Applied Energy 138 (2015) 337–345

Contents lists available at ScienceDirect

Applied Energy

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

An air-based corrugated cavity-receiver for solar parabolic trough concentrators

Roman Bader [a,*], Andrea Pedretti [b], Maurizio Barbato [c], Aldo Steinfeld [a,d]

[a] Department of Mechanical and Process Engineering, ETH Zurich, Sonneggstrasse 3, 8092 Zurich, Switzerland
[b] Airlight Energy Holding SA, Via Croce 1, 6710 Biasca, Switzerland
[c] Department of Innovative Technologies, SUPSI, Galleria 2, 6928 Manno, Switzerland
[d] Solar Technology Laboratory, Paul Scherrer Institute, WSAD/A7, 5232 Villigen PSI, Switzerland

HIGHLIGHTS

- We analyze a novel tubular cavity-receiver for solar parabolic trough collectors.
- Four-fold solar concentration ratio is reached compared to conventional receivers.
- Efficient operation at up to 500 °C is possible.
- The pumping power requirement is found to be acceptably low.

ARTICLE INFO

Article history:
Received 18 October 2012
Received in revised form 10 October 2014
Accepted 22 October 2014

Keywords:
Solar
Parabolic trough
Cavity
Receiver
Air
Heat transfer

ABSTRACT

A tubular cavity-receiver that uses air as the heat transfer fluid is evaluated numerically using a validated heat transfer model. The receiver is designed for use on a large-span (9 m net concentrator aperture width) solar parabolic trough concentrator. Through the combination of a parabolic primary concentrator with a nonimaging secondary concentrator, the collector reaches a solar concentration ratio of 97.5. Four different receiver configurations are considered, with smooth or V-corrugated absorber tube and single- or double-glazed aperture window. The collector's performance is characterized by its optical efficiency and heat loss. The optical efficiency is determined with the Monte Carlo ray-tracing method. Radiative heat exchange inside the receiver is calculated with the net radiation method. The 2D steady-state energy equation, which couples conductive, convective, and radiative heat transfer, is solved for the solid domains of the receiver cross-section, using finite-volume techniques. Simulations for Sevilla/Spain at the summer solstice at solar noon (direct normal solar irradiance: 847 W m^{-2}, solar incidence angle: 13.9°) yield collector efficiencies between 60% and 65% at a heat transfer fluid temperature of 125 °C and between 37% and 42% at 500 °C, depending on the receiver configuration. The optical losses amount to more than 30% of the incident solar radiation and constitute the largest source of energy loss. For a 200 m long collector module operated between 300 and 500 °C, the isentropic pumping power required to pump the HTF through the receiver is between 11 and 17 kW.

1. Introduction

A novel solar parabolic trough collector has been proposed, with the goal to substantially cut the costs per unit of collected solar power through an increase in size of the collector and the consequent use of low-cost materials, including polymer mirror membranes, concrete frames, and air as the heat transfer fluid (HTF) [1–3]. The use of air further avoids operating temperature constraints due to chemical instability of the HTF and allows for the direct coupling of the solar collector with a packed bed thermal storage [4–6]. However, the lower volumetric heat capacity and convective heat transfer of air compared to conventional HTFs, such as thermo-oil and molten salt, require receivers with larger tube diameter and higher heat transfer area than those found in conventional receivers of solar parabolic trough collectors [7]. A tubular cavity-receiver design allows for an increased tube diameter and higher heat transfer area without increasing the re-radiation area (in this case the cavity's aperture area). In combination with a linear secondary concentrator, a cavity-receiver can reach four-fold solar concentration ratio and, hence, four times

* Corresponding author at: Research School of Engineering, The Australian National University, Canberra, ACT 2601, Australia. Tel.: +61 2 612 53852.
E-mail address: roman.bader@anu.edu.au (R. Bader).

http://dx.doi.org/10.1016/j.apenergy.2014.10.050
0306-2619/

Nomenclature

Latin symbols

A	absorptance
CSR	circumsolar ratio
D_h	hydraulic diameter (m)
d	thickness (m)
E	emittance
F	view factor
G_b	direct normal solar irradiance (W m^{-2})
h	convective heat transfer coefficient (W m^{-2} K^{-1})
$h_{corrugation}$	corrugation depth (m)
i, j, k	indices
K	incidence angle modifier
m	mass flow rate (kg s^{-1}); number of segments in discretization
N_{nodes}	total number of discretization nodes in receiver cross-section
N_{seg}	total number of surface segments in enclosure
Nu	Nusselt number
n	number of segments in discretization
p	pressure (Pa)
Q'	heat rate per unit of receiver length (W m^{-1})
q	heat flux (W m^{-2})
R	reflectance
R_{cavity}	cavity inner radius (m)
R_{HTF}	specific gas constant of air (J kg^{-1} K^{-1})
T	temperature (K)
V	transmittance
W	mechanical power (W)
$w_{cavity\ aperture}$	cavity aperture width (m)
$w_{concentrator,net}$	net aperture width of the primary concentrator (9 m)
x,y	Cartesian coordinates

