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Numerical Study on the Effect of Using CuO-Water Nanofluid as a Heat Transfer Fluid on the Performance of the Parabolic Trough Solar Collector

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ABSTRACT

This research displays a numerical study of the effect of CuO-water nanofluid, as heat transfer fluid in compare with the use of pure water, on the performance of a parabolic trough solar collector. The numerical model was implemented by applying the energy balances of the heat collection element (HCE) of a parabolic solar collector in one dimension. The efficiency and the heat losses were calculated, once with pure water and once more with CuO-water nanofluid as a heat transfer fluid (HTF) at different CuO nanoparticles concentrations. The flow rate, ambient temperature, solar radiation, and wind speed were constant. The copper oxide nanoparticle concentrations used in the model were 1%, 3% and 5% of the HTF volume. Numerical results indicate that the using of CuO nanoparticles in suspension with water result in enhancing the efficiency by about 0.444%, 1.26% and 2% in average, and the heat losses have been decreased to about 4.44%, 12.6% and 20% in average at CuO concentrations of 1%, 3% and 5% respectively. The results are also showed that the performance enhancement factor (PEF) of the solar collector was improved by about 13.26% at a concentration of 5%.

1. Introduction

Applying solar technologies is one of the ways to facing global warming since they are causing no greenhouse gases emission [1,2]. Parabolic trough collector system (PTC) can be used to this purpose. It consists of a parabolic reflector which reflects incident solar radiation to the focal line where a heat transfer fluid flow inside the receiver tube to absorb the solar energy reflected by the reflector [3,4].

Many studies were executed to developing heat transfer from the receiver tube to the heat transfer fluid. One of the heat transfer limitations that has to be enhances is the low thermal conductivity of the heat transfer fluid [5-8]. Therefore, a novel nanofluid has been used where the thermal properties of the base fluid improved by adding small quantities of specified metal nanoparticles to the base fluid. These additives can cause high improvement in the thermal properties of the base fluids when dispersed in uniform and stable manner. Nanofluids are heat transfer fluids such as water, oil and ethylene glycol which contain small concentrations of

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suspending nanoparticles with average sizes below 100 nm to get a high heat transfer performance [9]. Numerous researchers are studies developing heat transfer in parabolic solar collectors by using nanofluid applications. Basbous *et al.,* [10] numerically studied the effect of Al_2O_3-Syltherm800 nanofluid on thermal performance of a parabolic trough solar collector. Based on results, the convection coefficient between the receiver and the heat transfer fluid was highly improved and the heat losses decreased of about 10%. Bellos *et al.,* [11] studied the enhancement in efficiency for commercial parabolic collector IST-PTC by using pressurized water and thermal oil with nanoparticles. Moreover, in order to make the flow more turbulence and to increase the heat transfer area, an absorber tube with a dimpled sine geometry is used and tested. The thermal efficiency and the heat transfer coefficient were improved as the final results shown. The using of nanofluids results in enhancing the efficiency of the collector by 4.25%. Ebrahimnia-Bajestan *et al.,* [12] investigated experimentally and numerically the effect of water-based TiO_2 nanofluid flowing inside a regularly heated tube on the convection heat transfer in the laminar flow. Also, the dynamic viscosity and thermal conductivity of the nanofluid were measured and displayed at different temperatures and nanoparticle concentrations. The results show that using TiO_2/water nanofluids result in an enhancement of 21% in average. The numerical results showed that the increasing in Reynolds number and nanoparticle concentration is lead to increasing in the convection heat transfer coefficient, while an opposite effect was noted with the particle size. Kasaeian *et al.,* [13] numerically examined the impact of Al_2O_3- synthetic oil nanofluid in a trough collector tube influenced by a uniform heat flux in case of mixed convection with fully developed turbulent. Different volume fractions lower than 5% of nanoparticles were used in the calculations, while a 500 K and 20 psig were taken for the operational temperature and pressure respectively. The results show that, the Nusselt numbers and the convection coefficients are directly depending on the nanoparticle concentrations. Also, the increasing in Reynolds number caused the heat transfer coefficients to be increase. Ghasemi and Ahangar [14] numerically tested the performance of a parabolic trough collector by using the Cu-Water nanofluid as heat transfer fluid. The study was evaluated and compared the thermal efficiency, temperature field and mean outlet temperatures for the conventional solar collectors and collectors with nanofluid. The results show that the performance of the solar parabolic collector increases with the increasing in the nanoparticles concentrations.

