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Parametric analysis of a dual-piston type free-piston gasoline engine linear generator

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Abstract

The free-piston gasoline engine linear generator (FPELG) is linear type power device, which couples a linear internal combustion engine and a linear electrical generator together. Free-piston gasoline engine is commonly modeled using simplified zero-dimensional models. In this paper, parametric analysis of a dual-piston type FPELG is presented, aiming to find the piston operation characteristics and the performances of the FPELG during the generating process. Model validation was undertaken with the test data from a running prototype, which showed good agreement. The results show that with fixed stroke, lower moving mass would lead to higher indicated power. With fixed moving mass, shorter stroke would make higher indicated power. The FPELG was observed to be operated at high load, and it was prone to operate at relative low speed. The influence of combustion parameters, i.e. the ignition timing and the combustion efficiency to the FPELG engine performance were also investigated respectively.

Keywords: Free-piston engine; linear generator; numerical model; parametric analysis; engine performance;

1. Introduction

The free-piston engine gasoline linear generator (FPELG) is linear type power device, and has attracted research interests recently due to its simplicity and potential fuel efficiency [1]. It couples a linear internal combustion engine

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and a linear electrical generator together, the high-pressure and high-temperature gas after the heat release process in the combustion chamber drives the piston assembly reciprocate, then the generator converts part of the mover's kinetic energy to electricity [2]. According to the principle of the free-piston engines, they are known to have a greater thermal efficiency than conventional internal combustion engine [3].

Researchers at Sandia National Laboratories presented an analysis on the steady-state performance of a FPE with a zero-dimensional numerical model [4]. Hansson et al. investigated the resonant behaviour of a dual piston type FPELG. They linearized the system after expanding the equation around an equilibrium point [5]. Roskilly et al. at Newcastle University provided a simulation investigation of a FPELG and they discussed the feasibility of the implemented models [6]. In addition, they developed a numerical model to investigate the techno-feasibility of operating the designed FPELG prototype using a two- or four-stroke thermodynamic cycle [7]. Boru Jia and Zhengxing Zuo et al. at Beijing institute of technology presented a linearization of the dynamic equation for a dual piston type FPELG, and simplified it to a one-degree forced vibration system with viscous damping and the solution to the vibration system was solved [8]. Ocktaeck Lim et al. presented an experimental and simulation study of a FPELG included of a two-stroke free-piston engine, the linear generators and the compressors [9]. Jaeheun Kim et al. simulated the effect of the combustion parameters on the piston dynamics and engine performance for a dual piston type FPELG[10]. Researchers at Toyota Central R&D Labs Inc. developed the FPELG prototype, the piston dynamics and its effect on combustion were analyzed [11, 12]. Siqing Chang at Nanjing Institute of Technology designed the FPELG prototype, the output power was 2.2 kW and efficiency was 32% [13].

Free-piston engine is commonly modelled using simplified zero-dimensional models. While there has very few model validation reported due to the limited test data available from operating prototypes. In this paper, parametric analysis of a dual-piston type free-piston engine generator is presented, along with the model validation results against test data from the FPELG prototype. The engine design parameters, the engine operation conditions and the combustion parameters will be varied and simulated separately, aiming to find the piston operation characteristics and the performances of the FPELG during the generating process. Furthermore, according to the results, it can guide the FPELG design and the operating process.

2. Model description and validation

2.1. Model description

In order to explore parametric analysis the piston operation characteristics and the performance of the FPELG, the numerical model is established based on the Newton's second Law. The forces acting on the pistons of the FPELG during the generating process was shown in Fig. 1.

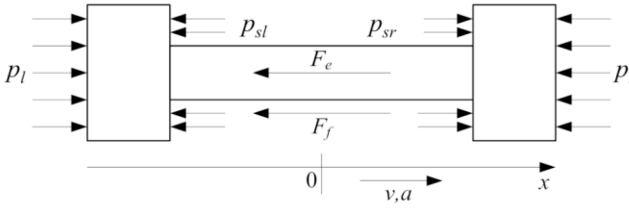

Fig. 1. Forces acting on the mover of the FPELG

Note: p_l and p_r are the in-cylinder gas pressure of the left and right respectively; P_{sl} and P_{sr} are the scavenging box gas pressure of the left and right respectively; F_f is the mechanical friction force and F_e is the force output from the linear electric generator [8].

