Simulation of Free Piston Linear Engine Motion with Different Intake and Exhaust Port Positions

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ABSTRACT

This paper presents the simulation of two-stroke free piston linear engine motion consisting of combustion and air-kickback chambers. Dynamic and gas-dynamic models for slider-piston motion are presented. The influence of the intake and exhaust port position on the engine performance is explored. Using chosen variables, the motion of two-stroke free piston linear engine is successfully simulated. It shows that the velocity profile of slider-piston motion is far from sinusoidal. The comparison between both simulations with different location of intake and exhaust port reveals that the slider-piston velocities are not similar. Slider-piston velocity of model 2 with intake and exhaust port at the top of the cylinder block is 19 % higher than model 1 (intake and exhaust port at the cylindrical-side of cylinder block). Although that was the case, model 1 has an advantage in a way that it does not need valves and open-close valves system.

Keywords: Free piston engine, two-stroke engine, intake, exhaust, dynamic

ABSTRAK

Kertas kerja ini membentangkan simulasi gerakan enjin linear omboh bebas dua lejang yang mempunyai satu kebuk pembakaran dan satu kebuk udara-tendang balik. Model dinamik dan model dinamik-gas untuk gerakan gelangsar-omboh dibentangkan. Kajian ini menyelidiki kesan kedudukan pancarongga masukan dan ekszos terhadap pergerakan enjin. Hasil simulasi menunjukkan bahawa profil halaju gelangsar-omboh tidak menyerupai bentuk sinus. Perbandingan hasil simulasi antara kedua model yang berbeza kedudukan pancarongga mendapati bahawa halaju kedua-dua model adalah tidak serupa. Halaju gelangsar-omboh model 2 dengan pancarongga terletak di hujung atas silinder adalah 19 % lebih tinggi daripada model 1 yang pancarongga di sisi silinder. Walaupun demikian, model 1 memiliki satu kelebihan iaitu tidak memerlukan injap dan sistem injap buka-tutup.

Kata kunci: Enjin linear omboh bebas, enjin dua lejang, masukan, ekzos, dinamik

INTRODUCTION

A free piston linear engine is a machine with linear motion as the primary motion. The crankshaft that is present on the conventional engine has been elimi-nated. This linear engine has benefits in terms of efficiency, weight reduction, robustness, variable compression operation and multi-fuel possibilities (Arshad et al. 2002). Free piston linear engines offer the potential to provide power without the conversion of linear to rotary motion (Nandkumar 1998).

The most common model that has been developed for free piston linear engine systems is dual opposed combustion chamber model (Atkinson et al. 1999; Goldsborough & Blarigan 1999; Houdyschell 2000; Nandkumar 1998). Meanwhile, the other model of free piston linear engine that have been developed is the model that replace one combustion chamber with other device. For example, a free piston linear engine that uses a spring as the opposite to combustion chamber has been developed (Annen et al. 2002). Air cushion chamber has been developed by Aichlmayr et al. (2002) in order to replace a combustion chamber in common free piston linear engine.

This paper focuses on free piston linear engine that one combustion chamber is replaced with air kickback chamber. The influences of different location of intake and exhaust ports in modeled engine are investigated for the system. The previous work, Fonna et al. (2004), clearly shows that the motion profile of linear engine is far from sinusoidal.

MODELED ENGINE

Free piston linear engine could be divided into two categories. The first category is free piston linear engine with intake and exhaust port at the cylindrical-side of the cylinder block. This engine is shown in Figure 1. The second category is engine with intake and exhaust port at the top of the cylinder block that is shown in Figure 2. These engines are called model 1 and model 2, respectively, for easy identification. The models are two-stroke spark ignition engine. The free piston linear engines consist of five main parts, i.e. combustion chamber, scavenging chamber, kickback chamber, generator housing, and slider-piston. The term slider-piston is used to describe the unity of combustion piston, kickback piston, connecting rod, and permanent magnet. The rod connects the two oppositely placed pistons and also acts as a prime mover for linear engine. The permanent magnet is placed at certain position of the rod. The two opposing pistons have different diameter. Kickback piston diameter is larger than combustion piston diameter to ensure the generated force in the kickback chamber is adequate to push back slider-piston.

