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Development of a beta-type Stirling engine with rhombic-drive mechanism using a modified non-ideal adiabatic model

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HIGHLIGHTS

- An efficient and practical model is presented to predict performance of Stirling engine.
- A 500-W prototype Stirling engine is successfully built based on the parametric study by the model.
- Experiments are conducted on shaft power of the engine for verifying the model.
- Effects of pressure drops, mechanical loss, mesh number, heating temperature and rotation speed are evaluated.

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ABSTRACT

The aim of this study is to develop an efficient theoretical model that can more accurately predict the performance of the designed engine. The developed model is practically further applied in the development of a 500-W engine. The theoretical model is built by modifying the existing non-ideal adiabatic analysis to more accurately predict performance the designed engine. In this model, pressure drops in the heater, the regenerator and the cooler caused by fluid friction, channel sudden expansion and sudden contraction are taken into consideration. Furthermore, an empirical formula for the mechanical loss as a function of rotation speed of the engine is obtained by experiments and introduced into the model. The shaft power, indicated power, and thermal efficiency of the engine are determined. Furthermore, a prototype engine is then built and tested to validate the model. Experimental measurements on the power output are conducted in this study. It is found that maximum shaft power of the prototype engine can reach 556 W at rotation speed of 1665 rpm and at a heating temperature of 1100 K.

1. Introduction

Stirling engines have prospects of being developed into low-pollution, quiet power-producing machines with excellent fuel economy and multi-fuel capability [1–3]. Therefore, Stirling engines have received much attention from researchers in energy-related fields in recent years. In theory, an ideal Stirling engine with a perfect regeneration is capable of reaching a thermal efficiency equal to the Carnot cycle efficiency. The compatibility of Stirling engines with various external thermal resources makes them suitable for engineering applications with waste heat, biomass fuel, hydrogen, radioisotope, solar energy, geothermal energy, fossil fuel, etc. For example, a Stirling engine can be used as an electrical power source based on radioisotope nuclear energy in deep space missions [4]. Stirling engines can also be applied in the air-independent-propulsion (AIP) system of submarines based on combustion of diesel oil [5,6]. In residential applications, Stirling engines can be integrated with a micro-combined-heat-and-power (μ-CHP) system [7–9], in which the Stirling engines are driven by combusting natural gas to generate electrical power; in parallel, another part of the combustion heat is used to produce hot water. In addition, the application of Stirling engines is extended to concentrating solar power (CSP) systems; in such a system, a Stirling engine is integrated with a solar dish, which focuses the solar radiation onto a thermal receiver of the Stirling engine, thereby converting solar energy into electricity via the Stirling engine [10,11]. The dish-Stirling CSP system has exhibited high overall efficiency exceeding 29% [10,12]. Its potential for high efficiency and the ability to use a wide variety of fuels has made the Stirling engine a potential alternative power source with widespread use. Therefore, a number of design methods and testing results have also been reported in recent years [13–15].

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Nomenclature

C_0, C_1, C_2, C_3	coefficients in Eq. (16)
c_v, c_p	constant-volume and constant-pressure specific heat (J/kg K)
c_{pr}	specific heat of the regenerator matrix (J/kg K)
e_d, e_p	dimensionless design parameter of the rhombic-drive mechanism
f	number of cycles per second (1/s)
m	mass of working fluid (kg)
$\dot{m}$	mass flow rate (kg/s)
m_{rw}	mass of the regenerator matrix (kg)
p	pressure (Pa)
p_0	charged pressure (Pa)
Δp	pressure drop (Pa)
$\dot{Q}_{in}$	total heat input (W)
R	specific gas constant (J/kg K)
R	thermal resistance (K/W)
t	time (s)
T	temperature (K)
V	volume (m^3)
V_s	volume swept by the displacer (m^3)
$\dot{W}_i$	indicated power (W)
$\dot{W}_{loss}$	mechanical loss (W)
$\dot{W}_s$	shaft power (W)

