

Numerical Study of Copper Oxide/Therminol VP-1 Nanofluid in Solar Parabolic Trough Collector



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1 Introduction

Enhancement in the heat transfer enhancement in heat exchanging devices like solar collectors, heat exchangers, etc., plays an important role in system design. There are several types of solar collectors which concentrate the solar radiation energy to give high fluid temperature. In PTC, incident solar radiation is converted to heat energy which is transported by the fluids which circulate through the collector. The linear tube is placed at the focal line of the parabolic reflector. The fluid which transports the heat is referred to as heat transfer fluids (HTF). The concentration of solar energy enables outlet fluid temperatures to achieve as high as 400 °C in PTC, which is suitable for power generation. In industrial processes, water, ethylene glycol, synthetic oil, molten salts, etc., are used as HTF. The thermal efficiency of PTC can be further improved by using HTF having more thermal conductivity as compared to conventional fluids. The thermal efficiency can be ameliorated considering nanofluid as HTF, due to their improved thermal characteristics of it. Use of additional coatings and fins to collector tube increases thermal collection. The application of evacuated tubes decreases heat loss and increases the thermal yield.

The metallic oxide-based nanofluid is used in many heat exchange applications. Nanofluids are an amalgamation of a nanoparticle (fine size and shape) and conventional fluids. Nanoparticles are oxides of metals having good thermal conductivity; the fine particles are uniformly spread in the conventional fluid. There are two methods for the preparation of nanofluids. The conventional fluids are referred to as base fluids. It is anticipated that the thermal conductivity of the nanofluids will have better concerning conventional fluid. The use of nanofluids may have sedimentation problems, and pressure drops may increase also. Due to certain advantages, nanofluids are used in the solar collector as compared to base fluids. This makes a

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more efficient heat collection system. The surface area is increased due to nano-size particles. This shows very high scattering and absorption of the incident radiation range as compared to the base fluids. The output temperature is increased; hence, the collector efficiency will also be increased. To get a similar collector efficiency, complex manufacturing processes are required for creating efficient heat transfer surfaces. This can be easily used in place of the based fluid in the existing system with different volumetric fractions as per requirement [1]. The thermal characteristics of nanofluid depend on properties of solid particles and base fluid [2–15]. The application of nanofluids will lead to internal fouling and erosion of the channel. The modeling of nanofluids flow in the PTC for heat transfer study has remained in the interest of the researcher. He et al. [16] considered a single-phase model (SPM) and used changed values of thermal and fluid flow properties of nanofluids. These properties are thermal conductivity, density, viscosity, etc., in the modeling. A comparative study of SPM and a two-phase model (TPM) were done numerically. TPM is more preferable in comparison with SPM, due to good agreement with experimental results. The heat transfer rate prediction using PTM was closer (about 8% error) with respect to the single-phase model (about 16% error) as compared to the experimental result for Cu-water nanofluid [16]. Davarnejad and Jamshidzadeh [4] used three models, namely (a) SPM (b) TPM and (c) volume of fluids. TPM's result was a very good larger volume fraction of nanoparticle in the fluids. The results based on single-phase are about 11% with respect to the experimental result. In this research, the work forced convection heat transfer, considering CuO/Thermal-VP1 the PTC by considering constant heat flux over horizontal receiver tube has been studied using finite volume based software, *ANSYS Fluent*. The volumetric fraction has been considered from 1 to 5%. Understanding the thermal behavior will help in the proper design of the collector.

2 Literature Survey

Several authors numerically studied PTC using constant heat flux on it. The heat transfer modeling has been done assuming 1-D. The mass and energy equations were solved simultaneously. The results were validated with Sandia National Laboratory's experiment data [14]. Kumar et al. studied CuO/water using single-phase modeling. The solar flux solar load cell and S2S radiation model were used [10]. Kaloudis et al. [17] simulated the SEGS LS2-module PTC with Al_2O_3 -Syltherm 800 nanofluid. For nanofluid modeling, TPM was used and validated with experimental data. The maximum relative error in the collector efficiency was 7.3%. For simulation, up to 4%, volumetric concentrations were used, and at this concentration, collector efficiency was amplified by 10%. Basbous et al. [18] coded for modeling of PT by considering all type losses from PTC. They used Al_2O_3 -Syltherm 800 as HCE. Considering losses, the heat transfer coefficient can be elevated up to 18%. The useful heat gain of a PTC depends on the heat loss from the operating temperature of

