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Dynamic simulation of thermal-lag Stirling engines

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HIGHLIGHTS

- First paper dealing with numerical dynamic simulation of thermal-lag Stirling engine.
- Observation on different operating modes, such as rotating, swinging, swinging-to-rotate, and swinging-to-decay modes.
- Study of influential factors affecting the operating modes.
- Optimal geometrical or operating parameters of the baseline case for maximum engine power are found.

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ABSTRACT

The present study is concerned with dynamic simulation of thermal-lag Stirling engines. A dynamic model is built and incorporated with a thermodynamic model to study the engine start process. A prototype engine is designed and simulated by using the dynamic model. In the simulation, different operating modes, including rotating mode, swinging mode, swinging-to-rotate mode, and swinging-to-decay mode, have been observed. The rotating mode is desired and can be achieved if the operating parameters are properly designed. In a poor design, the engine may switch to the swinging or even the swinging-to-decay mode. In addition, it is found that geometric parameters, such as bore size, stroke, and volume of working spaces, also determine the operating mode of the engine. Brake thermal efficiency of the engine is monotonically reduced by increasing engine speed. However, study of the dependence of the shaft power of the engine speed shows that there exists a maximum value of the shaft power at an optimal operating engine speed. The optimal engine speed leading to maximum shaft power is significantly influenced by the geometrical parameters.

1. Introduction

Among the promising renewable energy technologies, dish concentrating solar power (DCSP) systems have received increasing attention from the researchers in recent years. The DCSP system uses a dish to concentrate a large area of sunlight or solar thermal energy onto a small area. Electrical power is produced when the concentrated sunlight is converted to high-temperature thermal energy to drive a Stirling engine that is connected to an electrical power generator [1–3]. Besides, Stirling engine can also use natural gas or gasoline as auxiliary heat source so that it may still delivery power whenever the sunlight is not available. In some other cases, the engine is applied as a micro combined heat and power generation (micro-CHP) unit [4,5] that can supply both residential hot water and electricity.

Earlier technological development in the Stirling engines was reviewed by Thombare and Verma [6]. Traditionally, Stirling engines are categorized into three types namely, α-, β- and γ-types, in terms of the arrangement of pistons, displacers, and cylinders. In the three types of engines, typically an engine is equipped with at least two moving parts, a piston plus a displacer or another piston, in the cylinders. Relative movement between the two moving parts inside the cylinders is so carefully designed by assigning a phase angle between the two moving parts that the working fluid is able to flow back-and-forth between the heating and the cooling parts of the engine to absorb or reject heat, respectively, and then complete a thermodynamic cycle. Cheng and Yang [7] discussed the major differences in the configurations among the three types of engines and investigated relative performance of them. Rochelle and Grosu [8] took into consideration effects of exo-irreversibility and imperfect regeneration in the Schmidt model and obtained analytical solutions for engine speed, power and efficiency for different types of Stirling engine.

In 1979, Ceperley [9] presented a pistonless thermoacoustic Stirling engine which was demonstrated by using a looped tube