In this paper we are used a different volumetric fraction of CuO nanoparticles in water as a base fluid. A numerical study is implemented to study the effect of using CuO-water nanofluid on the performance and the efficiency of a parabolic trough solar collector.

2. Numerical Model

A numerical model implemented in EES was used in this study. An energy balance was used in the numerical model between the atmosphere and HTF, the terms in the energy balance was predicted by using all necessary equations and correlations, depending on type of the collector, condition of the HCE, ambient conditions, and optical properties. Energy balance analysis with steady situation assumption in one dimension is illustrated in Figure 1 for a HCE cross section, with the presence of the glass envelope. The incident solar radiation on the reflector is reflected to the focal point where it is converted to thermal energy and absorbed into the glass cover ($q_{5SolAbs}$) and coating layer on the surface of the absorber ($q_{3SolAbs}$). The energy absorbed by the coating layer is transferred in two way, way one by conduction across the absorber wall (q_{23cond}) and then transmitted by convection (q_{12conv}) to the HTF; way two is to glass cover by convection (q_{34conv}) and radiation (q_{34conv}). The energy is then transmitted by conduction (q_{45cond}) across the wall of the glass cover and then, the conducted energy (q_{45cond}) in addition of the absorbed energy within the glass cover ($q_{5SolAbs}$) is transferred into the atmosphere by convection (q_{56conv}) and radiation (q_{57rad}) [15,16]. The numerical analysis assumed uniform heat fluxes, temperatures and thermophysical properties around the boundary of the HCE. According to Figure 1, the conserving of energy was used for each surface of the HCE cross section to determine the equations of the energy balance analysis [17].

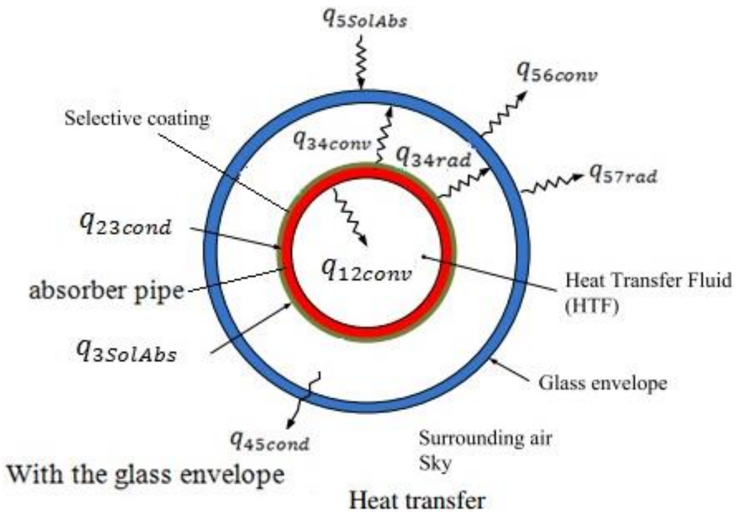

Fig. 1. Energy balance

$$q_{12conv} = q_{23cond} \tag{1}$$

$$q_{3solabs} = q_{34conv} + q_{34rad} + q_{23cond} \tag{2}$$

$$q_{34conv} + q_{34rad} = q_{45cond} \tag{3}$$

$$q_{45cond} + q_{5solabs} = q_{56conv} + q_{57rad} \tag{4}$$

$$q_{HeatLoss} = q_{56conv} + q_{57rad} \tag{5}$$

Note that the solar absorptance $q_{3SolAbs}$ and $q_{5SolAbs}$ are treated as heat flux terms.

The heat convected from the internal wall of the absorber into the HTF is evaluated from Newton's law of cooling as following [18].

$$q_{12conv} = h_1\, D_2\, \pi\, (T_2 - T_1) \tag{6}$$

With

$$h_1 = Nu_{D2} \frac{k_1}{D_2} \tag{7}$$

When the flow is turbulent or transitional (Re > 2300) the convective heat transfer is evaluated by the following Nusselt number correlation developed by Alberti *et al.,* [19].

$$Nu_{D2} = \frac{\frac{f_2}{8}(Re_{D2}-1000)Pr_1}{1+12.7\sqrt{\frac{f_2}{8}}\left(Pr_1^{2/3}-1\right)} \tag{8}$$

With

$$f_2 = (0.790 \;\; ln\, Re_{D2} - 1.64)^{-2} \tag{9}$$

The valid range of the correlation above is (0.5 < Pr_1 < 2000) and (3000 < Re_{D2} < 5000000). All properties of the fluid are calculated based on the mean temperature, T_1 of the HTF. Regular heat flux and a smooth inner surface of the absorber wall are assumed in the correlation above.

