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Thermal model of a DISH/STIRLING system

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Abstract

This paper presents a global thermal model of the energy conversion of the 10 kW_{el} Eurodish Dish/Stirling unit erected at the CNRS-PROMES laboratory in Odeillo. Using optical measurements made by DLR, the losses by parabola reflectivity and spillage are calculated. A nodal method is used to calculate the heat losses in the cavity by conduction, convection, reflection and thermal radiation. A thermodynamic analysis of a SOLO Stirling 161 engine is made. The Stirling engine is divided in 32 control-volumes and equations of ideal gas, mass and energy conservation are written for each control-volume. The differential equation system is resolved by an iterative method developed using Matlab™ programming environment. Temperature, mass, density of working gas, heat transfers and the mechanical power are calculated for one Stirling engine cycle of 40 ms and for a constant Direct Normal Irradiation (DNI). The model gives consistent results correctly fitting with experimental measurements.

Keywords

Eurodish, Dish/Stirling system, thermal model, heat losses, energy balance, Stirling engine

Nomenclature

Aw heat transfer area, *m²*

C_p massic heat capacity at constant pressure, $J.kg^{-1}.K^{-1}$

C_v volumic heat capacity at constant pressure, $J.kg^{-1}.K^{-1}$

D diameter, *m*

Diss thermal dissipation, *J*

DNI direct normal insolation, $W.m^{-2}$

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k thermal conductivity, $W.K^{-1}.m^{-1}$

f receiver tilt angle, *rad*

fr friction factor

F view factor

Gr Grashof number

h convective heat transfer coefficient, $W.m^{-2}.K^{-1}$

J radiosity, $W.m^{-2}$

L length, *m*

m working gas mass, *kg*

$\dot{m}$ mass flow, $kg.s^{-1}$

$\overline{M}$ molar mass, $g.mol^{-1}$

n number of regenerator screens or absorber tubes

Nu Nusselt number

P pressure, *Pa*

δQ heat power in Stirling engine, *W*

R ideal gas constant, $J.kg^{-1}.K^{-1}$

Re Reynolds number

Rt thermal resistance, $K.W^{-1}$

S area, m^2

S_{sec} section area, m^2

t time, *s*

T température, *K*

th thickness, *m*

V volume, m^3

v velocity, $m.s^{-1}$

δW mechanical power, *W*

Greek symbols

π_T Stirling cycle period, *s*

ε total hemisherical emissivity

φ solar flux, $W.m^{-2}$

Θ heat power in the receiver, *W*

ΔP pressure drop, *Pa*

ϕ phase angle, *rad*

σ Stefan – Boltzman constant, $W.m^{-2}.K^{-4}$

λ wavelength, μm

ρ_m mass density, $kg.m^{-3}$

ρ reflectivity

ω angular speed, $rad.s^{-1}$

<u>*subscripts*</u>

a aperture

am ambient air

C compression control-volume

cav cavity

cer ceramic

d dead

inc inconel

E “expansion” control-volume

H “absorber” control-volume

par parabola

sec section

w wall

<u>*exponents*</u>

Cd conduction

Cdv conduction and convection

Cv convection

LT low temperature, spectrum radiation emitted by a black body at a temperature about 1100 K

rad thermal radiation

S solar spectral band [λ<3 μm]

Introduction

Since July 2004, a 10 kW_{el} Dish/Stirling unit is in operation at the CNRS-PROMES (French National Centre for Scientific Research) Laboratory in Odeillo in France. This system is one of the several Country Reference Units of the Envirodish project. It is a Eurodish system developed by the Deutsches Zentrum für Luft und Raumfahrt (DLR) and Schlaich Bergermann und Partner (SBP) for solar electricity generation using a Stirling engine externally heated by concentrated solar radiation (Keck et al., 2006). During these two years, the system has accumulated 2500 operation hours and 14.6 MWh_{el} of electricity production with a power record of 11.1 kW_{el} at 974 W/m² direct normal insolation, which corresponds to a net solar-to-electric efficiency of 21.6%.

The present work is focused on the thermal modeling. The objective is to evaluate the optical and thermal losses of the system components, to identify the key parameters of the design and to study their influence on the performances of the components according to various operation conditions. The dynamic simulation of the system is also targeted in order to evaluate the efficiency on a daily, monthly or annual basis.