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Nomenclature

$\vec{a}_A$, $\vec{a}_P$	acceleration of joints A and P (m/s^2)
a_{Py}	y component of $\vec{a}_P$ (m/s^2)
a_{PAcgx}, a_{PAcgy}	x and y components of acceleration of center of gravity of link PA (m/s^2)
$\vec{AP}$	vector (m)
A_r	cross section void area of regenerative heater (m^2)
c_p	constant pressure specific heat (J/kg K)
c_v	constant volume specific heat (J/kg K)
d_p	diameter of working space and piston (bore size) (m)
D_h	hydraulic diameter of regenerative channel (m)
e_x, e_y	components of vector $\vec{AP}$ (m)
f	Fanning friction factor
f_{gt}, f_{tg}	direction factor
F_{Ax}, F_{Ay}, F_{Px}, F_{Py}	x and y components of forces acting on joints A and P (N)
F_f	friction force at interface between piston and cylinder (N)
G	acceleration of gravity (m/s^2)
h_r	heat transfer coefficient of regenerative heater (W/m^2 K)
I_f, I_{PA}	moment of inertia of flywheel and link PA (kg m^2)
$\vec{i}$, $\vec{j}$, $\vec{k}$	unit vectors in Cartesian coordinate
k	conductivity of working fluid (W/m K)
K_f	empirical coefficient in relation between velocity and pressure drop
l_{PA}, l_r	length of link PA and regenerative heater (m)
m_g, m_t	mass of working fluid in gas chamber and thermal buffer chamber (kg)
m_p, m_{PA}	mass of piston and link PA (kg)
Nu	Nusselt number in regenerative heater
$\vec{OA}$	vector (m)
p_b, p_g, p_t	pressure in buffer, gas, and thermal buffer chambers (Pa)
p_0, p_{b0}	charged pressure in working space and buffer chamber (Pa)
P	perimeter of regenerative heater channel (m)
$\dot{Q}_{in}$	total heat input rate (W)
$\dot{Q}_{in,g}$, $\dot{Q}_{in,r}$, $\dot{Q}_{in,t}$	rate of heat transfer in gas chamber, regenerative heater, and thermal buffer chamber (W)
r	offset distance from crank to shaft center (m)
R	gas constant (J/K mol)
Re	Reynolds number of flow through regenerative heater
R_c, R_g, R_t	thermal resistance in compression chamber, gas chamber, and thermal buffer chamber (K/W)
R_l	total thermal resistance in cold side (K/W)
s	coefficient in Eq. (18) (m^{-1})
t	time (s)
T	temperature of working fluid through regenerative heater (K)
T_g, T_t	temperature of working fluid in gas chamber and thermal buffer chamber (K)
T'_g, T'_t	temperature of working fluid at outlet of regenerative heater in gas chamber and thermal buffer chamber (K)
T_{wg}, T_{wt}	wall temperatures of gas chamber and thermal buffer chamber (K)
T_{wr}	temperature of regenerative porous matrix (K)
$\bar{u}$	average velocity of working fluid through regenerative heater (m/s)
$\vec{v}_A$, $\vec{v}_P$	velocity of joints A and P (m/s)
v_{Py}	y components of $\vec{v}_P$ (m/s)
V_c, V_g, V_t	volume of compression chamber, gas chamber, and thermal buffer chamber (m^3)
$\dot{W}_i$, $\dot{W}_s$	indicated power and shaft power (W)
Greek symbols	
γ	c_p/c_v
Γ_f, Γ_O	torque due to friction force and working gas pressure (N m)
Γ_{load}	loading torque (N m)
η_t, η_m, η_b	thermal, mechanical, and brake thermal efficiencies
μ	dynamic viscosity of working fluid (N s/m^2)
μ_p	coefficient of kinetic friction
ϕ	crank angle (rad)
ρ_{tg}	density of working fluid through regenerative heater (kg/m^3)
τ	period (s)
ω_O, ω_{PA}	angular velocity of flywheel and link PA (rad/s)
ω_{ave}	average angular velocity of flywheel (rad/s)
$\dot{\omega}_O$, $\dot{\omega}_{PA}$	angular acceleration of flywheel and link PA (rad/s^2)
Superscripts	
0	initial
k	time step

with a differentially heated regenerator. The prototype engine with no moving piston revealed a concept that the traveling-wave thermoacoustic oscillations can be regarded as a kind of heat engine. Lately, Swift [10,11] and Swift and Garrett [12] found that the thermoacoustic concepts could be equally applicable to some reciprocating heat engines. Hamaguchi et al. [13] developed a simple engine called "pulse tube engine", which is equipped with one single reciprocating piston and does not utilize the natural frequency as in the thermoacoustic Stirling engines. Yoshida et al. [14] measured work flux density distribution over the cross section of the pulse tube to elucidate the work source of the engine. They found that the pulse tube engine belongs to the standing wave engine group and the work source resides not in the porous stack but in the pulse tube.

On the other hand, one other concept of the Stirling engine was proposed by Chen and West [15] and the engine was entitled "thermal-lag Stirling engine" afterward. The thermal-lag Stirling engine requires no displacer and its connected link, and instead, a porous-medium stack is fixed in the cylinder as a static regenerative heater. The first prototype of the thermal-lag engine and its performance testing were made by Tailer [16,17]. He stated that the engine experiences imperfect cooling and heating processes in the working spaces, and the imperfect cooling and heating cause a time lag in the gas temperature response as following the movement of the piston. Thus, the engine was named "thermal-lag" engine. A numerical model for the thermal-lag engine was proposed by Arques [18]. In this report, it could be observed that the areas of the thermodynamic cycles of the thermal-lag engine plotted in the p–V diagrams are rather small. It seemingly implies that that the thermal-lag engines are not capable of producing appreciable work output. Recently, Cheng and Yang [19] made a prototype thermal-lag engine of 15 W and also developed a thermodynamic model for predicting transient variations of the thermodynamic properties in the individual working spaces of the engine. Effects of geometrical and operating parameters, such as heating and cooling temperatures, volumes of the chambers, thermal resistances, stroke of piston, and bore size on indicated power output and thermal efficiency were also evaluated.

Similar to the pulse-tube engine, the thermal-lag engine has only one moving part (piston) and a static part (regenerative