Greek symbols

ε_d, ε_p	dimensionless design parameter of the rhombic drive mechanism
γ	specific heats ratio (c_p/c_v)
η	overall thermal efficiency
θ	crank angle (rad)
ω	engine speed (rad/s)

Superscript

i	time step

Subscript

ave	average value
c	compression space
d	displacer
dead	dead volume
e	expansion space
f	friction
h	heater
i	ideal
ni	non-ideal
j	number of element
l	cooler
m	minor
p	piston
r	regenerator
rm	regenerator matrix
w	wall

In general, Stirling engines can be classified into alpha-type, beta-type and gamma-type engines in terms of their configurations and number of pistons and displacers. Kirkley [16] optimized these three types of Stirling engines and found that the beta-type engine has better performance than other types. To design the geometric parameters of a Stirling engine for achieving higher indicated work, Walker [17] presented an optimal combination of phase angle and swept volume ratio. Recently, a comprehensive study of optimization of the three types of Stirling engines was performed by Cheng and Yang [18]. In their study, the authors presented design charts for the optimal combinations of phase angle and swept volume ratio with the help of an extended mechanical efficiency theorem proposed by Senft [19]. The obtained theoretical results confirmed the finding of Kirkley [16] that the beta-type Stirling engine exhibits relatively higher performance compared with the other two types.

Under these circumstances, this study is intended to develop an efficient theoretical model that can more accurately predict performance the designed engine. In this model, pressure drops in the components caused by fluid friction, channel sudden expansion and sudden contraction are taken into consideration. Furthermore, an empirical formula for the mechanical loss as a function of rotation speed of the engine is included in the model. The model is practically applied in the development of a real 500-W engine for verification. The 500-W beta-type Stirling engine is equipped with a rhombic-drive mechanism. The rhombic-drive mechanism is adopted in the design because it leads to coaxial motions for both the piston and the displacer, thereby significantly reducing the friction at the peripheries of both the piston and the displacer. Optimization of the rhombic-drive mechanism used in the beta-type Stirling engine was performed more recently by Cheng and Yang [20] based on a dimensionless analysis. Note that the 500-W beta-type Stirling engine with the rhombic-drive mechanism is developed to verify the theoretical model. Nonetheless, the present model is flexible and not limited to the engine of 500-W class nor to the rhombic-drive mechanism.

Theoretical models for analysis of the Stirling engine can be divided into first-, second- and third-order models, according to Martini [21]. To name a few, Schmidt theory [22] and finite time thermodynamic analysis [23] are referred to as the first-order model. In the model, an isothermal assumption is typically made, and the pressure drop due to friction is always neglected. The first-order model is simple, and the solutions can be yielded rather rapidly; however, the accuracy of the solution may not be satisfactory. Alternatively, the performance of Stirling engines can be predicted by using Computational Fluid Dynamics – CFD simulation, which is referred to as the third-order model. For example, Cheng and Chen [24] performed a numerical simulation of the thermofluid dynamics of a 1-kW beta-type Stirling engine using CFD analysis. In their simulation, a mathematical model consisting of a number of partial-differential equations was solved based on the framework of a commercial three-dimensional CFD package. The third-order model is able to provide more detailed information that cannot be easily obtained either from experiments or from computations by using the first- or second-order model. However, note that the CFD analysis is very time consuming. In the study of Ref. [25], the authors found that it might take a week or two to complete the simulation for just one case. Thus, it is impracticable to use the third-order model in the preliminary design process of the prototype engine. Herein, a more practical second-order model is developed in this study by considering a balance between accuracy and computational efforts.

The ideal adiabatic model presented by Urieli and Berchowitz [25] is a second-order model. In the ideal adiabatic model, the fluid temperatures in the heater, the regenerator, and the cooler are all fixed constant, and the expansion and compression spaces are assumed to be adiabatic. However, in practice, the temperatures in the heater, the regenerator, and the cooler should be varied with