Coupled dynamic–multidimensional modelling of free-piston engine combustion. ⋆

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Abstract

Free-piston engines are under investigation by a number of research groups worldwide, as an alternative to conventional technology in applications such as electric and hydraulic power generation. The piston dynamics of the free-piston engine differ significantly from those of conventional engines, and this may influence in-cylinder gas motion, combustion and emissions formation. Due to the complex interaction between mechanics and thermodynamics, the modelling of free-piston engines is not straight-forward. This paper presents a novel approach to the modelling of free-piston engines through the introduction of solution-dependent mesh motion in an engine CFD code. The particular features of free-piston engines are discussed, and the model for engine dynamics implemented in the CFD code is described. Finally, the coupled solver is demonstrated through the modelling of a spark ignited free-piston engine generator.

Key words: free-piston engine, dynamic modelling, CFD

1. Introduction

The increasing focus on the environmental impacts of using fossil fuels as an energy source has led to increased research efforts into innovative engine technology. A number of research projects are ongoing, both within academia and industry, with the aim of reducing engine emissions and improving fuel efficiency through unconventional engine design. One such technology under investigation is the free-piston engine, which is an old concept that has attracted attention recently mainly due to its simplicity and operational flexibility.

Multidimensional computational fluid dynamics (CFD) codes are widely used in the design and development of internal combustion engines due to their ability to investigate variables which are difficult or costly to measure in experimental tests. CFD codes can be very useful for investigating unconventional engine design, since the effects of conceptual changes can be investigated on a fundamental level. With the ever-increasing availability of computing power and an increase in the amount of scientific software released under open licences, such advanced computational tools are becoming increasingly available to researchers and engineers.

This paper presents a novel approach to the modelling of free-piston engine operating characteristics and performance through the implementation of solution-dependent mesh motion, along with a model for the piston dynamics, in an engine CFD code. The aim of the work is to develop a model that allows detailed investigations into the complex in-

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teraction between the in-cylinder processes and the piston dynamics in the free-piston engine, which is the main difference between this type of engine and conventional, crankshaft engines.

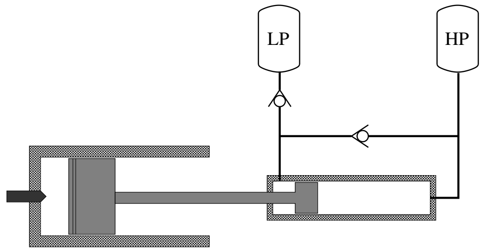
2. Free-piston engines

Free-piston engines are linear, 'crankless' engines, in which the piston motion is not restricted by a crankshaft but determined by the interaction of a number of forces acting on the mover. First proposed around 1930, the free-piston engine is an old concept. Such engines were in commercial use in the period 1930–1960 as air compressors in naval applications, and as gas generators feeding exhaust gas to a power turbine in large-scale power plants. These early free-piston engines were of the opposed piston type, with mechanical linkages connecting the two pistons, and their main advantages were the compactness of the units and the perfect dynamic balance. Attempts were made to use free-piston gas generators for vehicle propulsion, but without success. As conventional engine and gas turbine technology matured, the free-piston engine concept was abandoned in the early 1960s. A comprehensive review of free-piston engine history was presented by Mikalsen and Roskilly [1].

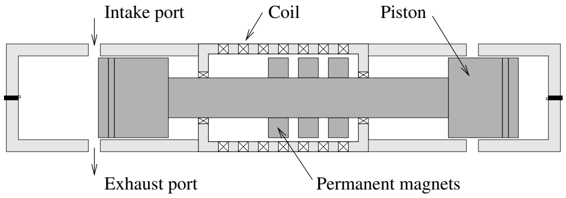
In recent years, the free-piston engine concept has again attracted attention among research engineers seeking to improve combustion engine efficiency and reduce emissions. The main attractions with the modern free-piston engines are the simplicity of the units, giving a compact engine with low frictional losses, and the operational flexibility through the variable compression ratio, potentially offering extensive multi-fuel and operation optimisation possibilities. Since the piston motion is not controlled by a crankshaft, the free-piston engine has low ignition timing control requirements, making it suitable for homogeneous charge compression ignition (HCCI) operation. Lean burn combustion at high compression ratios is possible with only minor penalties in engine mechanical efficiency, due to the low frictional losses in the free-piston engine.

