THERMAL EFFICIENCY AND ENTROPY GENERATION FOR A PARABOLIC TROUGH RECEIVER AT DIFFERENT CONCENTRATION RATIOS

Mwesigye A. [1,*], Huan Z. [2], Meyer J.P.[3] and Bello-Ochende T.[4]
*Author for correspondence
[1]School of Mechanical, Industrial and Aeronautical Engineering, University of the Witwatersrand
Private Bag 3, Wits 2050, Johannesburg, South Africa.
[2]Department of Mechanical Engineering, Mechatronics and Industrial Design, Tshwane University of Technology
Private Bag X680, Pretoria 0001, South Africa
[3]Department of Mechanical and Aeronautical Engineering, University of Pretoria
Private Bag X20, Hatfield 0028, South Africa
[4]Department of Mechanical Engineering, University of Cape Town
Private Bag X3, Rondebosch 7701, South Africa,
E-mail: Aggrey.Mwesigye@wits.ac.za

ABSTRACT

The use of high concentration ratio parabolic trough systems is emerging as one of the ways to further reduce the cost of energy from these systems. This paper reports on the thermal efficiency of a parabolic trough system and entropy generation in the parabolic trough receiver at different concentration ratios. In this study, the geometric concentration ratios considered were in the range 88-113, while the collector rim angle was taken as 80°. A combined Monte Carlo ray tracing and computational fluid dynamics procedure was implemented to obtain the numerical solution of the thermal and thermodynamic performance of the system. Results show that the thermal efficiency reduces by about 4.5% as the concentration ratio increases from 88 to 113 at a given flow rate, while the entropy generation rate is shown to increase with increasing concentration ratio. Results further show that there is potential for improved performance with heat transfer enhancement using Cu-Therminol®VP-1 nanofluid as the heat transfer fluid in the receiver's absorber tube. For a system with a concentration ratio of 113 using Cu-Therminol®VP-1 nanofluid as the heat transfer fluid, the thermal efficiency increases by about 12% as the nanoparticle volume fraction increases from 0 to 6%.

INTRODUCTION

The increasing concerns of climate change and the need to reduce the emission of greenhouse gases have resulted in increased research and development efforts to develop efficient, clean, renewable and sustainable energy systems. Concentrated solar power (CSP) is one of the renewable energy technologies that has seen significant growth over the last few decades [1]. The parabolic trough technology is the most technically and commercially mature of all the CSP technologies available. The parabolic trough technology has been in use since the early 1980s, following the construction and successful operation of the first solar electricity generating plants (SEGS) in Mojave desert in California [2]. Over 80% of the installed concentrated solar power plants are parabolic trough systems [3].

The thermal performance of the parabolic trough system significantly depends on the performance of the receiver tube also known as the heat collection element. As such, receiver thermal performance has been a subject of a number of investigations. Dudley et al. [4,5] investigated the performance of parabolic trough receivers of different configurations. They showed that evacuated receivers performed better than non-evacuated and bare receivers. Burkholder and Kutscher [6,7] performed steady state tests and obtained heat loss measurements for receivers at different absorber tube temperatures. Receiver heat loss was shown to increase with absorber tube temperature.

A number of research and development initiatives are in place to further reduce the cost of electricity from solar thermal systems [8]. The use of high concentration ratios is seen as one of the ways to reduce the cost of energy from parabolic trough systems. With high concentration ratios, it is expected that the number of drives and controls will reduce and therefore, the capital, operation and maintenance costs will also reduce. Already, a number of high concentration ratio systems have been developed such as the Ultimate Trough® collector [9] and the SkyTrough® collector technology [10].

In some studies, it has been shown that for a parabolic trough system with a given rim angle, the entropy generation rates in the receiver increase as the concentration ratios increase at any given flow rate and inlet temperature [11,12]. Apart from these studies, the thermal and thermodynamic performance of high concentration ratio parabolic trough systems has not been widely investigated. Therefore, in this study the performance of a parabolic trough receiver using Therminol®VP-1 as the heat transfer fluid is investigated to determine the thermal efficiency and entropy generation rates at different concentration ratios. In addition, the potential for improved thermal and thermodynamic performance with heat transfer enhancement using Cu-Therminol®VP-1 nanofluid is presented.

