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Heat transfer and thermodynamic performance of a parabolic trough receiver with centrally placed perforated plate inserts

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HIGHLIGHTS

- Heat transfer enhancement of a parabolic trough receiver with perforated plate inserts is studied.
- Effect of insert geometrical parameters on receiver thermal performance is investigated.
- Correlations for Nusselt number and friction factor performance are derived and presented.
- Performance evaluation using enhancement factors and collector modified thermal efficiency was demonstrated.
- Thermodynamic performance is investigated using the entropy generation minimization method.

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ABSTRACT

In this paper, a numerical investigation of thermal and thermodynamic performance of a receiver for a parabolic trough solar collector with perforated plate inserts is presented. The analysis was carried out for different perforated plate geometrical parameters including dimensionless plate orientation angle, the dimensionless plate spacing, and the dimensionless plate diameter. The Reynolds number varies in the range $1.02 \times 10^4 \leqslant Re \leqslant 7.38 \times 10^5$ depending on the heat transfer fluid temperature. The fluid temperatures used are 400 K, 500 K, 600 K and 650 K. The porosity of the plate was fixed at 0.65. The study shows that, for a given value of insert orientation, insert spacing and insert size, there is a range of Reynolds numbers for which the thermal performance of the receiver improves with the use of perforated plate inserts. In this range, the modified thermal efficiency increases between 1.2% and 8%. The thermodynamic performance of the receiver due to inclusion of perforated plate inserts is shown to improve for flow rates lower than 0.01205 m^3/s. Receiver temperature gradients are shown to reduce with the use of inserts. Correlations for Nusselt number and friction factor were also derived and presented.

1. Introduction

Parabolic trough solar collectors are one of the most technically and commercially developed technologies of the available concentrated solar power technologies [1,2]. The parabolic trough's linear receiver is a central component to the performance of the entire collector system. Its state and design greatly affects the performance of the entire collector system. The performance of the receiver is significantly affected by the thermal loss and heat transfer from the absorber tube to the working (heat transfer) fluid [3]. The conventional receiver consists of an evacuated glass envelope to minimize the convection heat loss and a selectively coated absorber tube to minimize the radiation heat loss [2]. Numerous studies have been carried out to characterize the thermal performance of the receiver and to determine the thermal loss at different receiver conditions [4–9]. From these studies, it has been shown that: the thermal loss is majorly dependent on the state of the annulus space between the glass cover and the absorber tube, the absorber tube selective coating, the temperature of the absorber tube, the wind speed and the heat transfer from the absorber tube to the heat transfer fluid.

With the availability of lightweight materials, the use of higher concentration ratios has become feasible [10]. Higher concentration ratios ensure shorter and less expensive collectors given the reduction in the number of drives and connections required. However, larger concentration ratios mean increased entropy generation rates [11], increased absorber tube circumferential temperature gradients as well higher peak temperatures.

