The design and simulation of a two-stroke free-piston compression ignition engine for electrical power generation. ⋆

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Abstract

Free-piston engines are under investigation by a number of research groups worldwide due to their potential advantages in terms of fuel efficiency and engine emissions. Some prototypes have emerged, mainly aimed for vehicle propulsion, and have reported favourable performance compared to conventional technology. This paper describes the design of a modular compression ignition free-piston engine generator, applicable to electric power generation in large-scale systems. The development of a full-cycle engine simulation model is described, and extensive simulation results are presented, giving insight into engine operating characteristics and performance. The operating characteristics of the free-piston engine was found to differ significantly from those of conventional engines, giving potential advantages in terms of fuel efficiency and emissions formation due to fast power stroke expansion. Effects of varying engine stroke length and compression ratio were not found to give any large advantages.

Key words: free-piston, two-stroke, diesel, linear combustion engine generator

1. Introduction

In light of the increased interest in the all-electric ship concept and with the aim of increasing engine efficiency and reducing exhaust emissions, work on the “more electric engine” concept was started in 1999 within the School of Marine Science and Technology, at the University of Newcastle upon Tyne. This work was subsequently supported by BAE Systems, Engineering and Physical Sciences Research Council (EPSRC) and The Institute of Marine Engineering, Science and Technology (IMarEST), and resulted in a small scale prototype system being developed. Currently a more detailed and comprehensive design study is being conducted to investigate potential large-scale applications.

The “more electric engine” concept is based on a free-piston internal combustion engine coupled to a linear electric machine. Free-piston engines have received more interest in recent years after being practically abandoned since the 1960’s. The modern developments include electric generators aimed for hybrid-electric automotive vehicles and hydraulic pumps aimed for off-highway vehicles with high hydraulic loads. Some prototypes of these machines have reported favourable performance compared to conventional technology.

1.1. *The free-piston engine*

The free-piston engine concept was first presented by Pescara [1] in 1928, and since then a number of free-piston designs have been proposed. Common for these is that the piston motion is not restricted

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by the motion of a rotating crankshaft, as known from conventional engines, but that the piston is free to move between its endpoints, only influenced by the gas and load forces acting upon it. This gives the free-piston engine some distinct characteristics, most importantly variable stroke length and high control requirements.

Only a handful of successful free-piston engines been reported, despite a high interest in the concept for a number of years. Free-piston air compressors developed by the German company Junkers were used by the German navy during World War 2, supplying compressed air for launching torpedoes [2]. In the 1940's, free-piston engines found use as gas generators, feeding hot gas to a power turbine. This concept was employed in marine and stationary powerplants, most successful being a model developed by Société Industrielle Générale de Mécanique Appliquée (SIGMA) in France [3]. Both Ford and General Motors later developed prototype vehicles with small-scale free-piston gas generators as prime movers, but none of these made it past the prototype stage [4,5]. As the technology of both conventional engines and gas turbines matured, the interest in the free-piston engine vanished in early 1960's.

With the introduction of modern, microprocessor-based control methods, the free-piston engine has again caught interest among present-day engineers seeking to reduce engine emissions and increase efficiency. Most promising at the moment appears to be the hydraulic free-piston engine, where among others the Dutch company Innas BV has reported performance advantages over conventional technology [6]. A number of other research groups are also working on this type of engine, which is mainly aimed for off-highway vehicles such as forklift trucks and earth-moving machines. The coupling of a free-piston engine to a linear electric generator is also being investigated, amongst others by researchers at University of West Virginia [7]. The main application for such units are hybrid-electric vehicles.

A more comprehensive background study on the history and potential advantages of the free-piston engine was presented by Mikalsen and Roskilly [8].

Potential advantages of the free-piston engine concept include: vibration-free operation; lower production and maintenance costs and reduced frictional losses due to mechanical simplicity; reduced heat transfer losses and NO_x emission production due to faster power stoke expansion; higher part load efficiency and multi-fuel possibilities due to combustion optimisation flexibility (variable compression ratio and electronically controlled valve operation and fuel injection timing). This paper presents the design of the "more electric engine" and investigates the general performance of the unit. It aims to identify some of the potential advantages of free-piston engines over conventional technology through a full-cycle engine simulation model.

2. Engine design

2.1. *A brief description of the engine configuration*

An engine as shown in figure 1 is proposed. The main parts of the engine are a combustion cylinder, a bounce chamber cylinder and a linear electric machine. The engine will have only one moving part, rigidly connecting the two pistons and the translator of the linear alternator. The piston will move freely between its two endpoints, its motion being determined by the instantaneous balance of cylinder gas forces, electric machine force and frictional forces.

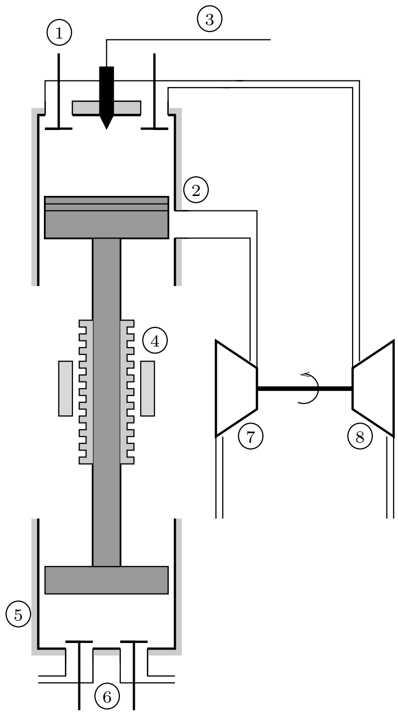


Fig. 1. Free-piston engine.

