Applied Thermal Engineering 145 (2018) 693–704

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Research Paper

Parametric analysis of a beta Stirling engine – A prime mover for distributed generation

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HIGHLIGHTS

- Multi-parameter analysis of beta-Stirling engine was presented.
- Hydrogen, helium and air were utilized as the working media.
- A mathematical model of the engine was developed and validated.
- The maximum output power varied within the range of 2.71–8.8 kW.
- The efficiency was in the range of 31.7–43.2%.

ARTICLE INFO

Keywords:
Distributed generation
Stirling engine
Mathematical modeling
Hydrogen

ABSTRACT

In this article, a multi-parameter analysis of a beta-Stirling engine has been presented. The calculations have considered the utilization of helium, hydrogen and air as the working media. The analysis was focused on the influence of the frequency (f) and the phase angle (φ), which were controlled in a wide range of values, on the operational parameters of the device i.e. output power and efficiency. The input parameters (f, φ) were selected as there are mechanical solutions to implement their change in real engines. The possibility of the regulation of the engine might significantly increase its application potential. Precise control and regulation of the engine is of great importance, especially in distributed generation technologies where the ability to operate under partial load is crucial. A mathematical model of the engine was presented and then validated using available data from the literature, including experimental studies for two types of working gas – hydrogen and helium. The maximum output power varied within the range of 2.71–8.8 kW and the efficiency was in the range of 31.7–43.2% depending on the working gas and the controlled parameters: frequency and phase angle.

1. Introduction

The Stirling engine, which was invented in 1816, is an external-combustion thermal engine. Its thermodynamic cycle, considering the idealized case, consists of two isothermal and two isochoric processes [1]. Basically, the engine consists of two pistons (called in certain configurations the power piston and the displacer), a heater, a cooler and a regenerative heat exchanger placed in between. Among the variety of the types of physical construction of the engine, its three classic configurations may be stated as α-, β- and γ-type engines. A detailed description of these may be found in the literature [1]. The classic Stirling engine configurations have been investigated by a number of researchers, resulting in a profound knowledge of their thermodynamic and kinematic behavior [2–4]. However, due to the increasing interest of certain groups of researchers in the applicability of the Stirling engine in the last few decades, a number of other configurations have been developed.

One of most interesting configurations of the engine, presented in the 20th century, is the rhombic drive Stirling engine, which was invented in 1959 [5]. Similar to the β-type, the power piston and displacer are placed in the common cylinder. However, unlike the classical configuration, two crankshafts are present. Furthermore, piston rods are connected to them with yokes. A pair of complete yokes connecting the piston and the crank forms a rhombus structure of changing angles during the motion of the yoke [5,6].

Due to the presence of a number of irreversibilities and unsteady thermal and fluid-flow phenomena, the development of an accurate mathematical representation of the Stirling engine is problematic. The first mathematical model of a Stirling engine was produced by Schmidt in 1871 [6]. Although the model was relatively simple and yielded a

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https://doi.org/10.1016/j.applthermaleng.2018.09.088
Received 26 July 2018; Received in revised form 20 September 2018; Accepted 21 September 2018

Nomenclature

A	area, m^2
c	specific heat, $J\cdot kg^{-1}\cdot K^{-1}$
C	coefficient
d	hydraulic diameter of the passage, m
f	frequency of the engine, Hz
h	convective heat transfer coefficient, $W\cdot m^{-2}\cdot K^{-1}$
l	linear characteristic dimension, m
m	mass, kg
$\dot{m}$	mass flow, $kg\cdot s^{-1}$
m’	mass flow at the interface between various spaces, $kg\cdot rad^{-1}$
M	total mass of the working medium inside the engine, kg
NTU	number of transfer units
P	output power of the engine, W
p	pressure, Pa
Q	heat, J
R	gas constant, $J\cdot kg^{-1}\cdot K^{-1}$
Re	Reynolds number
St	Stanton number
T	temperature, K
u	speed of gas during flow through the heat exchanger, $m\cdot s^{-1}$
V	volume, m^3
ΔV	change of volume affected by the change of cycle angle, $m^3\cdot rad^{-1}$
W	work, J
γ	adiabatic index
Δ	change
ε	effectiveness
η	efficiency
θ	cycle angle, degrees, radians
λ	conductance of the material, $W\cdot m^{-1}\cdot K^{-1}$
μ	dynamic viscosity, Pa·s
ρ	density of the gas, $m^3\cdot kg^{-1}$
φ	phase angle, degrees, radians