Free piston linear engine when coupled with linear alternator becomes a linear generator. When the slider-piston moves linearly, it will cause a disturbance of the field that was produced by the permanent magnet. Thus, an electromagnetic force will be in-duced in the coil of the linear alternator. This is the working principle of free piston linear generator that is producing electricity directly from the linear motion of the
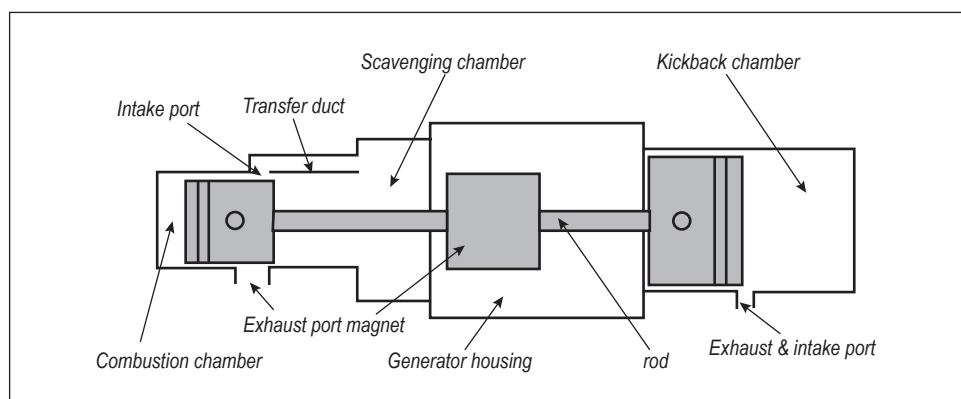

Figure 1. Free piston linear engine with intake and exhaust port at cylindrical-side of the cylinder block (model 1)
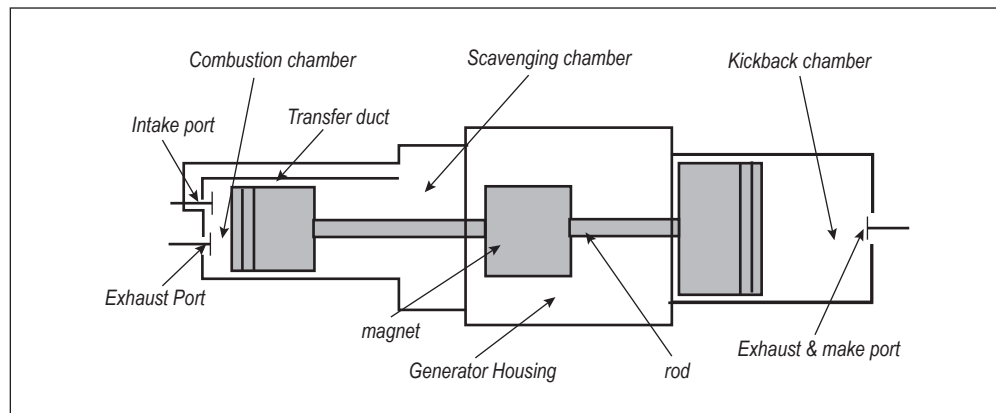
Figure 2. Free piston linear engine with intake and exhaust port at the top of the cylinder block (model 2

pistons. The salient feature of free piston linear engine is a mechanically unconstrained piston (Aichlmayr et al. 2002). Thus, the system obeys Newton's second law,

$$\sum F = m\frac{d^2x}{dt^2} \qquad (1)$$

where the left hand side of equation (1), i.e. $\sum F$, represents the s ummation of the forces that act in the plane of motion, the right hand side of equation (1), i.e. d^2x/dt^2, is the acceleration of slider-piston with x representing the displacement of sliderpiston and m is the slider-piston mass.