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Nomenclature

A	area, m^2
A_a	collector's projected aperture area, m^2
a_c	collector aperture width, m
A_r	absorber tube's projected area, m^2
Be	Bejan number
C_{2p}	inertial resistance factor, m^{-1}
c_p	specific heat capacity, $J\,kg^{-1}\,K^{-1}$
C_R	concentration ratio
d	perforated plate diameter, m
d_{gi}	glass cover inner diameter, m
d_{go}	glass cover outer diameter, m
d_{ri}	absorber tube inner diameter, m
d_{ro}	absorber tube outer diameter, m
DNI	direct normal irradiance, $W\,m^{-2}$
f	Darcy friction factor
h	heat transfer coefficient, $W\,m^{-2}\,K^{-1}$
h_w	glass cover outer heat transfer coefficient, $W\,m^{-2}\,K^{-1}$
I_b	direct solar radiation, $W\,m^{-2}$
k	turbulent kinetic energy per unit mass, $m^2\,s^{-2}$
L	receiver length, m
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number
$N_{s,en}$	entropy generation ratio = $S_{gen}/(S_{gen})_o$
P	pressure, Pa
p	perforated plate spacing, m
Pr	Prandtl number
q''	heat flux, $W\,m^{-2}$
$\dot{Q}_u$	heat transfer rate (W)
r	radial position, m
Re	Reynolds number
S_{gen}	entropy generation rate due to heat transfer and fluid friction, W/K
S'_{gen}	entropy generation rate per unit meter (W/m K)
S_m	momentum source term
$(S_{gen})_H$	entropy generation due to heat transfer, W/K
$(S_{gen})_F$	entropy generation due to fluid friction, W/K
S'''_{gen}	volumetric entropy generation, $W\,m^{-3}\,K^{-1}$
$(S'''_{gen})_F$	volumetric entropy generation due to fluid friction, $W\,m^{-3}\,K^{-1}$
$(S'''_{gen})_H$	volumetric entropy generation due to heat transfer, $W\,m^{-3}\,K^{-1}$
$S'''_{PROD,VD}$	volumetric entropy production by direct dissipation, $W\,m^{-3}\,K^{-1}$
$S'''_{PROD,TD}$	volumetric entropy production by turbulent dissipation, $W\,m^{-3}\,K^{-1}$
$S'''_{PROD,T}$	volumetric entropy production by heat transfer with mean temperatures, $W\,m^{-3}\,K^{-1}$
$S'''_{PROD,TG}$	volumetric entropy production by heat transfer with fluctuating temperatures, $W\,m^{-3}\,K^{-1}$
T	temperature, K
u,v,w	velocity components, m/s
V	volume, m^3
V_w	wind velocity, m/s
$\dot{V}$	volume flow rate, m^3/s
$\dot{W}_p$	pumping power, W
u_i, u_j	averaged velocity components, m/s
u'_i, u'_j	velocity fluctuations, m/s
x_i, x_j	spatial coordinates, m
x, y, z	Cartesian coordinates
y^+	dimensionless wall coordinate
$-\rho\overline{u'_i u'_j}$	Reynolds stresses, $N\,m^{-2}$
∇p	pressure drop, Pa
Δm	perforated plate thickness, m

Greek letters

α	thermal diffusivity, $m^2\,s^{-1}$
α_{abs}	absorber tube absorptivity
α_p	permeability of the perforated plate, m^2
α_t	turbulent thermal diffusivity, $m^2\,s^{-1}$
σ_e	slope error, mrad
$\sigma_{h.t}$	turbulent Prandtl number for energy
β	plate orientation angle, degrees
δ_{ij}	Kronecker delta
ε	turbulent dissipation rate, $m^2\,s^{-3}$
ξ	emissivity
ϕ	absorber tube temperature gradient, °C
φ_r	collector rim angle, degrees
ρ	density, $kg\,m^{-3}$
$\bar{\rho}$	collector reflectance
λ	fluid thermal conductivity, $W\,m^{-1}\,K^{-1}$
$\eta_{th,m}$	modified thermal efficiency, %
τ_g	glass cover transmissivity
τ_w	wall shear stress
θ	receiver angle, degrees
μ	viscosity, Pa s
μ_t	turbulent viscosity, Pa s
μ_τ	friction velocity, m/s
μ_{eff}	effective viscosity, Pa s
v	kinematic viscosity, $m^2\,s^{-1}$
χ	thermal enhancement factor = $Nu/(Nu)_o/(f/f_o)^{1/3}$

Subscripts

amb	ambient state
abs	absorber tube
abs, max	absorber tube maximum temperature
b	bulk fluid state
gi	inner glass cover wall
go	outer glass cover wall
i, j, k	general spatial indices
inlet	absorber tube inlet
max	maximum value
o	reference case (plain absorber tube – no inserts)
outlet	absorber tube outlet
ro	absorber tube outer wall
ri	absorber tube inner wall
sky	sky temperature
t	turbulent
w	wall

Superscripts

¯	mean value
~	dimensionless value
′	fluctuation from mean value

The presence of circumferential temperature gradients in the receiver's absorber tube is a major concern. At low flow rates, higher temperature gradients existing in the tube's circumference can cause bending of the tube and eventual breakage of the glass cover [12,13]. And the peak temperature in the absorber tube facilitate degradation of the heat transfer fluid especially as these temperatures increase above 673.15 K [14,15]. The degradation of the heat transfer fluid results in hydrogen permeation in the receiver's