Energy 53 (2013) 114–127

Contents lists available at SciVerse ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

Numerical investigation of entropy generation in a parabolic trough receiver at different concentration ratios

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ARTICLE INFO

Article history:
Received 13 September 2012
Received in revised form
1 January 2013
Accepted 3 March 2013
Available online 3 April 2013

Keywords:
Bejan number
Entropy generation
Fluid friction irreversibility
Heat transfer irreversibility
Parabolic trough
Receiver

ABSTRACT

This paper presents results of a numerical analysis of entropy generation in a parabolic trough receiver at different concentration ratios, inlet temperatures and flow rates. Using temperature dependent thermal properties of the heat transfer fluid, the entropy generation due to heat transfer across a finite temperature difference and entropy generation due to fluid friction in the receiver has been determined. Results show a reduction in the entropy generation rate as the inlet temperature increases and an increase in the entropy generation rate as the concentration ratio increases. Results further show that, there is an optimal flow rate at which the entropy generated is a minimum, for every combination of concentration ratio and inlet temperature. The optimal flow rates at which the entropy generated is minimum are presented for different flow rate and concentration ratio, and the results are the same irrespective of the inlet temperature considered. For the range of inlet temperatures, flow rates and concentration ratios considered, the Bejan number, which measures the contribution of entropy generation due to heat transfer irreversibility to the total entropy generation rate is about 1 at low flow rates and is between 0 and 0.24 at the highest flow rate.

1. Introduction

Solar resource is the world's most abundant source of energy with the potential to meet a significant portion of the world's energy requirements [1]. For high-temperature requirements, concentrated solar power (CSP) systems are usually used; the solar radiation collecting (receiving) area is larger than the heat collection area which reduces heat losses [1,2]. Parabolic trough collector technology is the most economic and commercially developed of the available concentrated solar power systems [3] especially after the construction of nine Solar Electric Generating Systems (SEGS) in the Mojave Desert in Southern California in the period between 1984 and 1990 [3,4]. Parabolic trough collectors consist of a reflecting element bent into a parabolic shape which focuses incoming solar radiation onto a tubular receiver or heat collection element together with supporting structures.

A number of studies have been carried out to determine the performance of parabolic trough collectors. Dudley et al. [5,6] used the AZTRAK rotating platform at SANDIA National Laboratories to study the performance of SEGS LS-2 and industrial solar technology solar collectors respectively. Liu et al. [7] developed an experimental platform to investigate parabolic trough performance. They obtained collector efficiencies between 40 and 60% and heat losses of about 220 W/m at an absorber-ambient temperature difference of 180 °C. Odeh and Morrison [8] developed a computer model for estimating the transient performance of a solar industrial water heating system. They have shown that for stable operation during transient radiation periods the thermal storage tank size should higher than 14.51 m^{-2} of the collector area. Lupfert et al. [9] measured the thermal losses of Solel UVAC and Schott PTR70 receivers, Burkholder and Kutscher [10,11] used steady-state tests to determine heat losses for Solel UVAC and Schott's PTR70 parabolic trough receivers respectively. The heat losses were found to increase as the absorber tube temperatures increased [9–11]. For example, the Solel UVAC receiver losses normalised per metre were between 15 and 460 W/m at average absorber temperatures between 100 °C and 450 °C [10]. Field measurements of glass temperatures were done using a solar-blind infrared camera by Price et al. [12] at the SEGS plants with over 12,000 receivers monitored. Forristall [13] developed a heat transfer model for determining the performance of a parabolic trough receiver implemented in Engineering Equation Solver (EES). The results were comparable with the experimental results of Dudley et al. [5].

For rim angles lower than 90°, only the lower half (lower half being the one facing the reflecting surface) receives concentrated

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http://dx.doi.org/10.1016/j.energy.2013.03.006