the heat absorber tube. An evacuated concentric glass was used, and it was numerically studied by Daniel et al. [19]. The effect of the evacuated tube with the selective coating with different wind velocity conditions was studied. Kasaeian et al. [20] modeled constant heat over PTC and used Al_2O_3 /synthetic oil nanofluid. For the velocity pressure compounding, SIMPLE algorithm is used. They considered three equations of the 3D Navier–Stokes equation. The heat transfer rate is related to nanoparticle concentration in the nanofluid and Reynolds number (Re). They considered the volume fraction of nanofluids up to 5%. The thermal and thermodynamic performances of a PTC with Al_2O_3 /synthetic oil nanofluid are presented based on the second law of thermodynamics and entropy generation [21]. With nanofluids, the heat transfer rate increased up to 76% when the volumetric fraction increased to 8% from 0%. Optimal value Re was calculated for minimum entropy generation in the receiver. Beyond the optimal value of Re, use of nanofluids is thermodynamically useless [22]. A code was developed in Fortran language based on finite difference method to study the thermal behavior of PTC. This shows the improvement in heat absorption using Cu-water nanofluid. Kasaeian et al. [23] considered constant solar heat flux for modeling of PTC using Al_2O_3 /synthetic oil nanofluid. The convection heat transfer coefficients increase with Reynolds number and particle concentration level in the nanofluid. Based on CFD simulation, overall efficiency by 0.125% Al_2O_3 water nanofluid is performed [24]. Experimental and simulation results were differed by 8% approximately. The thermal efficiency was found to improve by 12.5% with a nanoparticle concentration varying from 0 to 6%. Numerical simulation for heat transfer coefficient along with experimental results on titania nanofluid in a laminar flow condition (low Reynolds number) was performed at 0.6 vol% and 4000 W/m^2 heat flux. The coefficient of transfer of heat is much relying on the thermal conductive property and has a less effect from viscosity, Brownian force, and thermophoretic forces. Simulation of straight tubes showed marked improvements in the heat transfer coefficient using titania nanofluid at the entry portion of the tube. A simulation model with Syltherm 800 with dispersed alumina and cuprous oxide nanoparticles was performed, and 50% enhancement of heat transfer coefficient was obtained [2]. The efficiency of the model is decreased by 14% as the temperature of Syltherm oil 800 is increased from 25 to 325 °C. Numerical simulation of a PTC was carried out with CFD along with the MCRT technique. PTC with a concentration ratio of 113 was studied for its performance [25]. The results suggested receiver performance improvement by 12.5% at a nanoparticle concentration of 6%. Al_2O_3 and $\text{CuO}/\text{H}_2\text{O}$ nanofluids are considered as HTF, and results were obtained using the $k-\varepsilon$ model [26]. Wang et al. [27] simulated based on the finite element method to study PTC performances using Al_2O_3 /synthetic oil nanofluid. The temperature of the absorber tube using normal base fluid was high, which was reduced by Al_2O_3 /synthetic oil. It was also observed that the temperature gradients are low with respect to the base fluid. Numerical simulation of a PTC under conditions of varying solar flux was performed to investigate the absorber tube [28]. The circumferential temperature difference (CTD) decreases with the discharge rate and inlet temperature of fluids in the channel. However, it increases with the increase of DNI. At a velocity of 1 m/s and DNI (500 W/m^2), the circumferential temperature difference of 22 K was attained

in the model. The heat transfer study using CuO/water nanofluids in a rectangular shallow cavity was performed for 0–4 vol.%. As the volumetric fraction increased, the Nu values and the coefficient of transfer of heat are increased. Interesting studies like radial and axial temperature profiles have been attempted, and the mathematical model was developed to facilitate calculation of heat transfer of PTC [3]. Simulation for optical efficiency, cosine factor, and receiver efficiency was attempted, and under normal incidence conditions and a tracking error of 12 min radian angle, optical efficiency of 53% was attained [29]. The CuO/water nanofluid in a parabolic shaped trough collector was used, and this model was created and subjected to CFD simulation. Agreement of results of experimental and simulation studies was obtained with a marginal difference of 5.75% at a flow of 18 L per hour. The thermal efficiency in the model is improved by 6% when the HTF volumetric fraction was 0.01%. The fluid flow affects the heat transfer rate in the system [30–36].

3 Simulation of Parabolic Trough Collector

The numerical modeling was done by solving three equations, namely continuity equation (Eq. 1), momentum equation (Eq. 2), and energy conservation equation (Eq. 3) using ANSYS Fluent.

