Heat Transfer Enhancement in a Parabolic Trough Solar Receiver using Longitudinal Fins and Nanofluids (Postprint)

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Abstract

In this paper, we present a three dimensional numerical investigation of heat transfer in a parabolic trough collector receiver with longitudinal fins using different kinds of nanofluid, with an operational temperature of 573 K and nanoparticle concentration of 1% in volume. The outer surface of the absorber receives a non-uniform heat flux, which is obtained by using the Monte Carlo ray tracing technique. The numerical results are contrasted with empirical results available in the open literature. A significant improvement of heat transfer is derived when the Reynolds number varies in the range 2.57x10(4) <= Re <= 2.57x105, the tube-side Nusselt number increases from 1.3 to 1.8 times, also the metallic nanoparticles improve heat transfer greatly than other nanoparticles, combining both mechanisms provides better heat transfer and higher thermo-hydraulic performance.

Full Text

Preamble

Heat Transfer Enhancement in a Parabolic Trough Solar Receiver using Longitudinal Fins and Nanofluids

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Abstract

This paper presents a three-dimensional numerical investigation of heat transfer in a parabolic trough collector receiver equipped with longitudinal fins and utilizing various nanofluids. The study operates at a temperature of 573 K with a nanoparticle volume concentration of 1%. The outer surface of the absorber receives a non-uniform heat flux distribution obtained through the Monte Carlo ray tracing technique. Numerical results are validated against empirical correlations available in the open literature. Significant heat transfer improvement is achieved when the Reynolds number varies between 2.57×10^{4} and 2.57×10^{5}, with the tube-side Nusselt number increasing by a factor of 1.3 to 1.8. Metallic nanoparticles demonstrate superior heat transfer enhancement compared to other types, and the combined use of both mechanisms yields the best thermo-hydraulic performance.

Keywords: numerical study, Monte Carlo ray trace, parabolic trough collector, heat transfer, longitudinal fins, nanofluid

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Nomenclature

Symbol	Description	Units
A	Area	m^2
C	Specific heat	$J/(kg \cdot K)$
D	Diameter	m
DNI	Direct normal irradiance	W/m^2
f	Friction factor	-
h	Heat transfer coefficient	$W/(m^2 \cdot K)$
h_9	Glass cover outer heat transfer coefficient	$W/(m^2 \cdot K)$
k	Thermal conductivity	$W/(m \cdot K)$
L	Receiver length	m
$\dot{m}$	Mass flow rate	kg/s
Nu	Nusselt number	-
P	Pressure	Pa
PEC	Performance evaluation criteria	-
Pr	Prandtl number	-
q”	Heat flux	W/m^2
Re	Reynolds number	-
T	Temperature	K
v	Wind velocity	m/s
V	Volume	m^3
Greek Letters		
β	Particle volume concentration	-
	Emissivity	-
	Viscosity	$Pa \cdot s$

Symbol	Description	Units
	Density	kg/m^3
Subscripts		
a	Ambient	
b	Bulk fluid state	
bf	Base fluid	
f	Fluid	
g	Glass	
go	Glass cover outer wall	
i	Inlet	
nf	Nanofluid	
p	Particle	
w	Absorber tube inner wall	
sky	Sky temperature	
Abbreviations		
HTF	Heat transfer fluid	
PTC	Parabolic trough collector	
LCR	Local concentration ratio	
MCRT	Monte Carlo ray tracing	

Introduction

The surge in fossil fuel prices during the petroleum crisis has driven industrialized nations to pursue alternative and renewable energy sources, particularly solar energy. This energy source is considered both economical and clean. Among solar technologies, parabolic trough collectors (PTCs) have emerged as the most proven solar concentration technique and have been extensively studied for solar thermal power plants [1].

Recent research has focused on enhancing tube-side heat transfer in these devices through both numerical and experimental approaches, examining convective heat transfer mechanisms, geometric effects, and various working fluids. Aggrey et al. [2] numerically investigated the thermal performance of a PTC receiver with perforated plate inserts, demonstrating improved thermodynamic performance through reduced entropy generation rates and establishing relationships between Nusselt number, friction factor, and insert geometry. Wang et al. [3] numerically studied heat transfer enhancement in a direct steam generation PTC receiver by inserting metal foams, reporting significant effects from foam layout and dimensionless height while porosity showed minimal influence. Song et al. [4] analyzed PTC receivers with non-uniform heat flux and helical screw-tape inserts, finding that maximum absorber tube outer surface temperature increases with inlet temperature and solar irradiation. Cheng et al. [5] conducted a numerical study on heat transfer enhancement using unilateral longitudinal vortex generators inside PTC receivers, showing that average Nusselt number and friction factor increase with geometric parameters while thermal losses decrease.