In order to analyze the slider dynamics, the forces acting on the slider-piston are determined. These forces are due to the combustion cylinder pressure, the scavenging pressure, the air kickback cylinder pressure, the piston ring-wall friction, and electromagnetic force. The forces balance on free piston is given by equation (2) at combustion condition and equation (3) at air kickback condition.

$$P_c A_c - P_k A_k - P_s A_s - F_f - F_e = m\frac{d^2x}{dt^2} \quad (2)$$

$$P_k A_k - P_c A_c + P_s A_s - F_f - F_e = m\frac{d^2x}{dt^2} \quad (3)$$

where P_c is the instantaneous combustion chamber pressure, P_k is the instantaneous air-kickback chamber pressure, P_s is the instantaneous scavenging chamber pressure, A_c is the combustion piston area, A_k is the air-kickback piston area, A_s is the combustion piston back area, F_f is the friction force, and F_e is the electromagnetic force.

Assuming the engine is running within no load condition, i.e. no power delivered, the electromagnetic force may be neglected in this simulation. According to Atkinson et al. (1999), the friction force is not significant for linear engine. Thus, it can be neglected since the force is relatively very small due to no crankshaft system. Figure 3 is given to illustrate a force balance on free piston for simplified no load condition.

THE GAS-DYNAMIC MODEL

MODEL 1

The compression and expansion of each cylinder are modeled adiabatically and internally reversible. The process in combustion chamber is represented by Otto cycle, while in kickback chamber and scavenging chamber by the ideal gas adiabatic pro-cess (Figure 4). The dimensions of engine model 1, i.e. intake and exhaust port at cylindrical-surface of cylinder block, are also described in Figure 4. The effects of intake and exhaust port length to the generated pressure in each chamber are considered in the analysis.

For the initial condition, the slider is considered to move from the left (combustion chamber top dead center) to the right in positive x direction i.e. expansion process. Then, combustion. chamber is chosen as reference. At these conditions, the pressure in combustion chamber is P_{c3} as the
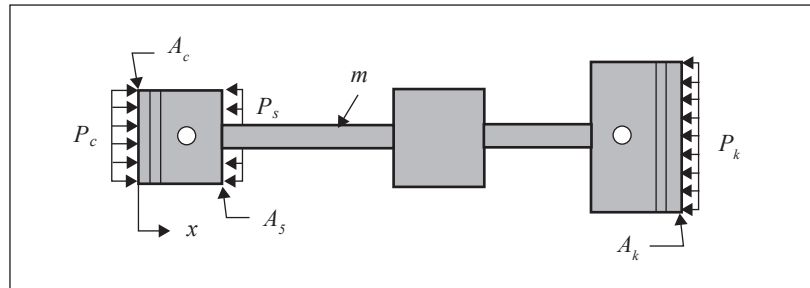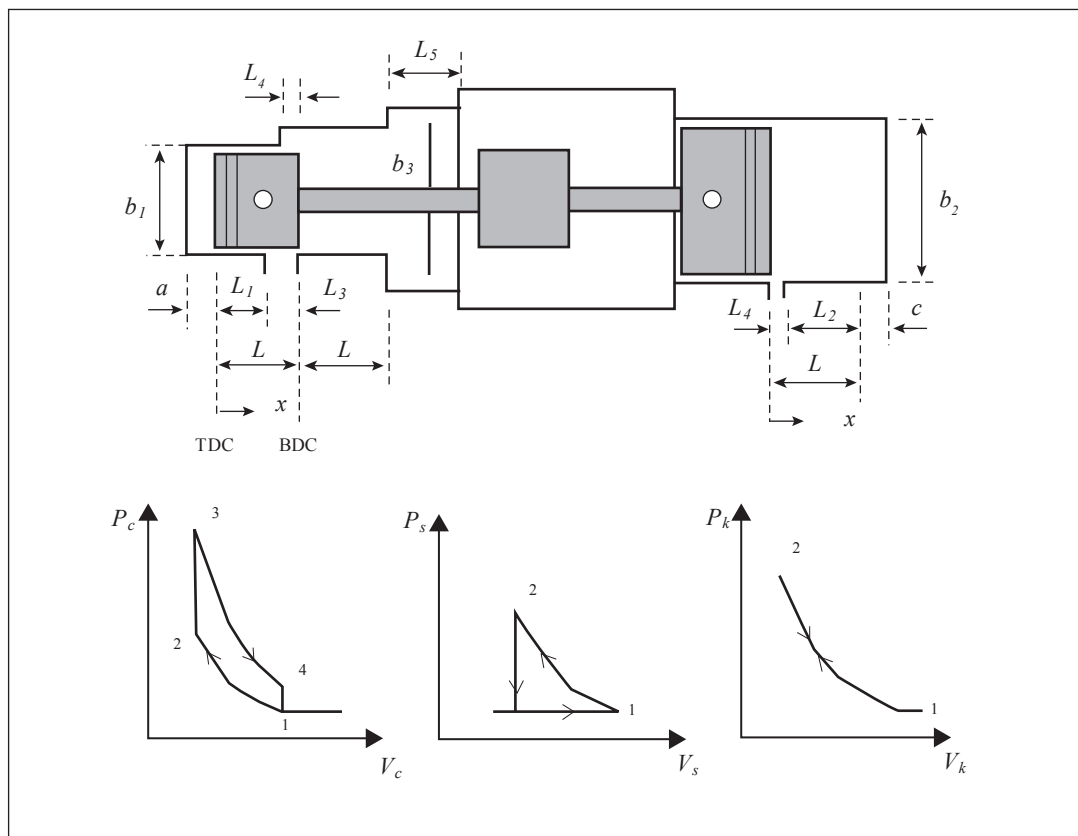
Figure 3. Slider free-body diagram