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_j)}{\partial x_j} = 0 \quad (1)$$

$$\frac{\partial(\rho u_i)}{\partial t} + \frac{\partial(\rho u_i \rho u_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\rho \delta_{ij} + \mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + \rho g_i \quad (2)$$

$$\frac{\partial(\rho C_p T)}{\partial t} + \frac{\partial(\rho u_i C_p T)}{\partial x_i} - \frac{\partial \left[\lambda \frac{\partial T}{\partial x_j} \right]}{\partial x_j} = S_T \quad (3)$$

In this effort, the absorber tube is modeled as a copper tube with nanofluid flowing through it. Single-phase model is used in the ANSYS Fluent for simulation in the laminar flow range.

4 Modeling of the PTC

This section covers the modeling using *Design Modeler* and solution using boundary conditions.

4.1 Geometry of PTC

The dimensions that are going to be used are given in Table 1. The PTC receiver is modeled as per the dimensions in the ANSYS *Design Modeler*. The current analysis is done by assuming that the absorber tube is not covered with the glass envelope. The model meshed, and respective domains are named as solid and fluid. The specification of heat absorber is as per commercial parabolic receivers. The boundary conditions were defined for the cylindrical wall, inlet, and outlet of the geometry. The heat transfer fluid (HTF) is nanofluid.

In the simulation, in ANSYS *Fluent*, the energy equation kept in ON position. All the simulations are done in the laminar range. So, the viscous model is kept as the laminar. In modeling, the absorber tube material is copper, and HTF is the CuO/Therminol VP1. The properties of the nanofluid are calculated using correlations given in equation no. 4–7. The density of nanofluid is calculated using Eq. (4)

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

The specific heat is calculated from

$$C_{p,nf} = \phi\rho_{np}C_{p,np} + (1 - \phi)\rho_{bf}C_{p,bf} \tag{5}$$

The thermal conductivity is given by the Maxwell model

$$k_{nf} = \frac{k_{np} + 2k_{bf} + 2\phi(k_{np} - k_{bf})}{k_{np} + k_{bf} - \phi(k_{np} - k_{bf})}k_{bf} \tag{6}$$

The viscosity of the nanofluid is given from the general Einstein’s formula

$$\mu_{nf} = (1 + 2.5\phi)\mu_{bf} \tag{7}$$

where subscripts *nf* are for nanofluids, *bf* is for base fluids, and *np* is for nanoparticles and ϕ is the volume fraction of nanofluids. The properties of the materials are given in Table 2.

Table 1 Geometric dimensions of PTC

I. D. of the absorber tube	0.0244 m
O. D. of Absorber tube	0.0254 m
Aperture width	1.22 m
Aperture length	6 m
Aperture area	7.32 m

Table 2 Thermal fluid flow properties of the materials

Material	Volume fraction percent (ϕ)	Density (ρ)	Thermal conductivity (k)	Specific heat (Cp)	Dynamic viscosity (μ)
Copper-Therminol VP1 nanofluid (30 °C)	0	1056	0.135	1575	0.00328000
	1	1110.44	0.13901	1488.1628	0.00336346
	2	1164.88	0.14310	1409.9428	0.00344992
	3	1219.32	0.14727	1339.1858	0.00353952
	4	1273.76	0.15152	1274.9349	0.00363242
	5	1328.2	0.15587	1216.3900	0.00372876
CuO	–	6500	20	535.6	–

4.2 Boundary Conditions

The receiver tube is modeled with the inner surface as stationary with the no-slip condition. The heat transfer has been considered only in the radial direction. It has been neglected in the other direction. The direct radiations from the sun are modeled as the constant heat flux on the receiver tube. The heat flux around the circumferential surface is assumed to be constant. In this case, the heat flux is taken as 10 kW/m². The initial condition of the velocity inlet and temperature of fluids were specified.

5 Results and Discussion

5.1 Effect of the Volume Fraction on Temperature

The volumetric fractions that were used in this work were 1–5%, and the length of the tube is considered 6 m. Normally, the volume fraction up to 6% is advised in the thermal application. Beyond this, the power required to maintain the flow is not justified.

