Operational Strategies for a Free Piston Energy Converter

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Abstract

This paper investigates how the losses in a Free Piston Energy Converter during start, stop and idling affects energy consumption and required power from the supply system.

Simulation results indicate that the electrical machine efficiency is the most critical factor during start. Moreover, by choosing the correct amplitude of the starting force, energy consumption during start can be reduced. When it comes to idling, friction is the most significant loss factor. Nevertheless, by compensating the mechanical loss for short time intervals using the generator force, the reciprocating motion can be kept alive for a rapid start without major energy consumption.

Keywords: energy conversion, hybrid vehicle, Free Piston Energy Converter, auxiliary power unit, energy, power, HCCI

1. Introduction

As the world is reaching the peak of oil production, the oil prices are rising and the environmental legislations for vehicles are getting harder there is a rising interest in alternative drivetrains and engines which can reduce fuel consumption and emissions. The Free Piston Energy Converter (FPEC) is such an engine, which converts chemical energy to electrical. The intended application for the FPEC is as a power unit in a series hybrid vehicle.

A hybrid electric vehicle's (HEV) performance is very dependant on the vehicle's energy management strategy. In fact, controlling the energy sources inappropriate may increase emissions and lower the system efficiency compared to a conventional vehicle. Therefore, to make efficient use of the FPEC in a HEV it is desirable to know which operational mode the FPEC should be in and what cost that mode is associated with. Furthermore, to design the supplying electrical system for optimal performance, the required energy and power for each mode must be known.

This paper studies the operational modes which are the most essential for a vehicle application with the exception of running: start, idling and to some extent stop. The associated cost will be energy consumption. Moreover losses, required power and time issues are considered.

2. Free Piston Energy Converter

The Free Piston Energy Converter consists of a linear electrical machine placed between two opposing combustion chambers as seen in Fig. 1. Combustion in the chambers makes the translator reciprocate and generate electrical energy.

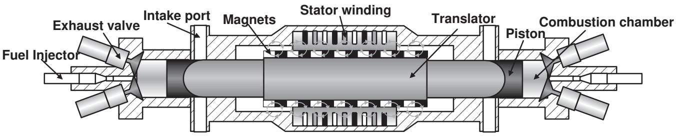

Fig. 1. Cross section of a Free Piston Energy Converter

As no crankshaft is limiting the translator motion the compression in the cylinders can be varied. This makes a modern combustion technique called Homogenous Charge Compression Ignition (HCCI) possible. In this combustion process the air-fuel mixture is injected at the beginning of the stroke and compressed until ignition, hence the name. HCCI has an indicated efficiency around 50%, low fuel consumption and low emissions especially of NOx and particulate.

One of the drawbacks of linear engines and HCCI has been that both the combustion process and the translator motion are harder to control than in a conventional engine. However, with the FPEC the electrical machine can be used as an extra actuator to improve controllability of the system.

During recent years the interest in FPECs has grown and prototypes have been built. One contribution can be seen in Fig. 2.

3. Operational modes

As there is a choice between idling and a stop followed by a start, it is useful to have some cost associated with these operational modes to know which to utilize. In this study energy consumption has been chosen but other costs could be emissions or fuel consumption.

3.1 Start-up The start scenario investigated is to inject the air-fuel mixture, close the exhaust valves

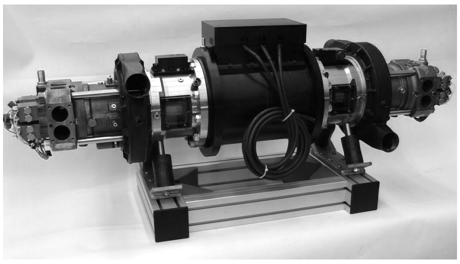

Fig. 2. Free Piston Energy Converter prototype developed in (3) (4) .

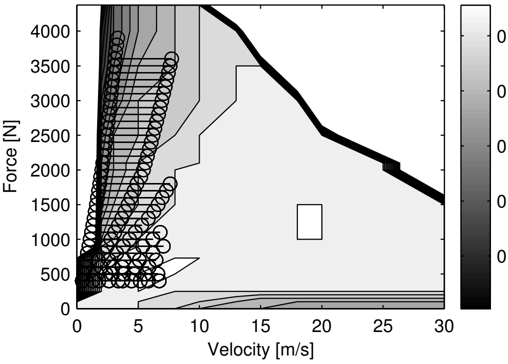

Fig. 3. Efficiency map (3) for the electrical machine with working points for each start-up case.

and use the electrical machine force to move the translator back and forth until the mixture ignites. Ignition is dependant on several parameters, the two most important are: pressure and temperature. However, it is not sufficient just to reach the required level of temperature and pressure. In addition this level or higher must be kept for a while to make the mixture ignite. That is, if the required levels of pressure and temperature are passed early in the stroke, they will rise further and the mixture may ignite. On the other hand, if the required level is reached near the end of a stroke, there may not be enough time for the mixture to ignite and one additional stroke is necessary.