NOMENCLATURE

a	[m]	Parabolic trough aperture width
A_c	[m^2]	Aperture area
Be	[-]	Bejan number
c_p	[J/kg/K]	Specific heat capacity
C_R	[-]	Concentration ratio
DNI	[W/m^2]	Direct normal irradiance
d_{ri}	[m]	Absorber tube inner diameter
d_{ro}	[m]	Absorber tube outer diameter
d_{gi}	[m]	Receiver tube glass cover inner diameter
d_{go}	[m]	Receiver tube glass cover outer diameter

f	[m]	Focal length
I_b	[W/m^2]	Direct normal irradiance
L	[m]	Length
$\dot{m}$	[kg/s]	Mass flow rate
$\dot{q}_u$	[W]	Rate of useful heat gain
$\dot{Q}_{loss}$	[W]	Rate of receiver thermal loss
Re	[-]	Reynolds number
S'_{gen}	[W/m K]	Entropy generation rate per unit meter
T	[K]	Temperature
$\dot{V}$	[m^3/h]	Volume flow rate
$\dot{W}_p$	[W]	Pumping power
x	[m]	Cartesian axis direction
y	[m]	Cartesian axis direction
z	[m]	Cartesian axis direction
ΔP	[Pa]	Pressure drop

Special characters

φ_r	[degrees]	Rim angle
ϕ	[%]	Nanoparticle volume fraction
σ	[W/m^2 K^4]	Stefan-Boltzmann constant
ε_{ro}	[-]	Absorber tube coating emissivity
ε_{gi}	[-]	Glass cover emissivity
η_{th}	[%]	Thermal efficiency
η_{el}	[%]	Electrical efficiency

Subscripts

Inlet	Inlet
Outlet	Outlet
ri	Absorber tube inner diameter
ro	Absorber tube outer diameter
gi	Receiver tube glass cover inner diameter

PHYSICAL AND THERMAL MODEL

A parabolic trough system consists of a collector in the form of a parabolically shaped mirror. At the focus of the collector is the receiver tube that receives the reflected solar radiation and converts it into useful heat energy that is transferred to the heat transfer fluid. The receiver consists of a steel absorber tube enclosed in an evacuated glass envelope. The parabolic trough system is a single axis tracking type collector, usually following the sun from east to west about the north to south axis. Figure 1 shows the 2-D view of the parabolic trough system. Figure 2 shows the 2-D cross-section view of the receiver tube and the corresponding thermal resistance network.

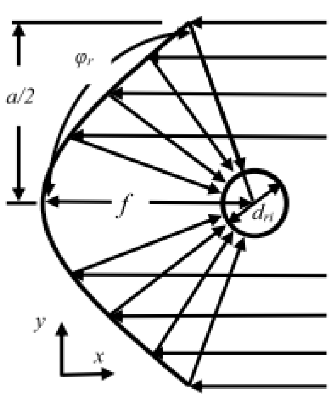

Figure 1 Parabolic trough collector system

The geometry of the parabolic trough collector is defined by the equation of the parabola as [13]

$$y^2 = 4fx \tag{1}$$

The focal length, f is also related to the rim angle, φ_r and aperture width, a as [13]

$$f = a/4\tan(\varphi_r/2) \tag{2}$$

The performance of the receiver significantly affects the performance of the entire parabolic trough system. Receiver thermal performance depends greatly on the receiver thermal loss. The thermal loss generally depends on the condition of the receiver. A receiver with an evacuated envelope performs better than a non-evacuated one as well as one without a glass cover (having a bare steel absorber tube) [4]. Also, the emissivity of the absorber tube's selective coating significantly influences the performance of the receiver [4]. Usually, the absorber tube is coated with a selective coating that gives it a high absorptance for the incoming solar radiation and lower emissivity for emitted infrared radiation. The general equations for the evaluation of receiver thermal loss are presented in Duffie and Beckman [13]. At steady state operating conditions, and for an evacuated receiver tube, the heat transferred from the absorber tube at a temperature T_{ro} to the receiver's glass cover at a temperature T_{gi} is equivalent to the heat transferred by conduction through the glass cover and is also equivalent to the heat transfer by combined radiation and convection heat transfer from the glass cover to the surroundings. As such, the receiver thermal loss can be taken as

$$\dot{Q}_{loss} = \frac{\pi d_{ro} L \sigma \left(T_{ro}^{\ 4} - T_{gi}^{\ 4}\right)}{\dfrac{1}{\varepsilon_{ro}} + \dfrac{1-\varepsilon_{gi}}{\varepsilon_{gi}}\left(\dfrac{d_{ro}}{d_{gi}}\right)} \tag{3}$$

