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Thermodynamic analysis of a beta-type Stirling engine with rhombic drive mechanism

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ABSTRACT

This paper presents a theoretical investigation on kinematic and thermodynamic analysis of a beta type Stirling engine with rhombic-drive mechanism. Variations in the hot and cold volumes of the engine were calculated using kinematic relations. Two different displacer cylinders were investigated: one of them had smooth inner surface and the other had axial slots grooved into the cylinder to increase the heat transfer area. The effects of the slots grooved into the displacer cylinder inner surface on the performance were calculated using nodal analysis in Fortran. The effects of working fluid mass on cyclic work were investigated using 200, 300 and 400 W/m^2 K convective heat transfer coefficients for smooth and grooved displacer cylinders. The variation of engine power with engine speed was obtained by using the same convective heat transfer coefficients and isothermal conditions. The convective heat transfer coefficient was predicted as 104 W/m^2 K using the experimental results measured from the prototype engine under atmospheric conditions. The variation in cyclic work determined by the experimental study was also compared with the theoretical results obtained for different convective heat transfer coefficients and isothermal conditions.

1. Introduction

The Stirling engine is an external combustion machine, invented by Robert Stirling in 1816. The thermodynamic cycle of the Stirling engine consists of two isothermal and two constant volume processes [1,3–5]. Since the Stirling engine is externally heated, it can be operated with solar radiation, geothermal energy, natural gas, biomass, fossil fuels and all kinds of energy sources [2,6,7]. In Stirling engines, mostly air, helium and hydrogen are used as the working fluid [8,9]. Stirling engines have some advantages such as high thermal efficiency, low noise, low pollutant levels, and long life when compared with internal combustion engines [8–11].

Kinematic Stirling engines are classified into three different mechanical arrangements as α, β, and γ [3,5,12]. α-type engines use two pistons situated with a 90° phase angle in separate cylinders. Cylinders can be configured in the form of a V or placed as parallel to each other. In β-type Stirling engines, the cycle is conducted by a piston and a displacer operating concentrically in the same cylinder. In γ-type engines, a piston and a displacer are situated in separate cylinders. The displacer cylinder provides heating and cooling of the working fluid. A regenerator can be placed inside or outside the displacer [3,12–14].

Different driving mechanisms such as a crankshaft, Ross-yoke, lever-drive, rhombic-drive, Scotch-yoke or swash-plate are used in Stirling engines. The cycle obtained by these driving mechanisms is different from the ideal Stirling cycle depending on the mechanical configuration. Cyclic work is the most important parameter in engine design [15]. Thermodynamic analyses are quite significant in determining the engine performance and parameters that affect the performance. Various analysis methods have been developed since the invention of Stirling engines and engine performance has been estimated during the design process.

The first analysis for determining the thermodynamic performance of Stirling engines was developed by Gustav Schmidt in 1871 and thermodynamic analyses of α, β, and γ type engines were conducted [3,6]. The nodal analysis method was developed in 1967 by Finkelstein. In this analysis, the total volume of the engine was divided into 13 nodal volumes: the expansion volume (1), heater volumes (3), regenerator volumes (5), cooler volumes (3), and cold volume (1). Instantaneous temperatures, system pressure, cyclic work, temperatures and thermal efficiency are calculated in the nodal volumes [15–17]. In 1978, Martini developed a five-zone nodal-isothermal analysis. In this model, the temperature of the working fluid in the nodal volumes was assumed to be equal to the wall temperature [18]. In 1994, Ladas and Ibrahim conducted

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Nomenclature

A_i	nodal values of heat transfer surface (m^2)
c_p	specific heat at constant pressure ($J\,kg^{-1}\,K^{-1}$)
c_v	specific heat at constant volume ($J\,kg^{-1}\,K^{-1}$)
E_i	enthalpy flow in and out of the nodal volumes (J)
e_c	length of cold volume (m)
e_h	length of hot volume (m)
$\frac{h_p}{9}$	distance between piston top and piston rod top (m)
h_d	displacer length (m)
h_i	nodal values of convective heat transfer coefficient ($Wm^{-2}\,K^{-1}$)
l_{dr}	length of displacer rod (m)
l_{pr}	length of piston rod (m)
l_{rh}	length of displacer and piston connecting rods (m)
m_t	total mass of working fluid (kg)
m_i	nodal values of working fluid mass (kg)
Δm_i	variation of nodal mass within a time step (kg)
R	gas constant ($J\,kg^{-1}\,K^{-1}$)
R_{cr}	radius of crankshaft (m)
T_i	nodal values of working fluid temperature (K)
T_C	cold-end temperature of displacer cylinder (K)
T_H	hot-end temperature of displacer cylinder (K)
$T_{w,i}$	nodal values of heat transfer surface temperature (K)
t	time (s)
Δt	period of time steps (s)
U_c	length from cylinder top to rhombic center (m)
V_i	nodal values of volume (m^3)
βr	angle made by connecting rod with vertical (rad)
θ	total crankshaft rotation (rad)