The surface temperature of the heat receiver tube and the bulk fluid temperature are directly proportional to the volumetric concentration. The surface temperature of the receiver tube increases in the axial direction in the HTF flow direction (Fig. 1). As the fluids flow in the forward direction, the heat transfer rate is high owing to the lower temperature of the fluids; hence, lower surface temperature of the tube is observed. The effect of volumetric fractions is also crucial. At any of the locations along with the flow, the surface temperature of the tube is inversely proportional to the volumetric fraction. The mean outlet bulk temperature is maximum for a maximum volumetric fraction (5%) of nanofluids (Fig. 2). The heat transfer coefficient initially is very high in thermal entry length, and it decreases later on and finally becomes almost constant along the flow direction. The heat transfer coefficient also increases

Fig. 1 Surface temperature variation along the length of absorber tube at $Re = 1500$

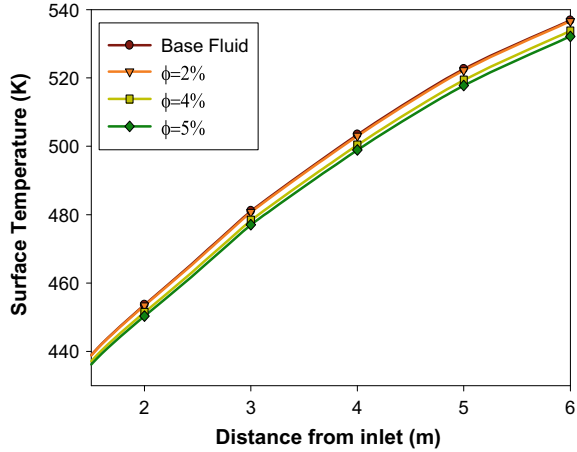
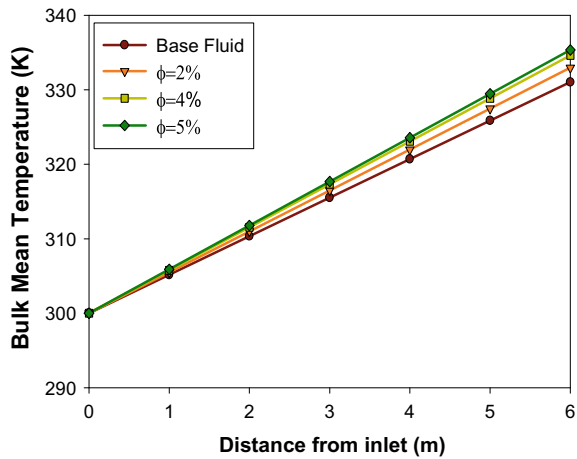


Fig. 2 Bulk mean temperature variation along the length of tube at $Re = 1500$



nanoparticle concentration in the nanofluid. It is the maximum for volumetric fraction 5 and lowest for base fluid. In the analysis, results have been presented for Reynolds number of 1500. The surface heat transfer coefficient is shown in Fig. 3.

5.2 Effect of Volume Fraction on Pressure Drop in the Tube

There is a need to simultaneously study the pressure drop due to the increase in the volumetric fraction. The density and viscosity result in the increase in fluid friction and thus pressure drop. Hence, there will be an additional pressure drop in the flow, which is disadvantageous, because it will have more pump power work.

Fig. 3 Surface heat transfer coefficient in the fully developed region at $Re = 1500$

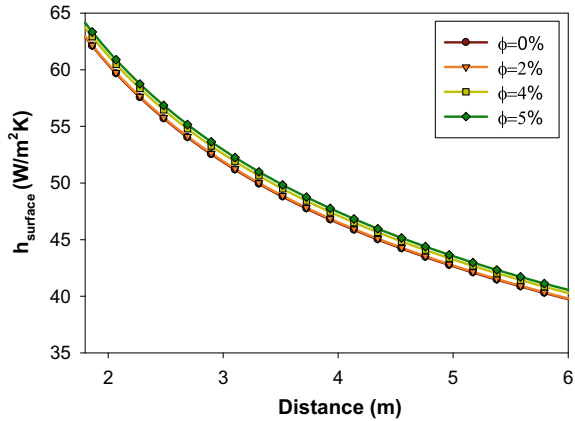
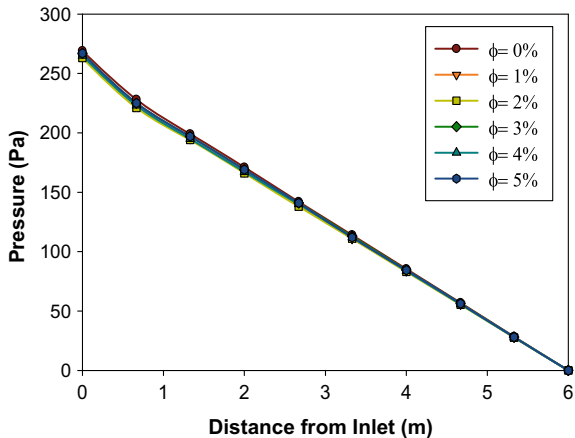