Figure 1 shows the proposed engine design, with:
① Exhaust poppet valves
② Scavenging ports
③ Common rail fuel injection

④ Linear alternator
⑤ Bounce chamber
⑥ Bounce chamber pressure control valves
⑦ Turbocharger compressor
⑧ Turbocharger turbine

The engine will operate on a turbocharged two-stroke diesel cycle with direct injection and electronically controlled fuel injection, utilising modern common rail technology. Scavenging is provided through scavenging ports in the cylinder liner and electronically controlled exhaust poppet valves in the cylinder head.

The bounce chamber will be a closed cylinder with pressure control valves to regulate the amount of gas trapped in the bounce chamber, and thereby the gas pressure force on the piston. Pressurised air will be provided to the pressure control valves to allow control of the bounce chamber pressure at all operating conditions.

The linear electrical machine is assumed to be a permanent magnet machine, as suggested by Arshad et al. [9]. Appropriate power electronics will be employed to allow control of the electric load force and to allow operation of the electric machine as a motor for starting purposes.

The engine is intended to be modular in the way that a number of such cylinders can be operated in parallel to achieve power outputs required in large-scale applications. A computer control system is to control the engines for performance optimisation and synchronisation of piston motions to cancel out vibrations. Operation of a number of cylinders on a common turbocharger is possible.

2.2. *Engine control*

The piston motion in conventional engines is controlled by the crankshaft, ensuring a sufficient compression ratio for autoignition in one end and sufficient time for scavenging in the other. The lack of this mechanism gives the free-piston engine the possible advantageous feature of variable compression ratio, but requires a piston motion control system to be established.

In addition, for the engine to be feasible in a large scale, it must be balanced. A great advantage of the free-piston engine is that vibrations can be completely cancelled out by making two identical pistons operate in opposite phases. The opposed piston free-piston engines developed in the mid-20th century had mechanical linkages rigidly connecting two pistons operating with a common combustion chamber. The present design, however, seeks to use electronic control to synchronise pairs of engine units to cancel out vibrations.

2.2.1. *Control objectives*

The primary control objectives for the free-piston engine includes the control of top dead centre (TDC) and bottom dead centre (BDC) positions. The TDC position must be controlled within tight limits, as overshoot may lead to mechanical contact between the piston and the cylinder head, which may be fatal for the engine. In addition to this, control of engine speed is required in order to synchronise the operation of multiple engines to minimise vibrations.

Control variables available are the mass of injected fuel (fuel rack position), and bounce chamber pressure. The electric load force is considered a disturbance input, although the implementation of appropriate power electronics may allow some control of this variable. (Fuel injection timing proved in the simulations to only have a minor influence on the main control objectives, and this variable was therefore saved for operation optimisation purposes.)

Secondary control objectives include the control of valve timings, fuel injection timing and compression ratio, the latter to provide the TDC setpoint to the primary controllers, in order to optimise engine operation for all operating conditions.

2.2.2. *Control strategy*

Johansen et al. [10] propose a control strategy for a free-piston gas generator with some similarities to the engine presented here. Mass of fuel per injection and mass of air in the bounce chamber are used to control BDC position and TDC position respectively, whereas engine speed control is achieved by changing engine stroke length by adjusting TDC and BDC setpoints. They further present experimental results showing the feasibility of this approach.

In this study, a somewhat similar control approach have been tested using the simulation model described below, and appears to be feasible. Further work on engine control topics is currently in progress, and for the purpose of the work presented here it is assumed that a satisfactory engine control system can be realised.

3. Modelling

The amount of research specifically addressing the modelling of free-piston engines is small and most authors tend to use well-known models used in conventional engine modelling to model free-piston engines. The engine processes are similar, but the free-piston engine does have some specific operating characteristics compared to the conventional engine, which need to be treated with some caution. This relates in particular to the combustion process, but the particular piston motion profile of the free-piston engine is also likely to influence in-cylinder heat transfer and scavenging.

In this section, the modelling of the free-piston engine generator is presented, and the feasibility of the implemented models is discussed.

3.1. *Piston dynamics*

The piston motion can be derived from Newtons 2nd law, giving

$$F_C - F_L - F_{fr} - F_R = m_p \frac{\mathrm{d}^2 x}{\mathrm{d}t^2} \tag{1}$$

where
x is piston position
m_p is moving mass
F_C is gas force from combustion cylinder
F_R is gas force from bounce chamber
F_L is electric load force
F_{fr} is friction force.

3.2. *Diesel engine submodels*

The submodels for the simulation of the combustion cylinder are based on existing single-zone models commonly used in the modelling of conventional diesel engines.

3.2.1. *Ignition delay*

Correlations for ignition delay based on the Arrhenius equation for reaction rate are widely used in internal combustion engine modelling and a correlation of this type described by Stone [11] has been used:

$$t_{id} = \frac{3.52\ \exp(2100/T)}{p^{1.022}} \tag{2}$$

where
t_{id} is ignition delay [ms]
p is in-cylinder gas pressure [bar]
T is in-cylinder gas temperature [K].

The variables p and T are the mean values during the ignition delay, and to account for the changing conditions in the ignition delay period the following correlation needs to be satisfied:

$$\int_{t_{si}}^{t_{si}+t_{id}} \left(\frac{1}{\tau}\right) \mathrm{d}t = 1 \tag{3}$$

where
t_{si} is the time of start of injection [ms]
τ is the ignition delay at time t [ms].

3.2.2. *Combustion*

The modelling of the heat release in the free-piston engine is one of the factors with the highest degree of uncertainty in the simulation model. The piston motion in the free-piston engine differs significantly from that of conventional engines, and very little research exists on how this influences the combustion process.

Some reports have indicated that the heat release rate is significantly higher in the free-piston engine compared to conventional engines. Somhorst and Achten [12] present measurements of heat release rate taken in the Innas Free-Piston Engine and state that the pressure gradient reaches values two to five times higher than in comparable conventional engines. Tikkanen et al. [13] present similar experimental results, and both groups indicate that combustion takes place predominantly in one pre-mixed phase in the free-piston engine.