For laminar flow (Re < 2300) case of the HCE, the Nusselt number is constant. Its value was 4.36 when the flow is a pipe flow [20].

The heat conduction across the wall of the absorber is calculated by using Fourier's law of conduction across a hollow cylinder as following [21]:

$$q_{23\,cond} \; = \; 2\pi k_{23} \; (T_2 \; - \; T_3) \; / \; ln(D_3 \; / \; D_2) \tag{10}$$

The conductivity of the absorber is specified based on the material type. For this study, the absorber material was taken to be copper (400 W/m-K) [21].

The heat energy is transported from the absorber to the glass envelope by convection and radiation. When the HCE annulus is under vacuum as in this study (pressure <=1 torr), the convection heat transfer occurs by free molecular conduction [22].

$$q_{34conv} \; = \; \pi D_3 h_{34} \; (T_3 \; - \; T_4) \tag{11}$$

With

$$h_{34} = \frac{k_{std}}{\left(\frac{D_3}{2*\ln\left(\frac{D_4}{D_3}\right)} + b\lambda\left(\frac{D_3}{D_4} + 1\right) \right)} \tag{12}$$

$$b = \frac{(2-a)(9\gamma-5)}{2a(\gamma+1)} \tag{13}$$

$$\lambda = \frac{2.331E(-20)(T_{34}+273.15)}{(P_a \delta^2)} \tag{14}$$

Where T_{34} = average temperature (T_3 + T_4)/2 (°C).

Table 1 lists Heat Transfer Constants of the air in the Annulus space.

Table 1
Heat Transfer Constants of the air in the Annulus space [23]

$\boldsymbol{K_{std}}$ ***[W/m.K]***	$\boldsymbol{b}$	$\boldsymbol{\lambda}$ [cm]	$\boldsymbol{\gamma}$	$\boldsymbol{\delta}$ [cm]
0.02551	1.571	88.67	1.39	$3.53*10^{-8}$

T_{1ave} = 300 °C, Insolation = 940 W/m²

The amount of heat transferred by radiation from the receiver tube to the glass cover (q_{34rad}) is evaluated by the correlation below [24].

$$q_{34rad} = \frac{\sigma\pi D_3 ({T_3}^4 - {T_4}^4)}{(1/\varepsilon_3 + (1-\varepsilon_4) D_3/(\varepsilon_4 D_4))} \tag{15}$$

Luz Black Chrome Selective coating was used in this study. Emissivity of the Selective coating *(ε_3)* is calculated based on the temperature of the coating layer with the equation below [25]:

$$\varepsilon_3 = 0.0005333\ (T_3 + 273.15) - 0.0856 \tag{16}$$

The equation used for the conduction across the wall of the absorber is used for the conduction through the glass envelope. The thermal conductivity of the glass cover is assumed to be constant with a value of 1.04 $\frac{W}{m.K}$ (Pyrex glass) [26].

Finally, energy within the glass cover is transmitted to the environment by convection and radiation. Newton's law of cooling is used to calculate the convective energy [27].

$$q_{56conv} = h_{56}\pi D_5(T_5 - T_6) \tag{17}$$

With

$$h_{56} = \frac{k_{56}}{D_5} Nu_{D5} \tag{18}$$

The Nusselt number for forced convection is determined by using of Zhukauskas' correlation for an isothermal cylinder exposed to external forced convection flow acted in normal direction [21,28].

$$Nu_{D5} = C{Re_{D5}}^m {Pr_6}^n \left(\frac{Pr_6}{Pr_5}\right)^{1/4} \tag{19}$$

With

C	m	Re_D
0.75	0.4	1 – 40
0.51	0.5	40 – 1000
0.26	0.6	1000 – 200000
0.076	0.7	200000 – 1000000

