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Thermal performance evaluation of various nanofluids with non-uniform heating for parabolic trough collectors

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ABSTRACT

Nanoparticles when used even in relatively low concentrations, can significantly alter the thermal properties of a base working fluid, thereby substantially enhancing the thermal performance of a power generation system. In the present study, numerical simulations were performed for a solar collector to test the effectiveness of six non-metallic nanoparticles, namely aluminum oxide (Al₂O₃), cerium oxide (CeO₂), copper oxide (CuO), ferric oxide (Fe₂O₃), titanium dioxide (TiO₂) and Silicon dioxide (SiO₂). These nanoparticles were dispersed individually in three different base working fluids; therminol VP-1 (at 400 K), water (at 400 K) and molten salt (at 600 K) to form different nanofluids. Each of these nanofluids were then examined with three different volume fractions (2%, 4% and 6%) in addition to the pure base working fluid case for a range of Reynolds number (Re=10⁴-10⁵). For the simulation the Monte Carlo Ray Tracing (MCRT) model was used to represent the non-uniform heat flux around the absorber tube of the Parabolic Trough Collector (PTC). The results show that the enhancement of the thermal and the hydraulic performances depended upon the combination of the nanoparticles and the base working fluid. Silicon dioxide, SiO₂, however, was found to be the most efficient nanoparticle regardless of the choice of the base working fluid for all the tested volume fractions. For example, for typical operating conditions for SiO₂ with a volume fraction (VF) of 6%, the average Nusselt number of the water-based mixture was enhanced by 32.4%, with a thermal efficiency improvement of 5.11% and performance evaluation criterion (PEC) of 1.313. On the other hand, for a molten salt-based SiO₂ mixture, the average Nusselt number was improved by 21.36% and thermal efficiency by 9.92% with a PEC of 1.155. Finally, the corresponding improvements with therminol VP-1-base fluid were 15.6%, 9.18% and 1.21 for the average Nusselt number, thermal efficiency and PEC respectively.

1. Introduction

Concentrated Solar Power Technologies (CSPT) allow harvesting clean, renewable and sustainable energy from sources that are...
### Nomenclature

#### Latin symbols
- $A_a$: Collector aperture area, m$^2$
- $C_{p,nf}$: Nanofluid specific heat capacity, J/kg.K
- $C_{p,f}$: Fluid specific heat capacity, J/kg.K
- $C_{p,s}$: Solid specific heat capacity, J/kg.K
- $D_i$: Inner diameter of the absorber, m
- $D_o$: Outer diameter of the absorber, m
- $D_{oe}$: Outer surface diameter of the glass envelope, m
- $E_a$: Available solar exergy, W
- $E_u$: Useful output exergy, W
- $f$: Flow friction factor
- $f_o$: Friction factor of pure fluid
- $f_L$: The focal line, m
- $f_μ$: Damping function
- $G_b$: Solar beam radiation, W/m$^2$
- $h_{conv}$: Ambient convection heat transfer coefficient, W/m$^2$. K
- $k$: Fluid thermal conductivity, W/m.K
- $k_f$: Fluid thermal conductivity, W/m.K
- $k_{nf}$: Nanofluid thermal conductivity, W/m.K
- $k_s$: Solid thermal conductivity, W/m.K
- $L$: Solar tube length, m
- $\dot{m}$: Mass flow rate, kg/s
- $Nu$: Nusselt number
- $Nu_o$: Nusselt number of the pure fluid,
- $Pr$: Prandtl number,
- $Q_{loss}$: Thermal losses, W
- $Q_a$: Available solar energy, W
- $Q_u$: Useful thermal energy, W
- $Re$: Reynolds number,
- $T^+$: Mean non-dimensional temperature
- $T_{am}$: Ambient temperature, K
- $T_{ave}$: Average fluid temperatures of inlet and outlet, K
- $T_f$: Mean fluid temperature, K
- $T_{in}$: Inlet fluid temperature, K
- $T_o$: Outer surface temperature of the solar receiver, K
- $T_{oe}$: Outer surface temperature of the glass envelope, K
- $T_{out}$: Outlet fluid temperature, K
- $T_{sky}$: Sky temperature, K
- $T_{sun}$: Sun temperature, K
- $T_w$: Inner wall temperature, K
- $U$: Bulk fluid velocity, m/s
- $U_f$: Flow friction velocity, m/s
- $\dot{V}$: Volumetric flow rate, m$^3$/s
- $V_f$: Base fluid volume, m$^3$
- $V_s$: Solid particle volume, m$^3$
- $V_w$: Wind speed, m/s
- $w_a$: Aperture width, m
- $W_p$: Pumping power, W
- $y$: Distance from the solid wall to the first cell, m
- $y^+$: Mean non-dimensional distance

#### Greek symbols
- $ε_o$: Emissivity of the solar receiver
- $ε_{oe}$: Emissivity of the glass envelope
- $η_{el}$: Power-block electrical efficiency
- $η_{ex}$: Exergy efficiency
- $η_{overall}$: Overall collector efficiency
inexhaustible and are naturally replenished. Amongst the CSPT’s, the Parabolic trough collectors (PTC) technology is one of the most widely used solar thermal technology with a maximum working temperature of 398 °C, which is dictated by the performance of the thermal oils (such as Therminol VP-1, Syltherm 800, Therminol 66, Sandotherm, Dowtherm, etc.) currently being used in the PTC systems around the world [1]. At very high temperatures, the thermal performance of these oils gradually deteriorates at which point environmental toxicity and flammability also become a serious issue. Thus, there is still a need for alternative working fluids which are not only environmentally friendly but can also operate at higher-temperatures, resulting in higher thermodynamic efficiencies, such as molten salt mixtures (60% NaNO$_3$-40% KNO$_3$) that can withstand temperatures of up to 550 °C, see Ref. [2].

In the recent past a number of investigations have been carried out to assess the effect of different technologies on the thermal performance of heat transfer fluids used in the PTC’s mainly focusing on water and thermal oils for a short review see Abed and Afgan [3]. The main objective so far has been to enhance the heat transfer from the solar receiver to the heat transfer fluid, whilst at the same time decreasing the inner and outer absorber temperatures. By doing so, the overall thermal losses reduce, thereby increasing the overall collector efficiency. Furthermore, the reduction in the absorber temperature helps in alleviating the receiver deformation problems by decreasing the effective thermal stresses as reported by Ref. [4].

Nanofluids are composed of a mixture of very tiny particles possessing thermal properties different to those of the base working fluid. Such a mixture generally acts as a thermally more efficient working fluid with a higher Prandtl number. Nanofluids thus improve the thermal conductivity, increase the dynamic viscosity and lowers the specific heat capacity leading to an overall improvement of the resulting fluid performance compared to pure working fluid (see Ref. [5] for details). There is a considerable scope for further research in nanofluid technology to extend its use to a wide range of industrial and engineering applications. For instance Ref. [6] reported that using TiO$_2$/water nanofluid in a semi-circular micro-channel enhanced the heat transfer capability and decreased the entropy
generation. For the same application [7] found that TiO$_2$/water nanofluid improved the Nusselt number without considerably increasing the friction factor. [8,9] deduced that using SiO$_2$/water nanofluid enhanced the heat transfer in both a corrugated duct and a curved duct of a square cross-section. For a channel flow with a backward-facing step, [10] succeeded in enhancing the heat transfer performance by using Al$_2$O$_3$ water nanofluid. Moreover, [11] reported that the use of a CuO/water nanofluid in a micro-scale backward-facing step, led up to 30% heat transfer enhancement. [12,13] also reported similar findings that using Al$_2$O$_3$-water, SiO$_2$-water, and CuO–water nanofluid enhanced different parameters including energy, exergy, thermal and hydraulic performances for small scale pin-finned heat sinks.

