A computational study of free-piston diesel engine combustion. ⋆

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Abstract

This paper investigates the in-cylinder gas motion, combustion process and nitrogen oxide formation in a free-piston diesel engine and compares the results to those of a conventional engine, using a computational fluid dynamics engine model. Enhanced radial gas flow (squish and reverse squish) around top dead centre is found for the free-piston engine compared to a conventional engine, however it is found that this has only minor influence on the combustion process. A higher heat release rate from the pre-mixed combustion phase due to an increased ignition delay was found, along with potential reductions in nitrogen oxides emissions formation for the free-piston engine.

Key words: free-piston, diesel, combustion, linear engine, CFD

1. Introduction

The increasing focus on the environmental impacts of hydrocarbon fuel based power generation has led to increased research efforts into unconventional engine technology, with the aim of reducing engine emissions and improving efficiency. Much of this work is driven by the increasing interest into hybrid electric vehicle technology within the automotive industry, but others, such as the marine industry, face similar challenges to meet future consumer demands and governmental legislation.

Free-piston engines are linear, 'crankless' engines, in which output power is extracted by a linear load device directly coupled to the moving piston. The free-piston concept has potential advantages over conventional technology due to its simplicity, which allows a more compact unit with reduced frictional losses and maintenance costs. As the piston motion is not mechanically restricted by a crank mechanism, the free-piston engine has the valuable feature of variable compression ratio, which may provide extensive operation optimisation and multi-fuel possibilities.

1.1. *Free-piston engines: historical note*

The free-piston engine, originally proposed by Pescara [1], was topic of much research in the period 1930–1960, and such engines had some degree of commercial success as an alternative to conventional engines and gas turbines. These early free-piston engines were of the opposed piston type, with the two pistons being mechanically connected, making the engines perfectly balanced and essentially vibration-free. Toutant [2] described the first successful free-piston engine application, the Junkers free-piston air compressor, which was in use in German naval vessels providing compressed air for launching torpedoes.

McMullen and Payne [3] discussed the performance of free-piston engine gas generators, in which

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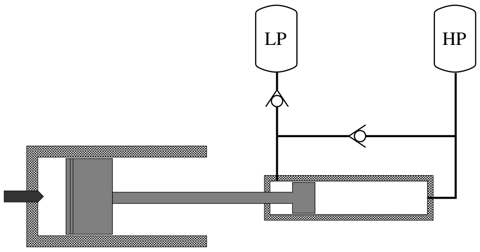
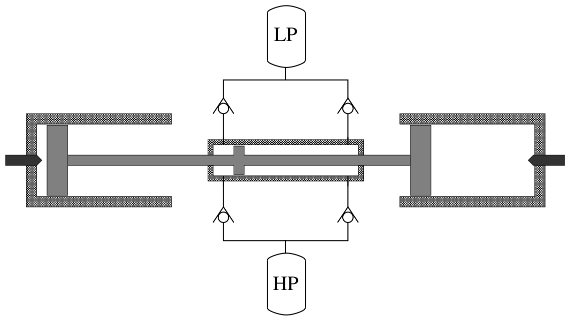


(a) Single piston free-piston engine.



(b) Dual piston free-piston engine.

Fig. 1. Modern free-piston engine configurations [6]. (Note that the illustrations are simplified for clarity.)

the load is extracted purely from an exhaust turbine. Such units saw use in stationary and marine powerplants, and both Ford [4] and General Motors [5] attempted to use free-piston gas generators for vehicle propulsion. However, as conventional engine and gas turbine technology matured, the free-piston engine concept was abandoned in the early 1960s.

1.2. *Modern free-piston engine applications*

Most modern applications of the free-piston engine concept differ from those of the mid-20th century in that they are single piston or dual piston configurations, and that a hydraulic cylinder or a linear electric machine is employed as load device. Figure 1 illustrates the principal configuration of hydraulic single piston and dual piston free-piston engines.

A number of free-piston engine applications have been proposed in recent years, however only a limited number of reports have presented experimental data from fully developed engines. Achten et al. [7] described the design and development of a single piston, hydraulic free-piston diesel engine, similar to that illustrated in Figure 1a. An advanced hydraulic control system allowed load control through 'pausing' the piston at bottom dead centre (BDC), giving very high part load efficiency. High piston acceleration around top dead centre (TDC) was reported, giving a power stroke expansion faster than that in conventional engines. Engine indicated efficiency in excess of 50 % and a fuel consumption advantage of 20 per cent compared to a conventional engine-pump unit, rising to 50 per cent at part load operation, were reported.