2.1. *Modern free-piston engine developments*

Modern free-piston engine developments are mainly of the single piston and dual piston types with power extracted through a linear load device, usually in the form of a hydraulic cylinder or a linear electric machine. Examples of modern free-piston engine configurations are shown in Figure 1. Single piston hydraulic free-piston engines, similar to that illustrated in Figure 1a, were presented by Achten et al. [2] and Brunner et al. [4]. The hydraulic cylinder acts as both load and rebound device using an advanced control system in the hydraulic circuit, and Achten et al. demonstrated how excellent engine performance can be obtained with this method. By varying the compression energy, the engine compression ratio can be controlled to optimise engine performance.



(a) Single piston hydraulic free-piston engine [2].



(b) Dual piston free-piston engine generator [3].

Fig. 1. Illustration of modern free-piston engine configurations.

Dual piston free-piston engines, such as that illustrated in Figure 1b, have been proposed by a number of authors, however only a limited number of reports have described experimental data from developed engines. Such engines potentially offer excellent power to weight ratios, but engine control has been reported to be a challenge. Tikkanen et al. [5] described the design of a 20 kW diesel powered hydraulic free-piston engine and test results from an engine prototype demonstrated the feasibility of the concept. High cycle-to-cycle variations were reported and the developers stated that further research into engine control issues are required.

Famouri et al. [3] and Clark et al. [6] described the development and testing of a spark ignited dual piston free-piston engine generator. The engine was reported to achieve 316 W power output operating at 23.1 Hz, with 36.5 mm bore and 50 mm maximum stroke. The design of a diesel powered dual piston free-piston engine generator was presented by Kleemann et al. [7].

Mikalsen and Roskilly [8] proposed a single piston free-piston engine generator with a gas-filled bounce chamber. The main motivation for choosing the single piston configuration was to improve engine controllability. Through replacing one firing cylinder in a dual piston engine with a variable-pressure bounce chamber, an additional control variable is available to obtain the accurate control required for operation optimisation of the engine.

2.2. *Free-piston engine operating characteristics*

The main difference between free-piston and conventional engines is the piston dynamics. In conventional engines the piston motion profile is determined by the crank system, and its high inertia makes the rotational speed of the engine constant in the time frame of one cycle. In the free-piston engine, the piston motion at any point in the cycle is determined by the instantaneous force balance on the mover. This allows the progress of the combustion process to influence the speed of expansion, meaning that the piston motion profile may vary for different operating conditions. As the load and rebound devices are directly coupled to the mover, their dynamic characteristics will also have a high influence on the piston dynamics.

The particular piston motion profile of the free-piston engine has been reported by a number of authors, including Achten et al. [2], Tikkanen et al. [5] and Mikalsen and Roskilly [8]. Figure 2 shows the simulated piston motion of a free-piston engine compared to that of a conventional engine. It is seen that the free-piston engine has a significantly higher piston acceleration shortly after TDC when the in-cylinder gas pressure is high, and that the time spent in the high-temperature parts of the cycle (around TDC) is shorter than that of the conventional engine. For the single piston free-piston engine, it can also be noted that the piston motion is asymmetric, leading to the engine spending slightly more time in the compression stroke than in the expansion stroke.

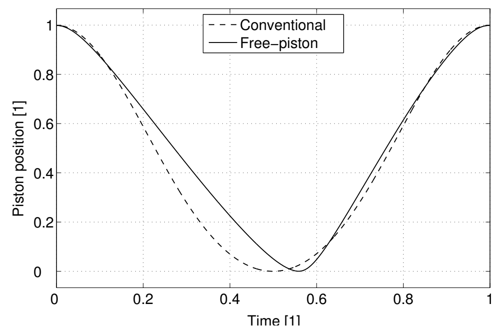


Fig. 2. Simulated piston motion of a single piston free-piston engine compared to that of a conventional engine [8]. (TDC at position 0.)