The nanofluid technology is also applicable in PTC systems, in which [14] reported that using 5% volume fraction of Al$_2$O$_3$–Synthetic oil enhanced the heat transfer coefficient by 11.5% for a single-phase model and 36% for a two-phase model. [15] reported that using 8% volume fraction of Al$_2$O$_3$–Syltherm 800 improved the overall collector efficiency by 7.6%. However, using the same nanofluid but with 2% volume fraction, [16] observed an increase of only 4.25% thermal efficiency. On the other hand, [17] reported a 10% improvement in thermal efficiency by using 4% volume fraction of Al$_2$O$_3$-Syltherm 800 nanofluid with a two-phase model and a uniform wall temperature over the solar receiver. Thus, nanofluids when added in small volume fractions help in improving the thermal efficiency but at the cost of greatly increasing the pumping power requirement; [18] showed that by using 5% volume fraction of Al$_2$O$_3$-Synthetic oil, the effect of nanofluids was very small compared to the increase in the pumping power requirement. Similarly, [19] showed that by using 5.5% volume fraction of CuO, SiO$_2$, TiO$_2$ and Al$_2$O$_3$ mixed with Therminol 55 led to improvements in the overall exergy efficiency by 3, 6, 9 and 11% respectively. [20] showed up to 32% improvement in the coefficient of heat transfer and 20–30% decrease in the entropy generation using 6% volume fraction of Cu-TherminolVP-1. Later, [21] found that using 4% volume fraction of Al$_2$O$_3$ and CuO dispersed in Syltherm 800 resulted in an improvement of thermal efficiency by 1.13 and 1.26% respectively. Another set of nanoparticles called Al$_2$O$_3$, Ag and Cu, dispersed in TherminolVP-1, led to the enhancement of the overall thermal efficiency by about 13.9% as reported by Ref. [22]. However, [23] reported that Ag dispersed in TherminolVP-1 provided better thermal performance than Cu. In the same year using Syltherm 800, Allouhi et al. [24] recorded an enhancement of the thermal energy by 1.46, 1.25 and 1.4% using 5% volume fraction of Al$_2$O$_3$, CuO and TiO$_2$ respectively. The authors also reported that the maximum increase in the exergy efficiency was 9.05%, for 3% volume fraction of CuO. The ref. [25] on the other hand, used only 6% volume fraction CuO but with two different base working fluids; Syltherm800 and Molten salt. The authors reported an increase in the Nusselt number by 13% with CuO-molten salt and 40% with CuO-Syltherm 800. [5] examined new types of nanoparticles called single-walled carbon nanotubes (SWCNT) dispersed in TherminolVP-1. Their results showed that 2.5% volume fraction of SWCNT improved the thermal efficiency by 4.4% and the heat transfer performance by 234%. [26] also examined a nanoparticle from the same family of carbon nanotubes called multi-walled carbon nanotube (MWCNT) mixed with Ethylene glycol and reported that the heat transfer coefficient was increased by 15% using 6% volume fraction MWCNT. [27] compared the thermal performance of Cu and CNT dispersed in Gallium (Ga) with different volume fractions of the nanoparticles. Their results showed a 34.5% augmentation in the heat transfer coefficient using Cu–Ga and 45.2% using CNT-Ga. [28] also reported similar findings that the thermal and exergy efficiencies increased by 2.76 and 2.6% respectively using 5% volume fraction of CuO-Syltherm 800.

[29] reported that 4% volume fraction of Fe$_3$O$_4$–Therminol 66 in the presence of a magnetic field enhanced the thermal efficiency by 4%. [30] found that using porous materials in annular space and 3% volume fraction of Al$_2$O$_3$ with Synthetic oil improved the heat transfer coefficient and thermal efficiency by 20 and 14% respectively. This enhancement resulted from a decrease in the absorber wall temperature which consequently led to the reduction of the radiation heat losses. [31] examined the effect of CuO-water nanofluid combined with the use of a metal foam inside the solar receiver. They deduced that by changing the volumetric flow rate from 20 to 100 Lph (liter per hour), the thermal efficiency was increased from 55.65 to 79.29%, when using 0.1% volume fraction of CuO-water and metal foam. [32] examined the effect of 4% volume fraction of Cu-Therminol 800 nanofluid on three different solar collector types; vacuumed-annular tube receiver, air-annular tube receiver and the bare receiver (without glass cover). They reported that the maximum thermal efficiency enhancement was achieved with the bare tube receiver configuration.

The utilization of hybrid nanofluids is also a promising approach for even stronger enhancements in the thermal performance. [33] showed that using 1.5% volume fraction of Al$_2$O$_3$ and TiO$_2$ dispersed in Oil enhanced the thermal efficiency by more than two times compared to the use of 3% volume fraction of Al$_2$O$_3$-Oil, or TiO$_2$-Oil alone. [34] examined three hybrid nanofluids; Ag–ZnO, Ag–TiO$_2$ and Ag–MgO mixed with Syltherm 800 with various volume fractions. Results revealed that the thermal efficiency was enhanced by 14% using 4% volume fraction of Ag–MgO hybrid nanofluid. The effect of nanofluids on the heat transfer and pressure drop can be jointly represented by a parameter called the performance evaluation criterion (PEC). In Ref. [35]; the authors numerically showed that using 6% volume fraction of TiO$_2$, Al$_2$O$_3$, CuO and Cu dispersed in water can improve the PEC by 1.214, 1.2, 1.18 and 1.155 respectively.

The aforementioned literature clearly shows that most of the numerical studies in the past have focused on utilizing thermal oils, whereas only a few studies exist for water and molten salts as the base working fluid. The literature clearly shows the thermal, hydraulic and thermodynamic performances including thermal efficiency, heat transfer coefficient, pressure drop, entropy generation and performance evaluation criteria compared to the performance of the base fluids. Most of these studies, however, have only considered one base working fluid and varied either the nanoparticles or just their concentration to test their effects. The current study thus differs from all these previous ones as it investigates the effect of nanofluids with different base-fluids (water, thermal oil and molten salts), which is still not fully covered in the literature. In the present paper six non-metallic nanoparticles namely aluminum oxide (Al$_2$O$_3$), cerium oxide (CeO$_2$), copper oxide (CuO), ferric oxide (Fe$_3$O$_4$), titanium dioxide (TiO$_2$) and Silicon dioxide (SiO$_2$) are examined by dispersing them individually in three different base fluids; therminol VP-1 (at 400 K), water (at 400 K) and molten salt (at 600 K). The choice of two different working temperatures 400 K (for therminol VP-1 and water) and 600 K for molten salt is based on the broader spectrum of the operating conditions of the wide range of PTC applications. It is typically suggested that the
inlet temperature of working fluid in most of the PTC applications is approximately 400 K. However, some PTC applications require higher inlet fluid temperature for the parabolic trough collectors. For these systems the working fluid temperature is commonly set at around 600 K.

In the present study the concentration ratio of the nanoparticles was varied from 2 to 6% by volume fraction for each nanofluid. Each configuration was then tested for a range of Reynolds (Re) numbers, 10000, 30000, 50000, 70000, and 100000. All cases were assumed to be steady state, fully three-dimensional and incompressible. The main objective of this study is to produce a parametric comparison between the hydraulic and thermal performances amongst the examined nanofluids and to determine the optimum operational conditions for every heat transfer fluid under realistic heat flux boundary conditions (i.e. non-uniform heat flux around the external surface of the absorber receiver in the circumferential direction). To the best of the authors’ knowledge, such assessment of the performance of six different nanoparticles dispersed in three different base fluids with the non-uniform heat flux distributions has not been reported anywhere in the literature.

2. Methodology and background of PTCs

Fig. 1a shows a typical parabolic trough collector. The configuration consists of a solar receiver positioned on the focal line of a parabolic trough, which is made up of mirrors shaped like a parabola. The solar receiver is usually made out of metal and is externally coated by a special material. Depending upon the configuration it may or may not be encapsulated by a glass envelope. The internal
space of the solar receiver is filled with an appropriate heat transfer working fluid. The thermal process in a PTC system depends on the thermal balance between this heat transfer fluid and its surrounding. The parabolic trough receives the solar energy from the sun and reflects it onto the solar receiver. Inside the solar receiver, the absorber tube absorbs part of the reflected energy and raises the temperature of the heat transfer fluid. The remaining part of the solar energy is returned back to the internal wall of the glass envelope via convection and radiation. The thermal energy is then transferred from the internal to the external wall of the glass envelope via conduction. The energy is then lost to the surroundings by radiation and to the ambient by convection through the glass envelope as shown in the thermal resistance model presented in Fig. 1b.

2.1. Geometric and thermal formulations of the PTC system

For the conventional PTC, the following parabolic formula represents the geometrical profile [36].

\[ x^2 = 4yf_l \]  

where \( x \) and \( y \) are the Cartesian co-ordinates of the parabolic surface and \( f_l \) represents the focal line, which can be represented as

\[ f_l = \frac{w_a}{4 \tan \left( \frac{\phi_r}{2} \right)} \]  

here the parameter \( w_a \) represents the aperture width of the trough collector and \( \phi_r \) is the rim angle.

