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A novel second-order thermal model of Stirling engines with consideration of losses due to the speed of the crack system

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ABSTRACT

Very accurate second-order thermal models have been developed for the thermal simulation of Stirling engines in recent years. One of the last ones is the comprehensive polytropic model of Stirling engine called the CPMS model. The accuracy of the CPMS model was found to be sufficient for the nominal operation of a prototype Stirling engine known as the GPU-3 engine. Nevertheless, the accuracy of the CPMS model was drastically reduced at high rotational speeds of the engine. In this paper, power loss and pressure change due to the inertial force of the crank system were integrated into the CPMS thermal model in order to compensate inaccuracy of the CPMS model at high rotational speeds. Moreover, the effect of rotational speed on the gas temperature in heater and cooler was also incorporated. A precise model for evaluating the mechanical friction loss was also employed and compared with the simple frictional model of the simple frictional model used in the CPMS. The model was examined on the GPU-3 engine, and it was found that it has superior accuracy compared to the previous thermal model over the entire working regime of the GPU-3 engine.

1. Introduction

The Stirling engine is a type of external heat engine that can be used to generate power from the thermal energy from various sources from fossil fuels to renewable heat sources such as the solar energy. On the other hand, it has the advantage of quieter operation compared to internal-combustion engines. On the other hand, it could be used in the reverse cycle as the Stirling cooler or refrigerator. Due to these advantages, Stirling systems are suitable alternatives for power generation and refrigeration in various application. For the proper design of Stirling engines/coolers, accurate thermal models that are able to predict the thermal performance of Stirling engines is required. First-order analytical model [1–5], second-order numerical models [6–13], and third-order models [14–19] are usually employed for this purpose. The first-order or closed-form models are those models that can easily estimate thermal performance of Stirling engine based on the operating parameters; however, they usually suffer from lack of enough accuracies required by designers. On the other hand, those are usually developed for Stirling cycles, not Stirling engines and coolers; therefore, those cannot be used to study the effect of engines/cooler's parameters. For accurate simulation of Stirling engines/coolers, numerical second-order and third-order models are used. The third-order models [14–19] are numerical models that are developed based on the computational fluid dynamics, CFD. Third-order models suffer from some limitation, including the high computational cost and the lack of generality that makes it impossible to be used to simulate every type of the engine. Instead, the third-order models should be developed for a specific type of engine, and the results cannot be extended to other engines. The second-order models are numerical zero-dimensional models that have sufficient accuracy for most cases. Moreover, they can be easily applied to various Stirling engines. In other words, second-order models can be utilized to simulate every type of Stirling engines/coolers with reasonable accuracy. On the other hand, due to much lower computation time compared to third-order models, these models can be used easily for design and optimization purpose. In second-order models, the engine is divided into the five compartments and governing equations of energy and mass are applied for each compartment as a function of the rotational angle of the crank system (time) while the spatial dependence of governing equations is ignored. Therefore, these models are also called as the zero-dimensional models. It means that the governing equations are not dependent on any coordinate. In these models, a system of boundary value ordinary differential equation, ODE, with respect to the crank angle (time) is obtained and converted into an initial value problem. Consequently, they are solved using the fourth order Runge-Kutta method. An early second order model was developed by the Urieli and Berchowitz [6] considering adiabatic expansion/

