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A transient one-dimensional numerical model for kinetic Stirling engine

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HIGHLIGHTS

- A non-equilibrium thermal mode with considering loses is adopted in Stirling engine.
- Good agreements are achieved for predicting various critical system parameters.
- Differences between helium and hydrogen systems are highlighted and analyzed.
- Pressure drop of helium system is much larger and more sensitive to frequency.

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ABSTRACT

A third-order numerical model based on one-dimensional computational fluid dynamics is developed for kinetic Stirling engines. Various loss mechanisms in Stirling engines, including gas spring hysteresis loss, shuttle loss, appendix displacer gap loss, gas leakage loss, finite speed loss, piston friction loss, pressure drop loss, heat conduction loss, mechanical loss and imperfect heat transfer, are considered and embedded into the basic control equations. The non-equilibrium thermal model is adopted for the regenerator to capture the oscillating features of the gas and solid temperatures. To improve the numerical stability and accuracy, the implicit second-order time difference scheme and the second-order upwind scheme are adopted for discretizing the time differential terms and convective terms, respectively. Experimental validations are then conducted on a beta-type Stirling engine with the extensive experimental data for diverse working conditions. The results show that the developed model has better accuracies than the previous second-order models. Good agreements are achieved for predicting various critical system parameters, including pressure-volume diagram, indicated power, brake power, indicated efficiency, brake efficiency and mechanical efficiency. In particular, both the experiments and simulations show that the Stirling engine charged with helium tends to have much lower optimal working frequencies and poorer performances compared to the hydrogen system. Based on the analyses of the losses, it reveals that the pressure drop in the flow channels plays a critical role in shaping the different behaviors. The pressure drop in the helium system is much larger and more sensitive to the frequency increase due to the much larger viscosity of gaseous helium. Hydrogen is a superior working gas for a Stirling engine. The transient characteristics of the oscillating flow and the associated thermal interactions between gas and solid in the regenerator are finally analyzed in order to have an insight of the complex thermodynamic process. The study provides a promising numerical approach in simulating Stirling engines for further understandings of their operating characteristics and the underling mechanisms.

1. Introduction

A Stirling engine is an external combustion engine based on cyclic compression and expansion of gas at different temperature levels. It attained substantial developments and commercial applications between 1816 and 1900. In twentieth century, however, due to the booming developments of internal combustion engines and oil industry, much less efforts were devoted into Stirling engines. In recent decades, with the growing energy needs and the threat of climate change, technologies for renewable energy and energy efficiency are becoming increasingly important for a sustainable future. Stirling engines have brought back worldwide attentions again because of their flexibility for fuel and promising prospects in solar power generation and low-grade heat recovery [1–4]. In the developments of Stirling engines, a precise model is

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Nomenclature

a	coefficient for finite speed pressure loss
$a_{1,2,3,4,5}$	coefficients for discretized momentum equation
A	cross sectional area, m^2
A_1	small area of cone joint, m^2
A_2	large area of cone joint, m^2
A_{wet}	contact area, m^2
b_k	coefficient for thermal conductivity
b_μ	coefficient for viscosity
$b_{1,2,3,4,5}$	coefficients for discretized gas energy equation
B	coefficient for discretized momentum equation
c	average molecular speed, m/s
c_p	isobaric specific heat, $J/(kg \cdot K)$
c_s	specific heat of solid, $J/(kg \cdot K)$
c_v	isochoric specific heat, $J/(kg \cdot K)$
$c_{1,2,3}$	coefficients for discretized solid energy equation
C	coefficient for discretized gas energy equation
d_h	hydraulic diameter, m
d_x	control volume length, m
D	coefficient for discretized solid energy equation
D_d	displacer diameter, m
D_p	power piston diameter, m
e	eccentricity of rhombic drive, m
E	effectiveness of drive mechanism
f	friction factor
f	frequency, Hz
g	power piston gap, m
$G_{1,2..7,8}$	coefficients for convection term
h	displacer gap, m
L	connecting rod length, m
L_d	displacer length, m
L_p	power piston length, m
k	thermal conductivity, $W/(m \cdot K)$
K	head loss coefficient
K_l	local head loss coefficient
m	mass, kg
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number
p	pressure, Pa
p_b	pressure in buffer space, Pa
p_m	mean pressure, Pa
Pr	Prandtl number
Q_{app}	appendix displacer gap losses, W
Q_h	heating power, W
Q_{hys}	gas spring hysteresis loss, W
Q_{leak}	gas leakage loss, W
Q_{sh}	shuttle heat transfer loss, W
Re	Reynolds number
r	crank radius, m
R_g	gas constant, $J/(kg \cdot K)$
$R_{1,2,3}$	coefficients for gas energy equation
$S_{1,2,3}$	coefficients for solid energy equation
t	time, s
T	temperature, K
T_a	ambient temperature, K
T_g	gas temperature, K
T_L	leakage gas temperature, K
$T_{1,2,3}$	coefficients for momentum equation
u	velocity, m/s
u_p	power piston speed, m/s
v_p	piston speed, m/s
V	volume, m^3
V_b	buffer space volume, m^3
V_c	compression space volume, m^3
V_{clc}	clearance volume of compression space, m^3
V_{cle}	clearance volume of expansion space, m^3
W_b	dissipated power in buffer space, W
W_{bra}	brake power, W
W_{com}	compression power, W
W_{exp}	expansion power, W
W_{ind}	indicated power, W
W_-	forced work, W
x	longitude coordinate, m
X_d	displacer displacement amplitude, m
X_i	coefficient for continuity equation
Y_i	coefficient for continuity equation

Greek symbols

α	heat transfer coefficient, $W/(m^2 \cdot K)$
α_{hys}	hysteresis heat transfer coefficient, $W/(m^2 \cdot K)$
γ	specific heat ratio
δp_{finite}	finite pressure drop, Pa
δp_{fric}	piston friction pressure drop, Pa
δt	time step, s
δx	node distance, m
Δp	pressure amplitude, Pa
ΔV_b	buffer space volume variation amplitude, m^3
η_{bra}	brake efficiency
η_{ind}	indicated efficiency
η_{mech}	mechanical efficiency
θ	phase difference, rad
λ	conductivity factor
μ	viscosity, $kg/(m \cdot s)$
ρ	density, kg/m^3
τ	period of one cycle, s
ϕ	porosity
ω	angular frequency, rad/s

Subscripts and superscripts

b	buffer space
cv	control volume
d	displacer
f	face
i	control volume number
k	time layer number
p	power piston
r	regenerator
s	solid
si	control volume number for solid
so	control volume number for solid
t	tube
w	wall

Abbreviations

PDMA	Penta-Diagonal Matrix Algorithm
TDMA	Tri-Diagonal Matrix Algorithm

highly desirable for predicting thermal performances and characterizing key operation features. Many models have been presented in recent years, including empirical [5–8], analytical [9–18] and numerical [19–54] approaches. For numerical models, they can be categorized into second-order [19–46], third-order [47–50] and multi-dimensional computational fluid dynamics [51–54] methods.

Empirical models correlate the output power and efficiency of a Stirling engine to the heater and cooler temperatures, piston displacement, engine speed and mean pressure based on