A faster combustion is, however, usually an unconditional advantage for engine performance. Hence, should the models used in this work underpredict the heat release rate, they will represent the more pessimistic predictions.

A single-zone combustion model described by Heywood [14] has been used:

$$m_b(t) = \beta f_1(t) + (1-\beta) f_2(t) \tag{4}$$

where
m_b is mass fraction burned [1]
t is time, non-dimensionalised by total time for combustion [1]
β is fraction of fuel burned in the pre-mixed phase [1].

f_1 is the pre-mixed burning function, given as

$$f_1 = 1 - (1 - t^{K_1})^{K_2} \tag{5}$$

and f_2 is the diffusion burning function

$$f_2 = 1 - \exp(-K_3 t^{K_4}). \tag{6}$$

The fraction of fuel burned in the pre-mixed phase, β, is a function including the ignition delay:

$$\beta = 1 - a\,\phi^b / {t_{id}}^c \tag{7}$$

where
ϕ is fuel-air equivalence ratio [1].

K_1, K_2, K_3, K_4, a, b and c are empirical coefficients. Values of these based on experimental data from a truck engine are presented by Heywood [14], and these have been implemented in the model. Total time allowed for combustion is set to 5 ms based on experimental results obtained from the engine against which the model was validated (see also Section 4.1).

3.2.3. *Scavenging*

Accurate modelling of the scavenging process is not possible in an early design phase, as this depends on design details. Estimating the scavenging performance is, however, of high interest to investigate its influence on engine performance, particularly due to the variable stroke which may have effects on scavenging performance.

A simple scavenging model is used to estimate the gas flows in and out of the cylinder. Exhaust blowdown is modelled at the parts of the cycle where exhaust valves are open but the scavenging ports have not yet been uncovered. The process is modelled as a polytropic expansion of the in-cylinder gas down to the exhaust back pressure, with mass flow through the exhaust valves modelled as flow through nozzles, with appropriate valve discharge coefficients [14].

Scavenging is modelled with a perfect displacement model. During scavenging the cylinder is assumed to comprise of two zones; one zone containing fresh air displaces the other zone with combustion products. The zones do not exchange mass or heat and only when all the combustion products have been fully displaced will fresh air flow out through the exhaust ports. Gas temperatures are calculated as the instantaneous mass-weighted average of fresh air and residual products in the cylinder. The thermodynamic consequences of poor scavenging on the next cycle are taken into account to allow simulation of engine transient operation.

3.2.4. *Turbocharger*

The turbocharger model relates the boost pressure and temperature to the exhaust temperature and exhaust back pressure. It receives as inputs: ambient conditions (inlet air temperature and atmospheric pressure), compressor and turbine efficiencies, and exhaust back pressure, all defined by the user. Using these together with exhaust gas temperature from the engine model, the turbocharger model predicts boost air temperature and pressure.

Standard equations for isentropic state changes, allowing isentropic efficiency losses, are used in the calculations. (The calculation procedure can be found in numerous textbooks, including Stone [11].) Turbocharger transient response to changes in engine operation is not modelled.

3.2.5. *Heat transfer*

In-cylinder heat transfer is modelled according to Hohenberg [15]:

$$\dot{Q} = \alpha A_s (T - T_s) \tag{8}$$

where
$\dot{Q}$ is heat flow rate [J/s]
α is heat transfer coefficient [W/m^{2}K]
A_s is in-cylinder surface area in contact with the gas [m^2]
T_s is average temperature of the in-cylinder surface area [K].

The heat transfer coefficient, α, is given by

$$\alpha = 130 V^{-0.06} \left(\frac{p}{10^5}\right)^{0.8} T^{-0.4} (\overline{v}_p + 1.4)^{0.8} \tag{9}$$

where
V is instantaneous cylinder volume [m^3]
$\overline{v}_p$ is mean piston speed [m/s].

Stone [11] suggests a average in-cylinder surface temperature, T_s, of 350 K, and this value is used in the simulations.

3.2.6. *Gas properties*

Having the instantaneous heat transfer rate to/from the in-cylinder gases (heat release from fuel burning minus heat transfer losses), cylinder pressure and temperature are calculated by applying the 1st law of thermodynamics on the cylinder charge:

$$\frac{\mathrm{d}U}{\mathrm{d}t} = \frac{\mathrm{d}Q}{\mathrm{d}t} - \frac{\mathrm{d}W}{\mathrm{d}t} \tag{10}$$

where
U is internal energy [J]
Q is heat transferred to the gas [J]
W is work done by the gas [J].

Assuming that the cylinder charge will behave as a ideal gas, the internal energy is a function of temperature only, giving

$$U - U_0 = mC_v(T - T_0) \tag{11}$$

where
m is mass of the in-cylinder gases [kg]
C_v is specific heat capacity at constant volume [J/kgK].

Further, the gas will follow the ideal gas law

$$pV = mRT \tag{12}$$

where
R is the gas constant [J/kgK].

Assuming that mass, specific heat, C_v and the gas constant, R, are constant, and letting $R/C_v = (\gamma - 1)$, the following correlations can be derived from equations 10, 11 and 12:

$$\frac{\mathrm{d}p}{\mathrm{d}t} = \frac{1}{V}\left((\gamma - 1)\frac{\mathrm{d}Q}{\mathrm{d}t} - p\gamma\frac{\mathrm{d}V}{\mathrm{d}t}\right) \tag{13}$$

$$\frac{\mathrm{d}T}{\mathrm{d}t} = \frac{1}{mC_v}\left(\frac{\mathrm{d}Q}{\mathrm{d}t} - p\frac{\mathrm{d}V}{\mathrm{d}t}\right) \tag{14}$$

where
γ is ratio of specific heats [1].

3.2.7. *Friction*

Friction losses in the free-piston engine are expected to be lower than for the conventional internal combustion engine due to the elimination of the crank mechanism. This removes a source of frictional losses, but also piston-cylinder friction is reduced since no side forces act on the piston in the free-piston engine.