The useful thermal energy (\( Q_u \)) collected by the working fluid is calculated from Ref. [37] as

\[ Q_u = \dot{m} C_p (T_{out} - T_{in}) \]  

where the parameter \( \dot{m} \) is the mass flow rate, \( T_{out} \) and \( T_{in} \) are the outlet and inlet fluid temperatures respectively. For the system the available solar energy (\( Q_s \)) is calculated as

\[ Q_s = A_a G_b \]  

where \( A_a \) represents the collector aperture area and \( G_b \) represents the solar beam radiation which for the current simulations is fixed at 1000 W/m². The overall collector efficiency can be calculated by considering the pumping power effect as per [38].

\[ \eta_{overall} = \frac{Q_u}{W_p/\eta_{el} Q_s} \]  

Here the parameter \( \eta_{el} \) represents the power block electrical efficiency which is taken as 32.7%. The variable \( W_p \) represents the pumping power which is given as

\[ W_p = \Delta P \dot{V} \]  

In the above equation, the parameter \( \dot{V} \) represents the volumetric flow rate through the solar receiver and \( \Delta P \) represents the flow pressure drop which is determined from the following expression as given in [39].

\[ \Delta P = \frac{f}{D_i} \frac{\rho U^2}{2} \]  

\[ f = \frac{8\tau}{D_i \rho U^2} \]  

here \( U \) is the bulk velocity of the fluid inside the absorber tube, \( \rho \) is the fluid density, \( D_i \) the inner diameter of the absorber, \( \tau \) the wall shear stress, \( L \) the solar tube length and \( f \) the friction factor. Now the useful output exergy \( E_u \) of the solar receiver can be calculated from Ref. [40] as

\[ E_u = Q_s - m C_p T_{am} \ln \left( \frac{T_{out}}{T_{in}} \right) - m T_{am} \frac{\Delta P}{\rho T_f} \]  

In the above formulation \( T_f \) represents the mean temperature of the working fluid and \( T_{am} \) is the ambient temperature. The last term in equation (9) is neglected, assuming that the ratio of the pressure drop to density and temperature is very small. According to Ref. [41]; the available solar exergy can be calculated as

\[ E_s = Q_s \left[ 1 - \frac{4}{3} \left( \frac{T_{am}}{T_{sun}} \right) + \frac{1}{3} \left( \frac{T_{am}}{T_{sun}} \right)^4 \right] \]  

here \( T_{sun} \) represents the sun temperature in its external layer which is known to be at 5800 K. The exergy efficiency for the solar receiver is then given as
\[ \eta_{oe} = \frac{E_s}{E} \]  

(11)  

For the thermal losses from the glass envelope to the environment, the following energy balance given by Ref. [42] is used

\[ Q_{loss} = \pi D_{oe} h_{oe}(T_{oe} - T_{am}) + \pi D_{oe} \sigma \varepsilon_{oe}(T_{oe}^4 - T_{sky}^4) \]  

(12)  

In the above formulation the subscripts oe represents the outer surface of the glass envelope and \( \sigma \) refers to the external wall of the solar receiver whereas \( T_{am} \) is the ambient temperature, \( \sigma \) is the Stefan-Boltzmann constant \((5.67 \times 10^{-8} \text{W/m}^2 \text{K}^4)\), \( \varepsilon_{oe} \) is the emissivity of the glass envelope and \( h_{oe} \) is the ambient convection heat transfer coefficient which can be expressed as per [42] as

\[ h_{oe} = 4V_w^{0.55} D_{oe}^{-0.42} \]  

(13)  

The variables \( D_{oe} \) and \( V_w \) are the outer diameter of the glass envelope and the wind speed respectively. Finally, the parameter \( T_{sky} \) represents the sky temperature which can be calculated using [43] as

\[ T_{sky} = 0.0552 T_{am}^{1.5} \]  

(14)  

It is worth noting that in the case of analyzing only the bare receiver (i.e. without the glass envelope, such as the present study), the thermal losses would take place directly from the solar receiver to the sky by radiation and to the ambient by convection and thus equation (12) can be rewritten as follows:

\[ Q_{loss} = \pi D_{oe} h_{oe}(T_{oe} - T_{am}) + \pi D_{oe} \sigma \varepsilon_{oe}(T_{oe}^4 - T_{am}^4) \]  

(15)  

The external surface emissivity of the solar receiver depends on the selected coating and mean external surface temperature \( T_{oe} \) of the solar receiver which is given by Ref. [44] as

\[ \varepsilon_{oe} = 0.062 + 2 \times 10^{-7} \times T_{oe}^2 \]  

(16)  

It should be noted here that temperature \( T_{oe} \) in the above equation is in \( ^\circC \). 

For the bare receiver, the convective heat transfer coefficient of the ambient reads as

\[ h_{oe} = 4V_w^{0.55} D_{oe}^{-0.42} \]  

(17)  

The heat transfer coefficient \( (h) \) inside the solar receiver can thus be calculated from the useful thermal energy as

\[ h = \frac{q}{(T_w - T_{ave})} \]  

(18)  

Here, the parameters \( q \) represents the heat flux, \( T_w \) is the inner wall temperature of the absorber tube and \( T_{ave} \) is the average temperature of the inlet and the outlet. For the calculations the Nusselt number \( (Nu) \), Prandtl number \( (Pr) \) and Reynolds number \( (Re) \) are calculated as

\[ Nu = \frac{hD_w}{k} \]  

(19)  

\[ Re = \frac{\rho U D_w}{\mu} \]  

(20)  

\[ Pr = \frac{\mu C_p}{k} \]  

(21)  

where, \( U, C_p, k, \mu \) and \( \rho \) are the fluid parameters, velocity, specific heat capacity, thermal conductivity, dynamic viscosity and density respectively. 

In solar receiver geometries when \( Re \geq 4000 \), the flow is fully turbulent and the corresponding \( Nu \) number can be predicted using the empirical correlation proposed by Ref. [45] as

\[ Nu = \begin{cases} \left( \frac{1}{3} \right) (Re - 1000) Pr & \text{for } 0.5 \leq Pr \leq 2000 \\ 1 + 12.7 \left( \frac{3}{2} \right) (Pr^{3/2} - 1) & \text{for } 3 \times 10^3 < Re < 5 \times 10^6 \end{cases} \]  

(22)  

This equation has been used to validate the numerical simulations performed in the current study. However, it must be noted here that the friction parameter \( f \) is strongly dependent upon the chosen \( Re \) number. For the comparison of the friction factor of the numerical simulations, the empirical correlation proposed by Ref. [46] as shown below is used.

\[ f = (0.75 \ln Re^{-1.54})^{-2} \text{ for } 3000 < Re < 5 \times 10^6 \]  

(23)
2.2 Fluid and material properties

The optical and geometrical characteristics along with the environment parameters used in the current study are clarified in Table 1. For simplicity, the glass envelope is entirely removed in the current study. Removing the glass envelop does indeed make the model simpler but glass while transparent to the incoming solar radiation, is opaque to the outgoing radiation from the collector. This has a strong effect on the thermal equilibrium, also known as the greenhouse effect. As a result, the numerical results will somewhat under-estimate the heat absorbed by the collector, [37]. However, since the objective of this study is not to look for absolute values but rather assess the relative performance of nanofluids, the removal of the glass envelope suffices the current modelling requirements.

321H stainless steel material is used for the solar absorber due to its strength and reduction in bending problem as recommended by Ref. [47]. All geometric and material properties of the model are given in Table 1.

2.2.1. Thermal properties of the working nanofluids

The first step of the nanofluids investigation is to calculate the thermal properties of the examined nanofluids for the single-phase approach which has been employed in this study. This is achieved by using information from the existing literature and the introduction of some assumptions. The nanofluid density (ρ_{nf}) is calculated depending on the classical form of the heterogeneous mixture proposed by Ref. [48]. Whereas, the specific heat capacity (C_{p,nf}) is derived from the thermal equilibrium between the solid particles and its surrounding base fluid as proposed by Refs. [49]. Nevertheless, several models are in use for determining the nanofluid viscosity and thermal conductivity. The model proposed by Ref. [50] is used for the dynamic viscosity (μ_{nf}) which has been derived based on the experimental data of [51] where they measured the viscosity of Al$_2$O$_3$-water. For the thermal conductivity (k_{nf}) the model proposed by Ref. [52] has been employed in the current study. This model has no limitations on the volume fraction and is based on the homogenous spherical solid-fluid mixture. The aforementioned models are presented below.

\[ \rho_{nf} = \rho_s \phi + \rho_f (1 - \phi) \]  
\[ C_{p,nf} = \frac{1}{\rho_{nf}} \left[ \rho_s C_{p,s} \phi + C_{p,f} \rho_f (1 - \phi) \right] \]  
\[ \mu_{nf} = \mu_s \left( 1 + 7.3 \phi + 123 \phi^2 \right) \]  
\[ k_{nf} = 0.25 \left( 3 \phi - 1 \right) k_s + (2 - 3 \phi) k_f + \sqrt{\Delta} \]

where: \[ \Delta = \left( 3 \phi - 1 \right) k_s + (2 - 3 \phi) k_f \]^2 + 8k_s k_f.

