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Modelling of a Stirling engine with parabolic dish for thermal to electric conversion of solar energy

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ABSTRACT

Stirling engines with parabolic dish for thermal to electric conversion of solar energy is one of the most promising solutions of renewable energy technologies in order to reduce the dependency from fossil fuels in electricity generation. This paper addresses the modelling and simulation of a solar powered Stirling engine system with parabolic dish and electric generator aiming to determine its energy production and efficiency. The model includes the solar radiation concentration system, the heat transfer in the thermal receiver, the thermal cycle and the mechanical and electric energy conversion. The thermodynamic and energy transfer processes in the engine are modelled in detail, including all the main processes occurring in the compression, expansion and regenerator spaces. Starting from a particular configuration, an optimization of the concentration factor is also carried out and the results for both the transient and steady state regimes are presented. It was found that using a directly illuminated thermal receiver without cavity the engine efficiency is close to 23.8% corresponding to a global efficiency of 10.4%. The components to be optimized are identified in order to increase the global efficiency of the system and the trade-off between system complexity and efficiency is discussed.

1. Introduction

Electric energy is essential to the human activities and to our society, and an increasing demand is expected in the coming years. In this scenario, the use of renewable energy sources for generating electricity is essential for reducing the dependency from fossil fuels and due to environmental concerns. Among the available technologies, Stirling engine systems powered by solar energy are one of the most promising solutions because they combine a readily available resource with a simple and efficient thermal cycle. Stirling engines present several advantages, such as high thermodynamic efficiency, the capability to use any heat source, including fuel, solar, geothermal or waste heat, and low level of noise. These are reasons why it has received the attention of researchers in the last years aiming its modelling, optimization and application in power generation systems [1–7]. This solution is particularly suitable for small or medium scale solar power plants [8] and its combination with thermal energy storage and the hybridization with other energy sources still are open fields of research to explore [9].

There are three main types of Stirling engines, the α-type, the β-type and the γ-type, depending on the compression and expansion chambers arrangement and on the working gas flow configuration. Recently, Cheng and Yang [3] reviewed the major differences between these three types and investigated its relative performance aiming the optimization of geometrical parameters. Another configuration known as thermal-lag Stirling engine comprises only one piston and a porous material stack fixed in the cylinder in place of the displacer [10]. Other types are the thermoacoustic [11] and the free piston Stirling engines coupled to linear electric generators, which are used in very specific applications, such as in the aerospace industry [12]. Stirling engines can also be classified as high temperature difference engines or low temperature difference engines depending on the range of temperatures of the hot and cold heat reservoirs, which, in turn, depends on the type of application. On the other hand, if the thermal cycle is reversed the Stirling machine can be used as heat pump in cryogenic cooling applications [13].

One of the first attempts of modelling the Stirling engine was made by Schmidt [14] circa fifty years after its invention, in which the expansion and compression of an ideal working gas in the engine is considered as isothermal processes and assuming a sinusoidal variation of the gas volume as function of the crank angle. Perfectly effective heat transfer and regeneration is also assumed which implies that the gas temperature is locally equal to the wall temperature in the expansion and compression chambers and in

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Nomenclature

A	area (m^2)
b	damping coefficient (N m s rad^{-1})
C	concentration factor (-)
c_p	mass heat transfer at constant pressure (J kg^{-1} K^{-1})
c_v	mass heat transfer at constant volume (J kg^{-1} K^{-1})
e	thickness (m)
f	parabolic focus (m)
F	strength (N)
g	gravitational constant (m s^{-2})
G_t	direct normal irradiance (W m^{-2})
h	heat transfer coefficient (W m^{-2} K^{-1})
H	length (m)
i	electric current (A)
J	moment of inertia (kg m^2)
k	thermal conductivity (W m^{-1} K^{-1})
L	width (m) or inductance (H)
m	mass (kg)
Nu	Nusselt number (-)
p	number of pole pairs (-)
P	pressure (Pa) or power (W)
Pr	Prandtl number (-)
Q	heat (W)
r	radius (m)
R	electrical resistance (Ω)
Ra	Rayleigh number (-)
R_g	constant of the gas (J kg^{-1} K^{-1})
S	absorbed radiation (W)
t	time (s)
T	temperature (K) or period (s)
V	volume (m^3) or voltage (V)
W	work (W)
(X, y)	distance (m)
z	distance or height (m)

Greek symbols

α	absorptivity (-)
β	expansion coefficient (K^{-1})
Δ	variation (-)
ε	emissivity or heat exchanger effectiveness (-)
η	efficiency (-)
θ	crank angle of piston (rad)
λ	amplitude of the flux induced by the permanent magnets (V s)
ρ	density (kg m^{-3}) or reflectivity (-)
σ	Stefan Boltzmann constant (W m^{-2} K^{-4})
τ	torque (N m)
υ	kinetic viscosity (m^2 s^{-1})
φ	ratio (-)
Ψ	phase difference between piston and displacer (rad)
ω	frequency (rad s^{-1})

Subscripts

a	environment
abs	absorption
c	compression space
cir	circular
cl	cylinder
conv	convection
cyc	cycle
d	displacer or axis
dis	parabolic dish
elec	electric
elecm	electromagnetic
f	flywheel
fd	connection between flywheel and displacer
fp	connection between flywheel and piston
glo	global
H	hot source
i	inlet or outlet of expansion space
ir	irradiated by the sun
j	inlet or outlet of compression space
l	connecting rod
L	cold source
ld	displacer and connecting rod group
m	engine housing or average value
max	maximum
mec	mechanical
opt	optic
p	piston
q	axis
r	regenerator
rad	radiation
s	stator windings
sf	static friction
T	total
ther	thermal
x	cylinder wall
0	initial condition

the regenerator. However, adiabatic processes describe better the real thermodynamic phenomena inside the engine. In an ideal adiabatic analysis it is assumed that the compression and expansion spaces are thermally insulated and that heat input and output to the cycle occur in the heater and in the cooler, respectively, which are modelled as separate zones from the expansion and compression spaces. Perfect heat regeneration is also assumed. Finkelstein [15] proposed a semi-adiabatic analysis accounting for heat transfer in the expansion and compression spaces with constant wall temperature, implying finite heat transfer coefficients and thus predicting lower efficiencies due to the irreversibility associated with heat transfer across a finite temperature difference. Urieli and Berchowitz [16] improved the adiabatic modelling by considering the effects of the imperfect processes in the heat exchangers and regenerator, including the pressure drop, which is known as Simple model. Chen and Griffin [17] presented in 1983 a comprehensive review of the isothermal and adiabatic modelling approaches, while improved versions of the models developed by Finkelstein and by Urieli and Berchowitz still are used nowadays [18–20]. Timoumi et al. [21] further enhanced the adiabatic model by considering the energy losses associated with the pressure drop in the heat exchangers, with the internal and external heat conduction, with the shuttle effect in the displacer and with the gas spring hysteresis, whereas the mechanical losses due to the friction between the moving parts were not considered.

Another approach uses the concept of finite time thermodynamics [22,23] for endoreversible machines in which only the external irreversibilities associated to heat transfer are included [24]. This analysis can also be used in the case of irreversible machines further considering the internal irreversibilities of a Stirling engine such as friction, pressure drop and entropy generation [25–27]. A similar approach uses the concept of finite speed thermodynamics and the so-called “direct method” for closed systems considering the effects of both internal and external irreversibilities [28,29], which involves the direct integration of equations based on the first law for finite speed processes. Ahmadi et al.