A breakdown of engine friction mechanisms in four-stroke spark ignition and diesel engines is presented by Heywood [14]. This data is used, with the following adjustments:

- Pumping work is left out as the cycle is two stroke.
- Compression and gas load are left out as their effects are accounted for elsewhere in the simulation model.
- Crankshaft friction is left out as it is non-existent in the free-piston engine.

This leaves frictional losses from the piston itself and the piston rings, along with valve gear and auxiliaries, giving a friction mean effective pressure (fmep) of around 120 kPa. Using this, a friction force is found. The friction force is assumed to be constant over the full stroke, an acceptable simplification since the magnitude of the friction forces will be significantly lower than the other forces that act on the piston and will not have a large effect on the engine dynamics.

3.3. *Linear alternator*

The design of the electric machine will have a high influence on the performance of the free-piston engine generator since the alternator translator forms a part of the moving mass. For the engine to be feasible and for it to be a realistic alternative to conventional technology, it is essential that a translator with sufficiently low weight can be found. Increasing moving mass reduces the bouncing frequency of the system and thereby the power to weight ratio.

Some reports have been published investigating the design of linear electric machines for use with free-piston engines. Arshad et al. [9] indicate, having undertaken an extensive study resulting in a number of publications, that an electric generator with 50 kW power output with efficiency of above 90% and a moving mass of less than 10 kg can be realised with the use of a transverse flux, permanent magnets machine. Similar longitudinal flux machines are reported to have moving masses in the order 10–15 kg.

In this study, a simple design procedure was developed to estimate the required magnet mass for a given free-piston engine generator, using basic magnetic circuit calculations. The predictions showed reasonable agreement with the data reported by Arshad et al., and, based on this, an estimation of magnet mass for the free-piston engine investigated in this study has been made. As will be shown below, the mass requirements for the electric machine are less stringent in a turbocharged diesel-powered free-piston engine compared to SI engines, due to the higher operating pressures giving increased operating frequencies.

The load force of the electric machine is assumed to be proportional to translator speed, however the simulation model allows the user to define any load force profile to be investigated.

3.4. *Bounce chamber*

The bounce chamber is modelled as a reversible isentropic compression and expansion of the air trapped within the bounce chamber cylinder. The force from the bounce chamber is proportional to the bounce chamber pressure, which is found using a user-defined start-of-compression pressure and the instantaneous bounce chamber compression ratio. The amount of air in the bounce chamber is allowed to vary by varying the given start-of-compression pressure, and using this the operating characteristics of the bounce chamber can be varied.

The frictional losses in the bounce chamber are likely to be lower than for the combustion cylinder since the bounce chamber is working with lower pressures and the sealing requirements are less stringent. A value of half the friction losses of the combustion cylinder was used for the bounce chamber.

4. Simulation results and discussion

A simulation program was written using the numerical computation software Matlab to investigate the influence of design parameters and engine input variables. Equation 1 is solved using standard forward Euler numerical integration to give piston speed and position, with the diesel engine submodels and the correlations for alternator load force and bounce chamber pressure being solved at each simulation step. The program allows investigation into both steady state outputs and transient response of engine performance.

The code takes the following input variables:

- Cylinder bore
- Moving mass
- Nominal stroke and compression ratio
- Ambient conditions
- Exhaust back pressure and turbocharger isentropic efficiencies
- Number of exhaust valves, dimensions and valve timing
- Scavenging ports height and width
- Fuel calorific value, mass of fuel per injection and injection timing
- Linear alternator force function

It predicts a number of variables, including:

- Speed
- Stroke
- Generator power output
- Cylinder temperature and pressure history
- Exhaust gas temperature and boost air pressure and temperature
- Fuel-air equivalence ratio
- Scavenging efficiency
- Predicted ignition delay, fuel burn rate and heat transfer rate
- Engine efficiency

4.1. *Model validation*

The output of the simulation model was compared to data from a Volvo TAD1240 six-cylinder, turbocharged diesel engine located at the Jones Engine Laboratory, University of Newcastle upon Tyne. The comparison was done with the aim of verifying that the model produces realistic results, and that it is able to predict real trends for varying engine operating conditions.

4.2. *Engine specifications*

A number of simulations were run with varying input variables, and the results are presented in the following sections. The engine specifications were chosen to be similar to those of the engine against which the simulation model was validated. Table 1 shows the main engine specifications.

Design stroke	0.150 m
Bore	0.131 m
Scavenging ports height	0.022 m
Nominal compression ratio	15:1
Piston mass	22 kg
Bounce chamber bore	0.150 m
Bounce chamber compression ratio	15:1
Exhaust back pressure	1.5×10^5 Pa

Table 1
Free-piston engine specifications.

Both the exhaust valve timing and fuel injection timing are variable in order to optimise engine performance, and controllers within the simulation model adjust these during operation. The exhaust valve opening timing is adjusted so that the in-cylinder pressure at the time when the scavenging ports are uncovered is equal to the boost pressure.

This maximises the work done on the piston and prevents backflow of exhaust gases into the intake manifold. The exhaust valves close at the same time as the piston covers the scavenging ports in the compression stroke. Fuel injection timing is set to its optimum value for high efficiency in all the simulated cases.

4.3. *Basic performance*

The basic performance parameters of the free-piston engine are summarised in table 2. The engine is operating at a fuel-air equivalence ratio of 0.60. The efficiency values presented below refer to the 'shaft' fuel efficiency of the combustion engine, and losses in the electric machine are therefore ignored.

Speed	30 Hz
Output power	44.4 kW
Mean piston speed	9 m/s
Scavenging efficiency	0.90
Boost pressure	1.68×10^5 Pa
Engine efficiency	0.42

Table 2
Simulated engine basic performance.