2.3. *Free-piston engine combustion*

Differences in the combustion process between free-piston and conventional engines have been reported by some authors. Somhorst and Achten [9] reported high heat release rates in a single piston hydraulic free-piston engine, with pressure gradients reaching values of two to five times those of a comparable conventional engine. Tikkanen et al. [5] reported the same behaviour in a dual piston free-piston engine and both groups stated that combustion takes place predominantly in a single, pre-mixed phase. It was suggested that this is due to the high piston velocities around TDC, increasing in-cylinder gas motion and turbulence levels [2,9].

2.4. *Modelling free-piston engines*

Free-piston engines are commonly modelled using zero-dimensional, single zone models developed for conventional engines. While such models can be useful for investigating basic engine performance and piston dynamics, they are unable to identify details of the engine operation such as in-cylinder gas motion and emissions formation. Furthermore, most of these models have been developed for and validated against conventional engines, and it is questionable whether they are suitable for modelling free-piston engines without modification. Examples of zero-dimensional modelling of free-piston engines can be found in the reports of Mikalsen and Roskilly [8], Atkinson et al. [10] and Fredriksson and Denbratt [11].

To fully investigate details of free-piston engine performance and potential advantages over conventional technology, a multidimensional simulation model is required. With such a model, the effects of the differences in piston motion profile will be accounted for and their influence on the in-cylinder processes can be investigated. Only very few studies have investigated free-piston combustion using multidimensional simulation models. Examples of such work include the reports of Fredriksson [12] and

Mikalsen and Roskilly [14]. These studies employed pre-determined piston motion profiles derived from simplified dynamic models, and the effects of varying operating variables (such as fuel injection timing) on the piston motion profile could therefore not be modelled. Kleemann et al. [7] and Fredriksson et al. [13] presented more advanced approaches, using detailed simulation models to obtain performance data such as scavenging performance and heat release rates and combining these with a dynamic piston motion model.

3. The coupled dynamic–multidimensional free-piston engine simulation model

Computational fluid dynamics simulation software is widely used in engine development and provides powerful tools for investigating details of engine in-cylinder processes. Engine CFD codes may be useful to identify the detailed influence of the particular piston motion profile in free-piston engines, however the mesh motion algorithms implemented in such software are usually time-dependent only, determining the piston motion from an analytic expression. In free-piston engines, the motion of the piston is influenced by the in-cylinder processes and solution-dependent mesh motion is therefore required.

Existing work on the CFD modelling of free-piston engines uses piston motion profiles obtained from a dynamic engine model, which are expressed mathematically as a function of time and implemented in the CFD code. While it is possible to investigate free-piston engine performance at a single operating point using such an approach, solution-dependent mesh motion must be introduced if the effects of major changes in engine operational variables are to be investigated. For example will changes in the fuel injection timing, the fuel properties or the compression energy all influence the piston motion.

3.1. *The CFD software*

The combustion simulations are based on the open source computational fluid dynamics toolkit OpenFOAM [15]. Written in the C++ programming language and released under the GNU General Public Licence, OpenFOAM gives the user considerable freedom to modify the code to suit specific needs. Implemented in the toolkit are the Weller model for premixed combustion [16], the Chalmers PaSR model for diesel combustion [17] and a CHEMKIN-compatible chemistry solver. Jasak et al. [18] described the toolkit and presented examples of the simulation of pre-mixed, spark ignition combustion and diesel spray and combustion.

3.2. *Modelling engine dynamics*

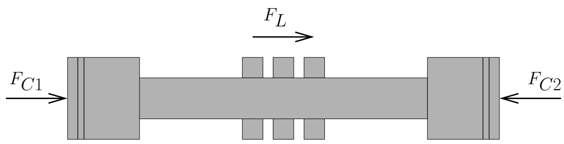


Fig. 3. Free body diagram of free-piston engine mover.

The motion of the mover in the free-piston engine is governed by the interaction of the forces acting on it, and can be described with Newton's 2nd law. Figure 3 shows the free body diagram of the mover in the free-piston engine illustrated in Figure 1b. The main forces acting on the mover are the cylinder pressure force, the force from the load device, the force from the rebound device (which is the second cylinder in dual piston engines) and frictional forces. Frictional forces are subordinate to the other three and do not have a large influence on engine dynamics, and they are therefore ignored here. The motion of the mover can be found with

$$F_{C1} + F_L - F_{C2} = m\frac{\mathrm{d}^2x}{\mathrm{d}t^2} \quad (1)$$

where m is the mass of the mover and x is its position. The pressure force from the combustion chamber is predicted by the CFD code, and the forces that need to be represented in the model are therefore the load force and the rebound force.