Figure 4. The dimension of engine and pressure volume diagrams for each chambers modeled for the simulation of model 1
(TDC: top dead center, BDC: bottom dead center)

result of instantaneous heat addition in the combustion process. Meanwhile the pressure in the kickback chamber and scavenging chamber are free-air pressure P_{k1} and P_{s1}, respectively. When the slider is moving to the right, the combustion pressure will follow *3-4* path of the $P_c V_c$ diagram and at point *4* the exhaust port s tarts to open, and the pressure suddenly drops to P_{c1} due to instantaneous heat rejection. The scavenging pressure will in-crease along *1-2* path in the $P_s V_s$ diagram and the pressure drops to P_{s1} from point *2* when the intake port starts to open. The kickback pressure mean-while remains at P_{k1} along L_4 because the port is still open and starting from point *1*, the pressure will increase following *1-2* path in the $P_k V_k$ diagram.

Considering the condition when the slider moves from r ight to left after the expansion stroke i.e. compression stroke, the combustion pressure is P_{c1} along L_3 and the pressure increase following *1 -2* path in $P_c V_c$ diagram starting from point *1*. The kickback pressure will take the same path as the expansion stroke; while the scavenging pressure remains at P_{s1} along compression stroke.

MODEL 2

The dimensions of engine model 2, i.e. intake and exhaust port at the top of cylinder block, are described in Figure 5. Similar to model 1, the slider is considered to move from left to right in positive *x* direction i.e. expansion process. At these conditions, the pressure in combustion chamber is P'_{c3} as a result of instantaneous heat addition in combustion process, while the pressure in kickback chamber and scavenging chamber are free-air pressure P'_{k1} and P'_{s1}, respectively. When the slider is moving to the right, the combustion pressure will follow *3-4* path in the $P'_c V'_c$ diagram and at point *4* the pressure suddenly drop to P'_{c1} because of instantaneous heat rejection. The scavenging pressure will increase along *1-2* path in the $P'_s V'_s$ diagram; while the kickback pressure will increase following *1-2* path in the $P'_k V'_k$ diagram.