Greek symbols

Δp	pressure drop (Pa)
$\Delta Q'$	residual of energy balance (W m^{-1})
ΔT	temperature difference (K)
δ_{kj}	Kronecker delta
ε	surface emissivity
$\varphi_{corrugation}$	corrugation opening angle (deg)
η	efficiency
κ	heat capacity ratio
ρ	density (kg m^{-3})
σ	Stefan-Boltzmann constant (W m^{-2} K^{-4})
θ	solar incidence angle on concentrator aperture with respect to normal (deg)

Abbreviations

C	corrugated
CPC	compound parabolic concentrator
DGW	double-glazed window
HTF	heat transfer fluid
rms	root mean square
S	smooth
SGW	single-glazed window

Subscripts

0	normal incidence angle
1,2,...	surface
in	incoming; inlet
out	outgoing; outlet
s	isentropic
w1	inner window
w2	outer window

lower re-radiation area compared to conventional solar receivers. On the other hand, the apparent emissivity of the cavity aperture is close to unity and hence approximately 10 times larger than the emissivity of the selective surface coating of conventional receivers.

The initial receiver design is shown Fig. 1a [2]. It consists of a black absorber tube enclosed by an insulated stainless steel cavity with asymmetric CPCs at the rectangular aperture. A 43 m long prototype of this receiver was tested to validate the heat transfer model of the receiver and to study the receiver efficiency under various operating conditions. The pressure drop across the receiver was experimentally measured and numerically simulated for air mass flow rates in the range 0.039–0.48 kg s^{-1}. For this range of mass flow rates, the measured pressure drop varied from 23 to 2710 Pa; the rms-difference between measured and simulated values was 8.5%. Off-sun steady-state tests were performed with receiver inlet temperatures in the range 185–542 °C and mass flow rates in the range 0.03–0.23 kg s^{-1}. The experimentally determined heat loss from the receiver ranged from 3.6 to 32.4 kW; the rms-difference between experimentally determined and numerically calculated heat loss values was 19%. It was shown in [2] that the prototype receiver efficiency falls below that of state-of-the-art receivers due to radiation spillage, reflection losses, convective and radiative heat losses through the open aperture, and conductive heat losses through the cavity insulation. In this paper, we present and analyze a modified cavity-receiver design that evolved from the initial prototype design by means of a numerical thermal analysis.

2. Receiver design

The receiver design analyzed in this paper is shown in Fig. 1b. The absorber tube has been eliminated and the HTF flows directly through the cavity, in order to provide a sufficiently large flow cross-section. The entire heat transfer surface (surface 1) is directly irradiated and its surface area can be enhanced with V-corrugations. The quartz glass window reduces the convective heat loss occurring at the aperture compared to the open aperture in the original receiver design. In addition, since glass is a poor transmitter for radiation emitted from a blackbody at <600 °C (Fig. 2), the window also retains most of the IR radiation that is emitted by surface 1 and incident on the aperture. A second window pane may be employed to act as an additional radiation trap for both the radiation emitted by surface 1 and by the inner window. A stagnant air layer between the two window panes of a double-glazed window reduces heat conduction through the window.

The receiver parameters are listed in Table 1. The cavity inner diameter, R_{cavity}, has been selected by considering that – for a given HTF mass flow rate m_{HTF} – a smaller cavity diameter leads to: (*i*) better convective heat transfer (i.e. higher $h \cdot D_h$) at surfaces 1 and 2, (*ii*) lower insulation material demand, and (*iii*) higher pumping power requirement. The insulation thickness, $d_{insulation}$, has been selected based on the heat loss determined for the initial prototype receiver design [2]. The corrugation opening angle, $\varphi_{corrugation}$, is an arbitrary choice. The number of corrugations has been selected large enough to avoid significant variation of the insulation thickness. The cavity aperture width, $w_{cavity\ aperture}$, is