The thermal model is composed of a radiation transfer model for the cavity, which is coupled to the solar flux distribution, and on a thermodynamic model for the Stirling engine. The results are compared to experimental data issued from a campaign of optical and power measurements made by DLR in 2005 under a direct solar irradiation of 906 W/m² (Reinalter et al., 2006).

1. Methodology

The system is divided into five components, each of them representing a stage of the conversion cascade (figure 1). The optical losses of the concentrator (reflectivity, spillage) are determined using the distribution of solar flux density in planes close to the aperture (Reinalter et al. 2006). For a given direct solar irradiation, the thermal model of the cavity and of the absorber estimates the energy provided to the working gas of the Stirling engine and the heat losses by reflection, thermal radiation and convection. A nodal method is used to simulate one Stirling engine cycle, yielding the estimation of the generated mechanical power. The generator efficiency and the consumptions of all single components (water pump, tracking system, cooling fan) are user-defined inputs.

2. Optical and power measurements

The effective parabola area considering the shadows is 52.9 m² and the mirror reflectivity is measured at 92.5%, a value somewhat lower than the design value (94 %). Reinalter et al. (2006) assessed the optical performance of the concentrator under elevated direct solar irradiation (906 W/m²), using the flux mapping system developed by DLR and described by Ulmer et al. (2002). The solar receiver is composed of a ceramic cylindrical cavity 30 cm in diameter and 12 cm in depth, the concentrated solar radiation enters through an aperture of 19 cm in diameter. The cavity walls are insulated using a Silica-based ceramic material. The hexagonal absorber composed of 78 tubes made of Inconel of 3 mm outer diameter is placed at the bottom of the cavity.

Figure 2 shows the solar flux map in the focal plane normalized to 1000 W/m² of direct normal irradiation (DNI) and 94% concentrator reflectivity. The aperture of the cavity is located in the focal plane, it has a diameter of 19 cm and it intercepts 85% of the incoming concentrated solar energy.

Figure 3 shows the solar flux distribution in the absorber plane, which is located 12 cm behind the aperture. It is observed that 78% of the concentrated solar energy directly hit the hexagonal absorber, while the remaining 7% hit the cavity wall. The flux distribution is rather inhomogeneous with a important peak of 1583 kW/m² and the mean solar flux on the absorber is measured at 702 kW/m².

3. Receiver model

A nodal method is used to calculate the heat losses by reflection, by thermal radiation and by convection out of the cavity, and by conduction trough the ceramic walls. The energy provided to the working gas flowing through the absorber tubes is also calculated over one complete cycle of the Stirling engine.

The receiver is divided into 11 control-volumes, named i = 1 to 11:

-8 for the absorber (i = 1 to 8) to take into account the inhomogeneous flux distribution

-2 for the irradiated and shadowed ceramic cavity walls (i = 9 and i = 10, respectively),

-1 for the aperture (i = 11).

Each insolated control-volume (6 for the absorber, and 1 for the ceramic walls) is assumed to receive a uniform solar flux density calculated from flux map in the absorber plane (figure 3) and given in the table 1. The highest solar flux density of 1068 kW/m², a value lower than the flux peak of 1583 kW/m² is located on the “absorber” control-volume 5.

Figure 4 shows a cross-section of the receiver and Figure 5 the 8 control volumes of the absorber. Note that both control-volumes 1 and 8 are not insolated; they are hidden behind front tubes.

For each control-volume i, the energy balance is written with one equation (Eq.1):

$$(\rho_m C_p V)_i \frac{dT_{w_i}}{dt} = \sum_j \Theta_{j-i} + \Theta_i \qquad \text{(Eq.1)}$$

Where Θ_{j-i} is the heat transferred by conduction, convection or radiation from each control-volume j (j = 1 to 11) to the control-volume i, and Θ_i is the concentrated solar energy hitting the control volume i.

3.1. Radiation transfer model

The radiation transfers between the control-volumes are calculated using a radiation balance applied to the solar receiver. The spectrum of electromagnetic radiation is divided in two bands in order to take into account the spectral dependence of the optical properties of Inconel and Ceramic. The first one (S) corresponds to the solar radiation spectrum (λ<3 µm) and the second one (LT) to the spectrum radiation emitted by a black body at a temperature about 1100 K (λ>3 µm). The optical properties of Inconel and Ceramic are given in table 2.