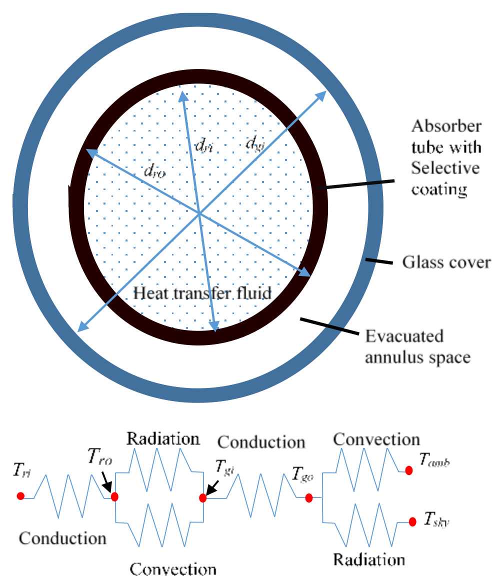

Figure 2 Receiver tube cross-section and thermal resistance network

In equation (3), ε_{gi} is the emissivity of the glass cover and ε_{ro} is the emissivity of the absorber tube. The emissivity of the glass cover, ε_{gi} was taken as 0.86 [14]. The emissivity of the absorber tube coating varies with temperature according to Burkholder and Kutscher's correlation [7] as

$$\varepsilon_{ro} = 0.062 + 2\times10^{-7} T_{ro}^2 \quad (4)$$

In equation (4), T_{ro} is the absorber tube temperature in ^{o}C. The thermal efficiency expression used incorporates the pumping power as suggested by Wirtz et al. [15] as

$$\eta_{th} = \frac{\dot{q}_u - \dot{W}_p / \eta_{el}}{I_b A_c} \quad (5)$$

In this, the electrical efficiency of the power block, η_{el} is taken as 32.7% [15]. The useful heat gain is, $\dot{q}_u = \dot{m} c_p \left(T_{outlet} - T_{inlet}\right)$ and the pumping power is $\dot{W}_p = \dot{V}\Delta P$.

SOLUTION PROCEDURE

The solution was obtained using both Monte-Carlo ray tracing and computational fluid dynamics. For Monte Carlo ray tracing, SolTrace [16], a ray tracing software from the U.S. National Renewable Energy Laboratory was used. In ray tracing analysis, the geometry and optical parameters of the system were specified. The sun shape was specified as a pillbox, 10^8 rays were set as the maximum number of generated sun rays while the desired number of ray interactions was set to 10^6. The collector slope error was taken as 3 mrad and specularity error of the mirror was taken as 0.5 mrad. The reflectivity of the mirror was taken as 0.97, the absorber tube absorptivity was taken as 0.96 and the transmissivity of the glass cover was taken as 0.96. The geometric concentration ratio, C_R was varied in turn from 88 to 113. C_R is defined as the ratio of the projected collector aperture area and the projected absorber tube area, i.e. $C_R = aL/d_{ro}L$. The absorber tube diameter, d_{ro}, was fixed at 80 mm and the collector aperture width, a was chosen as 7 m, 8 m and 9 m to give concentration ratios of 88, 100 and 113, respectively. The concentration ratios are selected to be higher than those used in current commercial plants, but comparable to those of the recently developed high concentration ratio systems [9, 10]. The rim angle was taken as φ_r = 80^{o}. The direct normal irradiance, DNI was set to 1 000 W/m^2.

The heat flux profiles obtained at each concentration ratio using ray tracing were coupled to a computational fluid dynamics code using user defined functions. This gave the thermal boundary condition on the receiver's absorber tube for the computational fluid dynamics analysis. The other boundary conditions used in the numerical analysis include: velocity inlet at the absorber tube inlet, pressure outlet at the absorber tube outlet, no-slip and no-penetration for all receiver tube walls, symmetry boundary condition for the annulus space inlet and outlet. A mixed radiation and convection boundary condition was used for the heat transfer between the receiver's glass cover, the ambient and the sky as specified in Mwesigye at al. [17]. The inlet velocities were selected to give a wide range flow rates between 1.22 m^3/h and 135 m^3/h so that the performance of the parabolic trough system at low and high flow rates is catered for. Moreover, flow rates in actual plants are within this range. For example, Forristall [14] developed a heat transfer model for a parabolic trough receiver in Engineering Equation Solver and compared his results with Test-Loop Data of SEGS plants at Kramer Junction, California (KJC) with flow rates between 21 m^3/h and 26 m^3/h. Other results in this study were presented for flow rates as high as 32 m^3/h.

