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프리피스톤 리니어엔진의 스프링경도에 따른 수치해석연구

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A Numerical Simulation for the Spring Hardness of a Free Piston Linear Engine

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Abstract >> This research numerically analyses the effects of the damping device on the operation characteristics of a free piston linear engine. In this paper, the free piston linear engine uses spring as a damping device. The investigated parameter is spring hardness which is varied at 0.5, 1, 2.9, and 14.7 N/mm. The effects of spring hardness on the dynamic characteristic, thermodynamic characteristic and electric power of the engine are investigated. Beside, the equivalent ratio is also changed to provide more information for this study. The simulation results show that, by increasing spring hardness from 0.5 to 14.7 N/mm, all of parameters related to dynamic characteristic such as piston velocity, acceleration, displacement, and frequency increase accordingly. Beside, the peak pressure in the cylinder and electric power are also increased when increasing spring hardness. The tendency is also observed at varied equivalent ratios.

Key words : Spring hardness(스프링 경도), Free piston(프리피스톤), Linear engine(리니어 엔진)

Nomenclature

- A : area of piston
- P_a : intake pressure
- P_l : pressure in the left cylinder
- P_r : pressure in the right cylinder
- F_f : friction force
- F_e : electric force
- F_{sl} : the left spring force
- F_{sr} : the right spring force
- m : mass
- x : displacement of piston
- x_0 : initial coordinate of piston
- a : acceleration of piston
- t : time
- P_{cl} : compression pressure in the left cylinder
- P_{cr} : compression pressure in the right cylinder
- x_{epl} : left closing coordinate of exhaust port
- x_{epr} : right closing coordinate of exhaust port
- x_m : the maximum stroke of piston
- x_{cl} : position of the left piston in compression
- x_{cr} : position of the right piston in compression
- p : instantaneous pressure in cylinder

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γ : specific heat ratio
V : instantaneous volume in cylinder
Q_c : heat released in combustion
Q_{in} : total input energy
Q_{ht} : heat transfer
χ : mass fraction burned
t_0 : time combustion begins
Δt : duration combustion
h : heat transfer coefficient
A_t : heat transfer area
T : instantaneous temperature in cylinder
T_w : wall temperature
P_{el} : expansion pressure in the left cylinder
P_{er} : expansion pressure in the right cylinder
P_{ecl} : pressure at the end of combustion in the left cylinder
P_{ecr} : pressure at the end of combustion in the right cylinder
n : polytropic exponent
x_{ecl} : position of the left piston at the end of combustion
x_{ecr} : position of the right piston at the end of combustion
x_{el} : position of the left piston in expansion
x_{er} : position of the right piston in expansion
W : average cylinder gas velocity
V_d : displaced volume
V_r : volume at reference state
P_r : pressure at reference state
T_r : temperature at reference state
P_m : motored cylinder pressure
$\overline{s_p}$: average speed of piston
k : spring hardness
Δx_l: deformation of the left spring
Δx_r: deformation of the right spring

1. Introduction

The crisis of global warming and shortage of fossil fuels are becoming important issues for human life over the world. There are many researchers are devoted to explore new conversion devices and environmental friendly fuels. The free piston linear engine (FPLE) is one of solutions to solve the crisis. The engine is the combination of two main components: one is the free piston engine, and another is the linear alternator. Unlike the conventional engines with crankshaft mechanism, the FPLE can optimize the combustion process through the variable compression ratios. Beside, the variation of compression ratio in FPLE also allows the engine to operate with multi-fuel. In general, the FPLE can be classified into three types including single piston, dual piston and four piston[1,5,6]. In which, the single piston engine has simple design with high controllability compared to the other free piston engine, however the dynamic balance is not good because it only has one piston. For four piston engine, the perfectly balanced design is the main advantage of this engine, however it also makes the engine complicated. For dual piston engine, the working piston provides the work to drive the compression process in the other cylinder, it allows a simple device with higher power / weight ratio[2]. Because the FPLE has no crankshaft, the operation of this engine is mainly controlled by electronic system. For the dual piston engine, the cylinder head may be stricken by piston crown if the control is not correct. In that case, a damping device (eg. spring) is used as one standby solution. Jin Xiao[3] presented a study about the motion characteristic of a FPLE. In which, the effect of damping coefficient to the motion characteristic of FPLE is investigated. The results showed that increasing the damping coefficient are effective ways to prevent engine damage.

