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Simulation on the effect of the combustion parameters on the piston dynamics and engine performance using the Wiebe function in a free piston engine

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HIGHLIGHTS

- ▶ A dual piston type free piston engine was separated into a single cylinder model to perform a precise parametric study.
- ▶ The spark timing, piston initial velocity and combustion duration were varied to analyze their effects on the performance.
- ▶ A relationship was found between the combustion phase at top dead center and the initial piston velocity condition.
- ▶ The spark timing should be tuned to satisfy specific mass fraction burned profile for best engine performance.

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ABSTRACT

Numerical simulations were conducted to observe the relationship between the combustion phase and the piston dynamics in a free piston engine. The simulations were conducted with commercial software, MATLAB/SIMULINK®. The Wiebe function was used to simulate the combustion process. The combustion parameters such as combustion duration and the spark timing were varied at various piston initial velocities at compression stroke. The indicated mean effective pressure (IMEP) and the mass fraction burned (MFB) were analyzed as indicators of the engine performance.

Under given combustion duration conditions, the minimum ignition advance for best torque (MBT) timing was first retarded towards top dead center (TDC) and consequently advanced away from TDC as the piston initial velocity increased. An MBT timing curve was plotted against various spark timings and piston initial velocities in a map. There existed a peak IMEP value along this MBT curve. Longer combustion duration brought negative effects on the IMEP due to larger deviation from the quasi-constant volume combustion, thus lowering the efficiency.

The velocity profile of the piston was plotted against the displacement with a contour of every 10% increment of mass fraction burned in order to provide a clear visualization of the combustion process. The spark timing had to be advanced with high piston initial velocity or a long combustion duration condition in order to complete the combustion process near TDC.

Mass fraction burned values at TDC showed a linear relationship with piston initial velocity values. This provided a general idea of how to vary the spark timing under a given piston initial velocity condition to achieve the best conversion efficiency from combustion to work regardless of the combustion duration.

1. Introduction

Massive efforts have been dedicated to improving engine efficiency to reduce CO_2 emissions with the aim of preventing global warming. Meanwhile, research on novel alternative high efficiency engines such as free piston engines has been conducted with the same goal of reducing fuel consumption and hazardous emissions. Free piston engines are not new concepts, having been first proposed in the early 1920s [1]. Free piston engines have, however, once again gained attention due to current energy security issues. Free piston engines with linear alternators are considered to be one of the most interesting research topics among several free piston engine types. The lack of a crankshaft mechanism allows the piston to move freely inside the cylinder liner without any mechanical constraints. This not only reduced mechanical friction losses, but

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also makes it possible to characterize the variable compression ratio with respect to the load condition. Thus, high efficiency can be achieved with precise control of the fuel quantity and the electric load [2,3].

The dynamics of the piston are determined by the resultant forces applied on the piston. Some major forces applied on the piston include the force by the in-cylinder pressure during compression and combustion, the electric load force, and the friction. Therefore, a careful and delicate set of piston dynamic controlling algorithms needs to be developed [4,5]. Prior to dealing with the controlling issue, however, the piston dynamics with respect to important engine parameters have to be studied. Minor changes in the engine configurations such as the mass of the piston moving assemblies or the friction force during motion may eventually result in large differences in the piston dynamics, and therefore the piston dynamic control and the combustion strategies may vary greatly from engine to engine of various research groups. Many research groups in this area have focused on the piston dynamics of their prototype engines, either on the basis of simulations [6–12] or experimental results [13]. Simulations are becoming complicated, with several sub-models being employed to estimate and predict the experimental results more accurately. The KIVA code was employed to estimate the combustion emissions with respect to the piston dynamics [14]. The potential of high efficiency of free piston engines with low hazardous emissions was suggested.