Because the scavenging box gas pressure of the left and right are the same, the joint force is zero. Thus, the mover dynamics equation is derived as follows:

$$m\frac{d^2x}{dt^2} = A\left(p_l - p_r\right) - F_f - F_e \tag{1}$$

Where, m is the moving mass of the piston assembly and the mover of the electric machine; x is the mover displacement; t is the time; A is the top area of the piston; $\frac{d^2x}{dt^2}$ is the mover acceleration.

According to the previous in our papers, F_f and e F_e are described as follows respectively:

$$\begin{cases} F_f = C_f \dfrac{dx}{dt} \\ F_e = k_{\mathrm{f}} \cdot k_{\varepsilon} \dfrac{1}{R_{\mathrm{S}} + R_{\mathrm{L}} + j \cdot L} \dfrac{dx}{dt} \end{cases} \tag{2}$$

Where, C_f is the coefficient of the friction; $\frac{dx}{dt}$ is the mover velocity; k_f is the thrust force constant; k_ε is the coefficient of the electromotive force of the generator; R_s is the resistance of the coil; R_L is the resistance of the external load; and L is the inductance of the generator [14, 15].

Based on the operation principle of the FPELG, the thermodynamics of the left is the same as that of the right. Therefore, the in-cylinder gas pressure changes equation of each combustion chamber is derived as following:

$$\frac{dp}{dt} = \frac{\gamma - 1}{V}\left(\frac{dQ_c}{dt} - \frac{dQ_h}{dt}\right) - \gamma \frac{p}{V}\frac{dV}{dt} \text{-} \gamma \frac{p}{m_{air}}\frac{d_{mair}}{dt} \tag{3}$$

Where, p is the in-cylinder gas pressure; γ is the ratio of heat capacities; V is the instantaneous cylinder volume; Q_c is the heat released from the combustion process; Q_h is the heat transferred to the cylinder wall; m_{air} is the in-cylinder gas mass.

According to the previous papers [14], the heat transfer to the cylinder wall is derived as following:

$$\frac{dQ_h}{dt} = 130V^{-0.06}\left(\frac{p(t)}{10^5}\right)^{0.8} T^{-0.4}\left(v_p + 1.4\right)^{0.8} A_{cyl}\left(T - T_w\right) \tag{4}$$

Where, h is the coefficient of heat transfer; A_{cyl} is area of the in-cylinder surface in contact with the gas, T_w is the average surfaces temperature of the cylinder wall. v_p is the average mover speed.

Based on our previous paper, the gas leakage mass is derived as following:

$$\frac{dm_{air}}{dt} = \begin{cases} \dfrac{C_D \times A_{leakage}}{\left(RT_0\right)^{0.5}} p(t)\left[\dfrac{p_s}{p(t)}\right]^{\frac{1}{\gamma}} \left\{\dfrac{2\gamma}{\gamma - 1}\left[1 - \left(\dfrac{p_s}{p(t)}\right)^{\frac{\gamma-1}{\gamma}}\right]\right\}^{0.5} & \dfrac{p_s}{p(t)} > \left(\dfrac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \\ \dfrac{C_D \times A_{leakage}}{\left(RT_0\right)^{0.5}} p(t)\gamma^{\frac{1}{2}}\left(\dfrac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} & \dfrac{p_s}{p(t)} \le \left(\dfrac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \end{cases} \tag{5}$$

Where, C_D is the discharge coefficient; $A_{leakage}$ is the leakage area; T_0 is the air temperature in the scavenging box and is assumed to the ambient temperature; p_s is the gas pressure in the pumping box.

As for the heat released from the combustion process, a time based Wiebe function is applied on the mass fraction burned during the combustion process, and is derived as following [14, 15]:

$$\frac{dQ_c}{dt} = a\frac{b+1}{C_d}\left(\frac{t - t_s^b}{C_d}\right)\exp\left(-a\left(\frac{t - t_s}{C_d}\right)^{b+1}\right)Q_{in} \tag{6}$$

Where, a and b are parameters in the Wiebe function respectively, and b is 5, a is 2. And will introduce in the following context. C_d is the combustion duration; t_s is the start timing during the combustion process.

2.2. Model validation

The prototype specification is identical with the input parameters used in the numerical model, which is summarized in Table 1. The engine was operated at stoichiometric air-fuel ratio (λ=1.0) with medium open throttle. Fig. 2 illustrates a comparison between simulated p-V diagrams with the experimental results. Both of the simulated compression and expansion processes are in good correspondence with test results, and the pressure difference can be controlled below 5%. The simulation model is of high accuracy to predict the actual engine performances, and the model is validated.