This paper presents a numerical simulation about FPLE with type of dual piston. The engine uses LPG as a fuel with component including 30% propane and 70% butane. Spring is used as a damping device in the engine. The paper focuses mainly on the numerical analysis of this engine through simulation models. The effects of spring hardness on the operation characteristic of the engine such as velocity, acceleration, displacement, peak pressure and electric power are investigated.

2. Working Principle

As shown in Fig. 1, the construction of FPLE includes two main components: free piston engine and linear alternator.

The free piston engine contains dual piston connected by connecting rod system. Spark plug is arranged at each cylinder head to ignite LPG-air mixture at the end of compression process. There are two intake ports and one exhaust port around circumference at the bottom of each cylinder. In order to increase intake pressure, the engine is designed with two compressor at two sides from the engine as shown in Fig. 1. Each compressor also includes dual piston which is connected by connecting rod system. Beside, in the compressor, the spring is arranged at each cylinder as a damping device. In order to provide intake mixture to the cylinder of engine in one way, a reed valve is arranged at each cylinder head of compressor. The second component is the linear alternator with the permanent magnet mounted on the connecting rod as the translator. The back iron made of Silicon steel material and the windings are arranged in the stator.

To start engine, the linear alternator will operate as a beginning device to drive free piston engine through connecting rod system. After certain frequencies, the combustion process will occur alternatively at each cylinder, forcing connecting rod to move back and forth. The movement of connecting rod will generate the current in the winding due to the magnetic flux linked with winding in stator is changed.

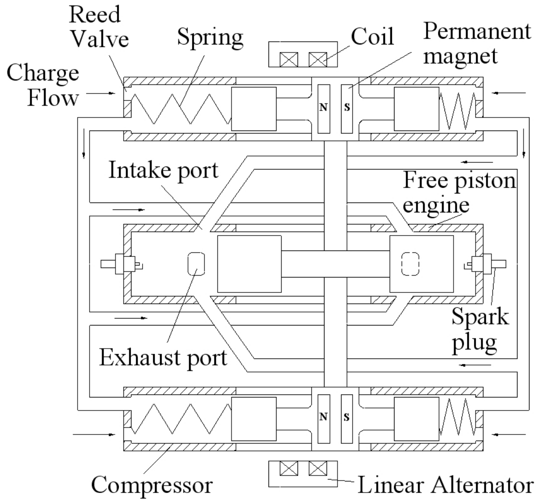

Fig. 1 Free body diagram of FPLE

3. Simulation Model

The objective of this section is to develop a numerical model that describes the operation of FPLE. There are three simulation models used in the analysis including dynamic model and thermodynamic model.

3.1 Dynamic model

The forces applied on the linear engine are expressed through a free body diagram as Fig. 2.

The dynamic model is described by equation that obeys Newton’s second law:

$$P_l A - P_r A - F_f - F_e + F_{sl} - F_{sr} = m\frac{d^2x}{dt^2} = ma \quad (1)$$

