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Thermodynamic and dynamic analysis of an alpha type Stirling engine and numerical treatment

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ABSTRACT

In this study, the nodal thermodynamic and dynamic analysis of an alpha type Stirling engine driven by Scotch-yoke mechanism is presented. The nodal thermodynamic section of the analysis is performed via 15 nodal volumes. The temperature variations in nodal volumes are calculated by means of the first law of the thermodynamics given for the open systems. The pressures in all of the nodal volumes are assumed to be equal and calculated via Schmidt relation. The momentary masses in nodal volumes are calculated via the perfect gas relation. The dynamic section of the analysis involves the motion equations of pistons and crankshaft. The motion equations are derived by means of the Newton method. In the derivation of the motion equations of pistons, the working fluid forces and friction forces are considered beside the inertia forces. In the derivation of motion equation of the crankshaft, moments of working fluid forces, moments of friction forces, the moment of external load and the moment of starter motor are considered as well as mass inertia moments. It is estimated that an engine having 1.8 L swept volume, 1000 K hot end temperature, 400 K cold end temperature, 3000 cm^2 total inner heat transfer area, 5.1 bar charge pressure and 2000 W/m^2 K inner heat transfer coefficient is capable of producing a shaft power above 2 kW. For these inputs and shaft power; the speed, speed fluctuation and torque are optimized as 138 rad/s, 16% and 14.9 N m respectively. The presented analysis is useful for engine development studies.

1. Introduction

Stirling engines are externally heated, closed cycle, piston type energy conversion machines invented in 1816 by Robert Stirling. The ideal theoretical cycle of Stirling engines consists of a constant temperature compression process, a constant volume heating process, a constant temperature expansion process and a constant volume cooling process [1]. The cycle is a regenerative cycle and its thermal efficiency is equal to the efficiency of the Carnot cycle [2]. The cycle of practical Stirling engines exhibits a considerable amount of deviations from the ideal theoretical cycle and as the result of these deviations, performance values of practical Stirling engines such as thermal efficiency, cyclic work generation, running speed, power output, specific power and torque become uncompetitive with Internal Combustion Engines. Stirling engines have too much application fields. Due to external heating property, Stirling engine enables the conversion of clean and renewable energies into useful energy forms. Among the clean and renewable energies, the solar energy comes first. There are some opinions that a hybrid engine with a higher thermal efficiency may be developed by integrating the Stirling and Internal Combustion Engines [3]. Beside thermal efficiency, the hybrid engines are considered to have better exhaust emissions. Stirling engines have also importance in space investigations as well [4]. The domestic Combined Heat and Power cogeneration systems [5]and, the Combined Cooling Heating and Power cogeneration systems [6] are also considered as application fields for Stirling engines. For the current situation, the development level of Stirling engine is not appropriate for commercial use but, too much investigations are undergoing to improve its development level. In the following paragraphs some of studies conducted within last decades are presented.

Costea et al. [7] developed an analytical model of estimating the performance of Stirling engines based on the first and second laws of the thermodynamics. Authors named this model as Finite Speed Thermodynamic analysis. From some aspects, the analysis resembles the Finite Time Thermodynamic analysis developed in 1975 by Curzon and Ahlborn [8]. The model presented by Costea et al. [7] directly connects the irreversibilities to the operation speed of the cycle. Beside the heat transfer irreversibilities, flow and mechanical frictions were taken into account. As the result of this study, authors stated that, the practical efficiency of Stirling cycle engines is about the half of the ideal Stirling