Nomenclature

a	collector's aperture width, m
A_a	collector's aperture area, m^2
A_c	absorber tube's cross-section area, m^2
A_r	projected absorber tube area, m^2
Be	Bejan Number = entropy generated due to heat transfer/total entropy generated
C_1, C_2, C_μ	turbulent model constants
c_f	skin friction coefficient
c_p	specific heat capacity, J kg^{-1} K^{-1}
C_R	concentration ratio
d_{gi}	inner glass diameter, m
d_{ri}	absorber inner diameter, m
d_{ro}	absorber outer diameter, m
d_r	absorber tube diameter, m
D	hydraulic diameter, m
DNI	direct normal irradiance, W/m^2
G	mass flux, kg s^{-1} m^{-2}
G_k	generation of turbulence kinetic energy due to mean velocity gradients, kg m^{-1} s^{-3}
g	acceleration due to gravity, m s^{-2}
h	heat transfer coefficient, W m^{-2}K^{-1}
I_b	direct solar radiation, W m^{-2}
k	turbulent kinetic energy, m^2 s^{-2}
L	receiver length, m
$\dot{m}$	fluid mass flow rate, kg/s
Nu	Nusselt number
N_s	non-dimensional entropy generation number
p	pressure, Pa
$\dot{q}$	heat transfer rate, W
q'	heat transfer rate per meter length, W m^{-1}
q''	heat flux, W m^{-2}
Q	heat transfer to the collector, W
Q^*	heat transfer from the sun to the collector, W
Q_o	collector heat losses = $Q^* - Q$, W
r	radial position, m
R	radius, m
Re	Reynolds number
S	modulus of the mean rate-of-strain tensor, s^{-1}
S_{ij}	rate of linear deformation tensor, s^{-1}
S_{gen}	entropy generation rate due to heat transfer and fluid friction in the receiver, W/K
S'_{gen}	entropy generation due to heat transfer and fluid friction per unit length of the receiver, W/mK
$S'_{gen,col}$	entropy generation per unit length of the parabolic trough collector, W/K
S'''_{gen}	volumetric entropy generation, W m^{-3}K^{-1}
$(S'''_{gen})_F$	entropy generation due to fluid friction, W m^{-3}K^{-1}
$(S'''_{gen})_H$	entropy generation due to heat transfer, W m^{-3}K^{-1}
$S'''_{PROD,VD}$	entropy production by direct dissipation, W m^{-3}K^{-1}
$S'''_{PROD,TD}$	entropy production by turbulent dissipation, W m^{-3}K^{-1}
$S'''_{PROD,T}$	entropy production by heat transfer with mean temperatures, W m^{-3}K^{-1}
$S'''_{PROD,TG}$	entropy production by heat transfer with fluctuating temperatures, W m^{-3}K^{-1}
T	temperature, K
T_o	ambient temperature
T_r	receiver temperature, K
T_s	apparent black body temperature of the sun, K
T_*	apparent temperature of the sun as an energy source = $^3/_4\ T_s$, K
u	velocity, m s^{-1}
U_∞	mean flow velocity, m s^{-1}
V	volume, m^3
$\dot{V}$	volumetric flow rate, m^3/s
u_i,u_j	averaged velocity components, m s^{-1}
u',v',w'	velocity fluctuations, m s^{-1}
u_τ	friction velocity, m s^{-1}
x_i, x_j	spatial coordinates, m
x,y,z	Cartesian coordinates
y^+	dimensionless wall coordinate
$-\rho\overline{u'_i u'_j}$	Reynolds stresses, Nm^{-2}

Greek letters

α	thermal diffusivity, m^2 s^{-1}
α_t	turbulent thermal diffusivity, m^2 s^{-1}
$\sigma_{h.t}$	turbulent Prandtl number for energy
σ_ε	turbulent Prandtl number for ε
σ_k	turbulent Prandtl number for k
δ_{ij}	Kronecker delta
ε	turbulent dissipation rate, m^2 s^{-3}
ξ	emissivity
η	turbulence model parameter = Sk/ε
η_0	optical efficiency, %
ρ	density, kg m^{-3}
τ_g	glass cover transmissivity
θ	angular position, degrees
λ	fluid thermal conductivity, Wm^{-1} K^{-1}
λ_{eff}	heat transfer fluid effective thermal conductivity, Wm^{-1} K^{-1}
μ	viscosity, Pa s
μ_t	turbulent viscosity, Pa s
μ_{eff}	effective viscosity, Pa s
ν	kinematic viscosity, m^2 s^{-1}

Subscripts

inlet	absorber tube inlet
i, j, k	general spatial indices
t	turbulent
w	wall
out	absorber tube outlet
bulk	bulk fluid state
d	diameter
ro	absorber tube outer wall
ri	absorber tube inner wall

solar radiation. The presence of a differential flux and thus differential temperature in the absorber tube's circumference has been noted in studies by Refs. [14,15]. Munoz and Abanades [16] investigated an internally helically finned absorber tube with a view of evening out the non-uniform absorber circumferential temperatures. Meanwhile, other receiver performance enhancements have been studied as reported in the studies by Hegazy [17], for externally finned receiver tubes; Reddy et al. [18], for a receiver with a porous fin and longitudinal fins and Kumar and Reddy [19], for a receiver with a porous disc at different angles.

Renewed interest in CSP in the last two decades has led to increased research and as a result improved plant components have been developed. Price et al. [3] present a review of the research and developments regarding parabolic trough collectors. With these developments, the cost of electricity from parabolic trough collectors has reduced significantly and further cost reductions are