A desirable starting procedure should be rapid so no major delay is noticed before power is delivered from the generator. At the same time, the power required to start should not exceed the rated values of the supplying components. Furthermore, a low energy demand may increase the total system efficiency.

Using closable intake valves or not will also affect the start strategy. If the valves are not closable the air-fuel mixture may be diluted if too many strokes are needed for ignition during start-up.

3.2 Idling

Idling mode can be useful if the power demand is low only for a short while and a rapid start-up is desired. From an emission point of view idling may also be favorable, as a transient start-up of a combustion engine usually results in high emissions.

When idling is required the fuel injection is stopped and the exhaust valves is closed, forcing the translator to bounce back and forth on the gases in the cylinders. This mode is very dependant on the mechanical losses in the system as they are the energy reducing factor. Nevertheless, if idling is a desirable mode, it would be possible to maintain a reciprocating translator motion without combustion by cancelling out the mechanical losses using the generator force.

3.3 Stop

When the system is stopped, time is not an issue but as much energy as possible should be regenerated and stored for a good energy economy. As friction and other losses will reduce the translator kinetic energy over time a rapid stop is desirable.

The available electrical force acting on the translator is not sufficient to stop the translator in one stroke, thus at least two strokes is required before the translator is drained of all energy.

4. System modeling

To investigate the operational modes a dynamic simulation model of a 45 kW FPEC is implemented using Matlab/Simulink. Reference (1) and (2) suggest models of Free Piston Engines and in this paper a similar approach is used.

The gas in the cylinders is assumed to be ideal,

$$p_i V_i = m_g R T_i, \cdots\cdots\cdots\cdots\cdots\cdots (1)$$

and the compression is assumed to be polytropic,

$$p_i V_i^{\gamma} = \text{const.} \cdots\cdots\cdots\cdots\cdots\cdots (2)$$

These two assumptions hold if the pressure is not too high. In (1)-(2) γ is the specific heat ratio, m_g is the gas mass, p_i is cylinder pressure, V_i is cylinder volume, R is the universal gas constant and T_i is the gas temperature. By using Newton's second law and adding the losses described below the differential equations shown in appendix 1 can be derived.

5. Losses

The most significant loss factors during start and stop will probably be friction and electrical machine efficiency. However, as this is a combustion engine the heat transfer to the cylinder walls is also considered.

5.1 Electrical machine efficiency

The electrical machine efficiency is modelled using an efficiency map from (3). However, when starting from standstill the losses in the electrical machine will probably be very high the first milliseconds due to inertia, transient currents etc. Therefore, the electrical machine efficiency is assumed to be only 25% during the first stroke. For the following strokes it is taken from the efficiency map given in Fig. 3.

5.2 Heat transfer to walls

Some of the heat generated will leak out through the cylinder walls and thus delay the ignition. This heat transfer is mainly dependent on the temperature difference between the gas and the cylinder walls. However, during start-up this temperature difference will not be as big as during combustion. Consequently, the heat transfer will not be a major loss. Here the heat transfer is modelled according to Hohenberg (1):

$$\frac{\partial Q_{loss}}{dt} = \alpha A_p (T(t) - T_w(t)) \cdots\cdots\cdots\cdots\cdots (3)$$

$$\alpha = 130 V(t)^{-0.06} \left(\frac{p(t)}{10^5}\right)^{0.8} T(t)^{-0.4} (\bar{V}_p + 1.4)^{0.8} (4)$$

$\bar{V}_p$ is the piston mean velocity, T_w is the cylinder wall temperature, Q_{loss} is the heat loss and A_p is the piston area.

5.3 Friction Friction is an essential parameter especially if the FPEC is in idling mode. As this will be the major loss factor, the size of the friction work will determine how long the system can reciprocate before reaching standstill.

Few empirical results are available for this type of engine, therefore friction is hard to predict. However, as the crankshaft mechanism is eliminated the friction should be quite low and originates from the friction between the cylinder wall and the piston skirt and piston rings (2).