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Nomenclature

General

A	area (m^2)
a	the distance between the center of gravity from the linkage and the center of the yoke (m)
$B_1,...,B_n$	constants values
c	the average speed of molecules ($m\,s^{-1}$)
c_n	polytropic specific heat ($kJ\,kg^{-1}\,K^{-1}$)
c_p	specific heat at constant pressure ($kJ\,kg^{-1}\,K^{-1}$)
c_v	specific heat at constant volume ($kJ\,kg^{-1}\,K^{-1}$)
D	diameter (m)
d	differential of a parameter or diameter (m)
e	the eccentricity of the crank system (m)
F	force (N)
fr	rotation frequency of engine (Hz)
f_{re}	Reynolds friction factor
$G_1,...,G_n$	constants values
$H_1,...,H_n$	constants values
I	mass moment of inertia ($kg\,m^2$)
i	gyration radius (m)
J	the annular gap between piston and cylinder (m) and or moment of inertia (N m)
k	gas conductivity
k_g	specific heat ratio [C_p/C_v]
L	length (m)
M	mass of the working fluid (kg)
m	mass of a component of the crank system (kg)
N_r	the rotation speed of the engine (rpm)
NTU	number of transfer units
$p_{b1},...,p_{bn}$	constants values
P	pressure (kPa)
Pr	Prandtl number
Q	heat transfer (kJ)
$q_{b1},...,q_{bn}$	constants values
R	universal gas constant (kJ/kg K)
r	crank radius (m)
Re	Reynolds number
S	stroke (m)
St	Stanton number
T	temperature (K) or torque (N m)
u	the velocity of gas flow ($m\,s^{-1}$)
u_p or $\dot{x}_p$	the linear velocity of the piston ($m\,s^{-1}$)
W	output work (kJ)
$\dot{W}$	power (kW)
V	volume (m^3)
R_{cond}	conduction resistance ($m\,K\,W^{-1}$)
$\ddot{x}_p$	piston acceleration ($m\,s^{-2}$)

Greek

η	efficiency
μ	viscosity (Pa s) and friction coefficient
ρ	density ($kg\,m^{-3}$)
ε	effectiveness
θ	crank-angle (deg)
γ	specific heat ratio (c_p/c_v)
ω	angular velocity ($rad\,s^{-1}$)

Subscript

b	brake, buffer
c	compression space
cc	crank-case
cj	crank-journal
ck	compression-cooler space interface
cp	crank-pin
cs	crank-pin bearing shell
d	displacer
dc	displacer's connecting rod
e	expansion space
f	friction
h	heater
he	heater-expansion space interface
ho	hot gas output
hi	hot gas input
$ind.$	indicated
k	cooler (kooler)
kr	cooler (kooler)-regenerator interface
$leak$	leakage
m	mechanical
$m\omega$	effect of inertial force of mass (m) at the rotational frequency (ω)
p	piston
pc	piston's connecting rod
pp	piston-pin
r	regenerator
rh	regenerator-heater interface
$Shuttle$	shuttle effect
0	environment condition
wh	heater wall
wk	cooler wall

Abbreviation

CFD	Computational Fluid Dynamics
CPMS	Comprehensive Polytropic Model of Stirling engine
GPU	Ground Power Unit
kWe	Kilowatts of electrical power
ODE	Ordinary Differential Equation
PSVL	Polytropic analysis of Stirling engine with Various Losses

compression processes as well as effects of non-ideal heat recovery of the regenerator, non-ideal heat transfer in cooler and heater, and pressure drops in heat exchangers. Their model was called as the Simple or Adiabatic model and later, this model was used by a number of researchers [7–12] to be modified for various loss mechanisms of real engines. In a new branch of the second-order models, the adiabatic expansion/compression processes of previous studies were substituted with polytropic processes [10–12]. In a more recent work, Babaelahi and Sayyaadi [25] developed a new thermal model called as PSVL (polytropic analysis of Stirling engine with various losses). In their work, the adiabatic or isothermal assumption of expansion/compression processes of previous works [15–24] was substituted with polytropic processes. In addition, the effects of the shuttle conduction heat loss, mass leakage from the working spaces to the crankcase, finite motion of the piston (based on finite speed thermodynamic model), mechanical friction, gas throttling, longitudinal conduction loss along the regenerator's wall, non-linear distribution of temperature along the regenerator, and non-isothermal behavior of the heater/cooler were considered in those generations of polytropic second-order model. In a modification on PSVL, Sayyaadi and Babaelahi [11] modified their