 1/T_s} + \frac{U_l(T_c - T_a)}{I_T}\frac{1 - T_a/T_c}{1 - T_a/T_s} + \frac{\dot{m}C_p T_a}{I_T(1 - T_a/T_s)}\left(\ln\frac{T_{f,out}}{T_{f,in}} - \frac{T_{f,out} - T_{f,in}}{T_c}\right)\right] \tag{23}$$

$$= 1 - \left[e_{opt} + e_{rp} + e_{pa} + e_{pf}\right] \tag{24}$$

All terms in brackets in Eqs. (23) and (24) represent exergy losses and their physical meanings are given as follows:

- ✓ e_{opt}: optical loss fraction of the absorbed solar radiation due to transmissivity of glazing and absorptance of the absorber.
- ✓ e_{rp}: a loss fraction when the solar radiation at T_s is absorbed by the absorber at T_c. (The high quality energy is degraded by absorption at low temperature.)
- ✓ e_{pa}: a fraction of the exergy leakage from the absorber to the surroundings.
- ✓ e_{pf}: Heat-conduction loss fraction accompanied with the heat transfer from the absorber to the fluid.

Two of the above loss fractions, e_{opt} and e_{pa}, correspond to the terms $(1 - \eta_o)$ and $U_l(T_c - T_a)/I_T$ in a well-known expression of energetic efficiency; the other two fractions have no corresponding term in the energetic analysis because they are not supposed as loss processes. It should be noted here that the term given for heat-conduction loss e_{pf} is closely related with the collector efficiency factor. Considering the correlations of temperature distribution in the collector, the following correlation will be obtained:

$$\frac{T_{f,out} - T_a - S/U_1}{T_{f,in} - T_a - S/U_1} = \exp\left(-\frac{U_1 A_p F'}{\dot{m} C_p}\right) \tag{25}$$

Here also, using the above equation, the component of outlet fluid temperature is omitted from Eq. (23) and the correlation of collector exergy efficiency is rephrased into the following form (Jafarkazemi and Ahmadifard, 2013):

$$\eta_{Ex} = \frac{\dot{m}C_p\left[\left(T_{f,in} - T_a - \frac{S}{U_1}\right)\left(\exp\left(-\frac{U_1 A_p F'}{\dot{m}C_p}\right) - 1\right)\right] - \dot{m}C_p\left[T_a\left(\frac{\exp\left(-\frac{U_1 A_p F'}{\dot{m}C_p}\right) - 1}{T_{f,in}}\left(T_{f,in} - T_a - \frac{S}{U_1}\right) + 1\right)\right]}{A_p I_T\left[1 - \left(\frac{T_a}{T_s}\right)\right]} \tag{26}$$

2.3 Input Data

The present study has some considerations to simplify the analysis. These are constants such as, absorbing fluid properties, exergy flow rate to be equal to the solar flux, area of the absorber plate, overall heat loss and other heat transfer coefficients. Furthermore, the fluid inlet temperature and the ambient temperature are also assumed to be constant and equal. Tables 1-3 (input data tables) list the properties of nanoparticles, the characteristic parameter of the solar collector and the analysis condition, respectively.

Table 1 Properties of different nanomaterial and base fluid (Moghaddami et al., 2011; Pandey and Nema, 2012).

Material	**Specific heat, C_p (J/kg.K)**	**Thermal conductivity, k (W/m.K)**	**Density, ρ (kg/m^3)**
Alumina (Al_2O_3)	773	40	3960
Copper oxide (CuO)	551	33	6000
Titanium oxide (TiO_2)	692	8.4	4230
Silicon di oxide (SiO_2)	765	36	3970
Water (H_2O), base fluid	4182	0.60	1000

Table 2 Characteristic parameters for two kinds of solar collectors (Suzuki, 1988).

Solar collector type	**Optical efficiency, η_o**	**Overall heat loss, U_1 ($W/m^2.K$)**	**Collector efficiency factor, F´**
Evacuated tube	0.47	1.1	0.99
Flat plate	0.82	5.0	0.97

3. Results and discussion

Table 3 includes selected environmental and analyses conditions for solar collector. Collector efficiency was determined using Eqs. (9), (10) and (26) and data from Tables 1-3. The results are shown in Table 4.