Reference (1) and (2) have utilized a model for a two-stroke engine with rolling element bearings and it is also used in this investigation. Since the shape of the friction force is unknown it is modelled using the Friction Mean Effective Pressure (FMEP) which is a mean value over the stroke.

$$FMEP = \frac{W_f}{V_d} \cdots\cdots\cdots\cdots (5)$$

$$W_f = F_f 2 L_{stroke} \cdots\cdots\cdots\cdots (6)$$

W_f is the friction work, V_d is the displaced cylinder volume and L_{stroke} is the stroke length. The FMEP can also be expressed as

$$FMEP = 6300 L_{stroke} f \cdots\cdots\cdots\cdots (7)$$

where f is the translator frequency. Combining (5)–(7) results in a mean friction force described by

$$F_{fric} = \frac{6300}{2} V_d f. \cdots\cdots\cdots\cdots (8)$$

5.4 Model characteristics Implemented in the simulation model are two other characteristics that may affect the result.

- Intake ports cannot be closed. As a result the cylinder pressure takes the outer pressure value every time the translator is uncovering the port. This introduces losses if the cylinder pressure is higher than the outer pressure at that time. However, simulations with closed inlet ports indicate no major difference in required energy and power for start.
- When the translator is reversing almost no power is delivered, therefore the generator force cannot load the translator near the turning points.

6. Results and discussion

6.1 Start Start of the FPEC using a constant force from 400 N–4000 N was simulated. Forces lower than 400 N were investigated initially; however, start-up became impossible for these cases as the friction force eventually exceeded the generator force.

Furthermore, it is assumed that an ignition can begin when the pressure rises above 35 bar and the temperature above 700 K. Just reaching these levels may not be sufficient as discussed earlier. Therefore the time with pressure and temperature above the desired levels was recorded.

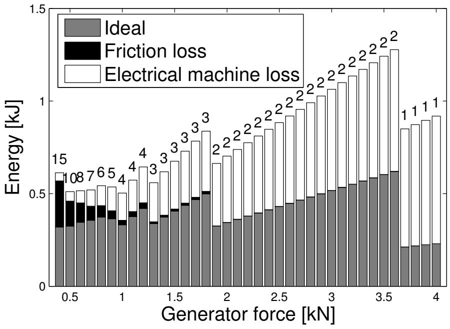

Fig. 4. Required electrical energy to start a combustion. Full bar is total energy consumption. The required number of strokes is presented above the bars.

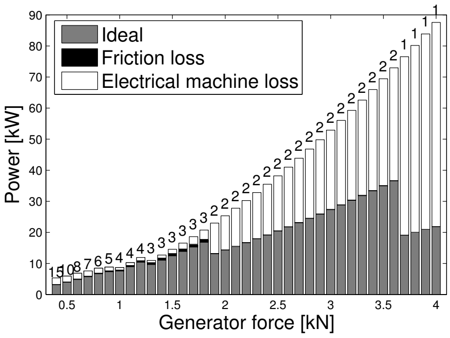

Fig. 5. Required peak electrical power to start a combustion. Full bar is total peak power. The required number of strokes is presented above the bars.

It can be seen in Fig. 4 that every time the required number of strokes to start combustion decreases so does the required energy. Consequently, the energy consumption can be minimized for a desired number of strokes. Furthermore, the total friction loss decreases with higher force and thus shorter start-up time. In fact it only has a minor influence for a two stroke or faster start-up. The heat loss during start is negligible and cannot even be discerned in the bars. This was an expected result due to the low temperature difference as discussed earlier.

The required power increases with shorter start-up time as seen in Fig. 5. The electrical machine efficiency has major effect on the required power. For a generator force over 2700 N the power exceeds 45 kW which is the maximal power of the FPEC. As the supplying system has to manage 45 kW, using a start power up to this level is acceptable, but it is not desirable to design the system for higher powers just to be able to start. However, the assumption of only 25% electrical machine efficiency the first ms affects the result heavily for the high-force cases and may be too pessimistic.

The time for start-up is presented in Fig. 6. It can be seen that for the same number of strokes no major time improvement is made by using a larger force. This supports the conclusion that the lowest possible force for a desired number of strokes should be used for low energy consumption.

A realistic scenario could be a two-stroke start-up scenario which results in a start time of approximately 50 ms. However, the time with required ignition level

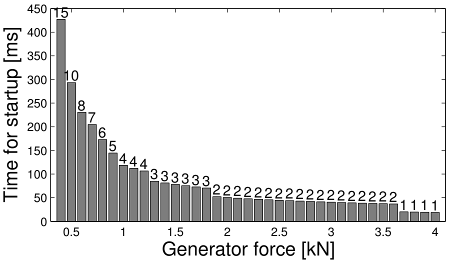

Fig. 6. Time to reach ignition level. The required number of strokes is presented above the bars.