Here the volume fraction (\phi) represents the ratio of solid particle volume (V_s), divided by the total volume, V_s + V_f as [53].

\[ \phi = \frac{V_s}{V_s + V_f} \]

In the models above, the subscript, nf represents nanofluids, s refers to the nanoparticle and f refers to the base fluid. The thermal properties of examined nanoparticles and the base fluids (water, therminol VP-1 and molten salt) are tabulated in Tables 2 and 3 respectively.

### Table 2
Thermal properties of particles examined in the current study.

<table>
<thead>
<tr>
<th>Type</th>
<th>( p ) (kg/m³)</th>
<th>( C_p ) (J/kg.k)</th>
<th>( k ) (w/m.k)</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Al$_2$O$_3$</td>
<td>3970</td>
<td>940</td>
<td>32.4</td>
<td>[37]</td>
</tr>
<tr>
<td>CuO</td>
<td>6320</td>
<td>532</td>
<td>77</td>
<td>[54]</td>
</tr>
<tr>
<td>TiO$_2$</td>
<td>4230</td>
<td>692</td>
<td>8.4</td>
<td>[24]</td>
</tr>
<tr>
<td>Fe$_3$O$_4$</td>
<td>5180</td>
<td>670</td>
<td>6.9</td>
<td>[37]</td>
</tr>
<tr>
<td>SiO$_2$</td>
<td>2200</td>
<td>765</td>
<td>1.4</td>
<td>[55]</td>
</tr>
<tr>
<td>CeO$_2$</td>
<td>6757</td>
<td>392.48</td>
<td>5.86</td>
<td>[37]</td>
</tr>
</tbody>
</table>
3. Numerical modelling

For the present study, open-source finite volume CFD software called Open Field Operation and Manipulation (OpenFOAM) is used as it is highly parallelizable and allows flexibility to easily modify boundary conditions, thermo-physical properties and numerical schemes. Furthermore, the OpenFOAM library named conjugated heat transfer multi region simple foam (chtMultiRegionSimpleFoam) is very well suited for the current work and was thus utilized. For the spatial discretization of the diffusive transport of all variables, the second order accurate central differencing scheme was used. The discretization of the convective transport was handled using the van Leer scheme [57] for the turbulent properties, while for the mean flow variables the second-order accurate upwind scheme was employed. Finally, the PISO algorithm was selected for the pressure velocity coupling, [58]. The boundary conditions used in the present study include:

i. A non-uniform heat flux distribution was applied over the external wall of the solar receiver. This boundary condition is a result of the Monte Carlo Ray Tracing (MCRT) model which was proposed by Ref. [17] as shown in Fig. 2. This boundary condition was implemented in OpenFOAM and completely described in Ref. [59].

ii. At the inlet, uniform velocity and temperature profiles were implemented with mean turbulent kinetic energy and dissipation rates which were calculated based on the prescribed Re number and the thermal properties of the nanofluids. The pressure was assumed to be zero gradient at the inlet.

iii. The pressure was set to be zero at the outlet with a zero gradient boundary condition for velocity, temperature and all turbulence parameters as it was assumed to be fully developed.

iv. For the solid walls, the no-slip boundary condition was used, with turbulent kinetic energy set to zero and a very large specific turbulence dissipation rate as suggested by the [60].

\[ \omega = \frac{60 \mu}{0.075 \rho y_1} \]  

(29)

Here, the \( \mu \) and \( \rho \) are the dynamic viscosity and density of the working fluids/nanofluids and \( y_1 \) is the distance from the solid wall to the first cell.

3.1. Mesh independence verification

The solar receiver’s length to diameter ratio is 60.606 and its inner and outer diameters are 66 and 70 mm respectively. For the simulations three different meshes (coarse mesh 0.8 million, medium mesh 1.8 million and fine mesh 2.4 million cells) were examined at the considered flow rates. The meshes were refined near walls for low Reynolds treatment with wall non-dimensional number \( y^+ \approx 1 \). A cross-section view of the medium mesh is shown in Fig. 3a. The cases selected for the mesh independence study were for water as the base working fluid with an inlet temperature (\( T_{in} \)) of 320 K. From Fig. 3b it can be seen, that the Nu number is not predicted well when using the coarse mesh for the higher Re cases where the flow is fully turbulent. However, both the medium and the fine mesh

<table>
<thead>
<tr>
<th>Fluid</th>
<th>T (K)</th>
<th>( \mu ) (Pa.s)</th>
<th>( \rho ) (kg/m(^3))</th>
<th>( C_p ) (J/kg.k)</th>
<th>k (w/m.k)</th>
<th>Pr</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>400</td>
<td>0.000217</td>
<td>937.21</td>
<td>4256</td>
<td>0.688</td>
<td>1.34</td>
<td>[39]</td>
</tr>
<tr>
<td>Therminol VP-1</td>
<td>400</td>
<td>0.000732</td>
<td>975.8</td>
<td>1850.5</td>
<td>0.1243</td>
<td>10.89</td>
<td>[20]</td>
</tr>
<tr>
<td>Molten salt</td>
<td>600</td>
<td>0.002713</td>
<td>1882.12</td>
<td>1499.22</td>
<td>0.5051</td>
<td>8.05</td>
<td>[56]</td>
</tr>
</tbody>
</table>

Table 3
Thermal properties of base fluids examined in the current study.
capture the thermal physics precisely over the entire tested Re number range. Thus, the medium mesh was chosen for all further computations.

3.2. Numerical model validation

For the present work, several steps were taken to validate the numerical results using two low-Reynolds number turbulence models; LS k-epsilon model [61] and $k-\omega$ SST model [60]. The output temperature of the working fluid has been validated through comparisons with the experimental data of [44]. These experiments were performed on a parabolic trough collector without glass envelope using Syltherm 800 oil as heat transfer fluid (HTF). The geometric and thermal boundary conditions are almost the same as the simulations as listed in Table 4.

For the temperature predictions Table 4 shows that the k-omega SST model performed somewhat better than the LS k-epsilon model, showing a mean deviation of only 0.67%. For the thermal efficiency comparisons, the $k-\omega$ SST turbulence model showed a better agreement with the experimental data, especially for the lower inlet temperature cases as shown in Fig. 4.

The numerical simulations were also validated through the average $Nu$ number for pure water as the HTF with the inlet temperature of 320 K and Prandtl ($Pr$) number of 3.77 against the experimental correlation of [45]. It can be observed from Fig. 5a that the k-omega SST model predictions are in very good agreement with the experimental data for all the tested Re numbers. The wall friction coefficients obtained via the $k-\omega$ SST model also showed a good agreement for the entire Re range with the experimental data (see Fig. 5b). The LS k-epsilon is basically an extended and modified version of the standard $k-\epsilon$ model which utilizes a damping function, $f_\mu$, to correct the diffusion and production through modifying the turbulent eddy viscosity. On the other hand, the $k-\omega$ SST model is a hybrid model which switches between the standard $k-\omega$ model near the wall regions to $k-\epsilon$ model in regions away from the walls.