Obtained results were compared with the data reported by Luminosu et al. (2005) which were reported for the open circuit mode of the solar collector and Farahat et al. (2009) which were reported for computer simulation of the solar collector. The comparison is represented in Table 5.

Table 3 Environmental and analysis conditions for the flat plate solar collector (Alim et al., 2013).

Parameters of collector	**Value**
Type	Black paint flat plate
Glazing	Single glass
Agent fluids	Water, Al_2O_3, SiO_2, TiO_2 and CuO nanofluids
Absorption area, A_p	1.51 m^2
Wind speed	20 m/s
Collector tilt, β_o	20°
Fluid inlet and ambient temperature, $T_{f,in} \approx T_a$	300 K
Apparent sun temperature, T_s	4350 K
Optical efficiency, η_o	0.84
Emissivity of the absorber plate, ε_p	0.95
Emissivity of the covers, ε_c	0.90
Glass thickness, t	4 mm
Insulation thermal conductivity, k_i	0.06 W/m·K
Incident solar energy per unit area of the absorber plate, I_T	500 W/m^2
Inner diameter of pipes, D_i	0.04 m

Table 4 Energy and exergy efficiency enhancement compared to the base fluid.

Absorbing medium	**Maximum η_{En} enhancement, (%)**		**Maximum η_{Ex} enhancement, (%)**	
	φ = 2% and diff. volume flow rate	*φ = 3.20% and V = 1 L/s*	*φ =2% and V= 2.40 L/s*	*φ = 3.20% and V = 1 L/s*
CuO nanofluid	38.46	13.25	15.52	13.18
TiO_2 nanofluid	28.84	8.40	5.63	8.10
SiO_2 nanofluid	28.84	6.40	5.63	6.41
Al_2O_3 nanofluid	28.84	6.40	5.63	6.41

Table 5 Comparison among the present analysis, experimental results and computer simulation.

	$T_{f, in}$ or T_a, (k)	I_T, (W/m^2)	S, (W/m^2)	ΔT, (K)	η_{Ex}, (%)
Present analysis (CuO)	300.00	500	420	62.00	3.35
Luminosu and Fara, (2005)	305.15	788	580	46.00	2.90
Farahat et al. (2009)	300.00	500	420	58.82	2.95

As can be seen, obtained results are in good agreement with the literature reports. **Table 5** allows the following points to be made:

- While the introduction of nanofluids increases viscosity, density and thermal conductivity, specific heat decreases substantially (Pandey and Nema, 2012)

- It is obvious that the lesser entropy generation number leads to higher exergy efficiency of the system. Theoretically one may expect that the entropy generation can be reduced with the application of nanoparticle water (Moghaddami et al., 2011).

- The heat transfer increases with the increase in concentration of nanoparticles (Lelea, 2011).

Figure 2 illustrates the changes in energy efficiency for the volume flow rate of 1 to 3.8 *L/min*. As can be seen, the energy efficiency increases steadily with the volume flow rate. Energy efficiency was determined using Eq. (9), input data tables, constant output temperature differences and 2% nanoparticles volume fraction. As shown in Figure 2, the energy efficiency of the solar collector increased substantially by 38.46% and 28.84% for CuO and Al_2O_3, respectively. Yousefi et al. (2012b) and Tyagi et al. (2009) reported similar results.

Figure 3 demonstrates the effect of output temperature on energy efficiency. Energy efficiency was calculated from Eq. (9). As can be seen, output temperature of a solar collector has substantial impact on energy efficiency. The reason for higher output temperature is the absorption of heat by the nanoparticles. As we know, specific heat is defined as the heat

required raising the temperature of a unit mass of a substance by one unit of temperature. It is clear from the definition that any substance, which has a lower specific heat, should result in increased temperature for equal heat flow.
The effect of nanoparticle volume fraction on the specific heat has been calculated in the study. This property tends to decrease when the volume fraction of nanoparticles is high. Therefore, the output temperature rises and the efficiency also increases. The specific heat of the nanofluid was determined using Eq. (3). The observed specific heats for the nanofluids are in the following order, CuO > TiO_2 > Al_2O_3 > SiO_2. However, despite their differences in specific heats, all these values are higher than water. Kamyar et al. (2012) and Sohel et al. (2013)'s observation suggested similar results.