Considering the slider moves from the right to the left after the expansion stroke i.e. compression stroke, the compression increases following *1-2* path in the $P'_c V'_c$ diagram starting at point *1*. The kickback pressure will take the same path as expansion stroke; meanwhile the scavenging pressure remains at P'_{s1} along compression stroke.
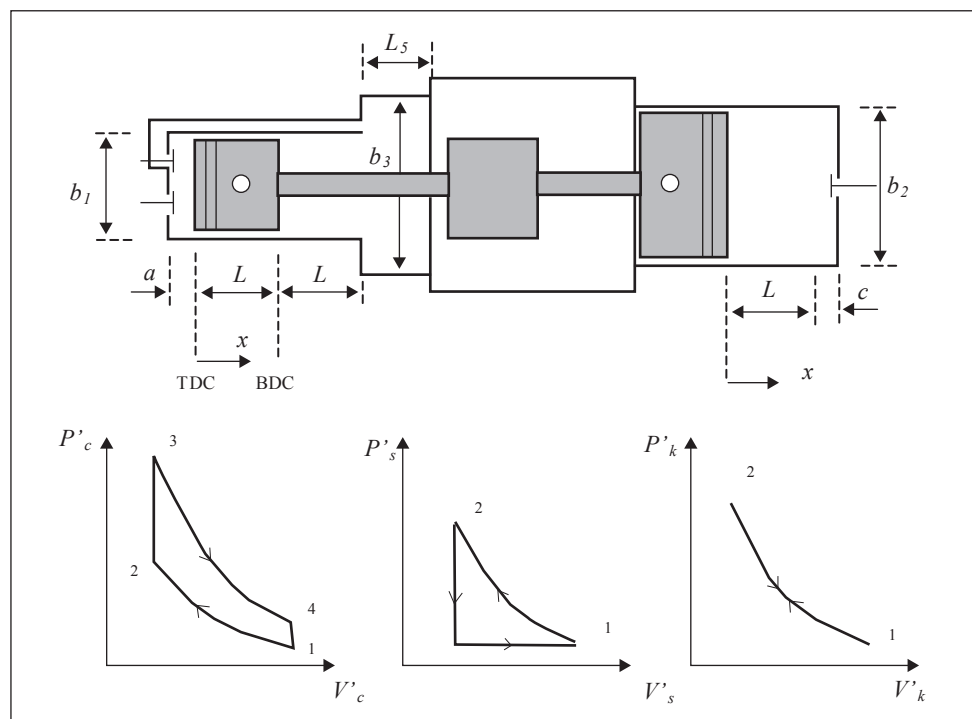

Figure 5. The dimension of engine and pressure volume diagrams for each chambers modeled for the simulation of model 2
(TDC: top dead center, BDC: bottom dead center)

In order to develop the pressure equations of each chamber, the processes are assumed to obey a polytropic process. The relationship between the pressure and the volume for polytropic process is described by (Heywood 1988):

$$PV^n = const \qquad (4)$$

The interdependence between the three thermodynamic properties pressure (P), volume (V), and temperature (T) is given by the state equation. For ideal gases, the state equation is given by (Heywood 1988):

$$PV = mRT \qquad (5)$$

where P is the gas pressure, V is the gas volume, m is the gas mass, R is the gas constant, and T is the absolute temperature. This is also known as the law of ideal gases.

RESULTS AND DISCUSSION

MODEL 1

The equations that described the pressures of each chambers are derived from equation (4) and/or equation (5). Equations (6) and (7) represent the force balance equations for expansion process and for compression process, respectively, in engine model 1, where P_{atm} is atmospheric pressure, Q_{in} is the heat addition to combustion chamber during combustion process, and n is the specific heat ratio.