The subsequent step was to validate two types of nanofluids with different volume fractions, both with $k-\omega$ SST model. The experimental data used for the comparisons were taken from Ref. [48]; where the authors examined the effects of nanofluids in a circular duct with a uniform heat flux distribution using water as a base fluid at 300 K inlet temperature. The nanofluid used for the experiments was $\gamma$-Al$_2$O$_3$-water with volume fractions of 1.34 and 2.78% (see Fig. 6a). The friction factor for the numerical simulations was also validated using TiO$_2$-water nanofluid at 2% volume fraction with the experimental measurements of [62]. Once again it can be seen that the results are in excellent agreement with the measurements for the entire Re number range, as shown in Fig. 6b. Based on the findings of all the validation cases, it was concluded that the $k-\omega$ SST model suffices as the turbulence model and that all the

### Table 4

<table>
<thead>
<tr>
<th>V (L/min)</th>
<th>$Pr$</th>
<th>$G_b$ (W/m$^2$)</th>
<th>$T_in$ (°C)</th>
<th>$T_out$ (°C)</th>
<th>$k$ omega SST $T_out$ (°C)</th>
<th>Deviation (%)</th>
<th>$LS$ k epsilon $T_out$ (°C)</th>
<th>Deviation (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>48.4</td>
<td>27.92</td>
<td>801.3</td>
<td>151.7</td>
<td>166.2</td>
<td>168.337</td>
<td>−1.286</td>
<td>171.347</td>
<td>−3.097</td>
</tr>
<tr>
<td>49.8</td>
<td>19.93</td>
<td>881.6</td>
<td>198.2</td>
<td>215.5</td>
<td>213.294</td>
<td>1.024</td>
<td>215.425</td>
<td>0.035</td>
</tr>
<tr>
<td>51.1</td>
<td>12.02</td>
<td>920.5</td>
<td>301</td>
<td>315.4</td>
<td>313.809</td>
<td>0.124</td>
<td>314.833</td>
<td>−0.201</td>
</tr>
<tr>
<td>55.6</td>
<td>11.37</td>
<td>929.4</td>
<td>314.8</td>
<td>324.8</td>
<td>325.340</td>
<td>−0.166</td>
<td>326.375</td>
<td>−0.485</td>
</tr>
<tr>
<td>55.8</td>
<td>8.95</td>
<td>940.4</td>
<td>395</td>
<td>395.12</td>
<td>395.466</td>
<td>−0.113</td>
<td>396.435</td>
<td>−0.363</td>
</tr>
<tr>
<td>50.9</td>
<td>14.85</td>
<td>935.7</td>
<td>252.1</td>
<td>268</td>
<td>266.868</td>
<td>0.422</td>
<td>267.360</td>
<td>0.239</td>
</tr>
<tr>
<td>39.8</td>
<td>42.7</td>
<td>817.5</td>
<td>101</td>
<td>120.8</td>
<td>127.859</td>
<td>−5.844</td>
<td>135.211</td>
<td>−12.757</td>
</tr>
<tr>
<td>50.1</td>
<td>19.32</td>
<td>854.5</td>
<td>203.1</td>
<td>219.2</td>
<td>217.292</td>
<td>0.870</td>
<td>219.364</td>
<td>−0.075</td>
</tr>
<tr>
<td>50</td>
<td>19.29</td>
<td>867.6</td>
<td>203.4</td>
<td>219.6</td>
<td>217.825</td>
<td>0.808</td>
<td>219.210</td>
<td>0.178</td>
</tr>
<tr>
<td>48.2</td>
<td>42.9</td>
<td>922</td>
<td>100.8</td>
<td>121.1</td>
<td>125.407</td>
<td>−3.558</td>
<td>131.945</td>
<td>−8.955</td>
</tr>
<tr>
<td>51.6</td>
<td>9.55</td>
<td>927.6</td>
<td>354.4</td>
<td>367.8</td>
<td>366.507</td>
<td>0.352</td>
<td>367.572</td>
<td>0.062</td>
</tr>
</tbody>
</table>

Aver. Deviations (%): −0.670 −2.311

Fig. 3. (a) Cross-section mesh presentation and (b) Mesh independence study of the current work.
Fig. 4. Validation of the present study with the thermal efficiency data of [44].

(a) Average Nusselt number of the present study compared with correlations proposed by [46] and [45]. (b) Friction factor compared with correlation proposed by [46].

Fig. 6. Validation the present model with experimental data of nanofluids (a) Nu number of [63] and (b) friction factor of [62].
adopted numerical procedures are adequate for the modelling requirements of the current cases.

4. Results and discussions

In this section, different variables regarding to the thermal performance and flow characteristics of considered nanofluids are presented. These parameters are heat transfer performance, pressure drop, performance evaluation criteria, thermal losses, overall collector efficiency and mean temperature profiles respectively.

4.1. Heat transfer performance

Adding nanoparticles to the base working fluid changes the thermal properties of the resulting nanofluid and consequently enhances the convective heat transfer performance. Fig. 7 illustrates the profiles of the $Nu$ number on the solar receiver as a function of $Re$ number for different nanofluids and their varying volume fractions. It is clearly seen that the enhancement in the heat transfer performance increases gradually with increasing nanoparticle volume fractions and $Re$ number, in line with the main findings of preceding

Fig. 7. Heat transfer performance of all nanoparticles at different Reynolds numbers for base fluids of water (W), molten salt (MS) and Therminol VP-1 (TO).
investigations [20,62]. The thermal enhancement with increasing particle volume fractions is attributed to the improved thermal properties of base working fluids. The Nu number continues to increase with increasing volume fractions since more thermal energy is transferred from the solar receiver to the working nanofluid, which leads to an increase in the useful thermal energy. It is further noted from these figures that the introduction of all the working nanofluids results in a similar thermal trend. The highest Nu numbers, however, have been achieved with the therminol VP-1 base fluid, followed by molten salt and then water. It is prudent to mention here that higher Pr numbers were obtained for all the examined nanofluids compared with base fluids as the tested nanoparticles had a lower specific heat capacity, a larger dynamic viscosity and a higher thermal conductivity than the base fluid.

It can also be noted from Fig. 7 that even though the use of therminol VP-1 results in the highest Nu numbers, in terms of relative improvement clearly water is the better choice. This is expected as the thermal conductivity of water is significantly higher than that of molten salt and therminol VP-1 by 1.36 and 5.5 times respectively. It is also noted here that in terms of the performance CeO$_2$ nanoparticles were always the least effective whereas SiO$_2$ nanoparticles produced the strongest thermal enhancement regardless of the choice of the base fluid. The relative effects of the nanoparticles tested were found to be the same for most of the base fluids, the only exception being the therminol VP-1 base fluid, where TiO$_2$ was found to be less effective than Fe$_2$O$_3$ and Al$_2$O$_3$ (see Table 5 for detailed comparisons). Due to the fact that water has the lowest Pr number, theNu number for water based nanofluids had the lowest values. Nevertheless, given that Nu ≡ h.D/k, a lower value for Nu, does not necessarily mean a lower value for the heat flux coefficient h.

It is very important to note that the effective Pr number of resulting nanofluids has a vital role on the thermal performance or Nu number. Since the inclusion of nanoparticles increases the Pr number of the nanofluid, it will also increase the Nu values. For the nanoparticles under consideration, the resulting nanofluids from SiO$_2$ with all base fluids produces larger Pr numbers at all volume fractions which in turn enhance the Nu number more than any other nanofluid, see Table 6 for more details. It is argued that the SiO$_2$ nanoparticle has the smallest thermal conductivity but with a moderate specific heat capacity compared to all other tested nanoparticles (as shown in Table 2). This result in much larger Pr numbers compared to the other nanoparticles regardless of the choice of the base working fluid. On the other hand, both molten salt and Therminol VP-1 have higher boiling point temperatures than water and hence show larger average values of the Nu number.

Based on the achieved results of considered nanofluids, three Nu number correlations are proposed in Table 7.

