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A complete model for dynamic simulation of a 1-kW class beta-type Stirling engine with rhombic-drive mechanism

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ABSTRACT

This study is aimed at development of a theoretical model by combining modified non-ideal adiabatic model and dynamic analyses in order to predict the dynamic behavior of a 1-kW class beta-type Stirling engine with rhombic drive mechanism during starting. All friction losses caused by piston rings, bearings, and seals are taken into consideration. The total torque can be expressed in a closed-form relation by including the effects of the inertia force, the gravity force, the pressure force and the friction forces. The transient variations of instantaneous angular velocity under different operating conditions are simulated. The variations of the engine speed, the shaft power, and the mechanical efficiency of the engine under time-varying output torque are also described. The results also indicate that engine has a minimum and a maximum operating speed, and a minimum initial speed for starting. A prototype engine is built and tested for validation of the present model. Experiments on the transient behavior of the prototype engine during starting and under time-varying operating conditions are conducted. The numerical predictions by the present model closely agree with the experiments. And the maximum power generated by the engine is 1358 W at 1313 rpm in the test.

1. Introduction

The first Stirling engine was invented by Robert Stirling in 1816 [1], and then a great variety of the Stirling engine have been developed during the past two hundred years. In theory, the theoretical efficiency of an ideal Stirling engine with a perfect regeneration is equal to the Carnot efficiency. Therefore, Stirling engine is efficient in wide range of power as compared with other existing heat engines. In addition, since the working medium in the Stirling engine is contained in a closed chamber, the engine is quiet even in the extreme environment like in space [2,3] or under water [4]. The Stirling engine may be compatible with a variety of external heat sources, such as radioisotope energy, solar energy, waste heat, combustion of fuel or hydrogen, geothermal energy, and so on. For example, one of the potential applications of Stirling engine is the concentrating solar power (CSP) system [5], which focuses the solar radiation to a focal point located at the thermal receiver of Stirling engine. Then, the high temperature concentrated solar energy drives the heat engine as well as the electric generator to produce the electric power. The capacity of individual CSP systems ranges from 2 to 50 kW, and the overall conversion efficiency exceed 29% [6]. One other possible application is called combined heat and power (CHP) system based on combustion of natural gas and liquefied petroleum gas. The CHP system provides process heating or hot water for a factory or building and also generates electric power by using Stirling engine. The overall efficiency of CHP system may exceed 80% [7]. Owing to these above advantages, the Stirling engines have come to the attention of a great number of researches. Therefore, numerous theoretical analyses and design methods have been presented. Reviews on the progress of the Stirling engine technology are available in Refs. [8,9].

The first theoretical model for estimating the work of Stirling engine was proposed by Schmidt in 1871 [10]. Urieli and Berchowitz [11] presented an ideal adiabatic model, in which the temperatures of working fluids in the heater, the regenerator, and the cooler are fixed, while in the expansion and the compression spaces the fluid temperatures are varying. The model can reflect the effects of phase angle, swept volume ratio, and dead volume ratio, but the relationship between the temperatures of working fluid and the

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Nomenclature

A	cross section area (m^2)
a	acceleration (m/sec^2)
c_v, c_p	constant-volume and constant-pressure specific heats (J/kg•K)
$c_{p,r}$	specific heat of regenerator matrix (J/kg•K)
e_d, e_p	dimensionless design parameters of the rhombic-drive mechanism
$\dot{E}$	energy (W)
F	force (N)
g	gravitational acceleration (m/sec^2)
I	moment of inertia ($kg \cdot m^2$)
$\vec{i}$, $\vec{j}$	unit vectors in x- and y-directions
l_d, l_l, l_p, l_u	lengths of links (m)
M	moment (N• m)
m	mass (kg)
$\dot{m}$	mass flow rate (kg/sec)
p	pressure (Pa)
Δp	pressure drop (Pa)
R	gas constant (J/kg• K)
R	thermal resistance (K/W)
r	radius (m)
t	time (sec)
T	temperature (K)
V	volume (m^3)
V_{es}	volume swept by displacer (m^3)
v	velocity (m/s)
$\dot{W}$	power produced by torque (W)
$\dot{W}_{out}$	shaft power (W)
x, y	rectangular coordinates

Greek Symbols

α	angular acceleration (rad/sec^2)
χ	dead volume ratio
ε_d, ε_p	dimensionless design parameters of rhombic drive mechanism
ϕ_{AB}	crank angle (rad)
γ	specific heats ratio (c_p/c_v)
η_m	mechanical efficiency
μ	coefficient of friction
τ	torque (N•m)
ω	angular velocity (rad/sec)
ξ	area ratio
ψ	angle of the counter weight (rad)
ζ	coefficient of Eq.(27) (sec/m)

Superscripts

i	time step
'	symmetrical parts

Subscripts

0	initial
A, B, C, D	joints
b	buffer space
bb	bearing bore
bf	bearing friction
c	compression space
cg	center of gravity
cri	critical
cw	counter weight
d	displacer
db	displacer bar
dp	pressure drop
e	expansion space
g	gear
gra	gravity
h	heater
in	input
j	index of link
k	index of bearing
l	cooler
$mass$	mass
n	normal
out	output
p	piston
pre	pressure
Δp	pressure drop
r	regenerator
rf	ring friction
s	shaft
sf	shaft friction
w	wall
x	x direction
y	y direction

heat transfers in the individual chambers are not able to predict. An analytical model which took the heat transfers in the working spaces and the imperfect regeneration into consideration was presented by Kaushik and Kumar [12,13] and later discussed by Tlili [14]. This model was named the finite time thermodynamic model by Kaushik and Kumar [12,13]. In this model, the thermodynamic cycle is divided into four processes, and each of the four processes lasts a limited period of time. The model can be used to predict the indicated power and the thermal efficiency of the engine; however, the effect of phase angle is not evaluated due to the assumption of isochoric processes. Therefore, a large number of numerical models were developed and improved the deficiencies of the earlier models. Yu and co-authors [15] proposed a non-ideal adiabatic model to predict the power and the efficiency of a double-acting Stirling engine. In this model, authors still assumed that the temperatures of working fluid in the heater and the cooler are constant, but they included the heat transfers between the working fluid and the wall boundaries during a thermodynamic cycle. In addition, the heat loss in the regenerator and the pressure loss in the heat exchanger were taken into account. Based on the non-ideal adiabatic model, Cheng, Yang and Lam [16] adopted Senft's theory to estimate the shaft power of Stirling engine, and successfully applied the model in development of a 300-W beta-type Stirling engine with rhombic-drive mechanism (BTSERD). Furthermore, Yang and co-authors [17] proposed a modified non-ideal adiabatic model to analyze a 500-W engine. Authors found that the modified model is capable of predicting the transient temperature variations in the working spaces by introducing the energy equations for all working spaces. However, since the above-mentioned thermodynamic analysis cannot lead to detailed information of the thermal and flow fields in the Stirling engine. To yield the detailed information of the thermal and flow fields, such as distributions of velocity, temperature, density and pressure in all the working spaces, one may need to acquire the computational fluid dynamic (CFD)