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Non-ideal Stirling engine thermodynamic model suitable for the integration into overall energy systems

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HIGHLIGHTS

- A numerical thermodynamic model for Stirling engine systems was developed.
- Thermodynamic equations were coupled with the heat transfer governing equations.
- The model was validated with experimental and numerical data.
- The brake power and engine efficiency at different conditions were calculated.
- Additional model results provide a deeper insight into the engine operation.

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ABSTRACT

The reliability of modelling and simulation of energy systems strongly depends on the prediction accuracy of each system component. This is the case of Stirling engine-based systems, where an accurate modelling of the engine performance is very important to understand the overall system behaviour. In this sense, many Stirling engine analyses with different approaches have been already developed. However, there is a lack of Stirling engine models suitable for the integration into overall system simulations. In this context, this paper aims to develop a rigorous Stirling engine model that could be easily integrated into combined heat and power schemes for the overall techno-economic analysis of these systems. The model developed considers a Stirling engine with adiabatic working spaces, isothermal heat exchangers, dead volumes, and imperfect regeneration. Additionally, it considers mechanical pumping losses due to friction, limited heat transfer and thermal losses on the heat exchangers. The model is suitable for different engine configurations (alpha beta and gamma engines). It was developed using Aspen Custom Modeller® (ACM®) as modelling software. The set of equations were solved using ACM® equation solver for steady-state operation. However, due to the dynamic behaviour of the cycle, a C++ code was integrated to solve iteratively a set of differential equations. This resulted in a cyclic steady-state model that calculates the power output and thermal requirements of the system. The predicted efficiency and power output were compared with the numerical model and the experimental work reported by the NASA Lewis Research Centre for the GPU-3 Stirling engine. This showed average absolute errors around ±4% for the brake power, and ±5% for the brake efficiency at different frequencies. However, the model also showed large errors (±15%) for these calculations at higher frequencies and low pressures. Additional results include the calculation of the cyclic expansion and compression work; the pressure drop and heat flow through the heat exchangers; the conductive, shuttle effect and regenerator thermal losses; the temperature and mass flow distribution along the system; and the power output and efficiency of the engine. These results show that the model allows an extensive study of different parameters of the engine and thus it is suitable for design optimization studies. In addition, it also presents the capability for the integration into overall Stirling engine combined heat and power systems and therefore will allow the performance evaluation of the engine integrated on these systems.

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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)
D_d	displacer diameter (m)
d_i	internal diameter of the tube (m)
d_{hy}	hydraulic diameter (m)
Err	error tolerance
Error1	absolute error calculated for T_c and T_e
Error2	absolute error calculated for T_k and T_h
Error3	absolute error calculated for T_{wh} and T_{wk}
f	friction factor coefficient
freq	engine frequency (1/s)
F_R	view factor
h	convective heat transfer coefficient (W/m^2K)
h_r	radiation heat transfer coefficient (W/m^2K)
h_{water}	water film heat transfer coefficient (W/m^2K)
J	annular gap cylinder displacer (m)
k	thermal conductivity (W/m K)
L	tube length (m)
m	mass of the fluid (kg)
M	molecular weight (kg/mol)
m_{water}	mass flow of the inlet water (kg/s)
NTU	number of transfer units
Nu	non-dimensional Nusselt number
P	pressure (Pa)
P_{br}	engine brake power (W)
Pr	non dimensional Prandtl number
Q	heat transfer rate (W)
Q_{hc}	heater heat transfer rate by cycle (J/cycle)
Q_{kc}	cooler heat transfer rate by cycle (J/cycle)
Q_{rc}	regenerator heat transfer rate by cycle (J/cycle)
Q_{ht}	total heating requirement for the engine (W)
Q_{kt}	total cooling requirement for the engine (W)
Q_{lossr}	heat loss due to imperfect regenerator (W)
Q_{lk}	heat loss due to internal conduction (W)
Q_{lsh}	heat loss due to shuttle conduction (W)
R	gas constant (J/kg K)
Re	non-dimensional Reynolds number
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)
St	non-dimensional Stanton number
SE	Stirling engine
T	temperature (K)
T_{ad}	adiabatic flame temperature of the fuel (K)
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)
u	mean velocity (m/s)
v	mean velocity (m/s)
V	volume (m^3)
W_c	compression work (J/cycle)
W_e	expansion work (J/cycle)
W_{el}	electric work output (W)
W_n	net work by cycle (J/cycle)
W_{ploss}	energy loss due to pressure drop (J/cycle)
W_{br}	engine net brake work (J/cycle)
ΔP	pressure drop (N/m^2)
Z	displacer stroke (m)

Acronyms

ACM	Aspen custom modeller
CHP	Combined heat and power
LeRC	Lewis Research Center

Subscripts

c	compression space
d	displacer
k	cooler space
e	expansion space
f	final value
h	heater space
r	regenerator space
i	inside section
o	outside section
w	wall
wg	wetted gas
pist	piston
0	initial value

Greek symbols

α	surface emissivity
γ	adiabatic constant
η_b	brake efficiency
σ	Stefan–Boltzmann constant ($W/m^2\ K^4$)
ε	regenerator effectiveness
ρ	fluid density (kg/m^3)
θ	crank angle (rad)
μ	viscosity (kg/m s)

1. Introduction

Increasing energy demands, strict environmental regulations and availability of different renewable resources requires the development of environmentally friendly energy systems. This needs an intensive research for the development of efficient and sustainable power technologies. In this scenario, fuel flexible and energy efficient technologies, such as the Stirling engine (SE), are becoming renewed solutions to address these requirements.

Research on these engines was extensive since its invention but the fast growth of parallel technologies, such as internal combustion engines, slowed its progress. However, the development of new materials combined with engineering advances and the potential advantages that the Stirling engine offers in terms of low emissions, low noise and fuel flexibility, has renewed the interest on its development [1–3]. In addition, the successfully integration with micro and small scale combined heat and power (CHP) systems [3–6] increased its potential to become a strong technological alternative. However, Stirling-CHP technology must overcome practical limitations, such as the lower electrical performance compared with other technologies [7]. These limitations arise from the complexities of the system, which must considers