### 4.2. Receiver hydraulic performance

Fig. 8 depicts the effect of varying volume fractions of various nanoparticles on the specific pressure drop as a function of Re number based on the thermal properties of nanofluids at an inlet temperature of 400 K for water and therminol VP-1 and 600 K for molten salt. The process of adding nanoparticles to the base fluids not only affects the fluid thermal conductivity but also increases the resulting fluid density and dynamic viscosity. This in turn leads to a reduction of hydraulic performance; due to friction, the pressure

<table>
<thead>
<tr>
<th>Table 5</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Relative thermal performance improvement with 6% volume fraction and Re = 100000.</strong></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>---</td>
</tr>
<tr>
<td><strong>Nanoparticle</strong></td>
</tr>
<tr>
<td>Water</td>
</tr>
<tr>
<td>Therminol VP-1</td>
</tr>
<tr>
<td>Molten salt</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 6</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Pr numbers of all nanofluids examined in the current study at 6% of volume fraction.</strong></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>---</td>
</tr>
<tr>
<td><strong>Base fluid</strong></td>
</tr>
<tr>
<td>Water</td>
</tr>
<tr>
<td>Molten salt</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 7</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Nu number correlations for all nanofluids examined in the current study.</strong></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>---</td>
</tr>
<tr>
<td><strong>Base fluid</strong></td>
</tr>
<tr>
<td>Water</td>
</tr>
<tr>
<td>Molten salt</td>
</tr>
<tr>
<td>Therminol VP-1</td>
</tr>
</tbody>
</table>
Fig. 8. Specific pressure drop of all nanoparticles at different Reynolds numbers for all base fluids.
drop increases and so does the pumping power requirement with the addition of nanoparticles. It can be observed from Fig. 8 that the specific pressure drop inside the solar receiver increases with increasing volume fraction of the nanoparticles which is in line with the previous literature [27,30]. However, the hydraulic behavior of nanofluids is relatively different from each other and is also dependent on the density and viscosity of the base working fluid. For example from Table 8, we can see that the largest pressure drop is with SiO$_2$ nanoparticles and the lowest pressure drop is with CeO$_2$ regardless of the base working fluid. In general, the specific pressure drop is higher when the base working fluid is molten salt regardless of the nanoparticle. Overall, one can conclude that the specific pressure drop is dependent upon both the density and viscosity of the base working fluid and the volume fraction of the nanoparticles.

Fig. 8. (continued).

<table>
<thead>
<tr>
<th>Nanoparticle Base fluid</th>
<th>Co$_2$O$_3$</th>
<th>CuO</th>
<th>Fe$_2$O$_3$</th>
<th>Al$_2$O$_3$</th>
<th>TiO$_2$</th>
<th>SiO$_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>163.8%</td>
<td>191.2%</td>
<td>186%</td>
<td>203%</td>
<td>199%</td>
<td>230%</td>
</tr>
<tr>
<td>Therminol VP-1</td>
<td>163%</td>
<td>168.5%</td>
<td>196%</td>
<td>200%</td>
<td>197%</td>
<td>231%</td>
</tr>
<tr>
<td>Molten salt</td>
<td>197%</td>
<td>212%</td>
<td>222%</td>
<td>233.5%</td>
<td>231.5%</td>
<td>252.7%</td>
</tr>
</tbody>
</table>
It is important to note here that using high concentration ratio of nanoparticles would lead to particle agglomerating inside the absorber tube resulting in a higher friction factor and larger pumping power requirements. Therefore, the volume fraction should be optimized and selected to provide an effective heat transfer performance with a reasonably acceptable pressure drop.

4.3. Performance evaluation criterion

One of the most important parameters that can be used to assess the thermal and frictional effects of any active or passive new technology (such as nanofluids, swirl generators etc.) is the performance evaluation criterion (PEC) given by Ref. [64]. This PEC represents the evaluation of heat transfer enhancement and pumping power augmentations of the considered technique compared to a base case and reads as

\[
PEC = \left( \frac{\text{Nu}}{\text{Nu}_0} \right)^{1/3} \left( \frac{f}{f_0} \right)
\]

Where \( \text{Nu} \) and \( f \) represent the Nusselt number and friction factor of the nanofluids respectively. Whereas \( \text{Nu}_0 \) and \( f_0 \) represent the Nusselt number and friction factor of the base working fluid without nanoparticles. All \( \text{Nu}, \text{Nu}_0, f \) and \( f_0 \) are based on the same \( Re \) number. Any value of the PEC over 1 means there are some thermal and hydraulic enhancements in the flow. Fig. 9 summarizes the PEC values for all examined nanofluids with 6% volume fraction, at a \( Re \) of 100000. It shows a PEC value of over 1 with the highest value of 1.313 being that of SiO\(_2\) mixed with water as the base working fluid. However, the PEC is reduced to 1.21 for therminol VP-1 and finally to 1.155 for using molten salt. The nanoparticles of CeO\(_2\) and CuO, on the other hand, showed the lowest PEC values for all the base working fluids.

4.4. Thermal losses

One of the most important purposes of using nanofluids is to reduce the external surface temperature of the solar receiver, which greatly minimizes both the modes of thermal losses; convection and radiation. This is because the changes in fluid properties which result from the introduction of nanoparticles, lead to higher values of the wall heat flux coefficients. Thus, the cooling nanofluid is able to further reduce the receiver temperature and attenuate any circumferential variations. A more uniform collector temperature leads to lower thermal losses and higher efficiencies.

The examined non-metallic nanofluids were also compared to assess their effect on thermal losses against \( Re \) number for water, therminol VP-1 and molten salts as shown in Fig. 10. It can be noted from this figure that the nanofluid cases show lower thermal losses compared to their respective base fluid cases. In fact, the greater the concentration of the nanoparticles, the lower the loss. The obvious reason behind this is the uniform temperature which is achieved via the use of the nanoparticles. Thus, one can conclude that the uniformity of the thermal gradient is the key in reducing the losses. Indeed, this inference is further substantiated when one observes the relationship between the \( Re \) number and the thermal loss profiles of any of the tested cases; at higher \( Re \) numbers, due to higher levels of turbulence, more mixing takes place which leads to higher levels of thermal uniformity and hence reduction of losses. This behavior is evident in Fig. 10.

One should also note here, that there are variations in the thermal capability of each type of nanofluids and that the same nanoparticles have also different levels of enhancement when mixed with different base working fluids. It is observed from Fig. 10, that the maximum reduction in the thermal losses were recorded when therminol VP-1 was used as the base fluid followed by water and molten salts, respectively.

![PEC Chart]

Fig. 9. Performance evaluation criterion of all examined nanofluids, TO: Therminol VP-1 and MS: molten salt.
At the lowest Reynolds number ($Re = 10000$) when the volume fraction was increased from 0 to 6% the reductions in the specific thermal losses were 9.8, 10.7, 11.65, 11.65, 11.9, and 12% for CeO$_2$-TO, CuO-TO, Fe$_2$O$_3$-TO, TiO$_2$-TO, Al$_2$O$_3$-TO, and SiO$_2$-TO respectively. Moreover, the levels of reduction in thermal losses from the same nanoparticles change when these are used with another base working fluid. For molten salts (MS) with the increasing volume fraction (0–6%), the specific thermal loss reductions were 1.125, 1.26, 1.35, 1.46, 1.38, and 1.53% respectively for CeO$_2$-MS, CuO-MS, Fe$_2$O$_3$-MS, Al$_2$O$_3$-MS, TiO$_2$-MS and SiO$_2$-MS. For water as the base fluid a similar trend was observed.

4.5. Overall collector efficiency

The overall efficiency of the PTC using all respective nanofluids is assessed by taking the influence of pumping power requirements into account, as indicted in the Methodology section. Figs. 11 and 12 present the variation of the overall collector efficiency due to the effect of loading various particle volume fractions and also different $Re$ numbers for therminol VP-1 (TO) and molten salt (MS). The overall efficiency starts to increase with the addition of nanoparticles and the highest efficiency is recorded for SiO$_2$ nanoparticles for both therminol VP-1 and molten salt. This increase in efficiency is a result of several factors, such as increase in the thermal performance, reduction in thermal losses and enhancement of the useful heat gain. Furthermore, Fig. 13, presenting the inner temperature of
the absorber tube using nanofluids with therminol VP-1 as a base fluid, confirms that the addition of nanofluids lowers the absorber temperature, thereby reducing thermal losses.

It can be further noted from Figs. 11 and 12, that gain in efficiency is higher at the lower \( Re \) numbers. The reason behind this is that at higher \( Re \) numbers, the pumping requirement increases due to an increase in the fluid velocity and turbulence. This is confirmed when one revisits the heat transfer performance (shown in Fig. 7) in conjunction with the pressure drop profiles (shown in Fig. 8). Here it is observed that by increasing the nanoparticles percentage, the heat transfer performance increases but so does the pressure drop. Thus, raising the power required for pumping the fluid and consequently an overall less gain in the efficiency. In other words, the heat transfer improvement is not enough to compensate for the increase in the power requirements.

The maximum enhancements in the overall collector efficiencies using therminol VP-1 with 6% volume fraction of nanoparticles at
Re number of 30000 were 9.18% for SiO$_2$-TO followed by 7.5% for Al$_2$O$_3$-TO and 6.4% for TiO$_2$-TO. On the other hand, the minimum contribution of 3.29% was obtained by CeO$_2$-TO. For the same conditions (Re number of 30000 and 6% volume fraction of nanoparticles), for molten salts, the maximum improvement in the collector efficiencies were found to be 9.92% for SiO$_2$-MS, 7% for TiO$_2$-MS and 6.8% for Al$_2$O$_3$-MS. A complete list of gain in efficiencies with all nanoparticles using water as a base fluid is shown in Table 9.