With

n= 0.37, when Pr ≤ 10

n= 0.36, when Pr >10

The correlation above is applicable when Pr_6 is greater than 0.7 and less than 500, Re_{D5} is greater than 1 and less than 10^6. The properties of the fluid are estimated based on the temperature of the atmosphere, T_6 except for Pr_5, which is determined based on the temperature of the external surface of the glass cover. The net energy radiated from the glass cover to the sky becomes [29,30]:

$$q_{57rad} = \sigma D_5 \pi \varepsilon_5 \left({T_5}^4 - {T_7}^4\right) \tag{20}$$

This correlation assumed that the sky is a large blackbody hollow and the glass envelope is a small convex gray object in that cavity. The effective temperature of the sky is assumed to be 8°C below the ambient temperature [27,28]. So as to simplify the model, the energy absorption by the glass envelope is assumed to be a heat flux. The absorption of energy in the glass envelope is estimated with the following equation [23,24]:

$$q_{5SolAbs} = q_{si}\eta_{env}\alpha_{env} \tag{21}$$

With

$$\eta_{env} = \varepsilon_1.\varepsilon_2.\varepsilon_3.\varepsilon_4.\varepsilon_5.\varepsilon_6.\rho_{cl}.K \tag{22}$$

Table 2 listed terms used in Eq. (22), except for K. To estimate the solar irradiation term (q_{si}), in Eq. (21), the direct normal solar irradiation must be multiplied by the width of the reflector aperture. The absorptance and emissivity of the glass envelope are constants (independent of temperature) (α = 0.02, ε = 0.86) [24].

When the solar radiation incident on the collector is not normal to the aperture, we need to consider the Incident angle modifier (K) [31].

$$K = \cos\theta + 0.000884\theta - 0.00005369\,\theta^2 \tag{23}$$

Where

θ, the incidence angle, is provided in degrees.

Table 2
The estimated values of the effective optical efficiency terms [32]

ε_1 = HCE shadowing	0.974
ε_2 = Error of tracking	0.994
ε_3 = Error of geometry (alignment of mirror)	0.98
ρ_{cl} = Reflectance of clean mirror	0.935
ε_4 = Mirrors are dirt	reflectivity/ρ_{cl}
ε_5 = HCE is dirt	(ε_4 + 1)/2
ε_6 = Unaccounted	0.96

The energy absorbed by the coating layer on the surface of the absorber is assumed to be a heat flux. Then the energy absorption in the absorber is estimated by the equation [23]:

$$q_{3SolAbs} = q_{si}\eta_{abs}\alpha_{abs} \tag{24}$$

With

$$\eta_{abs} = \eta_{env}\tau_{env} \tag{25}$$

In Eq. (25), η_{env} can be estimated from Eq. (22). The transmittance of the glass cover and the absorptance of the coating layer are constants (α_{abs}= 0.94, τ_{env}= 0.935) [24].

The collector efficiency of the PTC can be calculated by [33]

$$\eta_{th} = \frac{q_{12conv}}{q_{si}} \tag{26}$$

3. Validation of the Model

The results of the simulation were compared with the experimental data of Sandia National Laboratory so as to validate the numerical model. Comparisons between our model results and experimental results of SNL measured from June 1992 to January 1993 on the AZTRAK rotating test platform at SNL are offered in Figure 2 and Figure 3 [10]. The results from SNL are used in comparisons with the predicted results of collector efficiency and heat losses from this simulation for Black Chrome selective coatings when the annulus space is under vacuum. The results of this model were in acceptable agreement with the experimental data of SNL with average error of ±2.24 % for the collector efficiency and ±7.08 [W/m^2] for the heat losses. Therefore, the HCE performance simulation established in this study can be predicting performance of the parabolic trough solar collector over a wide range of parameters with a good accuracy.

