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MAXIMUM NET POWER OUTPUT FROM AN INTEGRATED DESIGN OF A SMALL-SCALE OPEN AND DIRECT SOLAR THERMAL BRAYTON CYCLE

by

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Submitted in partial fulfilment of the requirements for the degree
MASTER OF ENGINEERING (Mechanical Engineering)

in the

Faculty of Engineering, the Built Environment and Information Technology

UNIVERSITY OF PRETORIA
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February 2011

Abstract

Title:	Maximum net power output from an integrated design of a small-scale open and direct solar thermal Brayton cycle
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The geometry of the receiver and recuperator in a small-scale open and direct recuperative solar thermal Brayton cycle can be optimised in such a way that the system produces maximum net power output. The purpose of this work was to apply the second law of thermodynamics and entropy generation minimisation to optimise these geometries using an optimisation method. The dynamic trajectory optimisation method was used and off-the-shelf micro-turbines and a range of parabolic dish concentrator diameters were considered. A modified cavity receiver was used in the analysis with an assumed cavity wall construction method of either a circular tube or a rectangular channel. A maximum temperature constraint of 1 200 K was set for the receiver surface temperature. A counterflow plate-type recuperator was considered and the recuperator length was constrained to the length of the radius of the concentrator. Systems producing a steady-state net power output of 2 – 100 kW were analysed. The effect of various conditions, such as wind, receiver inclination and concentrator rim angle on the maximum net power output, and optimum geometry of the system were investigated. Forty-five different micro-turbines and seven concentrator diameters between 6 and 18 metres were considered. Results show the optimum geometries, optimum operating conditions and minimum entropy generation as a function of the system mass flow rate. The optimum receiver tube diameter was relatively large when compared with the receiver size. The optimum counterflow plate-type recuperator channel aspect ratio is a linear function of the optimum system mass flow rate for a constant recuperator height. The optimum recuperator length and optimum NTU are small at small system mass flow rates but increase as the system mass flow rate increases until the length constraint is reached. For the optimised systems with maximum net power output, the solar receiver is the main contributor to the total rate of minimum entropy generation. The contributions from the recuperator, compressor and turbine are next in line. Results show that the irreversibilities were spread throughout the system in such a way that the minimum internal irreversibility rate was almost three times the minimum external irreversibility rate for all optimum system geometries and for different concentrator diameters. For a specific environment and parameters, there exists

an optimum receiver and recuperator geometry so that the system can produce maximum net power output.

Acknowledgement

My Creator,

who did not come to judge, but to save.

Table of contents

List of figures

List of tables

Nomenclature

Alphabetic Symbols:		**Units**
a, a	Longer side of rectangle (channel width)	m
A	Area	m^2
b	Shorter side of rectangle (channel height)	m
c	Heat exchanger capacity ratio	-
c	Specific heat (c_p or c_v)	J/kgK
C	Concentration factor	-
CR	Concentration ratio	-
C_W	Optimum ratio of minimum internal to external irreversibility rate	-
d , d	Cavity receiver aperture diameter	m
D	Parabolic dish concentrator diameter	m
D , D	Cavity receiver diameter	m
D	Tube / rectangular channel diameter (D_{rec} , $D_{h,rec}$, $D_{h,reg}$)	m
e	Specific exergy	J/kg
$\hat{e}$	Specific energy	J/kg
e_p	Parabolic concentrator error	rad
E	Exergy	J
$\dot{E}$	Exergy rate	W
f	Darcy friction factor	-
F	Heat removal factor	-
F_D	External drag force	N
g	Gravitational constant	m/s^2
GF	Gradient vector of the objective function	-
Gr	Grashof number	-
h	Convection heat transfer coefficient	W/m^{2}K
h	Planck's constant (solar radiation)	-
h	Small increment	m
h	Specific enthalpy	J/kg