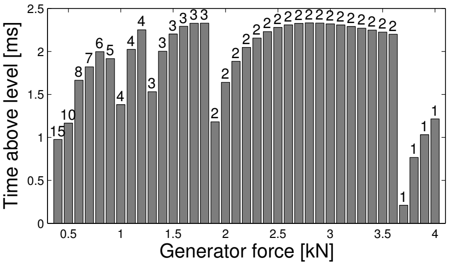

Fig. 7. Time with pressure and temperature above required level for ignition. The required number of strokes is presented above the bars.

Table 1. Initial values for idling simulation.

p_1	4.7 [MPa]	p_2	0.12 [MPa]	x_0	-6.3 [cm]	a_0	4000 [m/s2]
T_1	977 [K]	T_2	360 [K]	v_0	0 [m/s]		

is quite small for the first two-stroke case as seen in Fig. 7. Hence, a larger start-up force may be necessary to assure ignition.

6.2 Idling The simulation is started using the initial values shown in Table. 1 which corresponds to a 29kW working point. A combustion has just occurred in cylinder 1, the velocity is zero and the acceleration is high as the translator has just reversed.

The friction force, during the first stroke after combustion, is approximately 330 N and a friction work of 42 J is done. Consequently, the translator motion can be kept alive at an operating point near the FPEC rated power of 29 kW by pushing the translator with a force of 330 N. Doing this with a frequency of 60 Hz over a distance of 12 cm results in a energy consumption of 2400 J/s for idling. On the other hand, looking at the required energy to start in Fig. 4 it spans from 500–1300 J for the cases investigated. Therefore, idling for long time-intervals may not be efficient from an energy point of view. However, it may not be necessary to idle near this working point. Decreasing the reciprocating frequency during idling would lower the friction force and thus the required energy. This would still result in high efficiency as the electrical machine efficiency (Fig. 3) is acceptable over a large area. When deciding on a good idling working point combustion issues should also be considered and it is not the aim of this investigation.

It should be noted that the idling scenario's numerical result is heavily dependant on the friction model. But as the start-up scenarios use the same friction model the losses will be proportional and a comparison between these two modes is possible.

6.3 Stop When running stationary at 29 kW the maximal translator velocity is approximately 11 m/s which results in a kinetic energy of

$$W_k = \frac{mv^2}{2} \approx 544\text{J}. \cdots\cdots\cdots\cdots\cdots\cdots (9)$$

Consequently, this is the maximal energy that can be regenerated from the FPEC during a stop from this operating point.

7. Conclusions

Heat loss, and friction seems to have minor effect on the required start-up power and energy levels due to the rapid start-up time. Moreover, the required energy and power for start can be minimized by choosing the minimal force necessary for a desired number of start-up strokes. Furthermore, idling for long time intervals may be costly from an energy point of view.

It is planned to validate some of the results provided here on the prototype developed in (4).

References

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Appendix

1. Modeling

Subscript i refers to cylinder one or two.

$$\frac{dp_i}{dt} = -\frac{dQ_{loss}}{dt}\frac{p_i(t)}{m_g c_v T_i(t)} - \frac{p_i(t)\gamma}{V_i(t)}\frac{dV_i}{dt} \cdots\cdots \text{(A1)}$$

$$\frac{dV_1}{dt} = v(t)A_p \cdots\cdots \text{(A2)}$$

$$\frac{dV_2}{dt} = -v(t)A_p \cdots\cdots \text{(A3)}$$

$$\frac{dv}{dt} = \frac{A_p}{m_p}(p_1(t) - p_2(t)) + \frac{1}{m_p}(F_{el}(t) - F_{fric}(t)) \text{(A4)}$$

$$\frac{dx}{dt} = v(t) \cdots\cdots \text{(A5)}$$

Table A1. Variables and constants used in (A1)–(A5)

$p_i(t)$	pressure	[Pa]	$v(t)$	translator velocity	[m/s]
$T_i(t)$	temperature	[K]	$V_i(t)$	cylinder inner volume	$[m^3]$
$F_{el}(t)$	generator force	[N]	A_p	piston area	$[m^2]$
m_p	translator mass	[kg]	$x(t)$	translator position	[m]
γ	specific heat ratio	[-]	c_v	specific heat at const. vol.	$[\frac{J}{kg}K]$