Recently, a new class of fluids called nanofluids has been developed and tested. This term was proposed by Choi in 1995 [6] at Argonne National Laboratory to describe a liquid mixture containing a small concentration of nanometer-sized solid particles in suspension. Nanofluids exhibit enhanced thermo-physical properties, notably high thermal conductivity. Research on nanofluid applications has proliferated in recent years, with various investigators examining their effects on heat transfer enhancement in PTC receivers. Sokhansefat et al. [7] studied Al_2O_3/synthetic oil in a PTC tube, reporting heat transfer augmentation with increasing nanoparticle volume fraction and operational temperature. Risi et al. [8] investigated CuO+Ni/nitrogen gas in a PTC tube, demonstrating that above 0.3% volume fraction, pressure drop drawbacks outweigh thermal property benefits, with an optimized maximum solar-to-thermal efficiency of 62.5%.

The present work explores a compound enhancement technique for parabolic trough collectors based on combining nanofluids with two longitudinal fins on the tube side. A three-dimensional numerical model implemented in ANSYS Fluent solves the flow field and heat transfer in the enhanced geometry. The heat flux distribution around the absorber tube was obtained using the Monte Carlo ray tracing (MCRT) method. The first part of this study analyzes the effect of longitudinal fin inserts across Reynolds numbers ranging from 2.57×10^{4} to 2.57×10^{5}, depending on heat transfer fluid characteristics. The final part compares four different nanoparticle types at 1% volume concentration. The objective is to develop a comprehensive understanding of how heat transfer fluid properties and receiver geometries influence parabolic trough solar collector performance.

Physical Model

The investigation considers a simplified PTC receiver model where central rod and support effects are negligible. The solar collector geometry shown in Fig. 1 [Figure 1: see original paper] serves as the simulation model. The absorber tube is constructed from steel while the glass cover uses borosilicate glass. The annular space between tubes is modeled as a vacuum at low pressure and ambient temperature. Table 1 lists the receiver dimensions.

Table 1. Receiver Dimensions

Parameter	Value
Focal length	1.71 m
Aperture width	5.77 m

Parameter	Value
Absorber inner radius	3.2 cm
Absorber outer radius	3.5 cm
Glass cover inner radius	5.96 cm
Glass cover outer radius	6.25 cm
Absorber material	Steel
Glass envelope material	Borosilicate
Glass cover transmittance	-
Coating absorbance	-
Glass cover emissivity	-

To improve heat transfer inside the parabolic trough collector receiver, this study analyzes longitudinal fins inserted into the absorber tube. Fig. 2 [Figure 2: see original paper] illustrates the fin geometry utilized in this work.

Numerical Model

The computational domain was constructed and meshed using commercial software GAMBIT 2.4.6, which also specified the boundary conditions. The turbulence model employed is the k-ω SST [9]. The governing equations—continuity, momentum, energy, and other scalars—were solved using the finite volume solver Fluent 6.3.26 [10]. The finite volume technique discretizes the non-linear partial differential equations using a first-order upwind scheme, with a pressure-based solver handling the pressure-velocity coupling.

Heat Transfer Fluid Properties

The heat transfer fluid (HTF) used in the first part of this study is synthetic oil DOWTHERM A, a eutectic mixture of 73% diphenyl oxide ($C_2H_{10}O$) and 27% biphenyl (C_2H_{10}). This fluid exhibits favorable physical properties and low vapor pressure at maximum operating temperature [11]. The thermo-physical properties of DOWTHERM A as functions of temperature are provided in Appendix A. Table 2 shows the base fluid properties at the 573 K inlet temperature.