Table 1. Prototype specification

Parameters	Value	Unit
Bore	52.5	mm
Maximum stroke	70	mm
Piston and connecting rod mass	5	kg
External load resistance	28	Ω

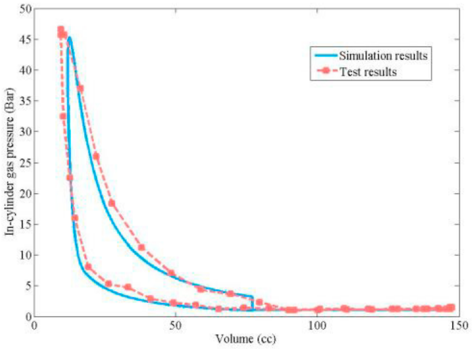

Fig. 2. Model validation results

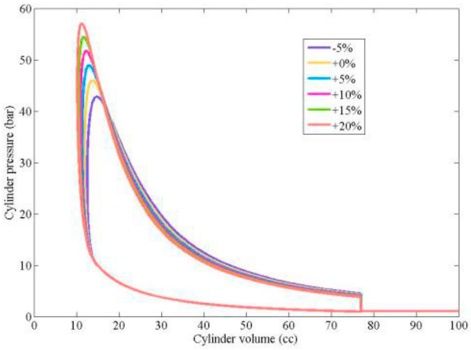

Fig. 3. P-V diagram with different moving mass

3. Simulation results and discussion

3.1. Engine design parameters

For the dual-piston type FPELG, the mover of the linear electric machine is connected with two pistons on both sides by two rods, which forms the only significant moving part of the FPELG. The initial value of the design moving mass is 5.0 kg, which is selected as a reference for a further simulation of -20% to +20% from the reference (4.0 kg to 6.0 kg). The other engine operation conditions remain unchanged. Fig. 3 shows the simulation results of the in-cylinder pressure with different moving mass with a step of 5%. It is observed that when the moving mass drops to -10% or more from the reference value (below 4.5 kg), engine misfire will happen. With relative higher moving mass, both of the peak cylinder pressure and achieved compression ratio are higher. While the heat release process is more close to a constant volume process when the moving mass is lower.

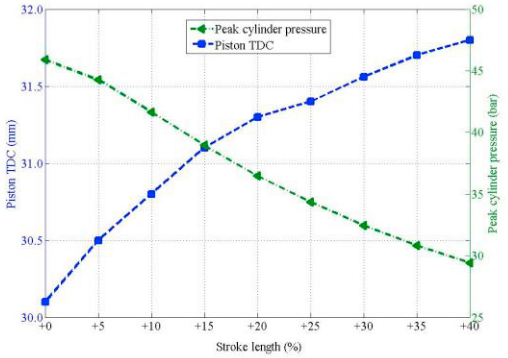

Fig. 4. Engine performance with various stroke

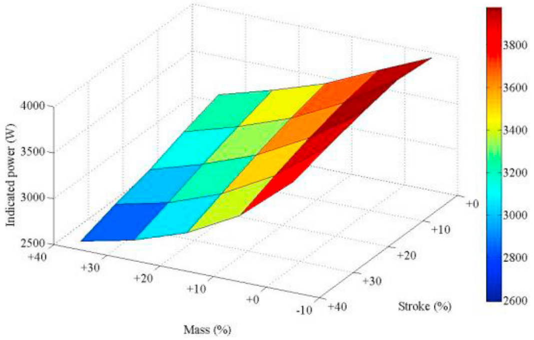

Fig. 5. Indicated power mass with various mass and stroke

As the piston and the mover are connected by a rod, the length of the rod will affect the stroke of the engine. A simulation is done with various stroke length of -40% to +40% (42.0 mm to 98.0 mm) from its initial value 0% (70.0 mm). The moving mass is set to 5.0 kg, and the other parameters stay the same. It is found that the peak in-cylinder pressure drops linearly when the stroke increases, while the achieve piston TDC increases gradually, as shown in Fig. 4. When the stroke drops below the 70.0 mm, the TDC achieved does not meet the required position for ignition, thus engine misfire occurs. However, if the stroke is longer than +40%, the peak in-cylinder pressure is lower than 30.0 bar, which means the indicated power and engine efficiency would be lower. As a result, the stroke is suggested to be controlled within 70.0 mm to 98.0 mm in order to avoid misfire and to achieve higher efficiency. As the bore is set to 52.5 mm, the bore stroke ratio is then suggested to be controlled within the range from 0.54 to 0.74, and the FPE engine is a typical long-stroke engine type.