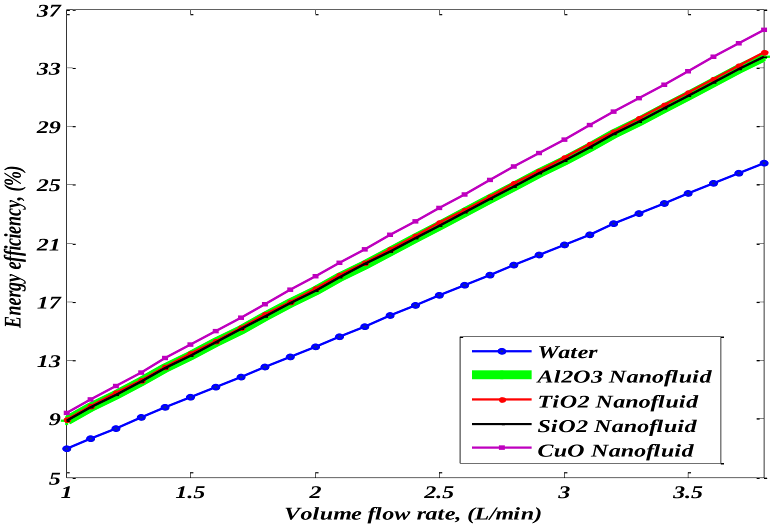

Figure 2. Effect of volume flow rate on energy efficiency

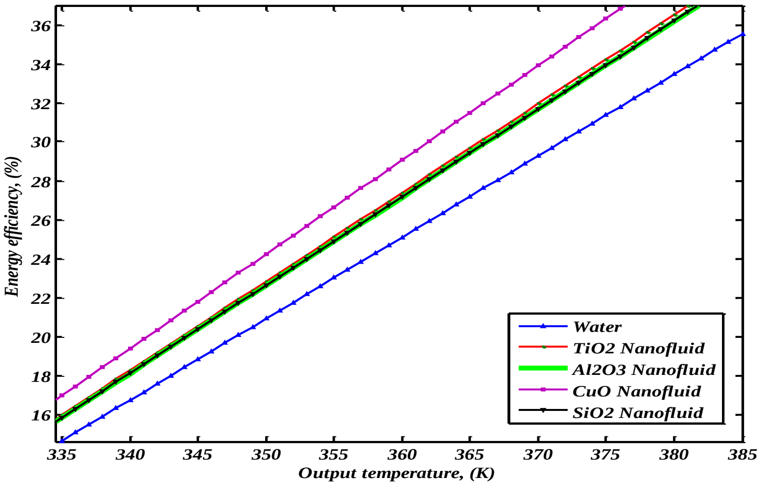

Figure 3. Effect of output temperature on collector exergy efficiency

Figure 4 represents the energy efficiency of solar collectors as a function of nanoparticles volume fraction. Eqs. (3), (4) and (9) were used in determining energy efficiency. As can be seen, efficiency of solar collectors goes up with the increment of nanoparticle percentage in the base fluid. For instance, 3.2% nanoparticles volume fraction increased the efficiency by 13.25 % for CuO nanofluid, 8.4% for TiO_2 nanofluid and 6.40% for Al_2O_3 and SiO_2.

The exergy analysis of a flat plat solar collector using different nanofluids was also carried out in the present study to evaluate the enhancement of exegetic efficiency when compared to a conventional collector. Figure 5 shows the behaviour of the exergy efficiency as a function of the volume flow rate of fluid. Exergy efficiency was calculated from Eq. (26). The analysis represents that the lowest efficiencies belonged with the collector operated by water. Therefore, a large amount of irreversibility belongs to the traditional solar collector. By using nanofluid in solar collectors as an absorbing medium, exergy efficiency can be increased. CuO nanofluid may be a good choice as an absorbing medium because of their higher exergy efficiency. Hamilton and Crosser model (1962), reported that the thermal conductivity of nanofluids is directly related to the volume fraction and the shape of the nanoparticle. It is expected that addition of nanoparticles leads to increased effective surface area for heat transfer. Additionally, the inherently higher thermal conductivity of nanoparticles will improve the thermal conductivity of the nanofluid. This could be the reason for improved exergy efficiency.