Table 2. Thermo-Physical Properties of DOWTHERM A at 573 K

Property	Value
Density (kg/m^3)	-
Specific heat ($J/kg \cdot K$)	-
Thermal conductivity ($W/m \cdot K$)	-
Viscosity ($mPa \cdot s$)	-

Boundary Conditions and Heat Flux Distribution

The absorber tube outer wall receives a non-uniform heat flux distribution obtained using the Monte Carlo ray tracing technique [12]. Fig. 3 [Figure 3: see original paper] illustrates the resulting local concentration ratio (LCR) distribution on the absorber outer surface cross-section. LCR is defined as the ratio of concentrated radiant flux at a local position to the direct normal irradiance (DNI), with a DNI of 1000 W/m^2 used in this study.

Symmetry boundary conditions are applied at the inlet and outlet of the annular space between absorber and glass cover. The outer glass cover uses a combined convection-radiation thermal boundary condition. Glass emissivity is approximately 0.83, while sky emissivity is determined using the Martin and Berdahl correlation [13]:

Sky temperature is calculated using the correlation [14]:

where ambient temperature is 300 K and Tdp is dew point temperature (K).

The convection heat transfer coefficient for the boundary condition is defined by the experimental correlation [15]:

where vw is wind speed (2 m/s in this study) and dgo is the glass envelope outer diameter.

Results and Discussion

Validation of Numerical Results

The average Nusselt number and heat transfer coefficient are defined as:

where q" is the average heat flux on the absorber tube inner wall, Tw is the average inner wall temperature, and Tb is the average bulk temperature of the HTF.

The Darcy friction factor for turbulent flow is defined as:

For validation, numerical friction factor results are compared against the correlations of Petukhov [16] and Blasius [17], while average Nusselt number results are compared with the Gnielinski [18] and Notter-Rouse [19] correlations:

Gnielinski correlation:

valid for $10^4 \leq$ Re $\leq$ 5\$×10^{6}\$ and 0.5 $\leq$ Pr $\leq$ 2000.

Notter-Rouse correlation:

Blasius correlation:

for Re $\leq 2\times10^{4}$, and for Re $\geq 2\times10^{4}$.

Fig. 4 [Figure 4: see original paper] shows excellent agreement for Darcy friction factor, with maximum deviation of 4.1% and minimum relative error of 0.7%. Fig. 4(b) demonstrates Nusselt number validation, showing maximum and minimum deviations of 4.8% and 0.64% respectively from the Gnielinski correlation, and 9.3% and 1.8% from the Notter-Rouse correlation. These results confirm good agreement between numerical and empirical data.

Effect of Longitudinal Fins on Heat Transfer

The finned absorber solution yields significantly higher Nusselt numbers compared to the smooth tube model. Fig. 5 [Figure 5: see original paper] presents Nusselt number evolution with Reynolds number for smooth and finned geometries (Case-1 and Case-2). Augmentations ranging from 1.3 to 1.8 times the plain tube values are observed, indicating substantial heat transfer enhancement from longitudinal fins.

Fig. 6 [Figure 6: see original paper] shows that the Darcy friction factor in finned tubes exceeds that of empty tubes and decreases with increasing Reynolds number. This elevated friction factor results from swirling flow induced by the longitudinal inserts acting as flow obstacles.

To evaluate overall enhancement, the thermal performance criterion (PEC) is defined as the ratio of dimensionless Nusselt number to dimensionless friction factor [20]:

where subscript 0 refers to the smooth tube baseline.

As expected, thermal performance decreases with increasing Reynolds number, as shown in Fig. 7 [Figure 7: see original paper]. An average PEC 1.5 is reported across the range $2.57\times10^{4} \leq$ Re $\leq 2.57\times10^{5}$. The triangular fin geometry exhibits slightly better performance than rectangular fins, indicating that geometric parameters significantly influence heat transfer improvement.

Circumferential Temperature Distribution

Fig. 8 [Figure 8: see original paper] presents temperature distributions on the middle cross-section of the absorber inner surface with and without fins for DNI = 1000 W/m^2, HTF inlet velocity = 1 m/s, and inlet temperature = 573 K. The finned tube shows higher absorber inner wall temperatures, particularly at fin locations. The HTF temperature increases by approximately 13 K with longitudinal fins. Fig. 9 [Figure 9: see original paper] reveals higher fluid temperatures at the bottom than the top due to non-uniform heat flux distribution concentrating at the lower region.