ANSYS Fluent® [18] software was used to numerically investigate the thermal and thermodynamic performance of the receiver tube. The Reynolds averaged Navier-stokes equations [18] together with the boundary conditions were solved using this software. The SIMPLE algorithm was used for pressure-velocity coupling, second order upwind schemes were used for integrating the governing equations together with the boundary conditions. The realisable k-ε model was used for turbulence closure while the enhanced wall treatment option was selected for modelling the near-wall regions. The solution was taken to be converged when the scaled residuals for continuity were less than 10^{-4}, scaled residuals for momentum, turbulent kinetic energy and turbulent dissipation rates were less than 10^{-5} and the scaled residuals for energy were less than 10^{-7}. More detailed information regarding the governing equations used and ray tracing analysis can be found in previous investigations, i.e. Mwesigye et al.[11,17].

Validation of the Numerical Model

The results of our study were compared with data available in literature for optical performance, receiver thermal loss and thermal efficiency. The optical performance and thermal performance validation were presented in an earlier study [11]. The variation of receiver thermal loss with temperature is shown in Figure 3 in comparison with results from Dreyer et al. [19]. As seen, the present study results are within less than 3% of the experimental results presented by Dreyer et al. [19]. This, together with validations done in an earlier study [11] show that our numerical model is an excellent approximation of the receiver thermal model.

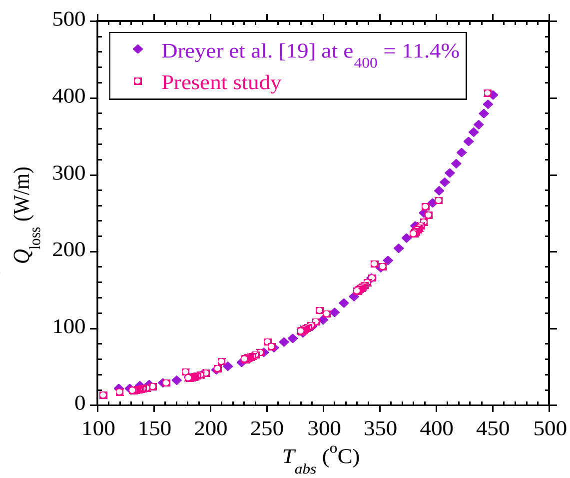

Figure 3 Comparison of current study receiver thermal model with experimental measurements by Dreyer et al. [19]

THERMAL PERFORMANCE

The thermal performance of the parabolic trough system is presented in terms of the receiver thermal/heat loss, the thermal efficiency and the energy transfer by heat from the receiver to the working fluid also called the useful heat gain. As shown in Figure 4, the receiver thermal loss increases as the concentration ratio increases. This is due to the fact that, at a given flow rate, higher concentration ratios result in higher absorber tube temperatures and thus a higher radiation heat loss between the absorber tube and the receiver's glass cover. Closely looking at the graph of receiver thermal loss shown in Figure 4, shows that the receiver thermal loss increases by about 22% at low flow rates of about 12 m^3/h, by about 9% at flow rates of about 25 $m^{3/}h$ and by about 5% at flow rates above 30 m^3/h as the concentration ratio increases from 88-113. Thus, the increase in receiver thermal loss is not significant at higher flow rates as concentration ratios increase. With higher concentration ratios, high flow rates will be necessary or heat transfer enhancement can be considered to reduce the receiver thermal loss considerably. The same trend shown in Figure 4 exists at other temperatures, but for a given flow rate, the thermal loss increases as the inlet temperatures increase owing to the higher radiation thermal losses at elevated temperatures.

With this increase in the receiver thermal loss as the concentration ratios increase, it is expected that the thermal efficiency will also reduce as concentration ratios increase. As shown in Figures 5 and 6 at inlet temperatures of 400 K and 600 K respectively, the thermal efficiency reduces as the concentration ratio increases. This is expected since at a given flow rate, higher concentration ratios will result in higher absorber tube temperatures thus a higher receiver thermal loss.