For these two spectral bands, the control-volume surfaces are assumed opaque diffuse and grey. The cavity aperture is considered as a black surface at ambient air temperature. The control-volume surfaces 1 and 8 are hidden and they do not participate in the radiation balance. For each control-volume surface i, radiosities in the solar spectral band and in the LT spectral band are respectively given by equations (Eq.2) and (Eq.3). Total radiosity J_i, given by equation (Eq.5) is the energy flux leaving the surface i composed of emitted plus reflected energy. In the solar spectral band the radiosity J_i^S is only composed of a reflected solar energy and in the LT spectral band the radiosity J_i^{LT} is build from the emitted energy and the reflected LT energy by the other control-volume surface i.

$$J_i^S = \rho_i^S.(\varphi_i + \sum_{j=1,\, j\neq 1,\, j\neq 8}^{11} F_{ij} J_j^S) \quad \text{(Eq.2)}$$

$$J_i^{LT} = \varepsilon_i \sigma T_{w_i}{}^4 + \rho_i^{LT}.(\sum_{j=1,\, j\neq 1,\, j\neq 8}^{11} F_{ij} J_j^{LT}) \quad \text{(Eq.3)}$$

The view factors F_{ij} are computed by integration of differential equations for discrete areas issued from Siegel and Howell (1992).

Knowing the radiosity, equation (Eq.4) gives the net radiation balance Θ_i^{rad} for each control-volume:

$$\Theta_i^{rad} = S_i(J_i - \varphi_i - \sum_{j=1,\, j\neq 1,\, j\neq 8}^{11} F_{ij} J_j) \quad \text{(Eq.4)}$$

Where J is the total radiosity of the surface i:

$$J = J^S + J^{LT} \quad \text{(Eq.5)}$$

The net radiation balance of the aperture, Θ_{11}^{rad}, represents the receiver heat losses by reflection and by thermal radiation.

3.2. Conductive and Convective heat losses in the cavity

Some works in the literature provide correlations to calculate Nusselt number in a cavity as a function of the Grashof number Gr, receiver tilt angle f, aperture diameter D_a and temperature walls T_w. The natural convective heat transfer coefficient h_{cav} used in the receiver model is derivated from several correlations (McDonald, 1995). The Stine and Mc Donald correlation for the Nusselt number Nu is given as follows:

$$Nu = 0.088 Gr^{1/3} (\frac{T_w}{T_{am}})^{0.18} (\cos f)^{2.47} (\frac{D_a}{L_{cav}})^s \quad \text{(Eq.6)}$$

And

$$s = 1.12 - 0.98 (\frac{D_a}{L_{cav}}) \quad \text{(Eq.7)}$$

The natural convective heat losses Θ_i^{Cv} through the aperture are given by equation (Eq.8):

$$\Theta_i^{Cv} = \sum_{i \neq 11} h_{cav} S_i (T_{wi} - T_{am}) \quad \text{(Eq.8)}$$

Where h_{cav} is derivated from Nusselt number:

$$h_{cav} = \frac{Nu k_{am}}{L_{cav}}$$

The conductive heat losses Θ^{Cdv} through the ceramic receiver walls and then evacuated to the ambient air by natural convection are calculated from thermal resistance Rt^{Cdv}. The ceramic thermal conductivity k_{cer} and the natural convective heat transfer coefficient h_{am} are assumed constant. The expression of conduction losses is given by:

$$\Theta^{Cdv} = \frac{(T_{wi} - T_{am})}{Rt^{Cdv}} \quad \text{(Eq.9)}$$

Where for the control-volumes 9 and 10:

$$Rt^{Cdv} = \frac{1}{h_{am} S_i} + \frac{1}{k_{cer} 2\pi L_i} Ln\left(\frac{D_{cav} + 2 th_{cer}}{D_{cav}}\right) \quad \text{(Eq.10)}$$

And for the control-volumes 1 to 8:

$$Rt^{Cdv} = \frac{1}{h_{am} S_i} + \frac{th_{cer}}{k_{cer} S_i} \quad \text{(Eq.11)}$$

3.3. Heat transfers in the absorber

The flux density of solar distribution on the absorber tubes is inhomogeneous inducing temperature gradients and low heat transfers by conduction. Conductive heat transfers Θ_{j-i}^{Cd} along the thin of the tubes between the 8 “absorber” control-volumes are calculated by equation (Eq.12).

$$\Theta_{j-i}^{Cd} = \frac{T_{wj} - T_{wi}}{Rt_{j-i}^{Cd}} \quad \text{(Eq.12)}$$