Rebound force

For single piston hydraulic free-piston engines, such as the one illustrated in Figure 1a, the rebound force will be provided by a hydraulic cylinder. With a constant hydraulic supply pressure, pressure losses across valves are commonly ignored and the force from the cylinder is assumed to be constant over the full stroke. For engines with a bounce chamber as rebound device, or dual piston engines where the opposite cylinder is driving the compression, the rebound force will depend on the bounce chamber gas pressure. In such engines the rebound force will be high at BDC when the bounce chamber pressure is high, and will decay as the bounce chamber pressure is reduced during the compression stroke. Figure 4 illustrates the force profiles of these two types of rebound device. The shaded areas represent the compression energy.

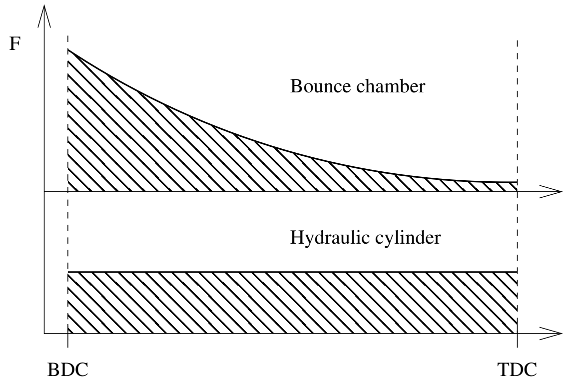


Fig. 4. Illustration of the force profiles of free-piston engine rebound devices.

Both types of rebound devices have been implemented in the model. In the case of a hydraulic rebound device, the rebound force is assumed to be constant over the full stroke, giving a rebound force $F_R = k$, where k is a constant. For a gas-filled rebound device, the gas in the bounce chamber is assumed to follow reversible polytropic compression and expansion, such that

$$pV^n = \text{constant}$$

where p is the bounce chamber pressure, V is bounce chamber volume and n is the polytropic coefficient. Hence, the pressure at any point can be found from the bounce chamber volume with

$$p = p_0 \left(\frac{V_0}{V} \right)^n$$

where subscript 0 indicates the initial conditions, here taken as the BDC position for the combustion cylinder.

The instantaneous bounce chamber cylinder volume can be described in terms of the initial volume V_0, the bounce chamber piston area A_p and the distance x which the piston has travelled from the initial condition:

$$V = V_0 + A_p x.$$

The nominal compression ratio R_C of the bounce chamber cylinder can be expressed as

$$R_C = \frac{V_0 + A_p\, S}{V_0}$$

where S is engine nominal stroke length, making $A_p S$ equal to swept volume. Using this, the bounce chamber pressure can be expressed in terms of piston position, initial bounce chamber pressure and bounce chamber compression ratio:

$$p = p_0 \left(\frac{S}{S + x(R_C - 1)} \right)^n .$$

Knowing that the bounce chamber force is proportional to the bounce chamber pressure, $F_R \propto p$, this expression can be modified to express rebound force directly.

Load force

The properties of the load device, such as moving mass and load force profile, will have a high influence on the engine dynamics, and different load devices will have different dynamic properties. A hydraulic cylinder working against a constant discharge pressure can, as above, be modelled as a constant load force acting against the motion of the mover. The load force profile of an electric machine will depend on details of the machine and the electric load coupled to it, however the load force in such devices is commonly assumed to be proportional to the speed of the mover. Figure 5 illustrates the force profiles of these two types of load devices. The shaded areas represent the output load energy.

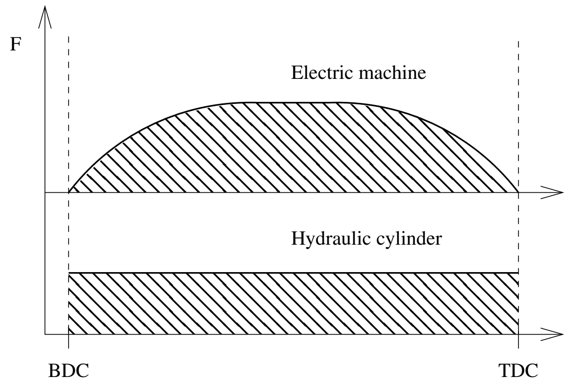


Fig. 5. Illustration of the force profiles of free-piston engine loads.