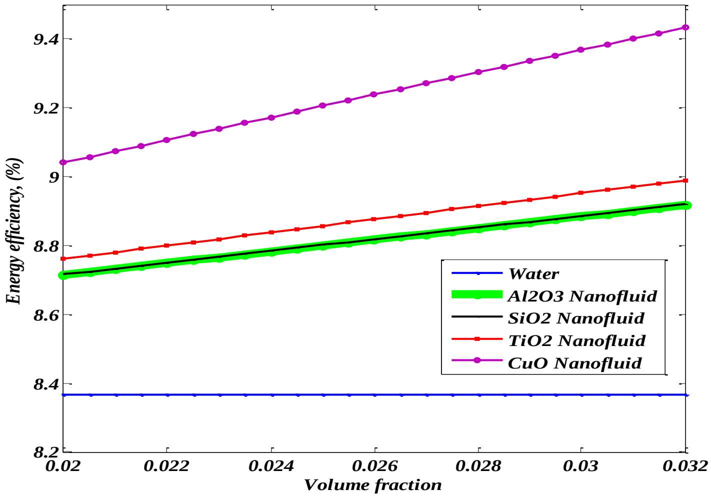

Figure 4. Effect of volume fraction on energy efficiency

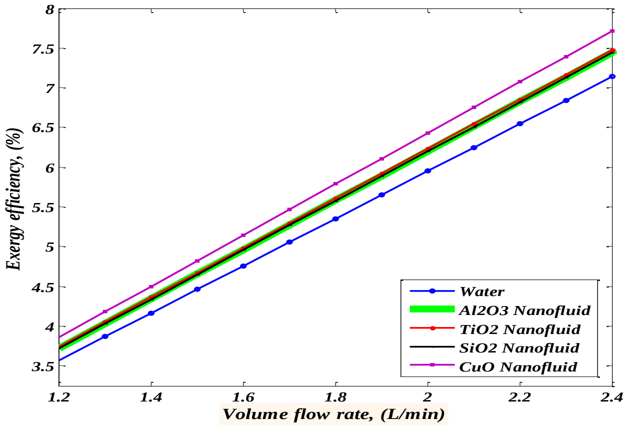

Figure 5. Effect of volume flow rate on exergy efficiency

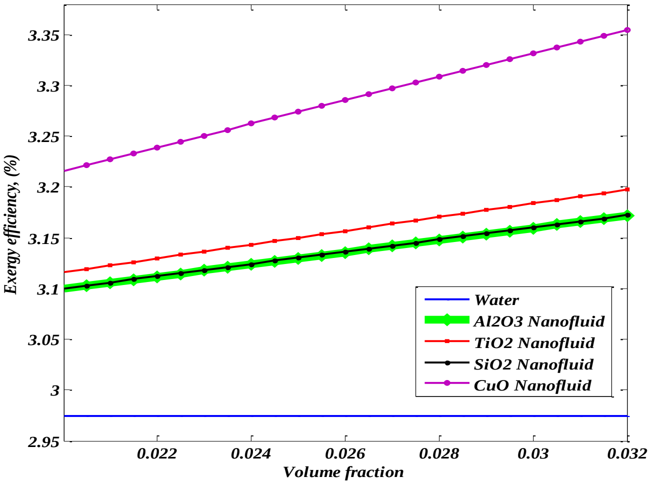

Figure 6. Effect of particle volume fraction on exergy efficiency.

Figure 6 shows the effect of nanoparticle volume fraction on exergy efficiency. For a fixed volume flow rate, the solar collector with CuO nanofluid exhibits highest exergy efficiency. For example, at 2.4 *L/s* volume flow rate, the exergy efficiency is higher than the conventional solar collector by 15.52%. Al_2O_3 and SiO_2 showed approximately equal exergy but higher than water. On the other hand, TiO_2 shows better performance compared to the base fluid, Al_2O_3 and SiO_2 nanofluids. The possible reason for this enhancement can be attributed to the following reasons: (I) the nanofluid with suspended nanoparticles increases the thermal conductivity of the mixture and (II) the convective heat transfer coefficient of the nanofluid is higher than that of the base fluid (water) at a given Reynolds number. The results are in good agreement with those obtained from Duangthongsuk and Wongwises (2009), Xuan and Li (2003) and He et al. (2007). According to the exergy efficiency equation, mass flow rate and specific heat have substantial impact on exergy efficiency when the collector absorber area is fixed.