It can clearly be seen that by adding different volume fraction of various nanoparticles, one obtains an improvement in the overall collector efficiency depending upon their respective thermal properties. A maximum enhancement of 5.11% in the overall collector efficiency was achieved by SiO$_2$–W whereas the minimum improvement (2.98%) was obtained by CeO$_2$–W.

Fig. 12. Effect of nanofluids on the overall collector efficiency using molten salt (MS) base fluid.
4.6. Thermal exergy efficiency

Another important aspect of the present study is to evaluate the effect of nanoparticles on the exergy efficiency. The exergy efficiency is an important parameter that should be taken into account since it represents the largest possible useful work extracted from the solar receiver. Fig. 14 presents the variation of the exergy efficiency due to the effect of various nanoparticles when mixed with molten salt (MS) and therminol VP-1 (TO). It is observed that the exergy efficiency increases with an increase in the volume fraction of the nanoparticles. Much like in the previous cases, however, this increase becomes less pronounced as the Re numbers is increased. In fact, the highest gain is at the lowest Re number of 30,000, regardless of the nanoparticles and the base working fluid. The increase in
the exergy efficiency with an increase in the volume fractions of the nanoparticle is due to several factors; mainly, increase in useful thermal exergy output from the solar receiver, reduction in thermal losses and enhancement of the useful heat gain.

For therminol VP-1 as the base fluid and the \( \text{Re} \) number of 30000, the maximum enhancement in the exergy efficiency was observed to be 9.02\% for SiO\(_2\) (with 6\% volume fraction), followed by 7.36\% for Al\(_2\)O\(_3\)-TO (6\% volume fraction) and 6.24\% for TiO\(_2\)-TO (6\% volume fraction). At the same conditions (6\% volume fraction and \( \text{Re} \) number of 30000) for molten salt as a base fluid, the exergy efficiency increase for various nanofluids is 10.08\% for SiO\(_2\)-MS, 7.035\% for TiO\(_2\)-MS and 6.96\% for Al\(_2\)O\(_3\)-MS. It should be noted here that improvement in the exergy efficiency with water as the base working fluid showed small improvements compared with other previous base fluids as presented in Table 10.

### Table 9
Overall collector efficiency (\%) of different nanofluids using water as a base fluid.

<table>
<thead>
<tr>
<th>Nanofluid</th>
<th>0%</th>
<th>2%</th>
<th>4%</th>
<th>6%</th>
<th>Enhancement (%) at ( \phi = 6% )</th>
</tr>
</thead>
<tbody>
<tr>
<td>SiO(_2)</td>
<td>35.18</td>
<td>36.22</td>
<td>36.57</td>
<td>36.98</td>
<td>5.11</td>
</tr>
<tr>
<td>TiO(_2)</td>
<td>35.18</td>
<td>36.08</td>
<td>36.28</td>
<td>36.53</td>
<td>3.83</td>
</tr>
<tr>
<td>Al(_2)O(_3)</td>
<td>35.18</td>
<td>36.08</td>
<td>36.27</td>
<td>36.51</td>
<td>3.78</td>
</tr>
<tr>
<td>Fe(_2)O(_3)</td>
<td>35.18</td>
<td>36.05</td>
<td>36.22</td>
<td>36.45</td>
<td>3.61</td>
</tr>
<tr>
<td>CuO</td>
<td>35.18</td>
<td>35.99</td>
<td>36.11</td>
<td>36.29</td>
<td>3.15</td>
</tr>
<tr>
<td>CeO(_2)</td>
<td>35.18</td>
<td>35.97</td>
<td>36.07</td>
<td>36.23</td>
<td>2.98</td>
</tr>
</tbody>
</table>

![Fig. 14. Effect of nanofluids on the exergy efficiency with molten salt (MS) and therminol (TO) base fluids at different Reynolds (Re) numbers.](image)

### Table 10
Thermal exergy efficiency (\%) of different nanofluids using water as a base fluid.

<table>
<thead>
<tr>
<th>Nanofluid</th>
<th>0%</th>
<th>2%</th>
<th>4%</th>
<th>6%</th>
<th>Enhancement (%) at ( \phi = 6% )</th>
</tr>
</thead>
<tbody>
<tr>
<td>SiO(_2)</td>
<td>25.51</td>
<td>26.27</td>
<td>26.51</td>
<td>26.81</td>
<td>5.09</td>
</tr>
<tr>
<td>TiO(_2)</td>
<td>25.51</td>
<td>26.15</td>
<td>26.31</td>
<td>26.48</td>
<td>3.80</td>
</tr>
<tr>
<td>Al(_2)O(_3)</td>
<td>25.51</td>
<td>26.16</td>
<td>26.30</td>
<td>26.47</td>
<td>3.76</td>
</tr>
<tr>
<td>Fe(_2)O(_3)</td>
<td>25.51</td>
<td>26.14</td>
<td>26.27</td>
<td>26.42</td>
<td>3.56</td>
</tr>
<tr>
<td>CuO</td>
<td>25.51</td>
<td>26.10</td>
<td>26.18</td>
<td>26.31</td>
<td>3.13</td>
</tr>
<tr>
<td>CeO(_2)</td>
<td>25.51</td>
<td>26.09</td>
<td>26.15</td>
<td>26.27</td>
<td>2.97</td>
</tr>
</tbody>
</table>
4.7. Mean temperature profiles

The effect of nanofluids on the mean non-dimensional temperature ($T^+$) was also considered in the current study. This is calculated from Ref. [66] as

$$T^+ = \frac{T_w - T}{T_f} \quad (31)$$

where the parameter $T_w$ is the mean temperature of the inner wall, $T$ the nanofluid temperature profile starting from the inner wall to the tube centre and $T_f$ the friction temperature which can be determined as

![Non-dimensional temperature profiles](image1)

![Non-dimensional temperature profiles](image2)

Fig. 15. Non-dimensional temperature profiles ($\langle \cdot \rangle$) against the non-dimensional wall distance ($\cdot$), resulting from nanofluids effect using therminol VP-1 as the base working fluid.
\[ T_i = \frac{q_v}{\rho C_p U_t} \]  

Here the parameters \( C_p, \rho \) and \( q_v \) represent the nanofluid specific heat capacity, nanofluid density and heat flux, respectively. The variable, \( U_t \), in equation (32), is the friction velocity which can be calculated depending on the wall shear stress and nanofluid density as

\[ U_t = \sqrt{\tau_w/\rho} \]  

For the simulations, the parameter, \( y^+ \), represents the mean non-dimensional distance which can be determined based on the friction velocity \( (U_t) \), nanofluid kinematic viscosity (\( \nu \)), and the first cell height (\( y \)), as shown below

\[ y^+ = \frac{y U_t}{\nu} \]  

The non-dimensional mean temperature \( (T') \) of all considered nanoparticles normalized by the friction temperature \( (T_f) \) is another parameter used to study the effect of nanofluids performance as shown in Fig. 15. These mean temperature profiles are plotted close to the absorber outlet (at \( L = 3.8 \) m) under uniform heat flux for all considered nanoparticles, using therminol VP-1 as the base fluid at a specific \( Re \) number (in this case \( Re = 10000 \)). It is observed from these profiles that the addition of nanoparticles leads to a gradual increase in the mean temperature profiles inside the logarithmic region, away from the walls. This is accompanied by a gradual decrease in the heat conduction sub-layer close to the wall (at smaller \( y^+ \) values) as the volume fraction of nanoparticles increases. This is due to the fact that by increasing the nanoparticles volume fraction, there is a gradual increase in the effective \( Pr \) number. An increase in the \( Pr \) number means that the near-wall sub-layer, across which heat conduction is the dominant mode of heat transfer instead of turbulence transport, is reduced in thickness.