For a hydraulic load device, the load force, F_L, is assumed to be constant, but with a direction opposite to that of the mover:

$$F_L = \begin{cases} -k & \text{if } \dot{x} > 0 \\ 0 & \text{if } \dot{x} = 0 \\ k & \text{if } \dot{x} < 0 \end{cases}$$

In the case of an electric machine, the load force is assumed to be proportional to the mover velocity:

$$F_L = -k\dot{x}.$$

Implementation into the CFD code

The simulations start at BDC, for which the computational mesh is defined and appropriate initial conditions, such as start-of-compression pressure, temperature and swirl level, are chosen. In the original code, the piston motion is calculated from an mathematical expression defining the position of the piston as a function of time, and the mesh is subsequently updated at each timestep. This expression was replaced by the solving of Equation 1 numerically. The piston acceleration is given by

$$\ddot{x} = (F_R - F_L - pA_p)/m \tag{2}$$

where F_R is the rebound force, F_L is the load force, p is gas pressure in the combustion cylinder and A_p is the combustion cylinder piston area. The expressions for F_R and F_L may be functions of x, depending on the choice of rebound and load devices, as discussed above. This equation is integrated at each timestep to give piston speed and position.

The dynamic model requires the user to specify

- the mass of the mover;
- the type of rebound device, along with the appropriate constants;
- the type of load device, along with the appropriate constants.

In addition to these come the ordinary input data required for the CFD simulation.

4. Simulation results

Due to the free-piston engine effectively being restricted to the two-stroke operating principle such engines are best suited for direct injection operation, and most modern free-piston engine developments are of the direct injection compression ignition type. Previous work by the authors [14] has indicated that the fuel efficiency and exhaust gas emissions advantages of spark ignited free-piston engines over conventional engines are limited. The simulation model will, however, be demonstrated using a pre-mixed, spark ignition engine solver, due to its significantly lower computational costs, allowing simulation over a wider range of operating conditions to fully investigate the capabilities of the coupled solver.

The results presented below are those predicted for a free-piston engine generator similar to that shown in Figure 1b. Simulations were run for a hydraulic free-piston engine and the same trends as those presented below were found, albeit with minor differences in the piston motion profile due to differences in mover mass (the mover of a hydraulic free-piston engine will have lower mass than that of an engine generator) and the dynamic characteristics of the load and rebound devices.

4.1. *Case setup*

A mesh describing a simplified piston and cylinder configuration was generated. The cylinder geometry is assumed to be symmetric around the cylinder axis, allowing a wedge geometry with cyclic boundary conditions to be used. The mesh describes a 30-degree sector and consists of 15,400 cells, equivalent to more than 180,000 cells for the full cylinder. The computational mesh is shown in Figure 6.

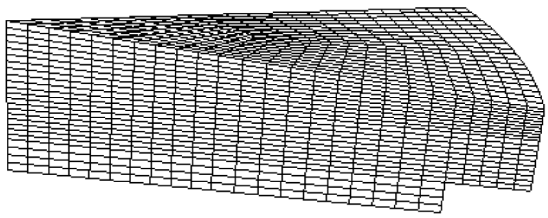
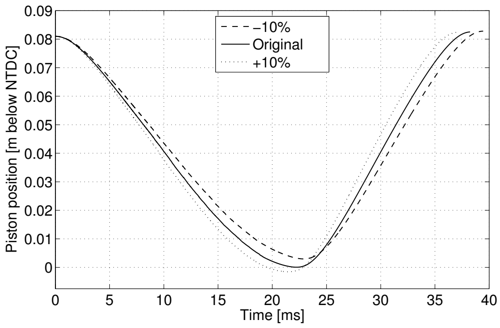


Fig. 6. Computational mesh used in the simulations.