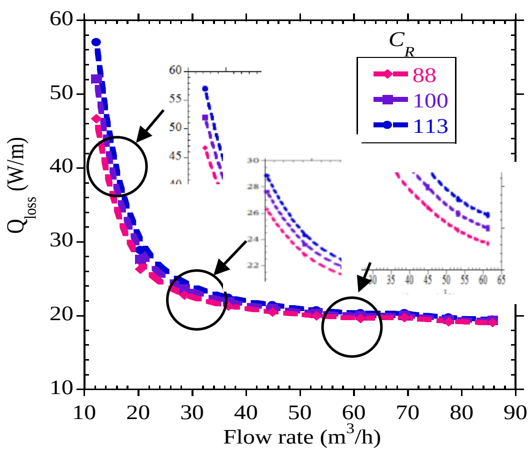

Figure 4 Receiver thermal loss as a function of flow rate and concentration ratio at an inlet temperature of 400 K

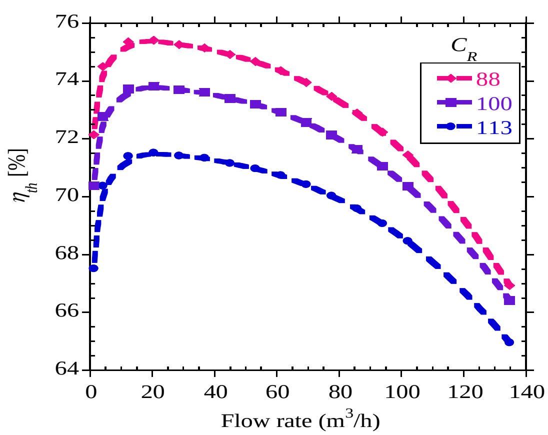

Figure 5 Thermal efficiency as a function of flow rate and concentration ratio at an inlet temperature of 400 K

The same trend exists at the other temperatures, but the thermal efficiency is higher at low inlet temperatures and lower at higher inlet temperatures for comparable values of flow rates. This is in line with the variation of receiver thermal loss with inlet temperature. The higher the inlet temperature, the higher the absorber tube temperature and thus the higher the receiver thermal loss.

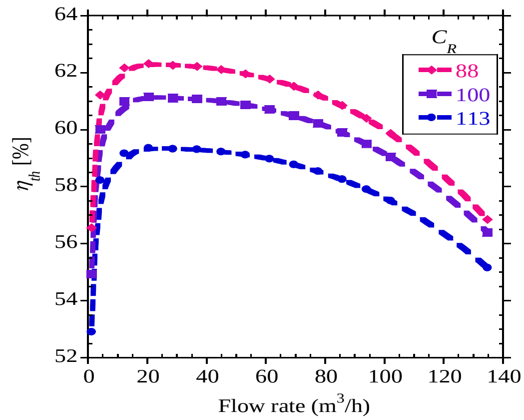

Figure 6 Thermal efficiency as a function of flow rate and concentration ratio at an inlet temperature of 600 K

Even though the increase in the concentration ratio results in higher receiver thermal loss and lower thermal efficiencies, the resulting heat transfer fluid temperatures will be higher and thus, the useful heat gain will increase as concentration ratios increase. This means that with higher concentration ratios, required heat transfer fluid outlet temperatures can be obtained with shorter solar collector assemblies. Figure 7 shows the variation of the useful heat gain as a function of concentration ratio and flow rate at an inlet temperature of 400 K.

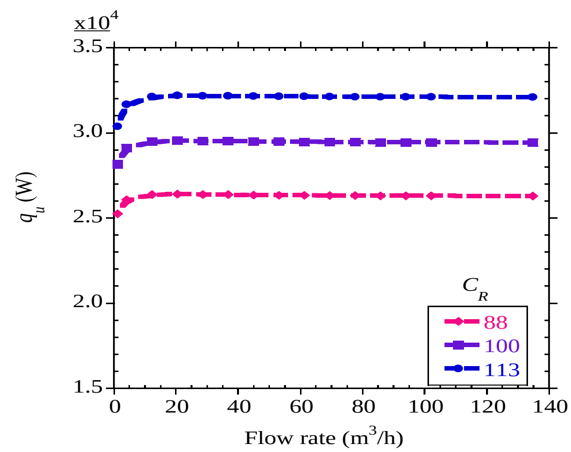

Figure 7 Useful heat as a function of flow rate and concentration ratio at an inlet temperature of 400 K

The useful heat gain is shown to increase as the flow rates increase up to about 25 m^3/ h and becomes constant thereafter. This is because the heat transfer fluid outlet temperature becomes almost constant as flow rates increase beyond this value. As expected, outlet temperatures increase as the concentration ratio increases. Thus, the useful heat gain increases as the concentration ratio increases as shown.