MODEL 2

Equations (8) and (9) represent the force balance equations for expansion process and for compression process respectively, in engine model 2. These equations show that compression ratio for model 2 is higher than model 1 since $L>L_1$ and $L>L_2$.

$$\left\{ P_{atm}\left(\frac{L_1+a}{a}\right)^n + \frac{Q_{in}\left(\frac{L_1+a}{a}\right)(n-1)}{(L_1+a)A_c} \right\}\left(\frac{a}{a+x}\right)^n A_c - P_{atm}\left(\frac{L_2+c}{c}\right)^n \left(\frac{c}{L_2+L_4+c-x}\right)^n A_k$$

$$- P_{atm}\left\{\frac{(b_3)^2 L_5 + (b_1)^2 2L}{(b_3)^2 L_5 + (b_1)^2 (2L-x)}\right\}^n A_s \quad -F_f - F_e = m\frac{d^2 x}{dt^2} \qquad (6)$$

$$P_{atm}\left(\frac{L_2+c}{c}\right)^n \left(\frac{c}{L_2+L_4+c-x}\right)^n A_k - P_{atm}\left(\frac{L_1+a}{a}\right)^n\left(\frac{a}{a+x}\right)^n A_c$$

$$+ P_{atm}A_s - F_f - F_e = m\frac{d^2 x}{dt^2} \qquad (7)$$

$$\left\{ P_{atm}\left(\frac{L+a}{a}\right)^n + \frac{Q_{in}\left(\frac{L+a}{a}\right)(n-1)}{(L+a)A_c} \right\}\left(\frac{a}{a+x}\right)^n A_c - P_{atm}\left(\frac{L+c}{c}\right)^n \left(\frac{c}{L+c-x}\right)^n A_k$$

$$- P_{atm}\left\{\frac{(b_3)^2 L_5 + (b_1)^2 2L}{(b_3)^2 L_5 + (b_1)^2 (2L-x)}\right\}^n A_s \quad -F_f - F_e = m\frac{d^2 x}{dt^2} \qquad (8)$$

$$P_{atm}\left(\frac{L+c}{c}\right)^n\left(\frac{c}{L+c-x}\right)^n A_k - P_{atm}\left(\frac{L+a}{a}\right)^n\left(\frac{a}{a+x}\right)^n A_c$$

$$+ P_{atm}A_s - F_f - F_e = m\frac{d^2x}{dt^2} \tag{9}$$

Table 1. Simulated engine specifications

Parameter	Model 1	Model 2
Effective combustion stroke	0.04 m (L_1)	0.07 m (L)
Effective kickback stroke	0.055 m (L_2)	0.07 m (L)
Exhaust length (L_3)	0.03 m	0.03 m
Intake length (L_4)	0.015 m	0.015 m
Back-scavenging length (L_5)	0.05 m	0.05 m
Specific heat ratio (n)	1.4	1.4
Mass of slider (m)	4.6 kg	4.6 kg
Free air pressure (P_1)	100 kPa	100 kPa
Combustion crevice length (a)	0.004 m	0.004 m
Kickback crevice length (c)	0.006 m	0.006 m
Combustion bore diameter (b_1)	0.076 m	0.076 m
Kickback bore diameter (b_2)	0.11 m	0.11 m
Back-scavenging diameter (b_3)	0.114 m	0.114 m
Heat addition (Q_{in})	300 J	300 J

In order to generate free piston simulation, some engine specifications have been chosen. These specifications are given in Table 1 including the geometry of the modeled engines.

By using the force balance equations, the simulations of the slider-piston motion are derived. The slider-piston velocity profiles of both models are shown in Figure 6. The velocity is taken as parameter because it gives a clear description about motion profile than acceleration. These velocity profiles are far from sinusoidal as for conventional engine. Comparing the velocity profile between simulation of model 1 and model 2 in Figure 6, model 2 gives higher velocity and also less time needed to complete one stroke than model 1. As shown in Figure 5, engine model 2 has different configuration than model 1 in term of the position of intake and exhaust port. This arrangement gives the higher effective volume for engine model 2. Thus, the work is higher for engine model 2 as represented by wider area under *PV* curve. Therefore, the velocity of slider-piston is also higher for engine model 2. From Figure 6, it can be concluded that the average velocity of slider-piston of model 2 is 19 % higher than model 1.