a finite-time thermodynamic analysis of a Stirling engine. They analyzed the effects of regeneration and engine speed on engine power and engine efficiency by using the conservation law of mass and the energy equations [19]. In 2000, Kaushik and Kumar determined the thermodynamic performance of a Stirling engine using finite-time thermodynamics. They investigated the effects of the temperature of different heat exchangers and heat source on engine power and engine efficiency [20]. In a study conducted by Karabulut et al. (2006) the power cylinder was concentrically situated in the displacer cylinder to increase the specific power and decrease the external volume of a gamma type Stirling engines. The thermodynamic performance characteristics of the engine were predicted by a nodal analysis program in Fortran [21]. Kongtragool and Wongwises (2006) investigated the performance of a Stirling engine with the dead volumes of the cold volume, hot volume and regenerator using an isothermal model [22]. Parlak et al. (2009) conducted a thermodynamic analysis of a gamma type Stirling engine using a quasi-steady flow model. The analysis was performed for five volumes: compression, expansion, cooler, heater and regenerator. The variations of temperature, pressure, mass and work were calculated by using the conservation law of mass and the energy equations [23]. Timoumi et al. (2008) investigated the effects of regenerator sizes and materials on the performance of a General Motors GPU-3 Stirling engine [24]. Eid (2009) conducted a thermodynamic analysis of a beta type Stirling engine with a regenerative displacer by using the Schmidt analysis method [13]. Cheng and Yu (2010) carried out the thermodynamic analysis of a beta type Stirling engine with a rhombic-drive mechanism by using a numerical model. The energy equations for control volumes were derived and solved by taking into account the non-isothermal effects, the effectiveness of the regenerative channel and the thermal resistance of the heating head [12].

In this study, kinematic and thermodynamic analysis of a beta type Stirling engine with a rhombic-drive mechanism was conducted using the nodal analysis method. In the analysis, the increase of the heat transfer surface area inside the displacer cylinder on engine performance was investigated. The performance of the engine was estimated for two different displacer cylinders, of which one has a smooth inner surface and the other has axial slots grooved into the cylinder. Engine performance parameters were also calculated for different values of the convective heat transfer coefficient, working fluid mass and engine speed. The theoretical results were compared with experimental data obtained from the prototype engine.

2. Test engine

Due to the lower side thrust and more silent operation [6], the rhombic drive mechanism was preferred and the engine was designed as beta type. The cycle is conducted by a piston and a displacer operating in the same cylinder. The rhombic drive mechanism consisted of a piston yoke, a displacer yoke, two pistons and two displacer connecting rods with equal length hexagon and two symmetric spur gears of equal diameter (Fig. 1). The spur gears rotate in opposite directions. The displacer rod is connected to the displacer yoke at the lower end of the equal length hexagon; the power piston rod is connected to the power piston yoke at the upper end of the equal length hexagon. The displacer and power piston yokes have linear motion. A displacer and a power piston rods are connected to pins on spur gears. The cold volume is between the power piston and the displacer, and the hot volume is

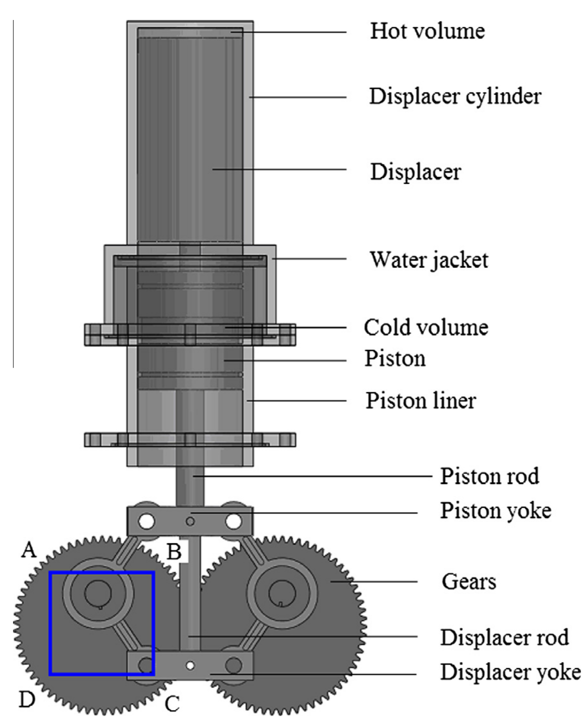

Fig. 1. Beta type Stirling engine with rhombic-drive mechanism.