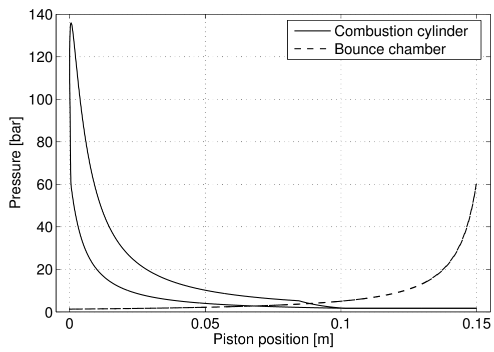


Fig. 2. Simulated combustion cylinder and bounce chamber pressure for one engine cycle. (TDC at piston position 0)

Figure 2 shows the simulated in-cylinder pressure of the combustion cylinder and bounce chamber for one engine cycle. The predicted combustion is seen to be close to a constant volume process. The bounce chamber is modelled as a reversible isentropic compression, as can also be seen from the figure.

Figure 3 shows the predicted piston dynamics of the free-piston engine. Figure 3a shows the simulated piston motion for one engine cycle, compared to a conventional engine (with crankshaft radius to connecting rod length ratio of 0.3) running at the same speed. Significant differences can be seen, the most important being that the free-piston engine spends shorter time around TDC, where the gas pressure and temperature are at their highest. This is emphasised in figure 3b, which shows the piston motion in the area close to TDC, normalised around the TDC position. It is seen that the expansion just after TDC is significantly faster for the free-piston engine.

It can also be seen from the piston motion trajectory that the free-piston engine motion is asymmetrical around TDC, and the engine spends more time in the compression than in the expansion phase of the cycle.

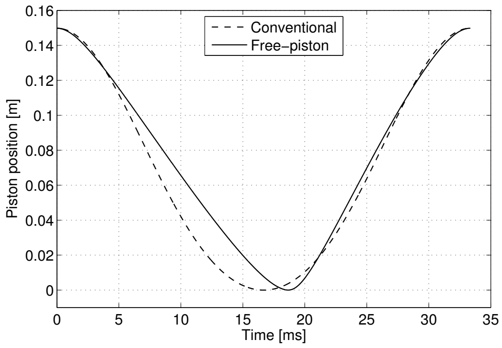
Figure 3c shows simulated piston speed in the free-piston engine and the piston speed profile of a conventional engine. It is seen that the peak piston speed is lower for the free-piston engine, and that the speed profile is significantly different. Compared to the conventional engine, the free-piston engine has a speed profile resembling a sawtooth wave, with rapid speed changes and periods of close to constant speed.

Figure 3d shows piston acceleration in the free-piston and conventional engine. As above, significant differences can be seen, particularly just after ignition where the unrestricted motion of the piston in the free-piston engine gives a very high acceleration when the in-cylinder pressure is high. Predicted peak piston acceleration in the free-piston engine is around 60 % higher than in the conventional engine.

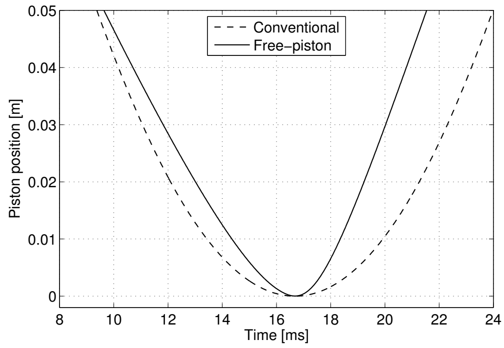
A curiosity of the free-piston engine is that, since the acceleration is directly proportional to in-cylinder pressure, events in the combustion chamber can be found on the acceleration graph. Both the fuel injection and the exhaust valve opening can be seen on the graph in figure 3d, at around $t = 18$ ms and $t = 27$ ms respectively.

4.4. *Engine geometry and design*

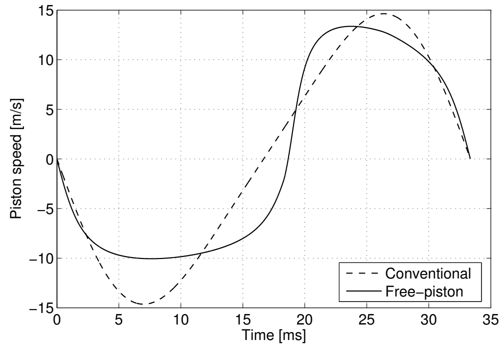
In an engine design process, knowledge of the influence of different engine geometric variables is of high importance. Although the influence of the main design parameters can be predicted based on experience from conventional engines, their effects on overall engine performance may be different in the free-piston engine due to the particular operating characteristics of this type of engine. This section investigates the influence of the main engine design



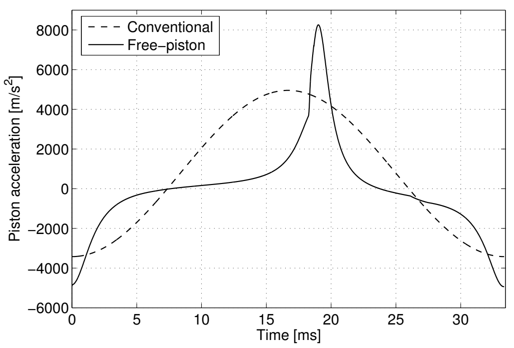
(a) Simulated piston motion for the free-piston engine compared to a conventional engine running at the same speed. (Piston position shown as distance below TDC.)



(b) Simulated piston motion close to TDC for a free-piston engine and a conventional engine, normalised around TDC position.



(c) Simulated piston speed profile for the free-piston engine compared to a conventional engine running at the same speed.



(d) Simulated piston acceleration profile of the free-piston engine compared to a conventional engine running at the same speed.

Fig. 3. Predicted piston dynamics of the free-piston engine.

parameters in the free-piston engine.