Where

$$Rt_{j-i}^{Cd} = \frac{L_j + L_i}{2} \cdot \frac{1}{k_{inc} S_{\sec j-i}} \quad \text{(Eq.13)}$$

And

$$S_{\sec j-i}=\frac{pi}{4}(D_{outter}^{2}-D_{inner}^{2}).n_{tubesi} \qquad \text{(Eq.14)}$$

3.4. Receiver heat losses

Finally, The energy balance for the aperture control-volume, 11, gives the total heat losses in the receiver.

$$\Theta_{cav}^{losses}=\begin{cases} -\Theta_{11}^{rad} & \text{Reflection and IR emission losses} \\ +\sum_{j=1}^{10}\Theta_{j-11}^{Cdv} & \text{Conduction losses through the ceramic cavity walls} \\ +\sum_{j=1}^{10}\Theta_{j-11}^{Cv} & \text{Convection losses} \end{cases} \qquad \text{(Eq.15)}$$

3.5. Absorber energy balance

For the absorber control-volumes, equation (Eq.1) can be written as follows:

$$(\rho_m C_p V)_{Hi}\frac{dT_{wHi}}{dt}=-\Theta_i^{ray}-\sum_{j}^{j\neq i}\Theta_{i-j}^{Cd}-\Theta_{i-11}^{Cdv}-\Theta_{i-11}^{Cv}-\delta Q_{Hi} \qquad \text{(Eq.16)}$$

Where δQ_H is the energy provided to the Stirling engine, depending on the pressure, the temperature, the mass density and the velocity of the working gas.

4. Stirling engine thermal model

A Stirling cycle machine operates on a closed regenerative thermodynamic cycle using a working gas, and subjects the gas to expansion and compression processes at different temperatures. Since its invention by Robert Stirling in 1816 different engine configurations have been developed (Thombare, 2006). The EURODISH Stirling engine, SOLO 161 is of the alpha-type. Alpha-type engine have two pistons in separated cylinders arranged in “V”, which are connected in series by three exchangers: a heater (here a solar absorber), a regenerator and a cooler.

Many works presents or uses thermal model more or less complex to approximate the real behaviour of the alpha-type Stirling engine. A review of these thermal models can be found in Dyson et al. (2004) and in Ercan et al. (2005).

For this study, a nodal analysis of the SOLO 161 is performed. The Stirling engine is divided in 32 control volumes as shown in figure 6. The control volumes of compression V_C and expansion V_E spaces are variable and expressed by equation (Eq.17) and (Eq.18). The other control-volumes have fixed volumes.

$$V_E(t)=\frac{V_E}{2}(\sin(\omega t)+1)+V_{E,d} \tag{Eq.17}$$

$$V_C(t)=\frac{V_C}{2}(\sin(\omega t-\phi)+1)+V_{C,d} \tag{Eq.18}$$

Mass equation (Eq.19), energy equation (Eq.20) and state equation (Eq.21) are written for each control-volume.

The following assumption are made in order to simplify the model equations:

1) The working gas is assumed to be an ideal gas
2) The total mass of the gas is constant within the Stirling engine
3) The instantaneous pressure is uniform
4) The compression and expansion spaces are adiabatic
5) The gas flow is one dimensional and quasi steady

$$\frac{dm_i}{dt}=(\dot{m}_{in}-\dot{m}_{out})_i \tag{Eq.19}$$

$$\delta Q_i+Diss_i+(C_{P(Tin)}T_{in}\dot{m}_{in}-C_{P(Tout)}T_{out}\dot{m}_{out})_i+\delta W_i=C_{v(T_i)}\frac{d(mT)_i}{dt} \tag{Eq.20}$$

$$PV_i=m_i\frac{R}{M}T_i \tag{Eq.21}$$

In the control-volumes of the compression and expansion spaces, the mechanical power δW_i is expressed by:

$$\delta W_i=-P\frac{dV_i}{dt} \tag{Eq.22}$$

And in the “heat exchanger” control volumes, the heat transfer δQ_i is given by:

$$\delta Q_i=h_i A_{wi}(T_{wi}-T_i) \tag{Eq.23}$$

h_i is the forced convective heat transfer coefficient calculated with two correlations. The classical Colburn correlation is used in the 8 “absorber” control-volumes and in the 10 “cooler” control-volumes. For the 10 “regenerator” control-volumes, the correlation given by Lemrani (1995) is used:

$$Nu_i = 0.42 \mathrm{Re}_i^{0.56} \quad \text{(Eq.24)}$$

$Diss_i$ is proportional to the pressure drop ΔP_i depending to the friction factor. As for the forced convective heat transfer coefficient, two correlations are used:

$$Diss_i = \Delta P_i \frac{P\overline{M}}{RT_i}\left(\frac{\dot{m}_{in}+\dot{m}_{out}}{2}\right)_i \quad \text{(Eq.25)}$$