Tikkanen et al. [8] described the performance of a dual piston hydraulic free-piston engine prototype. The piston motion was found to be different from that of conventional engines with piston acceleration peak values of more than double the values of comparable crankshaft engines. Cycle-to-cycle variations were reported, giving variations in compression ratio and in-cylinder gas pressure.

Clark et al. [9] and Famouri et al. [10] reported experimental results from a spark ignited dual piston engine-generator. The engine is reported to have achieved 316 W electric power output at 23.1 Hz, with 36.5 mm bore and 50 mm maximum stroke [10]. High cycle-to-cycle variations were reported, in particular at low loads [9].

Hibi and Ito [11] presented test results from an opposed piston, diesel powered hydraulic free-piston engine. A "hydraulic thermal efficiency" (the total conversion efficiency from fuel energy to hydraulic energy) of 31 per cent was reported, and the authors stated that this value stays practically constant even for very low loads.

A comprehensive review of free-piston engine history and operational characteristics of such engines was presented by Mikalsen and Roskilly [6].

1.3. *Free-piston engine combustion*

Although discussed briefly by some free-piston engine developers, very few studies investigating the details of the combustion process in free-piston engines have been reported. The differences in the piston motion profile between the free-piston engine and conventional engines have been documented by a number of authors, and the free-piston engine is known to have higher piston acceleration around TDC and a significantly faster power stroke expansion. This may influence the in-cylinder gas motion and the combustion process and, consequently, the performance of the engine.

1.3.1. *The combustion process in free-piston diesel engines*

Somhorst and Achten [12] investigated the combustion process in a hydraulic free-piston engine and reported differences when comparing the results to those of a conventional engine. The authors stated that most of the fuel in the free-piston engine burns in the pre-mixed phase, resulting in a very high rate of heat release with pressure gradients of two to five times those of a comparable conventional engine. It was suggested that this is due to the high piston velocities around TDC, increasing in-cylinder gas motion and turbulence levels [7,12]. Tikkanen et al. [8] reported the same behaviour in a dual piston free-piston engine and stated that combustion takes place predominantly in a single, pre-mixed phase.

Free-piston engine combustion was also studied in the mid-20th century free-piston engines and differences when compared to conventional engines are reported by a number of authors. Discussing the free-piston engine multi-fuel possibilities, Flynn [13] described the successful operation of a free-piston engine on a range of fuels including gasoline, diesel fuel, crude oil and even vegetable and animal oils, and stated that "It seems that these engines do not care whether they get fuel with octane or cetane numbers". Fleming and Bayer [14] reported abnormal combustion leading to mechanical problems in the free-piston engine, and stated that thermodynamic models had to be modified to fit free-piston engine data due to long ignition delays and high heat release rates. The authors described how the performance was improved with changes in the injection system, and further stated that when recreating the conditions under which the free-piston engine was successfully operated in a conventional engine, this engine would not run.

1.3.2. *HCCI mode operation*

The operational flexibility of the free-piston engine, with its variable compression ratio and lower requirements for accurate ignition timing control, suggest that this concept is well suited for alternative fuels and homogeneous charge compression ignition (HCCI) operation. A particular feature of the free-piston engine is that the progress of the combustion process influences the speed reversal at TDC and the start of the power stroke expansion. The free-piston engine should therefore be well suited for HCCI operation, since a delay in the pressure rise from combustion (late ignition) will give an increased compression ratio and an early pressure rise will reduce the maximum compression ratio. Due to these self-regulatory characteristics, the free-piston engine will be less sensitive to ignition timing, which is the main challenge associated with the use of HCCI in conventional engines.

The successful operation of a free-piston diesel engine in direct injection HCCI mode was described by Hibi and Ito [11]. Theoretical and computational investigations into the concept were presented by among others Kleemann et al. [15], Fredriksson [16] and Golovitchev et al. [17]. As this paper is concerned only with conventional diesel operation, the reader is referred to the reports mentioned above for further information on HCCI free-piston engines.

2. Engine modelling

In order to study the in-cylinder gas motion and the combustion process in a free-piston diesel engine, a multidimensional simulation was set up using the open source computational fluid dynamics (CFD) toolkit OpenFOAM [18]. Written in the object oriented C++ programming language and released under the GNU General Public Licence, OpenFOAM gives the user considerable power to modify and extend the code to suit specific needs. The toolkit caters for the simulation of a number of problems in continuum mechanics, including fluid flow, heat transfer, combustion, stress analysis and electromagnetics, and numerous ready-made solvers are supplied with the software.