While numerous simulation based studies have been carried out, parametric studies covering a wide range of engine combustion parameters during the engine operation are limited to a few cases. Studies aimed at obtaining general information about the relationships between various combustion parameters and the piston dynamics are insufficient despite their great importance. Because the free piston engine does not characterize a fixed stroke, the in-cylinder pressure trace during combustion not only affects the piston dynamics, but also strongly determines the efficiency of the thermodynamic cycle. Therefore, a simulation was carried out in order to provide a general view of the relationship between various combustion parameters and the piston dynamics to achieve the best engine performance. The engine has a dual piston type configuration, combining with a linear alternator to generate electricity. The effects of the spark timing with respect to the piston initial compression velocity and the combustion duration on the engine power output were studied based on a numerical simulation. The indicated mean effective pressure (IMEP) was analyzed as an indication of the engine power output. The simulation was conducted with commercial software, MATLAB/SIMULINK® (v. 2010a), which can calculate the dynamics of the piston. The zero-dimensional and single-zone models were used in the simulation. The model is designed to be as simple as possible to precisely examine the effects of individual parameters and prevent any other interference. The Wiebe function, which is simple but powerful, was used to simulate the combustion process.

2. Simulation methodology

Table 1
Engine specifications and scavenging system.

Bore (mm)	25
Maximum stroke (mm)	22
Piston assembly mass (kg)	0.77
Scavenging type	Cross-scavenged type

The specifications of the prototype free piston engine in the simulation are listed in Table 1. It is a dual piston type configuration combined with a linear alternator in the center. It has a bore size of 25 mm and maximum stroke of 22 mm. The mass of the moving piston assembly was 0.77 kg. A fundamental analysis was carried out using governing equations. Despite its dual piston type configuration, only one combustion chamber was simulated, which is similar to a rapid compression machine (RCM), for a precise parametric study. Simulations with both cylinders (the entire dual piston system) to examine the overall operation characteristics of a free piston engine can be found in the literatures. Generally, the initial compression velocity of the piston is determined by the previous combustion process taken place at the opposite side of the cylinder and the engine load condition. The initial compression velocity is important, because it influences the escaping velocity during the expansion and the successive combustion process. However, it is not easy to be decoupled from the interrelationship with other variables and to be carried out as an initial variable of interest in the dual piston type configuration. The free piston engine system was separated into a single combustion chamber model in order to gain freedom on setting the piston initial velocity condition. A relationship can be derived between the piston initial velocity during the compression and the escaping velocity after the expansion with respect to engine control parameters, such as spark timing and heat input energy. The escaping velocity after the expansion becomes the initial velocity condition during the compression of the opposite cylinder in the successive cycle. Therefore, the relationship derived above can be used iteratively.

The acceleration of the piston was derived with Newton's second law,

$$\ddot{y} = \frac{\sum p_i \cdot A_{piston} + \sum F_i}{M_{piston}} \tag{1}$$

where p_i is the pressure applied to the piston, A_{piston} is the area of the piston surface, and F_i is the summation of external forces, such as electric load force and friction force. The external resultant force was excluded to eliminate its influence on the piston dynamics. M_{piston} is the mass of the moving piston assembly. During the compression, the in-cylinder pressure satisfies an isentropic relation,

$$p_1(\pi r^2 y_1)^\gamma = p_2(\pi r^2 y_2)^\gamma \tag{2}$$

$$p_2 = p_1\left(\frac{y_1}{y_2}\right)^\gamma \tag{3}$$

where p_i is the in-cylinder pressure, y is the distance from the top dead center, and γ is the specific heat ratio. The force during the compression can be simply calculated through manipulation of the pressure value (Eq. (3)).

The in-cylinder pressure change due to the combustion can be obtained with the derivation form of the first law of thermodynamics,

$$\frac{dp}{dt} = -\gamma \cdot \frac{p}{V} \cdot \frac{dV}{dt} + (\gamma - 1)\frac{Q_{in}}{V}\frac{dx_b}{dt} \tag{4}$$

where Q_{in} is the input heat energy and x_b is the mass fraction burned.

The combustion process, which simulates the mass fraction burned, was conducted using the Wiebe function [15,16].

$$x_b = 1 - \exp\left[-a \cdot \left(\frac{t - t_s}{C_d}\right)^{b+1}\right] \tag{5}$$

where C_d is the combustion duration, t_s is the start of ignition, and a and b are constants. Constants of $a = 5$ and $b = 2$ are used. These constants are widely used for spark ignition engines in general and it has been proven that they correlate well with experimental data [17].

The coordinates of the piston assemblies were established as shown in Fig. 1. The maximum top dead center (TDC) was set as