All the design– and operational parameters in the free-piston engine are highly interconnected, and the system becomes more complicated in this sense than the conventional engine. For example, an increase in moving mass will not only lead to lower engine speed, it will also have effects on the stroke length which influences both the scavenging process and the compression ratio. Changing operational variables, such as fuel mass flow or boost pressure, will have the same effects. Due to this, a parametric study of the free-piston engine design and operational variables is not straight-forward. A number of prerequisites must be decided upon before such investigations can be undertaken.

In the simulations below, TDC and BDC positions are controlled to their nominal values by changing the bounce chamber pressure and fuel flow rate respectively. To ensure that the comparison is done on the same relative engine load, the electric load force is adjusted to achieve a fuel-air equivalence ratio of 0.60 for all cases. The height of the scavenging ports is further adjusted to achieve a scavenging efficiency of 0.90 to ensure that realistic engine designs are compared. Finally, the exhaust valve opening timing and fuel injection timing are optimised as described above.

4.4.1. *Moving mass*

The engine resembles a spring-mass system, and the moving mass is therefore of high importance to engine performance. A low moving mass is expected to give high engine speeds and vice versa. A reported concern with the free-piston engine generator is finding an electric machine with sufficiently low weight, and it is clear that if this cannot be found the engine will not achieve an acceptable power to weight ratio.

It is expected that a high moving mass leads to lower engine speed, which again leads to more time available for scavenging, allowing lower scavenging ports height, and closer to constant volume combustion, both increasing efficiency. It will, however, at the same time lead to higher peak temperatures and pressures, which increase heat transfer losses. An optimal engine speed for high efficiency may therefore be expected.

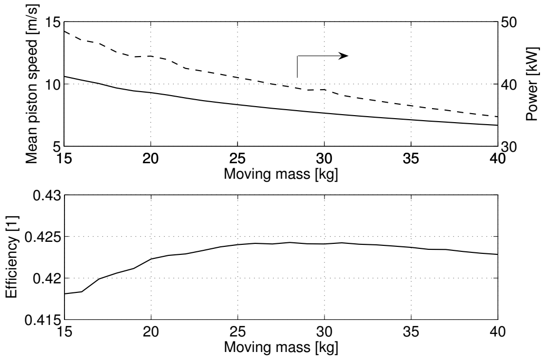


Fig. 4. Effects of changes in moving mass on engine performance.

Figure 4 shows the predicted effects of varying mass on engine performance. It is seen that the spring-mass analogy is valid, as increasing mass leads to a decrease in engine speed and power output.

The results for engine efficiency show that the predicted optimum moving mass for efficiency is relatively high. This indicates that the requirements of a low-mass electric machine are less stringent in a turbocharged diesel-powered free-piston engine, and that acceptable mean piston speeds and high efficiency can be achieved even for a high moving mass.

4.4.2. *Compression ratio*

The compression ratio is changeable during operation in the free-piston engine, but the nominal value must be decided on a design stage. Although a higher compression ratio gives higher theoretical cycle efficiency, it increases in-cylinder temperatures and pressures, increasing heat transfer losses and mechanical stress, and these effectively determine the maximum possible compression ratio.

Figure 5 shows the effects of variations in the nominal compression ratio of the free-piston engine. The simulations predict that an optimal compression ratio can be found, but only minor effects on engine efficiency are found for compression ratios between 15 and 25.

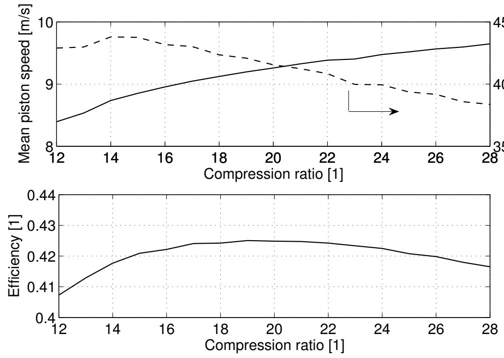


Fig. 5. Effect of varying nominal compression ratio on engine performance.

An effect that is not obvious is the decrease in engine power output together with increased engine speed. The latter is a result of the combustion cylinder "gas spring" being stiffer, which increases the bouncing frequency in the mass-spring system. The reduced power output is due to the fact that an increased compression ratio leads to an increased expansion ratio, which reduces the internal energy in the exhaust gases. While this may increase engine efficiency, it reduces the boost pressure and therefore also engine power output since the simulations are run with constant fuel-air equivalence ratio. In the simulations, the exhaust gas temperature drops by around 8 per cent between compression ratios of 15 to 25, giving a reduction in boost pressure by around 4 per cent in the same interval.

4.4.3. *Exhaust back pressure*

The exhaust back pressure of an engine is determined by the size of the turbocharger. The effects of varying exhaust back pressure on engine performance are investigated below, with the assumption that the turbocharger efficiencies are constant for all conditions.

An increase in exhaust back pressure is assumed to give higher engine speed, since the pressure forces increase. In addition, significantly higher power output is expected with higher boost pressure, since the engine is running at a constant fuel-air equivalence ratio.

Figure 6 shows the effects of varying exhaust back pressure on engine performance with constant equivalence ratio. It is seen that, along with the predicted increase in engine power output and speed, the efficiency improves with increasing exhaust back pressure. This is due to the fact that the contribution from frictional losses decreases at higher engine

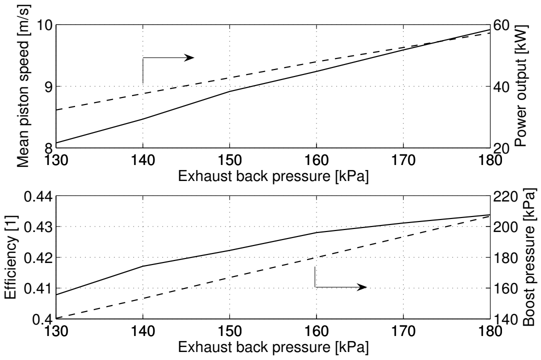


Fig. 6. Effects of changing exhaust back pressure on engine operation.

power outputs.