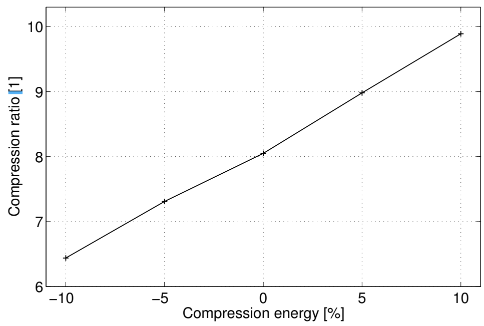
The engine has 81 mm bore, 81 mm stroke and a nominal compression ratio of 8:1. The fuel used was iso-octane at a fuel-air equivalence ratio of 1. Although the maximum compression ratio is limited due to fuel autoignition, this setup allows the coupled solver to be tested and demonstrated. Turbulence was modelled with a standard k–ϵ model with wall functions, and constant wall temperatures were used. Each simulation is run for one full cycle, i.e. between -180 and 180 degrees after TDC. As the engine is two-stroke, the in-cylinder pressure when the intake ports are open was assumed equal to the intake pressure (taken as atmospheric). With these specifications, the dual piston free-piston engine achieves an indicated power output of around 25 kW, with a speed of 26 Hz (equivalent to a mean piston speed of 4.2 m/s).

4.2. *Influence of compression energy*

The compression energy can be varied during operation in most proposed free-piston engine designs.

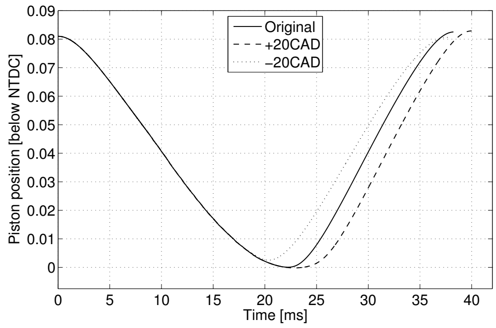


(a) Effect on piston motion. (Piston position shown as distance below nominal TDC (NTDC).)

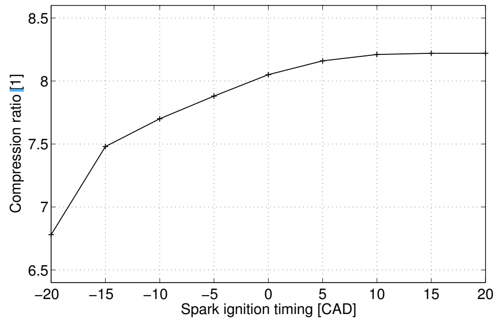


(b) Effect on engine compression ratio.

Fig. 7. Effects of variations in engine compression energy.



(a) Effect on piston motion. (Piston position shown as distance below nominal TDC (NTDC).)



(b) Effect on compression ratio.

Fig. 8. Effects of spark timing.

In hydraulic free-piston engines the hydraulic supply pressure may be variable, and in free-piston engines using a gas-filled bounce chamber a high-pressure gas supply may allow control of the amount of gas in the bounce chamber. In dual piston engines the compression energy will depend on the combustion energy from the opposite cylinder, which can readily be varied.

Figure 7 shows the influence of the compression energy on engine operation with the spark timing and all other variables kept constant. Figure 7a shows the effects of variations in the compression energy on the piston motion profile. It is seen that the compression energy influences both the TDC position (i.e. compression ratio), as one would expect, and also the cycle time. A 10 % change in the compression energy gives approximately a 1 Hz change in operating frequency. Although not a major change, this effect is of high importance if the frequency of the engine is to be controlled, for example to control the electric output or engine vibrations.

Figure 7b shows the influence of the compression energy on the compression ratio of the engine. It is seen that, provided that the compression energy can be accurately controlled, effective compression ratio control can be achieved in free-piston engines.

4.3. *Influence of ignition timing*

One of the main engine operational variables is the spark timing, and this is commonly electronically controlled in modern combustion engines to optimise engine fuel efficiency and reduce emissions formation. In the free-piston engine, it is expected that the spark timing has influence on the piston motion as it controls the release of the energy that slows the piston down around TDC and accelerates it downwards.

Figure 8 shows the influence of the spark timing on engine operation. Figure 8a shows the piston motion profiles for varying spark timing, with the spark timing varied with the time equivalent of 20 crank angle degrees (CAD). A clear influence on the piston motion profile is seen, which also influences the cycle time to some degree. The effect of the spark timing on engine compression ratio is shown in Figure 8b. The effect of advanced spark timing is a rapid decrease in compression ratio, as the high pressure from combustion slows down and returns the piston before it reaches the nominal TDC. Retarding the spark timing allows the piston to reach its nominal TDC, but increases the cycle time since the acceleration around TDC is reduced. As the compression energy is fixed, a significant increase in the compression ratio with retarded spark timing is not seen.