H	Recuperator height	m
I	Irradiance	W/m^2
$\dot{I}$	Irreversibility rate	W
k	Boltzmann's constant (solar radiation)	-
k	Gas constant (c_p/c_v)	-
k	Thermal conductivity	W/mK
K	Constant	-
KE	Kinetic energy	J
L	Length	m
L	Length of channel / tube (L_{rec} , L_{reg})	m
$\dot{m}$	Mass flow rate	kg/s
MT	Micro-turbine model number (Appendix D)	-
n	Number of recuperator flow channels	-
N	Number of fins	-
N_D	Optimum number of tube diameters between receiver edge and aperture edge	-
N_R	Optimum number of rectangular channels between receiver edge and aperture edge	-
NTU	Number of transfer units	-
Nu	Nusselt number	-
P , *p*, p	Pressure	Pa
PE	Potential energy	J
$\Pr$	Prandtl number	-
q	Specific heat	J/kg
Q, Q	Heat energy	J
$\dot{Q}$	Heat energy rate	W
$\dot{Q}_0$	Rate of solar energy lost due to convection and radiation	W
$\dot{Q}*$	Rate of solar energy intercepted by the cavity receiver aperture	W
$\dot{Q}_{net}$	Net solar energy rate available for receiver fluid	W
r	Pressure ratio (P_2/P_1)	-
R	Gas constant	J/kgK

Re	Reynolds number	-
refl	Mirror surface specular reflectivity	-
R_f	Fouling factor in the calculation of overall heat transfer coefficient	-
s, s	Specific entropy	J/kgK
S	Entropy	J/K
$\dot{S}$	Entropy rate	W/K
t	Thickness of recuperator separator plate	m
t	Time	s
T, T	Temperature	K, ℃
T^*	The apparent sun's temperature as an exergy source	
u	Specific internal energy	J/kg
U	Heat coefficient (U_L)	W/mK
U	Internal energy	J
U	Overall heat transfer coefficient	W/mK
v	Frequency (solar radiation)	Hz
v	Specific volume	m^3/kg
V	Velocity	m/s
w	Specific work	J/kg
w	Wind factor	-
W	Work	J
$\dot{W}$	Power	W
Wn	Receiver aperture diameter vector	m
x	Discrete least squares approximation constant	-
X	Optimisation vector	-
z	Height	m

Greek symbols:

α	Receiver absorptance	-
β	Inclination of receiver	-
ε	Effectiveness	-
ε	Exergetic efficiency	-
ε	Emissivity	-
ϕ	Horizontal inclination angle	-
φ	Angle	-

η	Efficiency	-
λ	Dimensionless parameter (λ_{BH})	-
λ	Wave length (solar radiation)	m
μ	Dynamic viscosity	kg/ms
ρ	Density	kg/m^3
υ	Volume	m^3
Ω_1	Solid angle subtended by the solar disc seen from earth	rad
Ω_2	Solid angle for the outgoing radiation	rad
ψ	Angle	-

Subscripts:

0, *0*	Surrounding / environment
0	Zero pressure (ideal gas) for c_p
1	State 1
2	State 2
a	Receiver aperture
A	Absorber
atm	Atmospheric
avg	Average
b	Boundary
b	Of blackbody radiation, final
B	Base
BH	Defined by Bahnke and Howard
c	Cold stream
c	Compressor
c	Cross-sectional
$conc$	Concentrator
$conv$	Convection
cv	Control volume
D	Based on receiver diameter
D	Destruction
e	Electric
e	Outlet
ext	External

F	Average of the heat transfer fluid
F	Fuel
$FirstLaw$	According to the first law of thermodynamics
gen	Generation
h	Hot stream
h	Hydraulic
$high$	Highest value on island of maximum compressor efficiency
i	Inlet
i	Inner
in, in	In
int	Internal
L	Loss due to convection and conduction
L	Loss
$loss$	Loss
low	Lowest value on island of maximum compressor efficiency
max	Maximum
min	Minimum
net	Net
o	Out/Outer
opt	Optimum
out, *out*	Out
p	For constant pressure
P	Product
rad	Radiation
rec	Receiver
reg	Recuperator
rim , rim	Concentrator rim
s	Shaft
s	Surface
$sngl$	For a single fin
t	Turbine
th , th	Thermal
v	For constant volume
v	Per unit frequency (solar radiation)

w	Cavity receiver inner surface
∞	Surrounding area / free-stream
ΔP	Due to fluid friction
ΔT	Due to temperature difference

Superscripts:

*	Solar
‘	Per unit length
.	Time rate of change
CH	Chemical