Engine performance with higher exhaust back pressures are not investigated as the upper range of the investigated values produced very high peak cylinder pressures (approaching 20 MPa). Operation on higher exhaust back pressures with constant fuel-air equivalence ratio is therefore not realistic. Further investigations to compare the performance of a highly turbocharged engine running at a low equivalence ratio compared to a moderately turbocharged engine running at high equivalence ratios may be worthwhile.

4.4.4. *Stroke to bore ratio*

The stroke to bore ratio is one of the core design variables in internal combustion engines, relating combustion chamber surface area to its volume and piston area to stroke length. Conventional engines with high stroke to bore ratios, such as those found in marine and stationary powerplants, are characterised by high efficiency but poor power density. Low stroke to bore ratios, commonly found in small-scale engines e.g. in automotive applications, allow higher engine speeds which increase power to weight ratio, but at the cost of higher fuel consumption.

Figure 7 shows the effects of varying stroke to bore ratio on engine performance with engine swept volume kept constant. Higher efficiencies are as expected obtained with an increasing stroke to bore ratio. Despite an increase in mean piston speed the engine output power is slightly reduced with increasing stroke to bore ratio due to a reduction in engine speed.

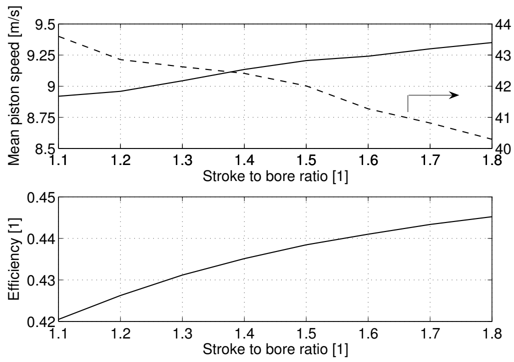


Fig. 7. Effects of varying free-piston engine stroke to bore ratio with constant swept volume and constant equivalence ratio.

4.5. *Effect of variable stroke*

The free-piston engine has the potentially valuable feature of variable stroke. This means that the compression ratio of the engine can be altered during operation, to optimise engine performance for any operating condition. This has been mentioned by a number of authors, and suggested advantages include increased part load efficiency and multi-fuel operation possibilities.

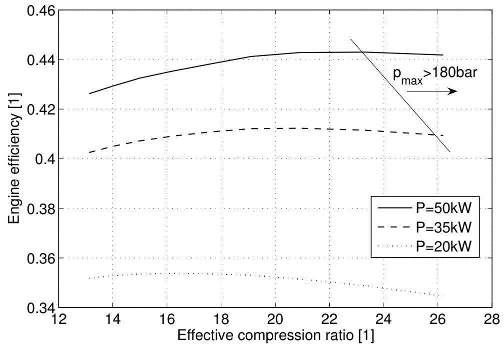


Fig. 8. Effect of changing effective compression ratio during engine operation at constant load.

Figure 8 shows the effect of changing the TDC setpoint for the engine (the figure shows the corresponding effective compression ratio) on engine efficiency at different loads. The simulations show that the optimum compression ratio varies only slightly with load and no advantage of increasing the compression ratio on low loads was found. For high loads a small increase in efficiency can be seen with increasing compression ratio, but in these cases the peak cylinder pressure will limit the maximum compression ratio. The compression ratios where peak cylinder pressure exceeded 18 MPa are shown.

Similar simulations were performed for BDC position, but varying the BDC setpoint produced no noticeable effect on engine performance.

4.6. *Effect of fuel burn rate*

As discussed above, some uncertainty exists regarding the accuracy of the combustion modelling as some authors have reported an increased fuel burn rate in free-piston engines. Furthermore, one of the claimed advantages of the free-piston engine is its multi-fuel possibilities, and it is therefore worthwhile to investigate how fuel burn rate influences engine performance.

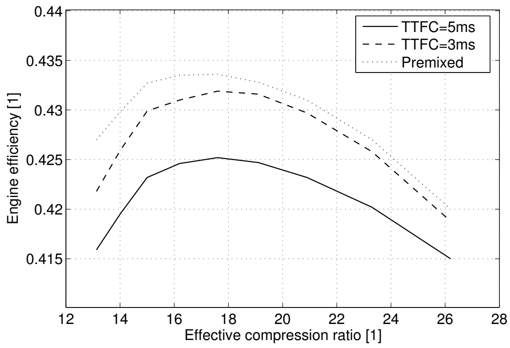


Fig. 9. Effect of total time for combustion (TTFC) on engine performance predictions.

Figure 9 shows the effect of changing fuel burn rate on engine performance. In addition to changing the total time allowed for combustion (but not the fuel burn pattern) in equation 4, the effect of assuming that all the fuel burns in a single pre-mixed phase is shown. Such an approach was adopted by Tikkanen et al. [16] and was supported by the experimental results of the same research group and those of Somhorst and Achten [12].

Some influence on engine performance is seen, and a more rapid combustion will, as expected, improve engine performance. It is further seen that the optimum compression ratio does not change significantly with varying fuel burn rate, indicating that the changeable the compression ratio is less beneficial than expected. It should, however, be noted that changing the compression ratio will lead to other effects that the current engine model is unable to identify, such as changes in squish effects, which may influence combustion.

The simulations showed that for high load (fuel-air equivalence ratio of 0.70), most of the fuel has been burned within around 2 ms with total time for combustion in equation 4 is set to 5 ms. With total time for combustion set to 3 ms most of the fuel burns in around 1.5 ms. Somhorst and Achten [12] report fuel burn rates even higher than this, stating that 90 % of the combustion in their free-piston engine finishes within 1 ms.