Fig. 4 Pressure drop along the length at $Re = 1500$



The optimization study is very important. Figure 4 shows the pressure drop along the length of the tube at different volumetric fractions of nanofluids. The pressure drop for base fluid and nanofluids shown in the graph in Fig. 4 is not very significant. The pressure drop in the tube is almost the same for all volumetric fraction. This is due to the very high Prandtl number of base fluids. Even the very little variation in the viscosity has been observed (Table 2).

5.3 Effect of Volume Fraction on Nu

The value of Nu along the length for the base fluid as well as the nanofluid at different volume fractions was obtained in the laminar flow range. The simulation data is compared with Shah correlations for Nu [37]. All the simulations were done

in the laminar range for the Re of 1500. The simulated data is per the correlations. For Therminol VP-1, simulation results are close to Hausen’s correlation, whereas it shows deviations with Shah’s correlation (Fig. 5). A similar trend is for nanofluids at a volume fraction of 1%.

The variation of Nu on the tube at volume fractions 2, 3, 4, and 5 is shown in Figs. 5, 6, 7, 8, 9, and 10, respectively. It is concluded that in the laminar flow region, CFD simulation provides very good results in the fully developed flow. In the present analysis, the hydrodynamic entry length is 1.83 m. In the developing flow region, the Nu number is very high, and it decreases very fast. This trend is visible from the graphs.