Figure 7 shows the slider-piston position versus stroke time for both modeled engine. It reveals that, both modeled engine give approximately the same stroke length, ~70mm, although they have different slider-piston velocities. The modeled engines have the same geometry. However, the effective combustion and kickback strokes are different. Therefore, each engines has different compression ratio, i.e. engine model 2 compression ratio is higher than engine model 1 since the effective stroke length of model 2 is longer than model 1. This is due to the fact that, the generated pressures/forces in each chamber, i.e. combustion chamber and kickback chamber, is higher for engine model 2. The increased pressure in kickback chamber in engine model 2 compensates the increased pressure in combustion chamber, thus the slider-

piston stroke remains the same as engine model 1. Figure 7 also shows that the stroke time is shorter for engine model 2 since the increased pressure would increase the sliderpiston acceleration.

The relationship between the slider-piston velocity and the slider-piston position for both modeled engines is illustrated in Figure 8. It gives a clear description that model 2 slider-piston velocity is higher than for model 1 and the stroke length for both modeled engine is approximately the same.

This fact revealed that model 2 would produce more power since the producing power is linear to engine speed. Although such was the case, model 2 has a disadvantage in a way that it needs valves and open-close valve system comparing to model 1. These would increase engine component number and mechanical losses. The results show good compromise with the previous work by Nandkumar (1998) although by a different engine model was used. Nandkumar (1998) also indicated that the increasing of compression ratio will increase engine speed/velocity.
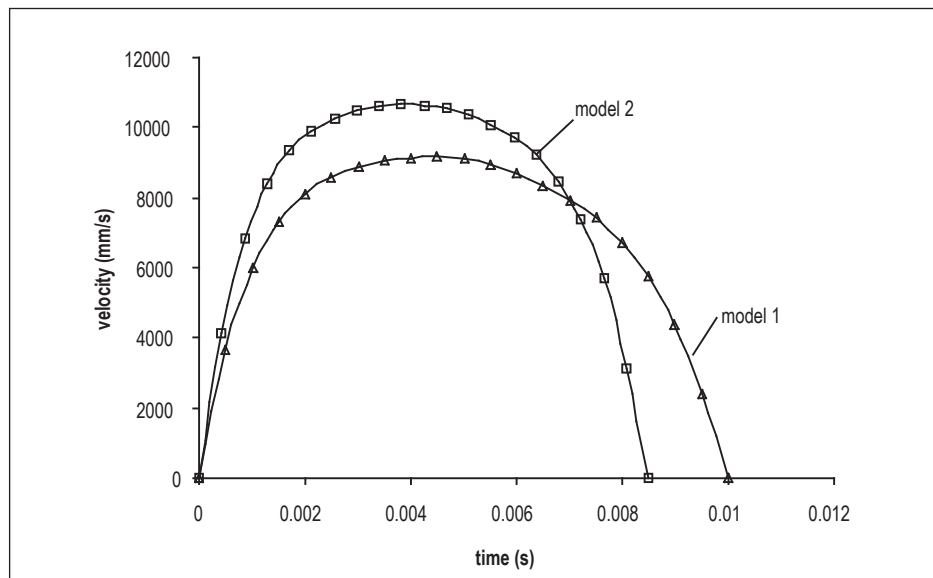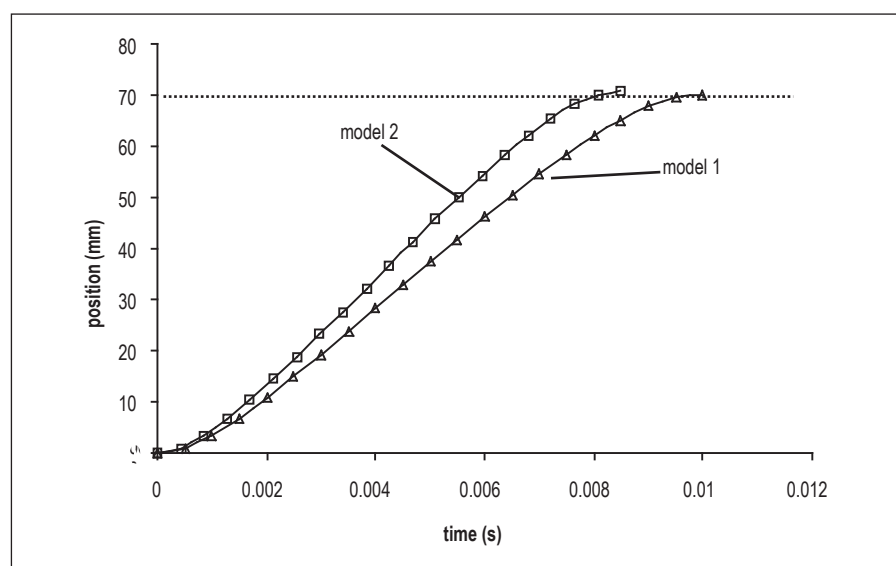