Where:

$$\Delta P_i \begin{cases} = f_{ri}\dfrac{1}{2}\rho_m v_i^2 n_i & f_{ri} = \dfrac{33.6}{\mathrm{Re}_i} + 0.337 \quad \textit{For the “regenerator” control volumes} \\ = f_{ri}\dfrac{1}{2}\dfrac{\rho_m v_i^2 L_i}{D_i} & f_{ri} \begin{cases} = \dfrac{64}{\mathrm{Re}_i} & \text{If } \mathrm{Re}_i < 2000 \\ & \textit{For the ”absorber” and “cooler” control volumes} \\ = 0.316 \mathrm{Re}_i^{-0.25} & \text{If } \mathrm{Re}_i > 2000 \end{cases} \end{cases} \quad \text{(Eq.26)}$$

5. Numerical Resolution

The differential equation system is resolved by an iterative method developed in Matlab. Temperature, mass, density of working gas, heat transfers and the mechanical power are calculated for one Stirling engine cycle of 40 ms and for a constant Direct Normal Irradiation. Periodic steady cyclic condition is obtained after few iteration cycles.

Mean value per cycle of the energy powers G are the integration of the discrete powers δG over the time and are calculated by equation (Eq.27):

$$G = \frac{\int_{cycle} \delta G dt}{\pi_T} \quad \text{(Eq.27)}$$

Few input data are necessary:

1) Geometric data of the receiver and Stirling engine.

2) Working gas characteristics of the Stirling engine.

3) EURODISH operation data: engine speed, DNI, total working gas mass.

it is important to point out that EURODISH system regulates the total working gas mass and consequently the cycle mean pressure, to keep constant the maximal absorber temperature at 1053 K. In the Stirling engine model, the total working gas mass is an input data and the absorber temperature is an output data. To solve this problem, a loop on the total working gas mass has been added in the numerical resolution in order to obtain a maximal absorber temperature of 1053 K over one Stirling cycle.

6. Results and discussion

Our numerical study is compared to experimental measurement given by Reinalter et al. (2006). The Eurodish operation data during the experimental measurements are provided in table 3. Figure 7 gives the model results and the experimental measurements of the solar energy dispatching. The “numerical errors” designation is the difference between the energy input and output of the Eurodish system: For the experimental measurements this term is due to measurement errors (Reinalter et al., 2006) and for the model results, it includes errors coming from numerical integrations. Figure 7 shows the considerable energy losses in the receiver in particular by spillage. The losses by spillage, conduction, convection and radiation are estimated at 13.5 kW, which represents 28 % of the solar energy incoming of the dish. 31 kW is provided to hydrogen and 19 kW is evacuated to the cooler. The developed model gives good results in agreement with the measurements particularly in the receiver. The efficiency of the receiver defined as the proportion of the thermal energy introduced into the engine to the power that enters in the cavity is estimated at 81.4 % and is measured at 82.6 %. The observed differences between the model results and the measurements are due to the Stirling model. The Stirling engine efficiency, which represents the conversion of thermal to mechanical power is calculated at 34.4 % and is measured at 39.2 %. This difference can be explained by correlations used to calculate the forced convective heat transfer coefficients in particular in the regenerator where the gas flow is complex (Bonnet, 2005). Finally, the model finds a net solar-to-electric efficiency of 19.1 % instead of 22.5 %, which gives a difference of 1.3 $kW_{el.}$

Figure 8 shows the absorber temperature distribution calculated by the model and measured by 20 thermocouples placed behind the absorber. The total hydrogen mass within the Stirling engine is adapted to obtain a maximal temperature about 1053 K. This maximal temperature is located on the surfaces 4 and 5. Figure 8 shows a good agreement between the model results and the measurements. The difference between the maximum and minimum temperature is calculated at 130 K when measurements gives about 100 K. The temperature of the expansion space is estimated at 914 K and the cycle pressure at 115 Bar.