THERMODYNAMIC PERFORMANCE

The thermodynamic performance was investigated by considering the irreversibilities present in the receiver and the subsequent entropy generation rates. The entropy generation rate is a sum of the heat transfer irreversibility and the fluid flow irreversibility. Detailed determination of these irreversibilities and the calculation procedure for the entropy generation rate is presented in Mwesigye et al.[11]. Figure 8 shows the variation of the entropy generation rate with Reynolds number and concentration ratio. The entropy generation rate is high at low values of Reynolds numbers due to the high heat transfer irreversibilities. It reduces as Reynolds numbers increase owing to the improved heat transfer and consequently reduced finite temperature differences attains a minimum and increases again with a further increase in the Reynolds number as fluid friction irreversibilities increase and become larger than heat transfer irreversibilities. The figure also shows that the entropy generation rate increases as the concentration ratio increases. This is in agreement with previous studies [11,12]. The increase in the entropy generation rate is attributed to the increasing finite temperature differences as the concentration ratios increase.

The increasing finite temperature difference results in an increase of the heat transfer irreversibility and thus the entropy generation rates. This can be shown by the plot of the Bejan number at different concentration ratios. The Bejan number is defined as the ratio of the heat transfer irreversibility to the total entropy generation rate. As shown in Figure 9, the Bejan number increases as the concentration ratio increases, indicating that the increase in entropy generation is mainly due to the increasing heat transfer irreversibility. Using high flow rates/Reynolds numbers reduces the heat transfer irreversibility at a given concentration ratio. Heat transfer enhancement can also be used to reduce the heat transfer irreversibilities and thus the entropy generation rates at a given concentration ratio.

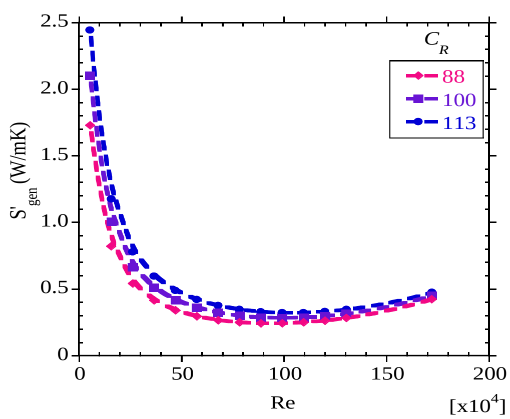

Figure 8 Entropy generation rate in the receiver as a function of Reynolds number and concentration ratio at an inlet temperature of 500 K

POTENTIAL FOR IMPROVED PERFORMANCE WITH CU-THERMINOL®VP-1 NANOFLUID

A number of studies on heat transfer enhancement in parabolic trough receivers have shown that there is potential for improved performance. The use of nanofluids for enhancement of the convective heat transfer performance of heat exchangers is another area that is receiving considerable attention. Some recent studies have shown that using nanofluids in parabolic trough receivers improves their performance [17,20,21]. In this study, the potential for improved performance with Cu-Therminol®VP-1 nanofluid is investigated preliminarily. A single phase homogeneous modelling approach was used for this purpose. This approach together with the equations for the determination of nanofluid density, specific heat, thermal conductivity and viscosity is described in Mwesigye et al. [17]. As shown in Figure 10, the use of nanofluids improves the thermal performance of the receiver. In this figure, the thermal efficiency obtained using equation (5) increases with Reynolds number, reaches a maximum at some Reynolds number and reduces again for any given value of volume fraction. The thermal efficiency is also shown to increase as the volume fraction increases, especially when Reynolds numbers are low. For the concentration ratios used in this study and depending on the inlet temperature considered, the thermal efficiency at higher volume fractions might be lower than that at lower volume fractions as Reynolds numbers increase. This occurs at much lower Reynolds numbers when the inlet temperature is lower and at higher Reynolds numbers when the inlet temperature is higher. For the range of parameters considered, the thermal efficiency increases by about 12% as the volume fraction increases from 0-6%. A higher increase in thermal efficiency occurs at lower Reynolds numbers and higher inlet temperatures.