Fig. 5 Nu along the length of PTC

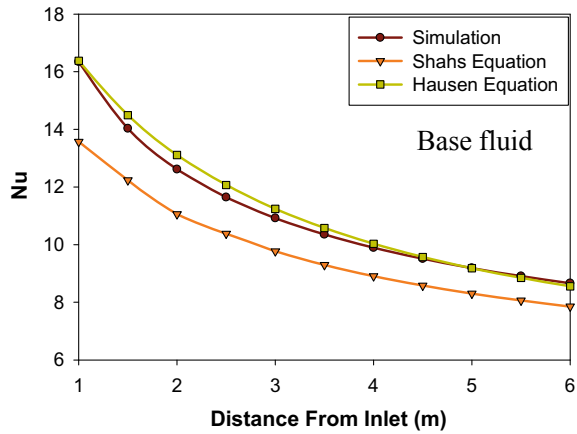


Fig. 6 Nu along the length CuO/Therminol

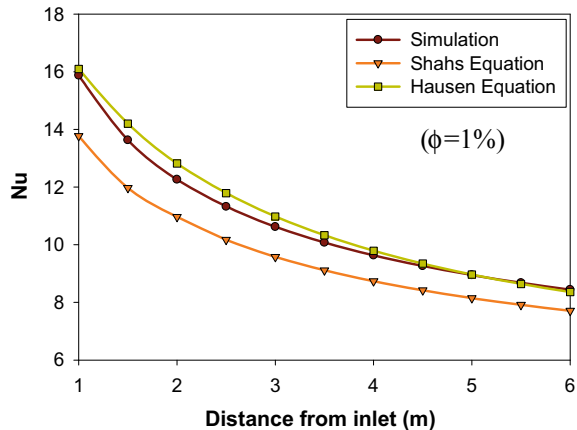


Fig. 7 Nu along the length CuO/Therminol VP-1

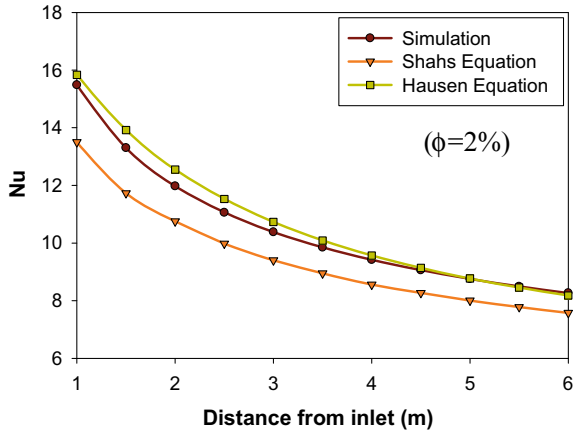


Fig. 8 Nualong the length CuO/Therminol VP-1

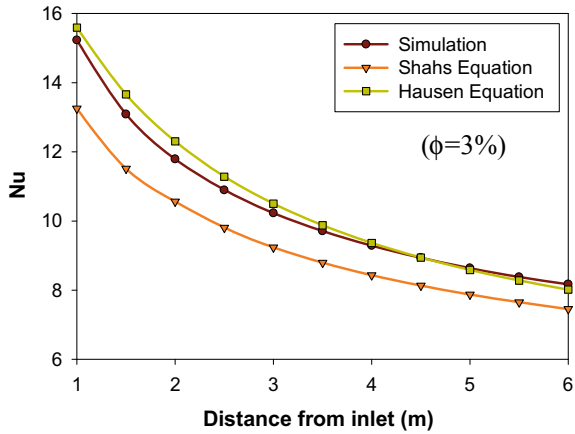


Fig. 9 Nu along the length CuO/Therminol ($\phi = 4\%$)

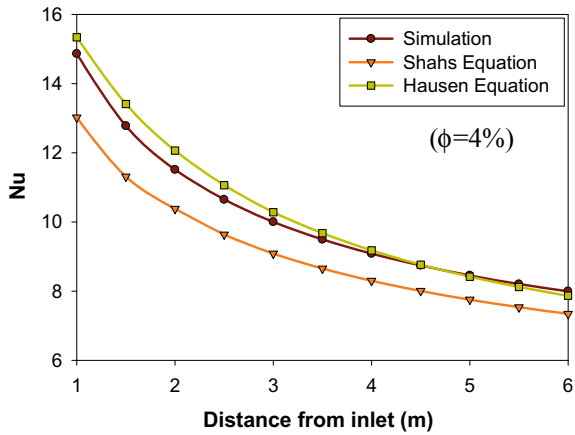
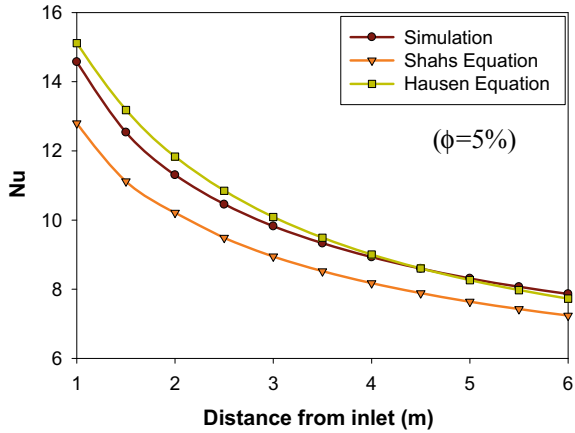


Fig. 10 Nu along the length
CuO/Therminol ($\phi = 5\%$)



6 Conclusion

The simulation of the heat receiver tube of a linear trough collector considering copper oxide/Therminol VP-1 nanofluid is investigated using *ANSYS Fluent* software. The effect of nanofluid volume fractions has been reported. The modeling work can be concluded as:

- (a) The volume fraction of the nanofluid plays a very crucial role in the thermal performance of the PTC. The thermal performance of the collector increases with an increase in the nanofluid volumetric concentration. The exit temperature of the nanofluid in the absorber is the increasing function of the volume fraction of the nanofluid for the same boundary conditions. As the volume fraction is increased, the absorber surface temperature decreases, and the bulk means that the temperature of the fluid increases. Also, the heat transfer coefficient increases.
- (b) The density and viscosity of the fluid increase with the volume fraction of the nanofluid; hence, pressure drop is also affected. This will increase fluid friction. An increase in the value of particle concentration leads to an increase in the pressure drop, and consequently, it increases the pumping power. It will increase the operating cost of the system. Therefore, the larger values of the viscosity and density are not desirable in our case. Volume fraction up to 5% is used. At higher values of volume fraction, it will increase the risk of sedimentation of the nanoparticles in the tube at lower discharge rates.
- (c) The mean bulk temperature of nanofluid depends on the discharge rate. It was seen that the mean bulk temperature of the nanofluid increases with the decrease in volume flow rate and vice versa.
- (d) The increase in the volume fraction causes a slight decrease in the variation of Nusselt number along the length for the same conditions of heat flux and Reynolds number.

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