Figure 9 shows the power provided to the Stirling engine, the power provided to the cooler and the electrical power as a function of the DNI. The maximal absorber temperature is constant at 1053 K. The model results are compared to experimental values of DNI, net electrical power and temperature in the cooler made automatically every second on the Eurodish system (Reinalter et al., 2006). The experimental net solar to electricity efficiency increases as a linear function of a DNI up to 800 W/m². When the insolation exceeds this level a blower in the receiver cavity is activated to reject additional heat and maintain the absorber at a fixed temperature of 1053 K. Model give good results. For the electrical power the difference with experimental values is less than 1 kW_{el} and is maximal at 700 W/m². At 800 W/m² the power provided to the cooler is evaluated at 16.5 kW and is measured at 17.5 kW. This energy could be recuperated for a cogeneration application. In this case, the overall efficiency (Onovwiona et al. 2006) increases to 58 %. For the power provided to the Stirling engine It can be seen a constant difference of 1 kW between the model and the experimental values.

Figure 10 gives the energy dispatching calculated by the model for two DNI values: 400 W/m² and 800 W/m². The concentrator efficiency, defined as the proportion of energy entering in the cavity to solar energy intercepted by the dish is constant at 78.6 %. The losses by conduction, convection and reflection and thermal radiation in the receiver increase from 5.2 kW at 400 W/m² to 6.5 kW at 800 W/m² DNI, which gives a receiver efficiency of 72 % and 80 %. The Stirling efficiency is constant at 34.4 % and finally the net solar electricity efficiency drop from 800 W/m² to 400W/m² due to the receiver efficiency decrease is 16.2 %.

7. Conclusion and Perspectives

This paper presented a thermal model of energy conversion of Eurodish Dish/Stirling. The model compared to experimental measurements gives good results and permits to study the detailed heat losses. Currently, the net conversion solar-electricity efficiency is about 21 % at a DNI value of

900 W/m² and at an ambient temperature of 20 °C. The model shows important heat losses in the cavity by spillage and radiation (reflection and IR-emission) and consequently an improvement potential of system efficiency.

Future works will focus on the study of the influence of cavity physical parameters in order to design a new solar receiver. Using the ray-tracing code SOLTRACE and parabola slope error data, a parabola and receiver simulation are made to estimate and study the solar flux distribution in different planes. Then, theses flux maps will be introduced in the presented model. Furthermore, the model is used to evaluate energy provided to the cooler in order to test a Eurodish Dish/Stirling for a cogeneration application.

Acknowledgements

The authors wish to thank Isabelle VERDIER from CREED, Philippe LEFEVRE from R&D EDF, Pascal STOUFFS from LATEP, Jean Michel GINESTE, Nicolas BOULLET and Emmanuel GUILLOT from PROMES, Wolfgang REINALTER, Steffen ULMER and Peter HELLER from DLR.

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Table 1: Measured mean solar flux for each "absorber" control-volume

Control-volume		1	2	3	4	5	6	7	8	Absorber
Mean Solar Flux	kW/m²	0	501	403	1052	1068	569	521	0	702

Table 2: Inconel and Ceramic properties

	ε	ρ	
	1073 K	ρ^S (λ<3 μm)	ρ^{LT} (λ>3 μm)
Inconel	0.88	0.07	0.14
Ceramic	0.9	0.15	0.12

Table 3: Eurodish operation data

working gas	hydrogen	
DNI	W/m²	906
S_{par}	m²	52.9
Solar energy	kW	48
P	bar	[130 – 140]
T_{wH} max	K	1053
T_E	K	[903 – 923]
[T_{in}-T_{out}] water cooler	K	295-305
T_{am}	K	293
Nt	rt.min^{-1}	1500

Figure captions

Figure 1: Model methodology

Figure 2: Solar flux map in the focal plane

Figure 3: Solar flux map in the absorber plane

Figure 4: Cross-section of the Eurodish receiver

Figure 5: Geometry and control volumes of the Eurodish absorber

Figure 6: Control-volumes in Stirling engine

Figure 7: Solar energy dispatching, Left: Model results, Right: experimental measurements (Reinalter et al. 2006)

Figure 8: Absorber temperature distribution: model results in underlined

Figure 9: Power provided to Stirling engine, power provided to the cooler and electrical power as function of DNI

Figure 10: Solar energy dispatching for two DNI values, Left: 800 W/m², Right: 400 W/m²