Effects of Nanofluids as Heat Transfer Fluid

Recent research has extensively investigated nanofluid effects on heat transfer enhancement in thermal engineering applications. Nanoparticle addition improves base fluid thermo-physical properties, with volume fraction significantly affecting heat transfer. This section compares four nanoparticle types for PTC performance: oxide ceramic (Al_2O_3), metal (Cu), metal carbide (SiC), and non-metal (C). Correlations for nanofluid thermo-physical properties are provided in Appendix A.

Fig. 10 [Figure 10: see original paper] shows local Nusselt number variation for the four nanofluids using DOWTHERM A as base fluid at 1% volume concentration and 13 nm particle diameter. Heat transfer enhancement arises from nanoparticles' higher thermal conductivity compared to conventional fluids. Nanoparticle type influences performance, with metallic nanoparticles providing superior enhancement.

Fig. 11 [Figure 11: see original paper] demonstrates that combining fins with nanofluids substantially improves heat transfer, though the fin insert effect dominates over nanoparticle presence. Both local Nusselt number and convective heat transfer coefficient decrease with axial distance.

To estimate compound technique performance, the Bergles et al. criteria [21] is applied under constant pumping power:

where h is the enhanced tube convective coefficient and h_{smooth} is the smooth absorber coefficient.

Fig. 12 [Figure 12: see original paper] shows efficiency decreases with Reynolds number, but the combined mechanisms provide superior heat transfer. The enhancement factor varies from 1.3 to 1.68 across $2.57\times10^{4} \le Re \le 2.57\times10^{5}$.

Conclusion

This paper investigated the influence of heat transfer fluid properties and receiver geometries (with and without fin inserts) on parabolic trough solar collector performance. Key conclusions include:

- Finned absorber Nusselt numbers are 1.3 to 1.8 times higher than smooth tube values.
- Finned absorber friction factors are 1.6 to 1.85 times greater than plain tubes.
- Fin geometric parameters significantly affect heat transfer improvement.
- Nanofluid thermo-physical properties depend on nanoparticle characteristics and type.
- Nusselt number response varies with nanoparticle type, with metallic nanoparticles providing the greatest enhancement.
- Combining both mechanisms (fins and nanofluid) yields the highest enhancement.

- Under similar conditions, nanofluid in finned absorbers offers superior heat transfer and thermo-hydraulic performance compared to smooth tubes with base fluid.

Appendix A

Thermo-Physical Properties of DOWTHERM A as a Function of Temperature [11]

Property	Correlation
Density (kg/m^3)	$1.493\times10^{3} - 3.332\times10^{0}\cdot T + 1.248\times10^{-2}\cdot T^{2} - 2.968\times10^{-5}\cdot T^{3} + 3.444\times10^{-8}\cdot T^{4} - 1.622\times10^{-11}\cdot T^{5}$
Specific heat (J/kg · K)	$-2.364\times10^{3} + 3.946\times10^{1}\cdot T - 1.703\times10^{-1}\cdot T^{2} + 3.904\times10^{-4}\cdot T^{3} - 4.422\times10^{-7}\cdot T^{4} + 1.979\times10^{-10}\cdot T^{5}$
Conductivity (W/m · K)	$1.856\times10^{-1} - 1.600\times10^{-4}\cdot T + 5.913\times10^{-12}\cdot T^{2}$
Viscosity (Pa · s)	$5.135\times10^{0} - 8.395\times10^{-2}\cdot T + 5.971\times10^{-4}\cdot T^{2} - 2.409\times10^{-6}\cdot T^{3} + 6.029\times10^{-9}\cdot T^{4} - 9.579\times10^{-12}\cdot T^{5}$

Thermo-Physical Properties of Nanofluids [22-25]

- **Density:** _{nf} = (1-β) _{bf} + β _p
- **Specific heat:** C_p,nf = [(1-β)(C_p)_{bf} + β(C_p)_p]/ _{nf}
- **Viscosity:** _{nf} = _{bf}/(1-β)^2.5
- **Thermal conductivity:** k_{nf} = k_{bf}·(k_p + 2k_{bf} - 2β(k_{bf} - k_p))/(k_p + 2k_{bf} + β(k_{bf} - k_p))

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Note: Figure translations are in progress. See original paper for figures.