4.7. *Comparison to conventional engines*

To investigate the effects of the particular operating characteristics of the free-piston engine compared to conventional engines, a crankshaft engine simulation subroutine was implemented in the simulation model. The subroutine uses the simulation results for a given free-piston engine configuration and runs a subsequent simulation for an engine with the same configuration and operational variables (bore, stroke, boost pressure, injected fuel mass etc.), but with a piston motion equal to that of a conventional engine. (I.e. a crankshaft-driven motion that is not influenced by the in-cylinder pressure and is described by a function of sine and cosine factors.) Using such an approach, some insight into the differences in performance between the free-piston and conventional engines can be gained.

The optimum point of fuel injection differs between the two engines due to the differences in piston motion profile, and in order to obtain a realistic comparison each engine is run with the injection timing giving best fuel efficiency. These optimum values were identified through simulations and it was found that the free-piston engine requires a slightly advanced injection timing compared to the conventional one.

Figure 10 shows the simulated efficiency of the free-piston engine compared to an identical conventional engine for varying engine load. Since no assumptions have been made for the mechanical efficiency of conventional engines, the comparison is done on the basis of indicated (in-cylinder) values. A slight fuel efficiency advantage is predicted for the free-piston engine over the full load range, and it will be shown below that this is due to reduced heat transfer losses. In addition to this come the potentially lower frictional losses in the free-piston engine.

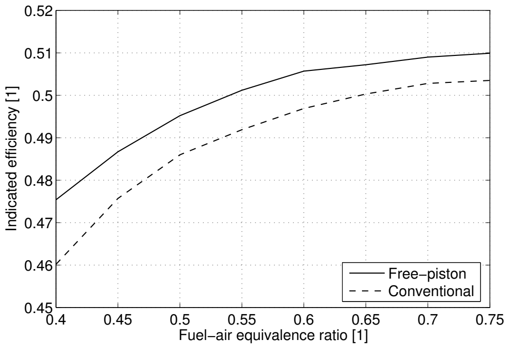


Fig. 10. Efficiency of the free-piston engine compared to a conventional engine for varying loads.

4.7.1. *In-cylinder gas temperature*

The fast power stroke expansion of the free-piston engine reduces the time which the gases spend in the high-temperature parts of the cycle, and this possibly reduces in-cylinder heat transfer losses and temperature-dependent emissions formation. Using the simulation model, the predicted heat transfer rate from the cylinder and in-cylinder gas temperature was compared for the free-piston engine and for a similar conventional engine.

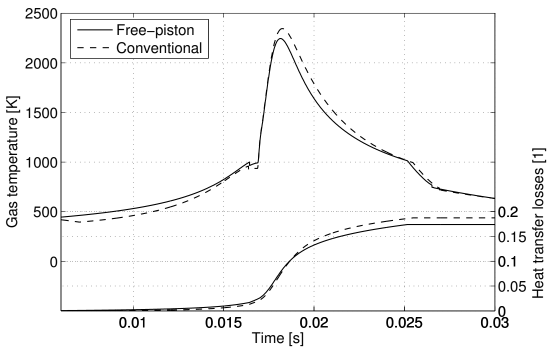


Fig. 11. In-cylinder gas temperature and heat transfer losses in free-piston and conventional engines.

Figure 11 shows the simulated in-cylinder gas temperature and the total heat lost to in-cylinder heat transfer for one cycle, the latter shown as a fraction of the injected fuel heat. The graphs are synchronised around the start of combustion. It can be seen that both peak temperature and temperature levels during expansion is lower in the free-piston engine, and that this results in lower heat transfer losses. The simulations predict that in the free-piston engine, 17.4 per cent of the fuel heat is lost to in-cylinder heat transfer. For the conventional engine this value is 18.7 per cent.

The simulations further show that the fast expansion of the free-piston engine makes it spend less time in the parts of the cycle where the in-cylinder gas temperatures are high. The amount of time per cycle that the gases in the free-piston engine hold a temperature of above 1500 K is 89 per cent of that in the conventional engine. (This time period is equal to 10.1 % of the cycle time in the free-piston engine and 11.3 % in the conventional engine.) For temperatures of above 2000 K, this value is 83 per cent (with such temperatures levels occurring for 4.5 % and 5.4 % of the cycle time respectively). In addition to reducing heat transfer losses, this particular feature of the free-piston engine may lead to a reduction in the formation of temperature-dependent emissions, most importantly nitric oxides.

4.7.2. *Limitations of the simulation model*

It should be noted that the existing simulation model, being a single-zone model, is unable to identify some factors that will influence engine performance and the predictions shown above. Differences in in-cylinder gas motion, in particular squish effects, due to the piston motion profile and varying compression ratio, will not have been identified, but such effects will undoubtedly have influence on both the combustion process and formation of emissions. Further research is therefore necessary to be able to identify the details of these effects, and these limitations should be kept in mind while reading the results presented above.

5. Conclusions

The design of a free-piston diesel engine generator was presented. A detailed simulation model was derived and extensive simulation results were presented, giving insight into the basic performance of the engine and effects of engine design parameters and operational variables on engine performance.

The performance parameters of the free-piston engine are highly interconnected, and changing one design parameter will influence many operational variables. The mass requirements for the electric machine, discussed by other authors, were found to be less stringent in the current design, due to higher operating pressures.

It was found that the free-piston engine has potential advantages over conventional technology in terms of fuel efficiency as a result of mechanical simplicity and faster power stroke expansion.

Lower temperature levels may reduce the formation of emissions. The flexibility of the variable stroke was not found to give any large advantages for normal engine operation, a result that was somewhat unexpected.

Much development work remains before the free-piston engine can become a realistic alternative to conventional technology. Most importantly, further research into engine control and free-piston engine combustion must be undertaken. Work on these topics is currently underway at the School of Marine Science and Technology at University of Newcastle upon Tyne.

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