Figure 6. Piston velocity versus stroke time

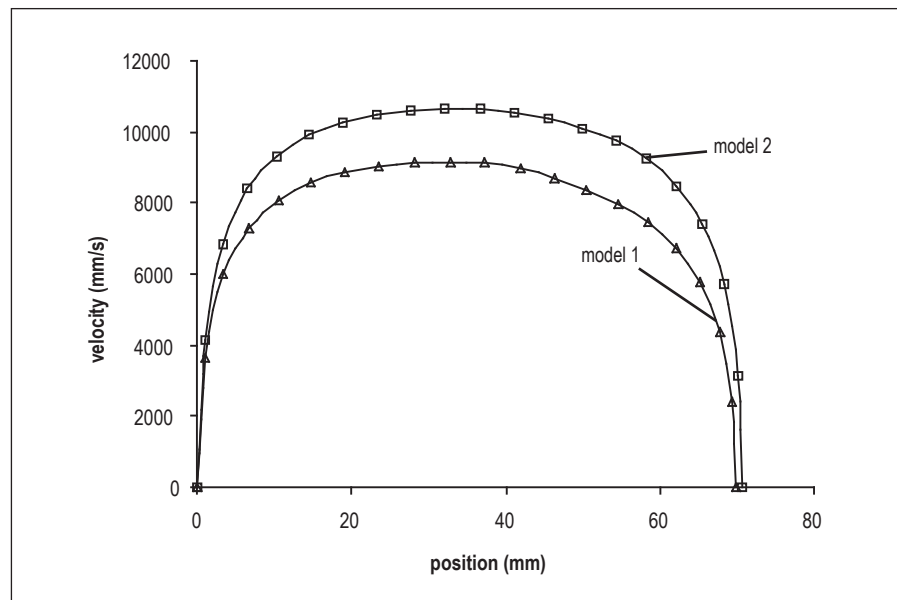**Figure 7.** Piston position versus stroke time

Figure 8. Piston velocity versus piston position

CONCLUSIONS

Dynamic and gas-dynamic model of free piston linear engine have been presented in this paper. The velocity and displacement profiles of slider-piston motion with respect to time are investigated for different modeled engines. The slider-piston motion of two-stroke free piston linear engine was successfully simulated. By comparing the results between engine model 1 and model 2, it revealed that the slider-piston velocity for the engines are not similar in completing one stroke, even for the same-modeled engine geometry. Slider-piston velocity of model 2 is 19 % higher than model 1. However, model 1 does not need valves and open-close valves system as for model 2.

ACKNOWLEDGMENTS

TERMINOLOGY

Symbol	Description	Units	Symbol	Description	Units
A	Area	m^2	Q_{in}	Heat addition	J
b	Diameter	m	R	Gas constant	kJ/(kmol.K)
F	Force	N	T	Temperature	K
m	mass	kg	t	Time	s
n	Specific heat ratio	-	V	Volume	m^3
P	Pressure	Pa	x	Displacement	m

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