Jasak et al. [19] described the toolkit and demonstrated applications of engine modelling. Examples of a premixed spark ignition engine simulation and the simulation of a diesel engine, including fuel spray and combustion, were presented. The implementation and validation of Lagrangian fuel spray modelling in the simulation code were described by Nordin [20]. Ignition and diffusion-controlled combustion was modelled with the Chalmers PaSR combustion model using complex chemistry in a CHEMKIN-compatible chemistry solver [21]. Other submodels implemented in the toolkit include the Rayleigh-Taylor Kelvin-Helmholtz spray breakup model [22] and the Ranz-Marshall correlation for droplet evaporation [23]. For further information about the toolkit, the reader is referred to [18–20].

2.1. *Engine configuration*

Mikalsen and Roskilly [24] presented the design of a free-piston engine intended for electric power generation in large scale systems such as marine powerplants. Proposed advantages of the concept included

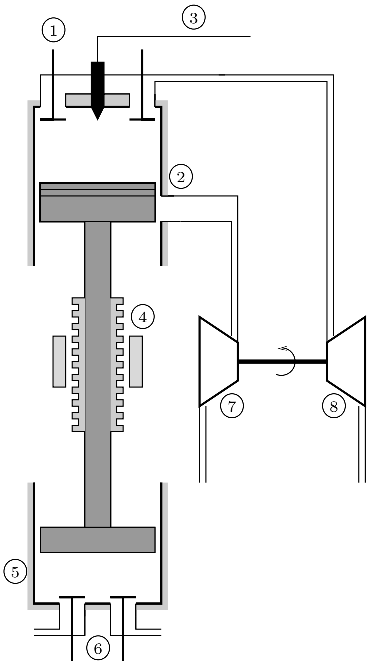
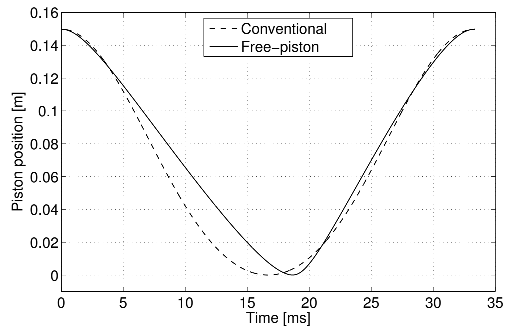


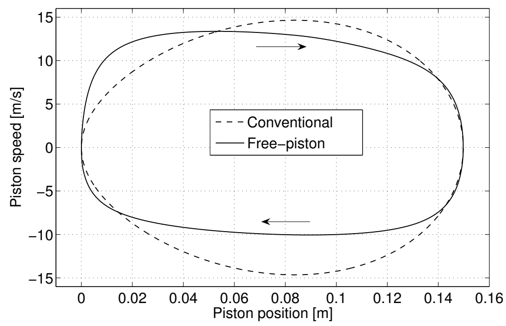
Fig. 2. Single piston free-piston engine generator [24].

Stroke	0.150 m
Bore	0.131 m
Operation	Two stroke
Nominal compression ratio	15:1
Speed	25 Hz
Boost pressure	1.68×10^5 Pa

Table 1
Free-piston engine specifications.



(a) Piston motion profile.



(b) Piston speed versus position.

Fig. 3. Simulated piston dynamics of a free-piston engine compared to that of a conventional engine [24]. (Piston position shown as distance below TDC.)

operational flexibility through variable compression ratio and reduced heat transfer and frictional losses, while engine control was found to be a potential challenge.

Figure 2 illustrates the basic engine design, with:
1. exhaust poppet valves;
2. scavenging ports;
3. common rail fuel supply;
4. linear alternator;
5. bounce chamber;
6. bounce chamber pressure control valves;
7. turbocharger compressor;
8. turbocharger turbine.

The main design specifications for the engine are listed in Table 1.

2.2. *Free-piston engine dynamics*

A full-cycle simulation model of the free-piston engine generator was presented in [24]. It was found that the piston motion profile of the free-piston engine differs from that of conventional engines, with the main difference being a significantly higher piston acceleration around TDC.

Figure 3 shows the predicted piston dynamics of the free-piston engine compared to that of a conventional engine. The significantly shorter time spent around TDC and the faster power stroke expansion for the free-piston engine are evident. Furthermore, it is seen that the peak piston velocity in the free-piston engine is lower than that of the conventional one. These factors may influence free-piston engine performance by affecting gas motion in the cylinder, heat transfer to the cylinder walls and volume change during combustion.

The piston motion profile shown in Figure 3a was fitted to a high-order polynomial through least square error fitting and implemented into the OpenFOAM code. This allows the simulation and direct comparison of free-piston and conventional engines with equal operating conditions.