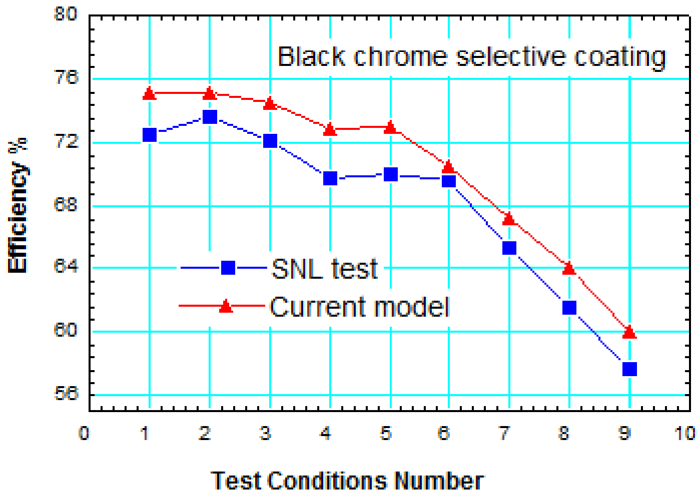

Fig. 2. Comparison of the efficiency between the current results and SNL results

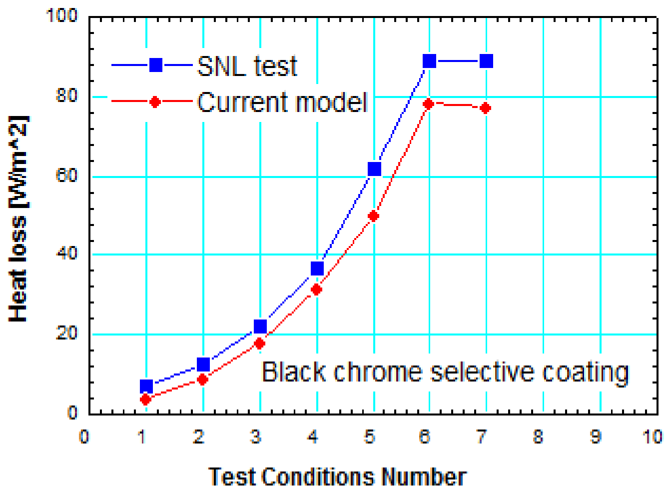

Fig. 3. Comparison of the heat losses between the current results and SNL results

4. Thermo Physical Properties of The Heat Transfer Fluid

Thermo physical properties of CuO-water were computed by using general correlations for a two phase fluid [10,11]:

$$\rho_{nf} = (1-\varphi)\rho_{bf} + \varphi\rho_{np} \tag{27}$$

$$\rho_{nf}Cp_{nf} = (1-\varphi)Cp_{bf}\rho_{bf} + \varphi Cp_{np}\rho_{np} \tag{28}$$

$$K_{nf} = \frac{K_{bf}[2K_{bf}+K_{np}-2\varphi(K_{bf}-K_{np})]}{2K_{bf}+K_{np}+\varphi(K_{bf}-K_{np})} \tag{29}$$

$$\mu_{nf} = \mu_{bf}(1+2.5\varphi+6.5\varphi^2) \tag{30}$$

The subscripts nf and np are refer to the nanofluid and nanoparticle respectively. ρ, Cp, K and μ are the density, specific heat capacity, thermal conductivity and dynamic viscosity respectively. Table 3 lists CuO thermo physical properties [12,25].

Table 3
Thermo physical properties of CuO

ρ [kg/m^3]	Cp [J/kg.K]	K [W/m.K]
6350	535.6	76.5

The properties of the base fluid and nanofluid are plotted in Figure 4 to Figure 7 based on the HTF average temperature and nanoparticles concentration. It is perceived from the figures that the using of CuO nanoparticles suspended in the water caused the thermo physical properties of the suspension to be improved. We can see that density, viscosity and thermal conductivity increase with the increasing in the concentration of nanoparticles, while the heat capacity decreases.

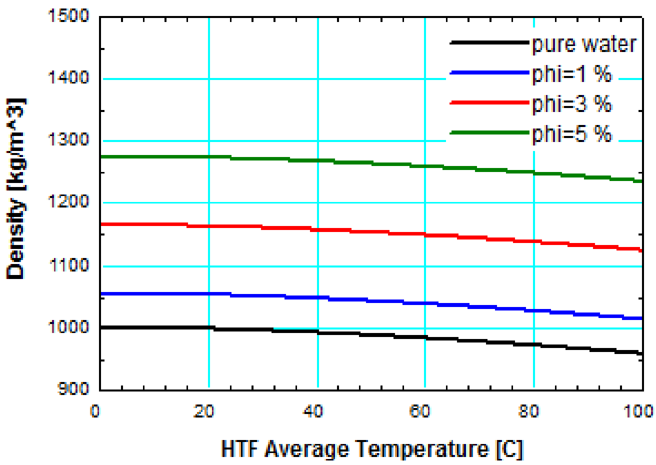

Fig. 4. Density of the CuO-water nanofluid at various nanoparticles concentrations