For a real prototype, the moving mass and the stroke length are coupled with each other, and both of them can affect the indicated power. According to the Fig. 5, it is observed that with fixed stroke, lower moving mass will lead to higher indicated power. With fixed moving mass, shorter stroke will make higher indicated power. The peak indicated power of approximately 4.0 kW is found to be achieved with the moving mass of 5.0 kg and the stroke length of 77.0 mm.

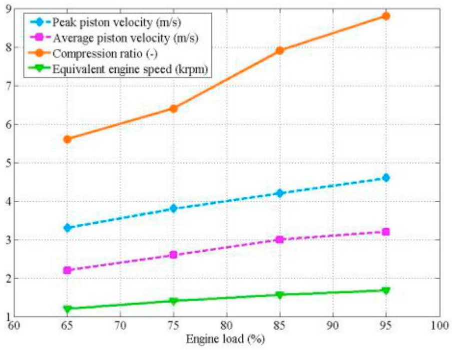

Fig. 6. Engine performance with different load

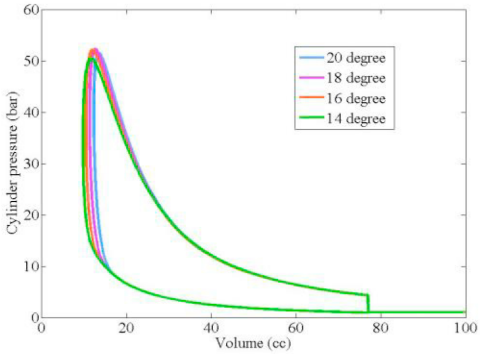

Fig. 7. p-V diagram with different ignition timing

3.2. Operation conditions

The engine performance can be varied with different operation conditions. Fig. 6 demonstrated the engine performance changing with engine load from 65% to 95% of the full load. A stoichiometric air/fuel mixture is assumed and the engine is operated in stable generating process. The ignition timing is fixed at 7.0 mm from the cylinder head. The maximum piston velocity, average velocity, and the compression ratio vary in positive correlation with the load. When the engine load drops below 65%, the compression ratio is lower than 6:1, which will be affect the combustion efficiency, and thus the FPELG is suggested to be operated at high load. The equivalent engine speed increases from 1200 rpm to nearly 1800 rpm with the engine load changes from 65% to 95%. It is found that the FPELG is prone to operate at relative low speed compared with the conventional engines. As result, the power output of the FPELG will be limited by its narrow speed range.

3.3. Combustion parameters

If the engine load is decided, the variable parameters for the combustion model are the ignition timing. Nevertheless, due to the elimination of the crankshaft mechanism, the piston displacement is used as a reference signal to decide the ignition timing. According to the equivalent ignition timing in crank angle from the reference [7], the series equivalent ignition timings in crank angle of 14°, 16°, 18° and 20° that correspond to 5.4 mm, 6.2 mm, 7.0 mm and 7.8 mm from the cylinder head respectively. As shown in Fig. 7, the TDC, or the compression ratio achieved is higher with earlier ignition timing. However, with the increase of ignition timing advance, the peak in-cylinder pressure fluctuated and reaches its maximum value with the ignition timing advance of 18°. The peak in-cylinder pressure changes from 50.5 bar to over 52.5 bar with slight change of the ignition timing advance. The heat release process of the FPELG is more close to a constant volume combustion process with later ignition timing, and the area enclosed by the p-V diagram is larger. As a result, the indicated work generated by the FPELG is higher with later ignition timing. As a result, the ignition timing is supposed to be a potential variable for the future stable operation control system.

4. Summary

In this paper, parametric analysis of a dual-piston type free-piston engine generator is presented, along with the model validation results. Model validation was undertaken with the test data from a running prototype, which showed good agreement.

It was observed that with relative higher moving mass, both of the peak cylinder pressure and achieved compression ratio are higher. While the heat release process was more close to a constant volume process when the moving mass was lower. And the FPELG engine was found to be a typical long-stroke engine type. With fixed stroke, lower moving mass would lead to higher indicated power. With fixed moving mass, shorter stroke would make higher indicated power. The maximum piston velocity, average velocity, and the compression ratio were found to vary in positive correlation with the load. It is found that the FPELG was prone to operate at relative low speed (approximately 1200 rpm to 1800 rpm) compared with the conventional engines, and the power output of the FPELG would be limited by its narrow speed range. The achieved piston TDC, or the compression ratio was higher with earlier ignition timing. While the indicated work was higher with later ignition timing.

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