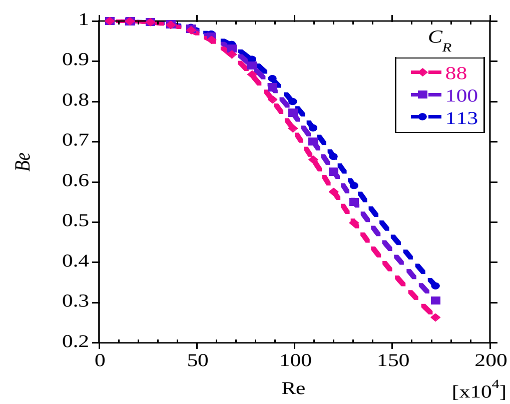

Figure 9 Bejan number as a function of Reynolds number and concentration ratio at an inlet temperature of 500 K

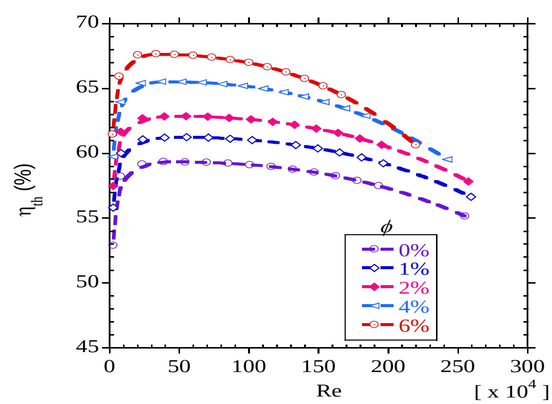

Figure 10 Thermal efficiency as a function of Reynolds number and volume fraction at an inlet temperature of 600 K

The potential for improved thermodynamic performance with heat transfer enhancement using Cu-Therminol®VP-1 nanofluid is shown using the entropy generation rate. The variation of the entropy generation rate with Reynolds number and volume fraction is shown in Figure 11. As shown, the entropy generation rate reduces as the volume fraction increases at low values of Reynolds numbers mainly due to improved heat transfer performance and reduced heat transfer irreversibilities. As Reynolds numbers increase, the entropy generation rate reduces, attains a minimum at some Reynolds number and increases again. Beyond the value of Reynolds number at which the entropy generation rate is a minimum, the entropy generation rate increases as the volume fraction increases. This is due to high fluid friction irreversibilities as the volume fraction increases at high Reynolds numbers.

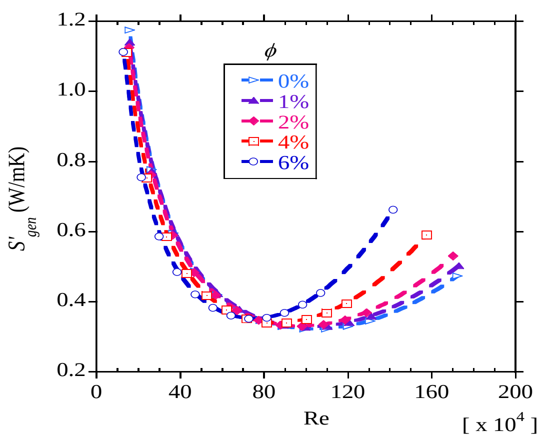

Figure 11 Entropy generation rate as a function of Reynolds number and volume fraction at an inlet temperature of 500 K

CONCLUSION

In this work, the influence of concentration ratio on the performance of a parabolic trough receiver is presented. Even though, the useful heat energy from the receiver increases with the increase in concentration ratio, the thermal efficiency of the receiver is shown to reduce with increasing concentration ratios at any given flow rate. The thermal efficiency reduces by about 4.5% as the concentration ratio increases from 88 to 113. The entropy generation rates were also shown to increase as the concentration ratio increased.

From preliminary results on heat transfer enhancement with Cu-Therminol®VP-1 nanofluid, the potential for improved performance was investigated. At a concentration ratio of 113, the thermal efficiency increased by up to 12% as the nanoparticle volume fraction increased from 0 to 6%. The highest increase in thermal efficiency was observed at the largest value of the inlet temperature considered in this study. The improvement in receiver thermal performance is mainly due to improved heat transfer performance as well as the subsequent reduction in receiver thermal loss due to lower absorber tube temperatures.

The thermodynamic performance of the receiver also improves with heat transfer enhancement provided the flow rate is lower than some value (about 45 m^3/h). Above this value, increasing the volume fraction makes the entropy generation rates higher than those in a receiver with only the base fluid.

