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Numerical simulation for the design analysis of kinematic Stirling engines

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HIGHLIGHTS

- A thermodynamic analysis for kinematic Stirling engines was presented.
- The analysis integrated thermal, mechanical and thermodynamic interactions.
- The analyses considered geometrical and operational parameters.
- The results allowed to map the performance of the engine.
- The analysis allow the design assessment of Stirling engines.

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ABSTRACT

The Stirling engine is a closed-cycle regenerative system that presents good theoretical properties. These include a high thermodynamic efficiency, low emissions levels thanks to a controlled external heat source, and multi-fuel capability among others. However, the performance of actual prototypes largely differs from the mentioned theoretical potential. Actual engine prototypes present low electrical power outputs and high energy losses. These are mainly attributed to the complex interaction between the different components of the engine, and the challenging heat transfer and fluid dynamics requirements. Furthermore, the integration of the engine into decentralized energy systems such as the Combined Heat and Power systems (CHP) entails additional complications. These has increased the need for engineering tools that could assess design improvements, considering a broader range of parameters that would influence the engine performance when integrated within overall systems. Following this trend, the current work aimed to implement an analysis that could integrate the thermodynamics, and the thermal and mechanical interactions that influence the performance of kinematic Stirling engines. In particular for their use in Combined Heat and Power systems.

The mentioned analysis was applied for the study of an engine prototype that presented very low experimental performance. The numerical methodology was selected for the identification of possible causes that limited the performance. This analysis is based on a second order Stirling engine model that was previously developed and validated. The simulation allowed to evaluate the effect that different design and operational parameters have on the engine performance, and consequently different performance curves were obtained. These curves allowed to identify ranges for the charged pressure, temperature ratio, heat exchangers dimensions, crank phase angle and crank mechanical effectiveness, where the engine performance was improved. In addition, the curves also permitted to recognise ranges were the design parameters could drastically reduce the brake power and efficiency. The results also showed that the design of the engine is affected by the conditions imposed by the CHP interactions, and that the engine could reach a brake power closer to 832 W with a corresponding brake efficiency of 26% when the adequate design parameters were considered. On the other hand, the performance could also be very low; as the reported in experimental tests, with brake power measurements ranging 52–120 W.

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Nomenclature

A	area (m^2)
A_o	external wet area of the tube (m^2)
C_f	non-dimensional friction coefficient
C_{fd}	form drag coefficient
C_{sf}	skin friction coefficient
C_p	constant pressure specific heat (J/kg K)
C_{pwater}	constant pressure specific heat for inlet water (J/kg K)
C_v	constant volume specific heat (J/kg K)
d	diameter (m)
d_{hy}	hydraulic diameter (m)
E	crank mechanism effectiveness
f	friction factor coefficient
freq	engine frequency (Hz)
F_R	view factor
h	convective heat transfer coefficient (W/m^2 K)
h_r	radiation heat transfer coefficient (W/m^2 K)
h_{water}	water film heat transfer coefficient (W/m^2 K)
k	thermal conductivity (W/m K)
K	piston to displacer swept volume ratio
K_w	thermal conductivity (W/m K)
L	length (m)
m	mass (kg)
n	number of flow resistance layers
m_{water}	mass flow of the inlet water (kg/s)
M	total mass of the working gas (kg)
NTU	number of transfer units
P	pressure level (Pa)
P_{ch}	engine charging pressure (bar)
P_{br}	engine brake power (W)
Q	heat transfer rate (W)
Q_{ht}	total heating requirement for the engine (W)
Q_{kt}	total cooling requirement for the engine (W)
Q_{lk}	heat loss due to internal conduction (W)
Q_{lsh}	heat loss due to shuttle conduction (W)
Q_{lossr}	heat loss due to regenerator inefficiency (W)
R	gas constant (J/kg K)
R_{ci}	conductive thermal resistance for tubes wall (K/W)
R_{fi}	fouling thermal resistance inside the tubes (K/W)
R_{fo}	fouling thermal resistance outside the tubes (K/W)
R_{hi}	convective thermal resistance inside the tubes (K/W)
t	time (s)
T	temperature (K)
T_{fM}	measured flame temperature (K)
T_{ratio}	cold to heat temperature ratio
T_{wi}	temperature at the internal wall of the tubes (K)
T_{wo}	temperature at the outer wall of the tubes (K)
T_{water_in}	inlet temperature of the water (K)
v	mean velocity (m/s)
V	volume (m^3)
V_{de}	total dead volume (m^3)
W	work flow per cycle (J/cycle)
W_i	engine indicated work (J/cycle)
W_s	engine shaft work (J/cycle)
W_{ploss}	energy loss due to pressure drop (J/cycle)
$W-$	engine forced work (J/cycle)
X	dead volume ratio

Acronyms

ACM	Aspen Custom Modeller
CHP	Combined Heat and Power
SE	Stirling engine

Subscripts

b	buffer space
c	compression space
d	displacer
e	expansion space
f	flame
h	heater space
$hous$	regenerator housing space
i	inside section
k	cooler space
M	measured values
o	outside section
r	regenerator space
w	wall
whe	heater wall
wk	cooler wall

Superscripts

+	positive variation
−	negative variation

Greek symbols

α_s	surface absorptivity
γ	adiabatic constant
η_b	brake efficiency
η_m	mechanical efficiency
η_t	thermal efficiency
σ	Stefan–Boltzmann constant (W/m^2 K^4)
ϵ	regenerator effectiveness
ρ	fluid density (kg/m^3)
θ	crank rotational angle (rad)
μ	viscosity (kg/m s)

1. Introduction

The Stirling engine is a closed-cycle regenerative system that presents good theoretical capacities, which include a high thermodynamic efficiency, low emissions levels thanks to a controlled external heat source, and multi-fuel capability among others [1,2]. These could place the engine in a good position to address the efficiency and fuel flexible requirements for actual energy technologies [1,3,4]. However, the performance of actual prototypes largely differs from the mentioned theoretical potential as reported by Thomas [5] and Klemes [6]. Actual engine prototypes present low electrical power outputs and large energy losses. These are mainly attributed to the challenging mechanical interaction, the complicated heat transfer, and the complex fluid dynamics through the different engine components [7]. These increased the need for engineering tools, such as numerical simulation, that could asses design improvements on the different components in order to increase the engine performance.

Simulation based analysis allows studying the influence of different parameters in the engine performance. The effect of operational variables like working gas temperatures, working fluid [8], engine frequency, compression ratio, and pressure levels have been presented by Strauss [9], Timouni [10,11], Tew [12], Garcia [13], Paul [14], Cheng [15] and others. In addition, the influence of design variables such as heat exchangers configurations [16–18], crank mechanism geometry [15,19,20], and regenerator materials [21,22] have been also reported. The studies have guided the development of Stirling engine prototypes. Some examples are