Indices

c	compression space, cooler
Carnot	carnot definition of efficiency
cov	carnot coverage
cs	cross-sectional area of a single passage in the heat exchanger
e	expansion space
fRe	Reynolds friction factor
h	heater
k	cooler
loss	loss of heat or power
max	maximal
mean	mean working pressure in the engine
min	minimal
n	number of heat exchanger in pressure loss calculations
p	at constant pressure
r, reg	regenerator
Suth	Sutherland constant
t	total
v	at constant volume
wk	wall of the cooler
wg	wetted area
wh	wall of the heater
0	reference parameter

closed-form solution, its inaccuracy proved the necessity of further research [7]. In the 1960s and the 1970s, a number of models that considered nodal analysis were presented [6]. Since the 1980s, a number of analytical and numerical Stirling engine models have been developed, including finite-time analysis-based models [8–10] and analytical model based on the inclusion of losses in an idealized analysis [7]. Due to its simplicity and utilization of the basic concepts of thermodynamics, combined with acceptable accuracy, the last of mentioned models, described in [7], has been used in a number of analyses, as well as forming the basis of more complex models [8,11]. A modified analytical model has been proposed in the literature [11], that included conductive losses in the heat exchangers and the influence of conduction through the displacer, combined with sophisticated conditional structures. A similar concept, proposing the addition of a non-linear extrapolation of temperature along the regenerative heat exchanger and a high-accuracy method for the integration of the differential pressure equation, has also been introduced [12,13]. Based on a modified analytical model, a more sophisticated set of equations have been derived in the literature [8], including the influence of pressure and power losses affected by the mechanical friction of the working gas and the non-ideal performance of the pistons. A useful 3rd order analysis method, including a derivation of formulae for real fluids, along with additional characteristic numbers for improved description of the fluid mechanics, as well as a significant part of the engine's kinematics, was introduced in the literature [14]. Furthermore a complex thermodynamic-mechanical model of a Stirling engine, based on a modified adiabatic analysis, with the introduction of losses in all the heat exchangers and a modified method for the computation of convective heat transfer coefficients has been proposed in the literature [15]. Moreover, since 2010, thermodynamic analysis models using computational fluid dynamics (CFD) tools have also been developed [3,16,17].

Profound research has led to a number of parametric analyses being performed. These concerned the investigation of the influence of a variety of geometrical and operational parameters on the maintenance of the Stirling engine. However, due to the difficulties in the introduction of a complex and accurate thermodynamic-mechanical model of novel Stirling engine constructions, there has been little general or extensive parametric research described in the literature. Temporary work has focused on the investigation of methods of improving the maintenance parameters of various constructions [4,18], as well as the development of novel designs [19,20].

An example of a second order cyclic analysis, including sensitivity analysis of a number of design and operational parameters, concerning the optimization of a beta configuration Stirling engine with a rhombic drive has been presented in the literature [2]. The results of the analysis proved the possibility of the design of an engine with satisfactory power output, filled with hydrogen at intermediate pressures.

A novel parallel-type regenerator, including small-diameter circular channels, that has proven the vital importance of direct heat conduction in the exchanger on overall engine losses has been investigated in the literature [21]. Further research on the effectiveness of the regenerator and its limitations caused by the uneven temperature distribution, carried out in the literature [22], emphasized the influence of a number of fluid flow phenomena on imperfect heat transport in the heat exchanger mentioned. Other research, presented in the literature [3], has indicated the influence of thermal radiation of the heat exchanger surfaces and the working media on the power output of the engine, and showed a vital rise in the modeled power output with improvement in the radiative heat transfer computation methods. A profound sensitivity analysis of a low-power beta Stirling engine, the results of which have been presented in the literature [18], suggested the necessity of a general redesign of the heat exchangers, compared to the classical