Though output temperature has a greater effect on energy efficiency, it also enhances absorber plate temperature which may cause exergy loss. As mentioned in many articles, the main reason of exergy loss is the difference between the absorber plate temperature and the sun temperature *(T_s)*. The increase in the absorber plate temperature leads to an increase in this difference and consequently a decrease in the collector exergy efficiency. Jafarkazemi et al. (2013) reported that increasing the flow rate to approximately 0.01 kg/s leads to a considerable decrease in the absorber plate's temperature. The decrease in temperature gradient between the absorber plate and the environment results in a decrease in the overall heat loss coefficient and subsequently, an increase in the thermal efficiency of the collector. Figure 7 directly supports that statement. Table 6 lists the parameters, or in other words ,a bird's eye view of the present study. According to the results shown in Table 6, it is expected that an efficient solar collector can be produced where nanofluids would be the absorbing medium.

It is very clear from Table 6 that the CuO nanofluid provides maximum efficiency for both energy and exergy. On the other hand, it also requires the highest surface area. This may cause higher costs. But in the case of other nanofluids, it required less area than the traditional solar collector which may reduce the cost.

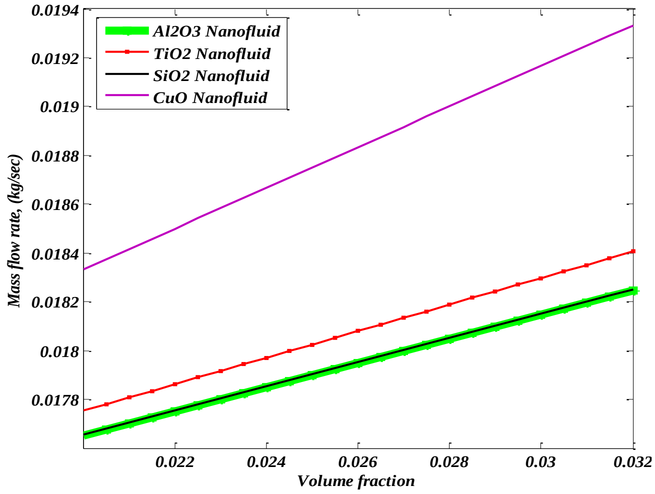

Figure 7. Effect of particle volume fraction on mass flow rate

Table 6 Analytical findings of a flat plate solar collector for different nanofluids and base fluid (equal nanoparticles volume fraction and volume flow rate).

Base fluid/ Nanofluid	**C_p, (J/kg.K)**	**A_p, (m^2)**	**Mass flow rate, enhancement, (%)**	**Energy efficiency enhancement, (%)**	**Exergy efficiency enhancement, (%)**
Water	4182.00	1.61	-	-	-
Al_2O_3	4113.82	1.51	9.47	28.84	4.25
TiO_2	4112.20	1.52	10.38	28.84	4.25
CuO	4109.38	2.24	15.95	38.46	15.52
SiO_2	4113.66	1.50	9.47	28.84	4.25

4. Conclusion

In the present study, we have focused on the benefits of using different nanofluids in a flat plate solar collector. We have studied the effects of volume flow rate, nanoparticles volume fraction, mass flow rate, density and specific heat on energy and exergy efficiency of the solar collector.

The following conclusions can be made based on the results:

a) Analytical outcomes revealed that CuO nanofluid could increase the energy and exergy efficiency of a flat plate solar collector in analogy with water as absorbing fluid, by 38.46% and 15.52%, respectively.

b) The study also remarked that the increment of volume fraction, mass flow rate and density could enhance both energy and exergy efficiency. For equal volume flow rate, mass flow rate could be increased by injecting nanoparticles in the base fluid. Specific heat is one of the most important parameters for efficiency improvement. By reducing specific heat, the efficiency of a flat plate solar collector can be enhanced, and it is very simple to reduce specific heat of a fluid by dispersing a small amount of nanoparticles.

c) From this study, we may conclude that the performance of a solar collector can be enhanced by converting the absorbing medium to nanofluid. On the basis of this study, CuO nanofluid may be a good option.

A number of assumptions have been made in this study to reduce the complexity of the analyses, such as the values of the overall heat loss coefficient, nanoparticle properties, inlet temperature, pressure drop, area of the absorber plate and heat removal factor are constant. Future study is required to address those issues.

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