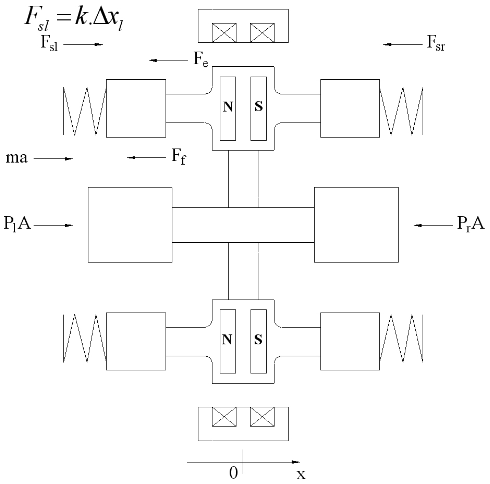

Fig. 2 Free body diagram of FPLE

Where:

$$F_{sl} = k.\Delta x_l \tag{2}$$

$$F_{sr} = k.\Delta x_r \tag{3}$$

The velocity and position of translator can be found by equations:

$$\frac{dx}{dt} = \left(\frac{dx}{dt}\right)_0 + \frac{d^2x}{dt^2}t \tag{4}$$

$$x = x_0 + \frac{dx}{dt}t + \frac{\left(\frac{d^2x}{dt^2}t^2\right)}{2} \tag{5}$$

3.2 Thermodynamic model

Thermodynamic model is described by processes such as compression, combustion, expansion and scavenging. In which, the scavenging is assumed as a perfect process.

The compression process is calculated between the time when exhaust port is closed and when the spark occurs. In the analysis, compression process is assumed to obey a polytropic equation. The pressure during compression process is given as follow:

$$P_{cl}(t) = P_a\left(\frac{x_m + x_{epl}}{x_m + x_{cl}(t)}\right)^n \tag{6}$$

$$P_{cr}(t) = P_a\left(\frac{x_m - x_{epr}}{x_m - x_{cr}(t)}\right)^n \tag{7}$$

The combustion process is assumed to occur immediately after the spark occurs, which means that the ignition delay is ignored. The pressure in the combustion process is calculated as follow:

$$\frac{dp}{dt} = -\gamma\frac{p}{V}\frac{dV}{dt} + \frac{\gamma-1}{V}\left(\frac{dQ_c}{dt}\right) \tag{8}$$

$$\frac{dQ_c}{dt} = \frac{dQ_{in}}{dt} - \frac{dQ_{ht}}{dt} \tag{9}$$

The heat release rate dQ_{in}/dt is calculated when the mass fraction burned is known. The mass fraction burned can be found through the Wiebe function. The Wiebe function is usually used to calculate the mass fraction burned versus crank angle for crankshaft engine. Because the FPLE has no crankshaft, the mass fraction burned is calculated as a function of time

$$\chi = 1 - \exp\left[-a\left(\frac{t-t_0}{\Delta t}\right)^{m+1}\right] \tag{10}$$

Where, a and m are adjustable parameter (according to Heywood[4], a = 5, m = 2)

By differential two sides of the equation (10), the mass fraction burned rate is given

$$\frac{d\chi}{dt} = a\frac{m+1}{\Delta t}\left(\frac{t-t_0}{\Delta t}\right)^m \exp\left[-a\left(\frac{t-t_0}{\Delta t}\right)^{m+1}\right] \quad (11)$$

From equation (11), the heat release rate can be calculated as follow

$$\frac{dQ_{in}}{dt} = a\frac{m+1}{\Delta t}\left(\frac{t-t_0}{\Delta t}\right)^m \exp\left[-a\left(\frac{t-t_0}{\Delta t}\right)^{m+1}\right].Q_{in} \quad (12)$$

The heat transfer rate is calculated by

$$\frac{dQ_{ht}}{dt} = h.A_t.(T-T_w) \quad (13)$$

In the equation (13), the heat transfer coefficient is calculated by

$$h = 3.26.B^{-0.2}.P^{0.8}.T^{-0.55}.W^{0.8} \quad (14)$$

Where

$$W = \left[C_1\bar{S}_p + C_2\frac{V_d T_r}{P_r V_r}(P-P_m)\right] \quad (15)$$

The expansion process is given from the end of combustion process until the exhaust port opens. The process is also assumed to obey a polytropic equation:

$$P_{el}(t) = P_{ecl}\left(\frac{x_m + x_{ecl}}{x_m + x_{el}(t)}\right)^n \quad (16)$$

$$P_{er}(t) = P_{ecr}\left(\frac{x_m - x_{ecr}}{x_m - x_{er}(t)}\right)^n \quad (17)$$