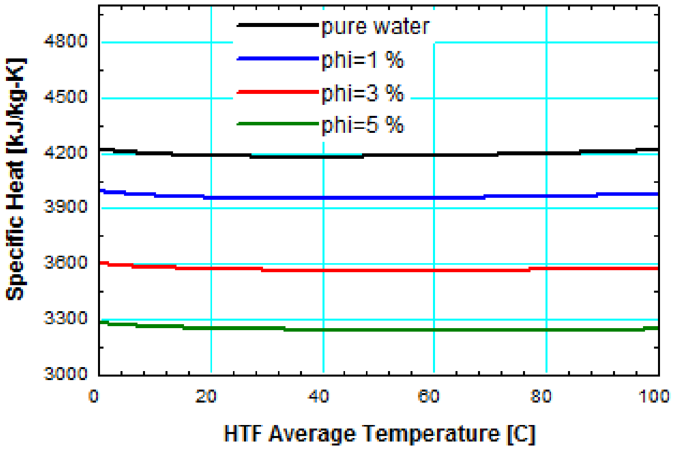

Fig. 5. Specific heat of the CuO-water nanofluid at various nanoparticles concentrations

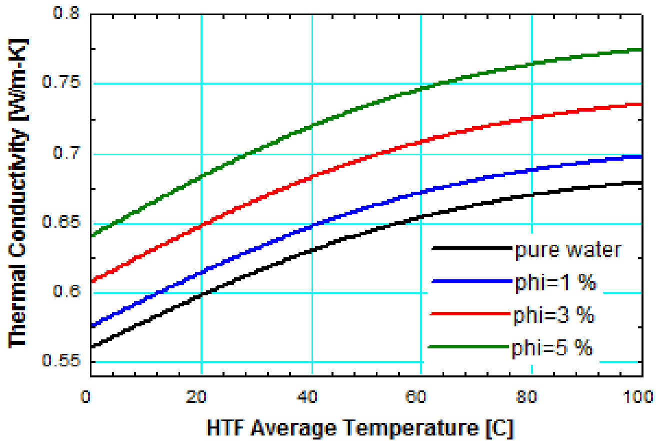

Fig. 6. Thermal conductivity of the CuO-water nanofluid at various nanoparticles concentrations

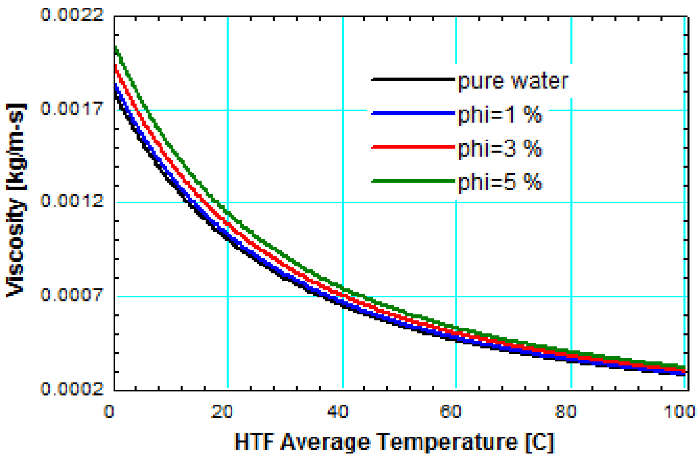

Fig. 7. Viscosity of the CuO-water nanofluid at various nanoparticles concentrations

5. Performance Enhancement Factor (PEF)

The performance enhancement factor is a measurement of the enhancement in the performance which is used to measure the performance enhancement of the solar collector resulted from the use of the CuO-water as a heat transfer fluid. The PEF is equal to the heat transfer coefficient ratio divided by the friction factor ratio as following [34]:

$$\mathrm{PEF} = \frac{\frac{\mathrm{h_{nf}}}{\mathrm{h_{bf}}}}{\left(\frac{\mathrm{f_{nf}}}{\mathrm{f_{bf}}}\right)^{1/3}} \tag{31}$$

The PEF was calculated to show the effect of the nanofluid on the performance and the results were illustrated in Figure 12.

