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Multi-objective optimization of Stirling engine using non-ideal adiabatic method

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In the recent years, remarkable attention is drawn to Stirling engine due to noticeable advantages, for instance a lot of resources such as biomass, fossil fuels and solar energy can be applied as heat source. Great numbers of studies are conducted on Stirling engines and non-ideal adiabatic method is one of them. In the present study, the efficiency and the power loss due to pressure drop into the heat exchangers are optimized for a Stirling system using non-ideal adiabatic analysis and the second-version Non-dominated Sorting Genetic Algorithm. The optimized answers are chosen from the results using three decision-making methods. The applied methods were compared at last and the best results were obtained for the technique for order preference by similarity to ideal solution decision making method.

1. Introduction

Increasing the price of fossil fuel, environmental pollution, and noise pollution due to the engines using this fuel, has caused additional motivation for research on other types of power generation. High thermal efficiency, minimal pollution, reliability, maximum utilization and clean combustion are the major features that we expect from the new engines. In the recent years, the Stirling engines have attracted a lot of attentions due to high capability [1,2]. A Stirling engine presents a reasonable theoretical efficiency which can be closer to the Carnot efficiency, comparing with other reciprocating thermal engines [3]. According to the Stirling engine's ability to achieve the highest possible efficiency, this engine always is noticed for the researchers. With this regard, the first classical analysis of Stirling engine was done by Schmidt [4]. Schmidt assumed that the gas is isothermal in the expansion and the compression spaces. This assumption reduces the complexity of the compression and the expansion spaces with considering constant temperatures in the spaces.

In the real engines, during the process, the working spaces tend to the adiabatic mode mostly. Based on this fact, Finkelstein carried out the first analysis of non-constant temperature of the Stirling engine in 1960. In the analysis presented by him, each part of the engine (cooler, heater, regenerator, expansion and compression spaces) was considered as a control volume and the conservation laws of mass and energy were analyzed using the equation of state [5]. Martini provided computer simulation of Stirling engine with considering five working spaces for it [6]. In this model, pressure drop and heat loss were intended. Also, Urieli and Berchowitz used a computer code for solving differential equations using Runge–Kutta fourth-order method. They presented an adiabatic model for non-ideal mode in order to improve the numerical predictions. In this method, the concept of the non-ideal regenerator was realized [7].

Abbas et al. developed a non-ideal adiabatic method. In their work, the regenerator space was divided into two parts and this model included losses such as the shuttle loss and regenerator loss [8]. Strauss and Dobson developed other model using the quasi-adiabatic analysis in which the losses of the regenerator were considered [9]. Granodos et al. developed the quasi-steady flow model to calculate the pressure drop through the heat exchanger channels. In their model, the engine was divided into 19 parts which 10 control volumes were dedicated to the regenerator [10]. Tlili et al. developed the second adiabatic model based on the quasi-steady flow. In their modeling approach, they considered the shuttle losses, the internal and external conduction losses in the regenerator, and the loss of energy that lead to pressure drop in the heat exchangers [11]. Parlak et al. carried out the thermal analysis on the Stirling engine with the Gamma structure. The quasi-steady flow analysis was performed to achieve more précised results. The thermal efficiency reached to 25% using nitrogen as working fluid under pressure of 6.5 bar and the heat source temperature of 873 K [12]. Considering the different methods of optimization in the energy issues, which have been carried out recently, it is worth to do multi-objective optimization of which, some are explained as followings.

Different engineering problems such as skyline computation and vehicle routing issues have applied multi-objective