Table 1 Parameters are used in simulation

Parameters	Value	Unit
Bore	30	mm
Reciprocating mass	0.974	kg
Exhaust port open	20	mm
Maximum stroke	35	mm
Polytropic exponent	1.4	
Intake pressure	1.1	bar
Intake temperature	300	K
Equivalent ratio	1.0	
Combustion duration	1.5	ms
The natural length of spring	60	mm
Spring hardness	Available	N/mm
Frictional coefficient	1.04	
Lower heating value of LPG	40.95	MJ/kg

4. Results and Discussion

The dynamic model and thermodynamic model are combined and solved by a Fortran program. A virtual prototype with parameters and initial conditions are listed in Table 1. These parameters are chosen base on operated conditions and specifications of actual FPLE.

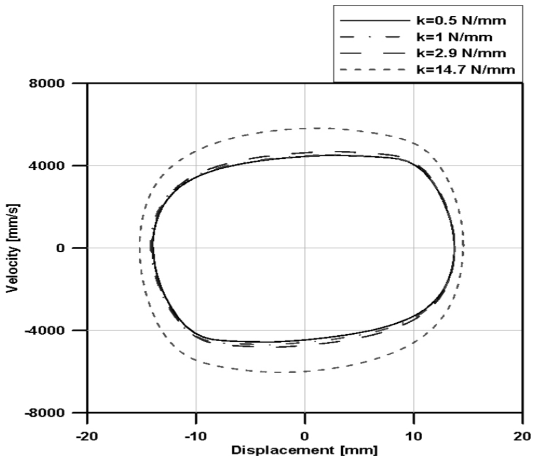

Fig. 3 Piston velocity versus displacement

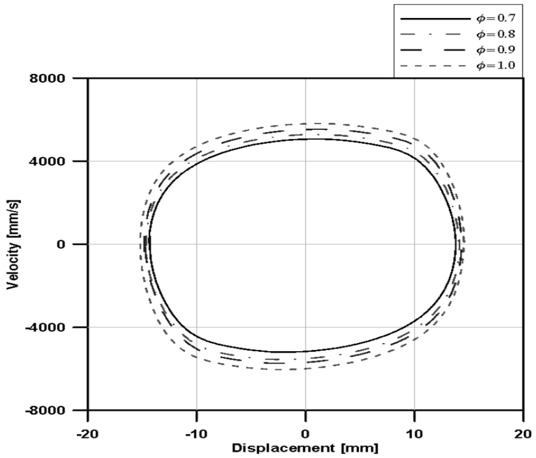

Fig. 4 Piston velocity versus displacement

4.1 The effect of spring hardness to the dynamic characteristic of FPLE

In this section, the effect of spring hardness to the dynamic characteristic of FPLE including velocity, displacement, acceleration and motion frequency are investigated. Beside, by keeping a certain spring hardness, the effect of equivalent ratio to the dynamic of FPLE is also investigated.

Fig. 3 describes the piston velocity versus displacement for different spring hardness including 0.5, 1, 2.9 and 14.7 N/mm. At the same initial conditions as shown in Table1, by increasing spring hardness from 0.5 N/mm to 14.7 N/mm, the piston velocity increase accordingly. This can be seen clearly at the highest spring hardness corresponding to 14.7 N/mm. Beside, the displacement of piston is also longer at higher spring hardness, it means that the compression ratio is increased.