6. Results and Discussion

This numerical work was executed to examine the performance of the PTSC by using CuO-water nanofluid as a heat transfer fluid. The dimensions and parameters used in this numerical model are enumerated in Table 4.

Table 4
List of parameters and PTC dimensions used in the numerical model

Parameters	Values	PTC dimensions	Values
I_b	700 W/m^2	Collector type	LS-2
T_{amb}=T_6	15 C	W	4.8235 m
v_6	1.5 m/s	D_2	0.066 m
θ	0 degree	D_3	0.070 m
$\dot{m}$	0.033 kg/s	D_4	0.109 m
Reflectivity	0.94	D_5	0.115 m

The efficiency and heat losses of the PTC were tested at three values of the concentrations of CuO nanoparticles (1%, 3%, and 5%). Figure 8 illustrates the efficiency of the base fl uid and nanofluid. The numerical results show that the suspension of CuO-water enhances the thermal performance of PTC. The enhancement in efficiency was about 0.444% in average at nanoparticle concentration of 1%, while the value above increased to about 1.26% in average at concentration of 3% and when we increased the concentration to 5%, the increment in efficiency was about 2%.

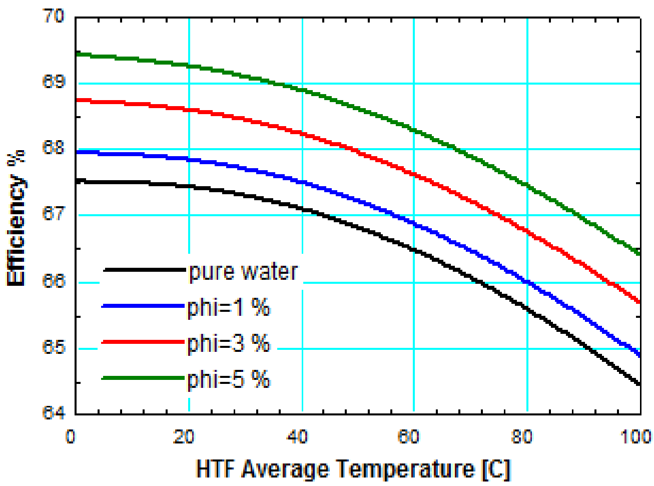

Fig. 8. Efficiency of PTC at different nanoparticle concentrations

As for heat losses, we can observe from Figure 9 that thermal losses decrease with increasing in the concentration of nanoparticles. The percentage of decrease in heat losses is about 4.44%, 12.6% and 20% in average when the concentration is equal to 1%, 3% and 5% respectively.

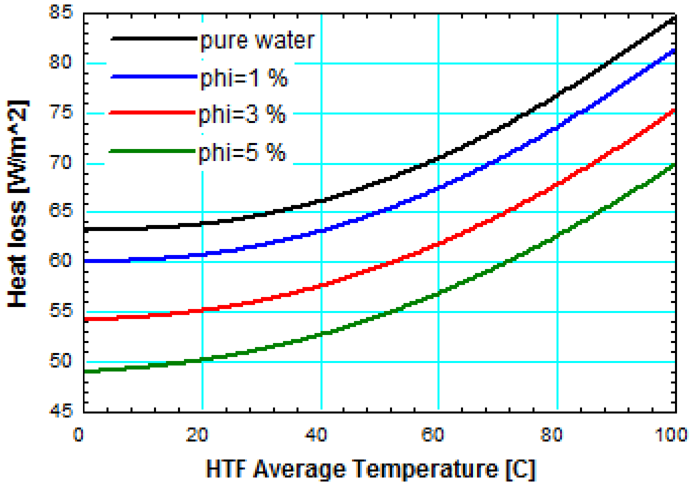

Fig. 9. Heat losses of PTC at different nanoparticle concentrations

The influence of nanoparticles on the heat transfer coefficient of the working fluid is illustrated in Figure 10. It is observed that the using of CuO-water suspension cause in enhancing heat transfer coefficient and that the coefficient of heat transfer is increase with increasing in the nanoparticles concentration. Also, the friction factor between the working fluid and the internal surface of the absorber is increase when the nanoparticles concentration increases which is justified the improvement in the heat transfer coefficient, see Figure 11. The maximum average enhancement in the heat transfer coefficient and the friction factor was 15.38% and 5.72% respectively, at nanoparticles concentration of 5%.