### 4.8. Comparison with previous studies

The heat transfer performance and the collector thermal efficiency of the present study are compared with existing data for a wide range of different nanofluid parameters, as illustrated in Table 11. It is apparent that the findings of the current study are in agreement with those found in the literature. In particular, the enhancement of thermal efficiency achieved in this study is very close to the results of [22] for the exact same conditions (base fluid, nanoparticles and volume fractions i.e. Therminol VP-1 with 6% volume fraction \( Al_2O_3 \)). More specifically, [22] proved that by using 6% volume fraction nanofluid of Therminol VP-1 with \( Al_2O_3 \) can lead to a 7.2% enhancement in the thermal efficiency, which is very close to the current estimates of 7.5% whereas when the nanoparticle is replaced by SiO\(_2\) with the same base fluid, the thermal efficiency enhancement is increased to 9.18%. Similarly, for the heat transfer performance, the current predictions are very close to the results of [30]; which use thermal oils with \( Al_2O_3 \) for slightly different nanoparticle volume fractions. It should be noted here that any differences that are observed in the present predictions compared to values found in

<table>
<thead>
<tr>
<th>Reference</th>
<th>Re</th>
<th>Tin (K)</th>
<th>Base fluid</th>
<th>N.P.</th>
<th>( \phi ) (%)</th>
<th>Enhancement (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>[15]</td>
<td>6710</td>
<td>350</td>
<td>Syltherm800</td>
<td>( Al_2O_3 )</td>
<td>8</td>
<td>7.6</td>
</tr>
<tr>
<td>[16]</td>
<td>–</td>
<td>627</td>
<td>Thermal oil</td>
<td>( Al_2O_3 )</td>
<td>2</td>
<td>8.1</td>
</tr>
<tr>
<td>[17]</td>
<td>–</td>
<td>298</td>
<td>Syltherm800</td>
<td>( Al_2O_3 )</td>
<td>4</td>
<td>10</td>
</tr>
<tr>
<td>[22]</td>
<td>( 2 \times 10^3 )</td>
<td>600</td>
<td>TherminolVP-1</td>
<td>( Al_2O_3 )</td>
<td>6</td>
<td>7.2</td>
</tr>
<tr>
<td>[21]</td>
<td>–</td>
<td>298</td>
<td>Syltherm800</td>
<td>CuO</td>
<td>4</td>
<td>1.26</td>
</tr>
<tr>
<td>[28]</td>
<td>698</td>
<td>573</td>
<td>Syltherm800</td>
<td>CuO</td>
<td>5</td>
<td>2.76</td>
</tr>
<tr>
<td>[30]</td>
<td>( 6 \times 10^3 )</td>
<td>500</td>
<td>Synthetic oil</td>
<td>( Al_2O_3 )</td>
<td>3</td>
<td>5</td>
</tr>
<tr>
<td>Present Study</td>
<td>( 3 \times 10^4 )</td>
<td>400</td>
<td>Therminol VP-1</td>
<td>( SiO_2 )</td>
<td>6</td>
<td>9.18</td>
</tr>
<tr>
<td>Present Study</td>
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<td>400</td>
<td>Therminol VP-1</td>
<td>( Al_2O_3 )</td>
<td>6</td>
<td>7.5</td>
</tr>
<tr>
<td>Present Study</td>
<td>( 3 \times 10^4 )</td>
<td>400</td>
<td>Water</td>
<td>( SiO_2 )</td>
<td>6</td>
<td>5.11</td>
</tr>
<tr>
<td>Present Study</td>
<td>( 3 \times 10^4 )</td>
<td>600</td>
<td>Molten salt</td>
<td>( SiO_2 )</td>
<td>6</td>
<td>9.92</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Reference</th>
<th>Re</th>
<th>Tin (K)</th>
<th>Base fluid</th>
<th>N.P.</th>
<th>( \phi ) (%)</th>
<th>Enhancement (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>[14]</td>
<td>–</td>
<td>300</td>
<td>Synthetic oil</td>
<td>( Al_2O_3 )</td>
<td>5</td>
<td>36</td>
</tr>
<tr>
<td>[16]</td>
<td>–</td>
<td>627</td>
<td>Thermal oil</td>
<td>( Al_2O_3 )</td>
<td>2</td>
<td>10.92</td>
</tr>
<tr>
<td>[18]</td>
<td>–</td>
<td>300</td>
<td>Synthetic oil</td>
<td>( Al_2O_3 )</td>
<td>5</td>
<td>1.65</td>
</tr>
<tr>
<td>[20]</td>
<td>–</td>
<td>400</td>
<td>Therminol VP-1</td>
<td>Cu</td>
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<td>32</td>
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<tr>
<td>[28]</td>
<td>698</td>
<td>298</td>
<td>Syltherm800</td>
<td>CuO</td>
<td>5</td>
<td>15.53</td>
</tr>
<tr>
<td>[30]</td>
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<td>500</td>
<td>Synthetic oil</td>
<td>( Al_2O_3 )</td>
<td>3</td>
<td>7</td>
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<tr>
<td>[65]</td>
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<td>298</td>
<td>Water</td>
<td>ZnO</td>
<td>4</td>
<td>14.4</td>
</tr>
<tr>
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<td>( 3 \times 10^4 )</td>
<td>400</td>
<td>Therminol VP-1</td>
<td>( SiO_2 )</td>
<td>6</td>
<td>15.6</td>
</tr>
<tr>
<td>Present Study</td>
<td>( 3 \times 10^4 )</td>
<td>400</td>
<td>Therminol VP-1</td>
<td>( Al_2O_3 )</td>
<td>4</td>
<td>7.32</td>
</tr>
<tr>
<td>Present Study</td>
<td>( 3 \times 10^4 )</td>
<td>400</td>
<td>Water</td>
<td>( SiO_2 )</td>
<td>6</td>
<td>32.4</td>
</tr>
<tr>
<td>Present Study</td>
<td>( 3 \times 10^4 )</td>
<td>600</td>
<td>Molten salt</td>
<td>( SiO_2 )</td>
<td>6</td>
<td>21.36</td>
</tr>
</tbody>
</table>
the literature are mainly due to small differences in the Re number, inlet temperature of the base fluid or the volume fraction of the nanofluid. Table 11 thus shows that the current results provided by the numerical simulations are acceptable and logical.

5. Conclusions

Introducing the parabolic trough collector application to the solar thermal field is promising approach and very useful technique to reduce the greenhouse emission and enhance the thermal performance of the solar thermal field. The main contribution of the present study is to investigate numerically the thermal performance and flow features inside a parabolic trough collector by presenting the capability of different non-metallic nanoparticles (Al₂O₃, CeO₂, CuO, Fe₂O₃, TiO₂ and SiO₂) dispersed in three different base fluids (therminol VP-1, water, and molten salt) over a range of Reynolds (Re) numbers and three volume fractions (2, 4 and 6%). A realistic non-uniform heat flux distribution in the circumferential direction over the absorber outer surface has been applied, using the Monte Carlo ray tracing technique found in the literature. The importance of the study is to compare the effect of different nanoparticles in the same base fluid and also to deduce the effect of these nanoparticles in different base fluids on the overall performance of the PTCs as summarized below

➢ For water based nanofluids the improvement in the Nusselt (Nu) ranged from 12.72% to 32.4% with 6% volume fraction of SiO₂ nanoparticle providing the highest improvement. Whereas for the molten salt and therminol VP-1 base nanofluids, the improvements in the Nu number was not as pronounced but the general trend was the same with SiO₂ being the most effective and CeO₂ the least.

➢ For molten salt base nanofluids the enhancement in the thermal efficiency ranged from 5.1% to 9.92% with 6% volume fraction of SiO₂ nanoparticle providing the largest enhancement and CeO₂ the least. On the other hand, for therminol VP-1 base nanofluids the enhancement in the thermal efficiency ranged from 3.29 to 9.18% with 6% volume fraction of SiO₂ nanoparticle providing the largest improvement and CeO₂ the least. For the water based nanofluid, no marked improvement was observed in terms of the thermal efficiency. However, SiO₂ still performed better than CeO₂.

➢ The improvement in the thermal exergy efficiency for the molten salt base fluids ranged from 5.2% to 10.08% with 6% volume fraction of SiO₂ nanoparticle giving the maximum improvement and CeO₂ the minimum. On the other hand, the improvement for therminol VP-1 base nanofluids in the thermal exergy efficiency ranged from 3.18% to 9.02% with 6% volume fraction of SiO₂ nanoparticle providing the largest improvement. Finally, for the water based nanofluids, once again the thermal exergy did not show huge gains.

➢ Based on the performance, it is concluded that regardless of the base working fluid SiO₂ is the best candidate from all perspectives (improvement in Nu number, thermal efficiency enhancement, exergy efficiency improvement) with CeO₂ being the worst.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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