REFERENCES

[1] Richter C., Teske S., and Short R., Concentrating solar power, Global outlook 09, *Greenpeace/SolarPACES/ESTELA*, 2009, pp.1-86

[2] Philibert C., and Frankl P., Technology roadmap: concentrating solar power, *IEA/OECD*, Paris, France, 2010, pp. 1- 45

[3] Price H., Lüpfert E., Kearney D., Zarza E., Cohen G., Gee R., and Mahony R., Advances in parabolic trough solar power technology, *Journal of Solar Energy Engineering*, Vol.124, 2002, pp.109-125

[4] Dudley E.V., Kolb J.G., Mahoney A.R., Mancini T.R., Sloan M., and Kearney D., Test results: SEGS LS-2 solar collector, S*andia National Laboratory Technical Report*, SAND94-1884, 1994

[5] Dudley E.V., Evans R.L., and Mathews W.C., Test results: Industrial Solar Technology parabolic trough solar collector, *Sandia National Laboratory Technical Report*, SAND94-1117, 1995

[6] Burkholder F., and Kutscher C., Heat-loss testing of Solel's UVAC3 parabolic trough receiver, *NREL Technical Report*, NREL/TP - 550-42394, 2008, pp.1-19

[7] Burkholder F., and Kutscher C., Heat loss testing of Schott's 2008 PTR70 parabolic trough receiver, *NREL Technical Report*, NREL/TP - 550-45633, 2009, pp.1-58

[8] SunShot Initiative, http://www1.eere.energy.gov/solar/sunshot/index.html, US Department of Energy [Accessed 12.12.2013]

[9] Schweitzer A., Schiel W., Birkle M., Nava P., Riffelmann K., Wohlfahrt A., and Kuhlmann G., ULTIMATE TROUGH® - Fabrication, erection and commissioning of the world's largest parabolic trough collector, *Energy Procedia*, Vol. 49, 2014, pp.1848-1857

[10] SkyFuel® 2014, Development of a high concentration, Low cost parabolic trough system for baseload CSP generation, Phase II Report, Prepared under contract to the Department of Energy, contract No. DE-EE0003584.000

[11] Mwesigye A., Bello-Ochende T., and Meyer J.P., Minimum entropy generation due to heat transfer and fluid friction in a parabolic trough receiver with non-uniform heat flux at different rim angles and concentration ratios, *Journal of Energy*, Vol.73, 2014, pp.606-617

[12] Mwesigye A., Bello-Ochende T., and Meyer J.P., Numerical investigation of entropy generation in a parabolic trough receiver at different concentration ratios, *Journal of Energy*, Vol.53, 2013, pp.114-127

[13] Duffie J.A. and Beckman W.A., Solar engineering of thermal processes, 3rd ed. Hoboken, New Jersey, *John Wiley and Sons Inc*., 2006

[14] Forristall R., Heat transfer analysis and modeling of a parabolic trough solar receiver implemented in Engineering Equation solver. *NREL Technical Report*, NREL/TP-550-34169, October 2003, pp.1-145

[15] Wirz M., Petit J., Haselbacher A., and Steinfeld A., Potential improvements in the optical and thermal efficiencies of parabolic trough concentrators, *Journal of Solar Energy*, Vol.107, 2014, pp.398-414

[16] NREL. SolTrace optical modelling software. SolTrace 2012;2012.7.9

[17] Mwesigye A., Huan Z., and Meyer J.P., Thermodynamic optimisation of the performance of a parabolic trough receiver using synthetic oil–Al2O3 nanofluid, *Journal of Applied Energy*, Vol.156, 2015, pp.398-412

[18] ANSYS® Academic research, release 15, ANSYS FLUENT, user guide, ANSYS, Inc

[19] Dreyer S., Eichel P., Gnaedig T., Hacker Z., Janker S., Kuckelkorn T., Silmy K., Pernpeintner J., and Luepfert E., Heat loss measurements on parabolic trough receivers. *Proceedings 16th SolarPACES conference* 2010, Perpignan, France, September 2010

[20] Taylor A.R., Phelan E.P., Otanicar P.T., Walker A.C., Nguyen M., Trimble S., and Prasher R., Applicability of nanofluids in high flux solar collectors, *Journal of Renewable Sustainable Energy*, 2011, pp. 023104:1-15

[21] Sokhansefat T., Kasaeian A.B., and Kowsary F., Heat transfer enhancement in parabolic trough collector tube using Al2O3/synthetic oil nanofluid. *Journal of Renewable and Sustainable Energy Reviews* Vol. 33, 2014, pp.636-644