4.4. *Engine operation optimisation*

Having a model which allows simulations to be run for a range of engine operating conditions, operation optimisation possibilities can be investigated. Such analyses are commonly used to determine the optimum operating point for conventional engines, where variables such as fuel-air ratio and ignition timing can be varied for a given engine load in order to optimise engine efficiency and/or emissions formation. In the free-piston engine, the main operating variables available also include the compression energy.

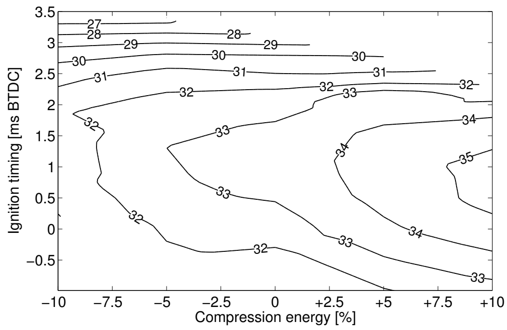


Fig. 9. Engine indicated efficiency (per cent) map for varying compression energy and spark timing.

Figure 9 shows an example of the predicted engine efficiency map for variations in spark timing and compression energy. It can be seen that an optimum ignition timing can be found for a given compression energy, and that increased compression ratio improves the efficiency, as one would expect.

With variable compression ratio, the peak in-cylinder pressure will be of high importance in the free-piston engine. Although the compression ratio in the spark ignited free-piston engine is likely to be limited by the fuel knock limit, in free-piston diesel engines the peak in-cylinder gas pressure is likely to be the limiting factor. Figure 10 shows the peak in-cylinder gas pressure for varying spark timing and compression energy. It can be seen that increased

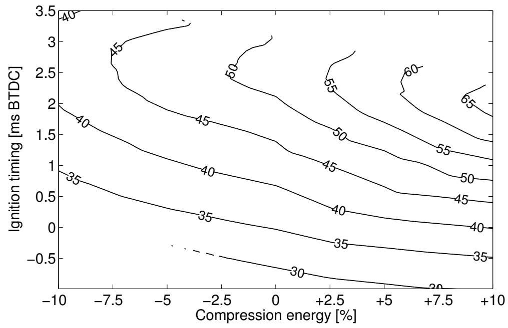


Fig. 10. Peak in-cylinder pressure (bar) for varying compression energy and spark timing.

compression energy leads to higher peak pressures, however for retarded ignition timings this effect is reduced. For highly advanced ignition timings, it can be seen that the peak pressure is reduced due to the early pressure rise reducing the maximum compression ratio.

The plots shown in Figures 9 and 10 are examples of operating maps that can give insight into the performance of the free-piston engine and its operation optimisation possibilities. Similar performance maps can be made for engine emissions formation or other engine operational variables of interest, and they can be compared to similar predictions for conventional engines in order to investigate potential advantages of the free-piston engine.

5. Conclusions

A coupled dynamic-multidimensional simulation model for the simulation of free-piston engines was described. The model is based on the OpenFOAM CFD toolkit, but incorporates solution-dependent mesh motion in order to model the influence of engine operational variables on the piston motion. The use of the coupled solver was demonstrated with the simulation of a spark ignited free-piston engine, and predictions of the influence of some operational variables on engine performance were presented.

The coupled solver provides a powerful tool to investigate the operating characteristics and performance of free-piston engines. The influence of poor ignition timing control in HCCI engines can be studied, along with the potential benefits of lean-burn, high compression ratio operation. In the case of free-piston diesel engines, the effects of changes in TDC position and compression ratio are expected to be higher than in spark ignited engines, due to the higher influence of the in-cylinder gas motion on the combustion process. With changes in TDC position, the clearance distance ('squish band') between the piston and cylinder head changes, influencing squish and reverse squish in the combustion chamber. With the coupled dynamic-multidimensional solver, detailed investigations into the effects of such changes on the combustion process and engine performance can be undertaken.

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