Fig. 4 shows the piston velocity versus displacement at the different equivalent ratios with the same spring hardness k = 14.7 N/mm. It can be observed

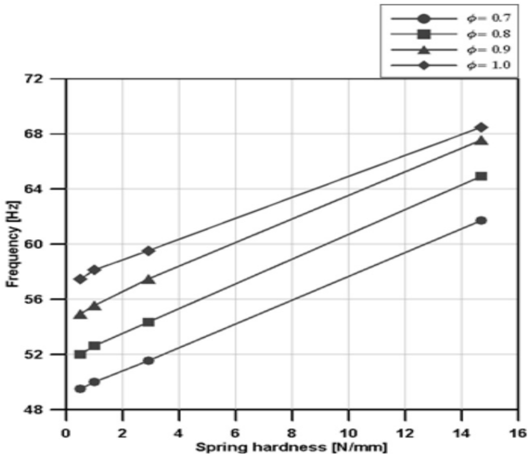

Fig. 6 Motion frequency versus spring hardness

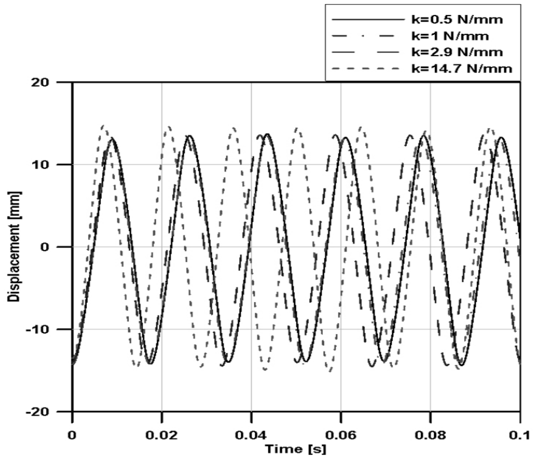

Fig. 5 Displacement versus time

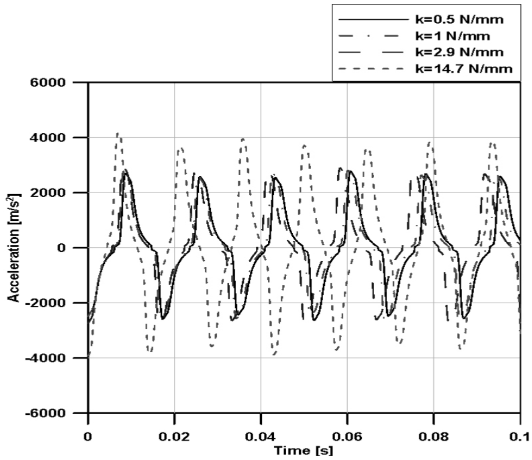

Fig. 7 Acceleration versus time

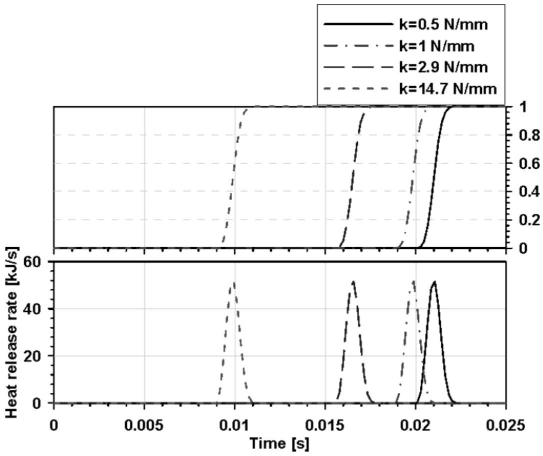

Fig. 8 Mass fraction burned and heat release rate versus time

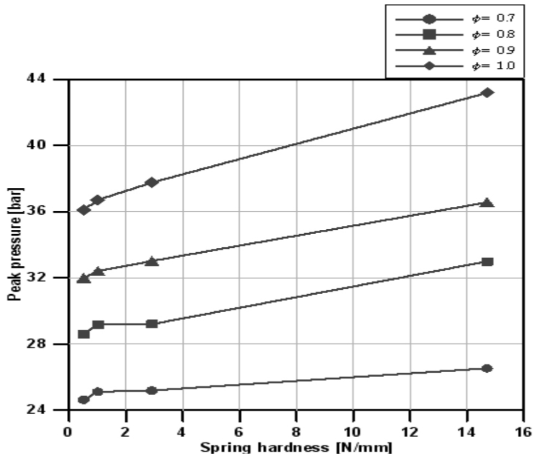

Fig. 9 Peak pressure versus spring hardness

that the piston velocity is reduced when decreasing equivalent ratio. Even though the piston velocity is changed, but the curves keep the same shape.