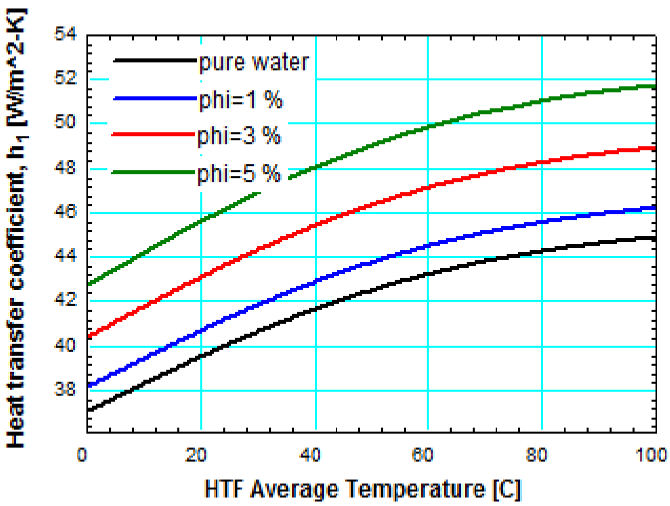

Fig. 10. Heat transfer coefficient at different nanoparticle concentrations

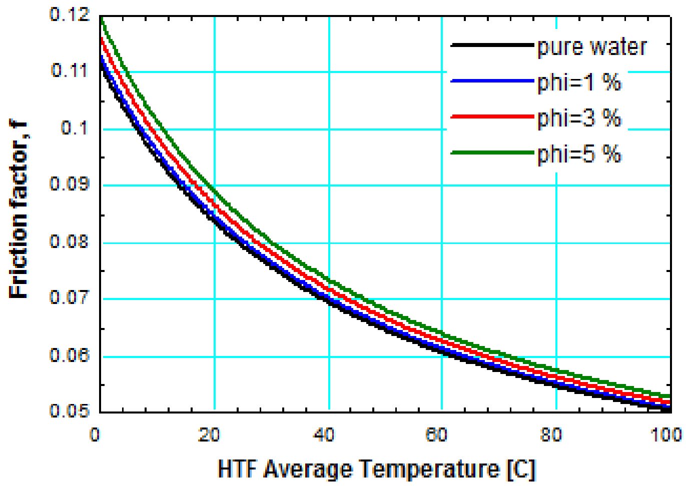

Fig. 11. Friction factor at different nanoparticle concentrations

Figure 12 show that the PEF is proportional with the nanoparticle concentration, where it is increased about 13.26% at concentration of 5%, and this is improved that the using of the nanoparticle suspension caused in enhancing the performance.

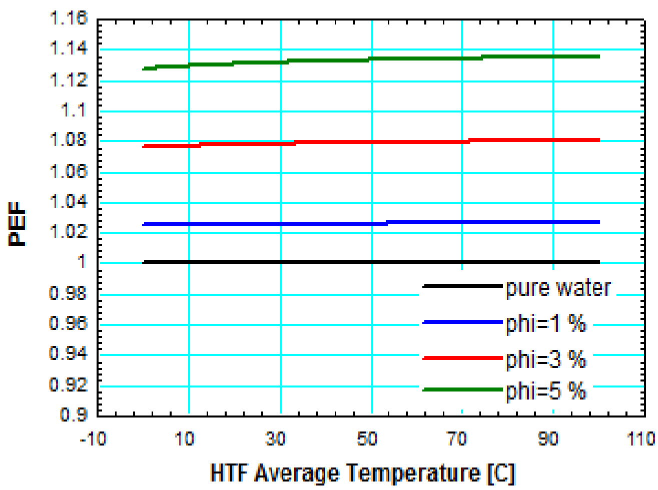

Fig. 12. PEF at different nanoparticle concentrations

7. Conclusion

This study implemented to study the enhancement of thermal performance of the PTSC by using CuO-water suspension as a heat transfer fluid. It was found that the use of the nanoparticles leads to the following results:

(i) Thermo physical properties of CuO-water suspension were greater than those of the pure fluid.
(ii) The efficiency of the PTC increases with the increasing in the nanoparticles concentration.
(iii) Using of CuO nanoparticles tend to decrease heat losses to about 20% at concentration of 5%.
(iv) Using of nanofluids as heat transfer fluid in the parabolic trough collector systems result in enhancing thermal performance.

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