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Nomenclature

Q_h	heat transfer to heater
Q_k	heat transfer to cooler
Q_r	heat transfer to regenerator
p	mean effective pressure
S	stroke
f_r	engine's frequency
T_h	temperature of the working fluid of hot side
T_H	temperature of the heat source
T_C	temperature of the heat sink
W_c	work done by compression space
V_{Se}	expansion swept volume
V_{Sc}	compression swept volume
V_{de}	expansion dead volume
V_{dc}	compression dead volume
A	heat transfer effective surface
A_{wg}	wetted area of the metal net of the regenerator
l_h	heater length
l_k	cooler length
l_r	regenerator length
V_r	heater volume
V_h	cooler volume
V_k	regenerator volume
f	friction factor
d_h	heater hydraulic diameter
d_k	cooler hydraulic diameter
d_r	regenerator hydraulic diameter
C_V	specific heat at constant volume
C_P	specific heat at constant pressure
G	working fluid mass flow
k	thermal conductivity
M	total mass of the working fluid
m	mass of the working fluid in the different parts of the engine
Re	Reynolds number
Pr	Prandtl number
St	Stanton number
$f(x)$	objectives function
X	vector of decision variables

Subscripts

rh	interference between regenerator and heater
kr	interference between cooler and regenerator
he	interference between heater and expansion chamber
ck	interference between compression chamber and cooler
h	heater
k	cooler
r	regenerator
c	compression chamber
e	expansion chamber
J	number of the inequality constraints
K	number of the equality constraints

Greek letter

ε	effectiveness of the regenerator
η	efficiency
$\varnothing$	crank angle
γ	specific heat ratio
μ	dynamic viscosity

optimization [13–15]. The solution rout of the multi-objective optimization cases is a highly achieved target which requires the simultaneous satisfaction of a number of different and conflicting objectives. Evolutionary algorithms (EA) were applied in an attempt to stochastically solve problems of this generic class during the 18th century [16]. Inquirying a set of solutions of which each satisfies the objectives at an acceptable level without being overcome by any other solution, is a reasonable solving method to a multi-objective quandary [17]. A possibly uncountable collection of solutions so-called Pareto frontier is universally represented by the multi-objective optimization. The evaluated vectors of the Pareto frontier illustrate the best feasible trade-offs in the objective function space. In this case, multi-objective optimization of various thermodynamic and energy systems have been of high interest attention of researchers these days [18–25].

For considering all the above-mentioned issues, we have studied two objective functions including the power loss due to pressure drop into the heat exchangers and the efficiency. In addition, the multi-objective optimization is conducted with four decision variables including the temperature of working fluid at hot side, frequency, the stroke and the mean effective pressure.

Considering the parameters that have been neglected by most of the previous works on the performance optimization of Stirling engine such as the temperature of working fluid at hot side, frequency, the stroke and the mean effective pressure, we selected our approach for the current study. In the previous studies, the multi-optimization and adiabatic methods have not been linked together for Stirling engines and the adiabatic systems have only been utilized for modeling. In this work, by implementing the multi-objective optimization algorithms, the power loss due to pressure drop into the heat exchangers and the efficiency are minimized and maximized, respectively.

2. Theoretical model

In the constant temperature of the Schmidt analysis, it is assumed that the compression and the expansion spaces during the engine performance remain under constant temperature conditions [26]. This assumption provides a contradiction that the heater, the regenerator, and the cooler do not contribute in the any net heat transfer engine cycle. For this reason, they seem to be of none use. The real engine working spaces tend to the adiabatic mode during the processes. This assumption requires that the net heat transfer during a cycle is supplied by heat exchanger.An adiabatic model is shown in Fig. 1 for better understanding. This model is composed of five parts which are connected together sequentially. In this model, the working fluid in the cooler and the heater are at constant temperature conditions namely T_k and T_h, respectively. Also, the wire matrix regenerator and the embedded fluid are placed in a linear distribution. In this model, the assumption is that the working spaces are at adiabatic conditions. Also, it is assumed that there is no leakage of the working fluid and the pressure drop is not created in the fluid flow path. The performed work under these assumptions, is dependent on the changes in the volume of the compression and the expansion spaces and heat is transferred between the outside and the working fluid in the cooler and the heater which are Q_K and Q_h, respectively.

The most common method to obtain the required differential equations and the heat transfer in the heater and the cooler, is writing the energy equation and the ideal gas equation for each control volume. The assumed control volume is shown in Fig. 2

With getting differential of the mass balance for the assumed system [7], Eq. (1) is obtained.

$$Dm_c + Dm_k + Dm_r + Dm_h + Dm_e = 0 \quad (1)$$