The relation between displacement of piston and time is shown in Fig. 5. At the highest spring hardness corresponding to k = 14.7 N/mm, the maximum displacement of piston is largest with average value around 15mm. Beside, the higher spring hardness, the shorter time for full stroke. This is due to the increase of piston velocity as shown in Fig. 3. As the result, the motion frequency of piston is increased when increasing spring hardness as shown in Fig. 6. The tendency also occurs at the lower equivalent ratios

Fig. 7 shows a plot of piston acceleration versus time with the effect of different spring hardness. It can be found that the acceleration of piston is improved significantly when spring hardness up to 14.7 N/mm.

4.2 The effect of spring hardness to the thermodynamic characteristic and electric power of FPLE

Fig. 8 describes the effect of spring hardness on the mass fraction burned and heat release rate in the combustion chamber of FPLE.

It can be found that, the higher spring hardness, the earlier combustion process occurs. This is due to the increase of piston acceleration which is shown in Fig. 7.

Because the movement of piston is almost symmetric as shown in Fig. 4, the pressure in left cylinder or right cylinder can be chosen as a analysis parameter. In this paper, the pressure in right cylinder is chosen to investigate under changing of spring hardness and equivalent ratio.

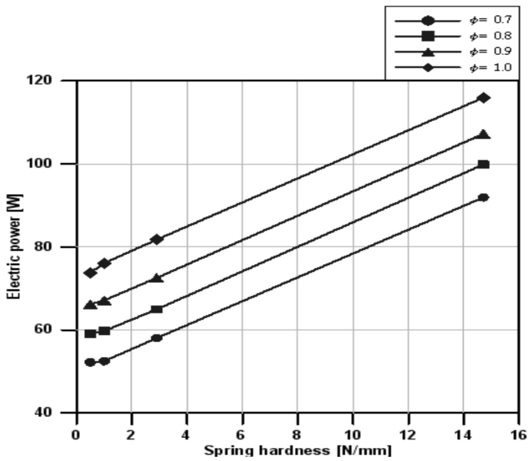

Fig. 10 Electric power versus spring hardness

Fig. 9 illustrates the effect of spring hardness to the peak pressure at different equivalent ratios. At the equivalent ratio of 1.0, by increasing spring hardness, the peak pressure increase respectively. This is because the compression ratio of the engine is increased due to increasing of piston velocity as shown in Fig. 3. At the same spring hardness, when decreasing equivalent ratio as well as the amount of fuel, the peak pressure is smaller as shown in Fig. 9. This is also reason to explain the reduction of piston velocity as shown in Fig. 4.

Fig. 10 shows a relation between spring hardness and electric power which is obtained from the reciprocation of piston. It can be found that the electric power increases when increasing spring hardness. This is explained by increasing of piston velocity as shown in Fig. 3.

5. Conclusion

A numerical simulation related to the damping device as spring of a FPLE has been investigated in this paper. The spring hardness was used as an investigated parameter in the simulation. The simulation results shows that

1) The dynamic characteristic of FPLE is better at higher spring hardness.
2) The higher spring hardness, the earlier combustion process occurs.
3) The peak pressure in the cylinder and electric power is improved when increasing spring hardness
4) Although the spring hardness up to 14.7 N/mm, the maximum displacement of piston is less than theory maximum stroke. It means that the hit between piston crown and cylinder head is avoided.

The study is base to choose a suitable spring for designing the FPLE. In the future, a further study will be carried out with the effect of more different parameters to operation of the engine. Beside, the performance of FPLE will be investigated with different fuels.

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