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Nomenclature

A_i	heat transfer area of nodal volumes (m^2)
A_p	top area of pistons (m^2)
A_{cc}	cold cylinder heat transfer area (m^2)
A_{hc}	hot cylinder heat transfer area (m^2)
A_{cs}	area augmentation in cold cylinder by slotting (m^2)
A_{hs}	area augmentation in hot cylinder by slotting (m^2)
C_v	specific heats at constant volume (J/kg K)
C_p	specific heats at constant pressure (J/kg K)
C_d	damping constant of piston due to lubrication (N s/m)
C_l	damping constant of crankshaft due to external loading (N m s/rad)
C_m	damping constant of crankshaft bearings due to lubrication (N m s/rad)
D	cylinder diameter (m)
F_h	total force exerting on hot piston (N)
F_c	total force exerting on cold piston (N)
F_∞	dry friction on the piston surface (N)
h_e	specific enthalpy of entering fluid into a nodal volume (J/kg)
h_o	specific enthalpy of outgoing fluid from a nodal volume (J/kg)
H_i^e	enthalpy flow into a nodal volume within the time step Δt (J)
H_i^o	enthalpy flow out of a nodal volume within the time step Δt (J)
I_{cr}	inertia moment of crankshaft and flywheel (m^2 kg)
p	working volume pressure (Pa)
p_d	engine block pressure (Pa)
p_{ch}	charge pressure (Pa)
Q_i	the heat exchange of the working fluid with solid boundaries of the nodal volume i within a time step of Δt (J)
Q_H	heat exchange of working fluid with heater during Δt (J)
ΔQ_R	heat exchange of working fluid with regenerator matrix during Δt (J)
Q_C	heat exchange of working fluid with cooler during Δt (J)
R	crank radius (m)
R_m	radius of crank journals (m)
$\mathfrak{R}$	gas constant (J/kg K)
L_p	piston length (m)
L_m	length of crank journals (m)
m	total value of working fluid mass (kg)
Δm_i	mass variation in a nodal volume within the time step (kg)
m_i	gas mass in nodal volumes (kg)
m_i^f	mass of working fluid in nodal volume i at pervious time step (kg)
m_e	mass flow into the nodal volume i during Δt (kg)
m_o	mass flow out of the nodal volume i during Δt (kg)
m_p	piston mass (kg)
M_h	moment of hot piston force (N m)
M_c	moment of cold piston force (N m)
M_s	starter moment (N m)
M_q	external load (N m)
Δt	length of time interval (s)
ΔT_i	temperature variation in a nodal volume within the time step Δt (K)
T_i	gas temperature in nodal volume i within the current time step (K)
T_i^f	gas temperature in nodal volume i within the previous time step (K)
T_i^w	wall temperature of a nodal volume within the time step Δt (K)
U	distance between cylinder top and crankshaft center, ($U = R + Z + \varepsilon$) (m)
$(\Delta U)_i$	internal energy variation in a nodal volume within the time step Δt (J)
ΔV_i	variation of a nodal volume within the time step Δt
V_{cc}	volume of cold cylinder, Fig. 1 (m^3)
V_{hc}	volume of hot cylinder, Fig. 1 (m^3)
V_i	value of a nodal volume (m^3)
V_w	total value of nodal volume (m^3)
W_i	work generation in the nodal volume i within the time step of Δt (J)
y_h	distance between coordinate origin and the top of hot piston, Fig. 2 (m)
y_c	distance between coordinate origin and the top of cold piston, Fig. 2 (m)
Z	distance between piston top and slot, Fig. 2 (m)

Greek symbols

ε	minimum distance between piston top and cylinder top (m)
δ	thickness of lubricant layer between piston and cylinder, or journal and bearing (m)
μ	dynamic viscosity of working fluid (N s/m^2)
ω	average speed (rad/s)
$\overline{\omega}$	average crankshaft speed (rad/s)
Ω	a dummy variable to avoid divergence
θ	crankshaft angle (rad)
λ_i	heat transfer coefficient at a nodal volume (W/m^2 K)

Subscripts

i	counter for nodal volumes
n	counter for time steps

cycle.

Kaushik and Kumar[9] conducted a finite time thermodynamic analysis of an endoreversible Stirling engine. The study intended to maximize the power output and corresponding thermal efficiency. For the case of 100% regenerator efficiency, the power output and thermal efficiency of the endoreversible Stirling engine is found to be equal to the power output and thermal efficiency of the Carnot cycle. Finkelstein and Organ [10] made a comprehensive examination of studies conducted before the year 2000 and published as a book. The book presents most of the Stirling driving mechanisms used before the year 2000 as well as theoretical analysis. By using Schmidt approximation, Senft [11] optimized the geometry of an alpha type Stirling engine with respect to the indicated and shaft works. The influences of cold to hot space temperature ratio, piston to displacer swept volume ratio, dead volume to displacer swept volume ratio and phase angle were examined. The optimal value of the piston to displacer swept volume ratio was found to be about 1. The optimal value of the phase angle was found to be in the range of 83°-91°.

Tanaka et al. [12] conducted an experimental study and determined friction and heat transfer characteristic of three different wire screens made of stainless steel which were named as WN50, WN100, WN150 and WN200. Here WN50, WN100, WN150, etc. indicates 50, 100 and 150 wires per inch of screen. For WN50 wire screen, the heat transfer coefficient range was determined as $1000 < h < 3000$ W/m^2 K while the fluid velocity is ranging from 1 m/s to 6 m/s.

Rogdakis et al. [13] developed thermodynamic and dynamic analysis of the beta type free piston Stirling engine. The dynamic analysis involves three motion equations; one of them is for displacer, other is

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