2.3. *Case setup*

A standard bowl-in-piston configuration was assumed, with a bowl diameter to bore ratio of 0.60 and a minimum clearance between the piston and cylinder head of 2 mm (equal to 1.33 % of the stroke length). The piston and cylinder were designed to be symmetric around the cylinder axis, allowing a wedge geometry with cyclic boundary conditions to be used. An eight-hole injector was assumed, and the mesh represents a sector of 45 degrees (1/8 of the cylinder) and consists of 60,000 cells, equivalent to 480,000 cells for the full cylinder. The mesh is shown in Figure 4.

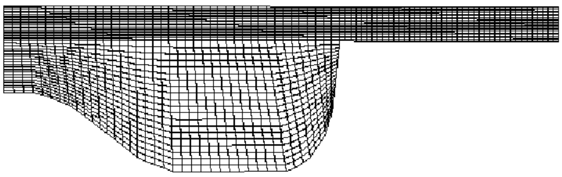


Fig. 4. Computational mesh describing the bowl-in-piston configuration.

Turbulence was modelled with a standard k-ϵ model and swirl was introduced at the start of compression with a swirl/rpm ratio of 3.0. Constant wall temperatures were used. Finally, as the engine operates on a two-stroke cycle, the simulations are run from the point of inlet ports closure (approximately 130 degrees before top dead centre).

2.4. *Model validation*

A grid sensitivity analysis was performed to ensure the accuracy of the predictions and the independence of the solution from the mesh density. The simulation model had previously been partially validated against experimental data from a Volvo TAD1260 turbocharged diesel engine located at the Swan Thermal Energy Systems Laboratory at Newcastle University. Experimental data from this engine including in-cylinder pressure plots, fuel efficiency, and nitrogen oxides emissions levels for a range of engine loads were compared to simulation results, and the CFD model showed good agreement with the experimental results. The free-piston engine investigated here has main design parameters similar to those of the Volvo engine, however as it is a two stroke engine a direct comparison to the experimental results is not possible.

3. Simulation results

A number of simulations were run to with the aim of identifying potential differences in the in-cylinder processes between free-piston and conventional engines. Cold-flow simulations were performed to investigate the influence of the piston motion profile on the in-cylinder gas motion. Fuel injection was subsequently introduced to investigate differences in the combustion process.

All the simulations below were performed using identical simulation settings with the only difference being the piston motion profile. This allows a direct comparison between the free-piston and the conventional engine at identical operating conditions, such as boost pressure, engine speed, and fuel-air equivalence ratio.

3.1. *In-cylinder gas motion*

In-cylinder gas motion has a high influence on engine performance in both spark ignition and compression ignition engines. The in-cylinder flow during combustion directly affects the fuel-air mixing, the combustion process and the heat transfer between the gases and cylinder walls, which all influence engine efficiency and emissions formation.

In-cylinder gas motion is typically divided into swirl effects and squish effects. Swirl is the rotational motion of the in-cylinder gases around the cylinder axis, and is a result of the intake system giving the gases an angular momentum as they enter the cylinder. Squish is the radial (inwards) motion of the gas as it is forced into the cylinder bowl when the piston approaches TDC and the clearance to the cylinder head decreases. These effects depend heavily on the piston velocity.

3.1.1. *Gas motion in the free-piston engine*

It has been suggested that the reported differences in the combustion process between free-piston and conventional engines are related to the in-cylinder gas motion. Such differences are likely to be found mainly in the squish effects, since swirl levels are decided predominantly by the initial swirl generated by the intake system. The swirl momentum losses during the compression stroke will not be largely influenced by the piston motion profile differences between the free-piston and conventional engines.

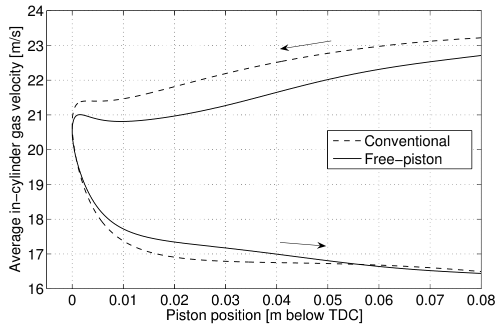
The squish effects occur at the last stages of the compression stroke, where the clearance between the piston and cylinder head decreases rapidly, and depend heavily on the instantaneous piston speed. Figure 3b showed that the piston speed in the conventional engine was higher than that of the free-piston engine for most of the compression stroke but that the free-piston engine piston speed was slightly higher very close to TDC. Differences in the squish effects between the two engines may therefore be expected.

As the piston speed is reversed at TDC and the piston accelerated downwards, a low-pressure region above the squish band is created and gas from the piston bowl flows back towards the cylinder liner region. This reverse squish is critical for the last stages of combustion when the swirl levels have decayed [25]. The significantly faster power stroke expansion in the free-piston engine may enhance this effect, and this may benefit the combustion process in the free-piston engine.

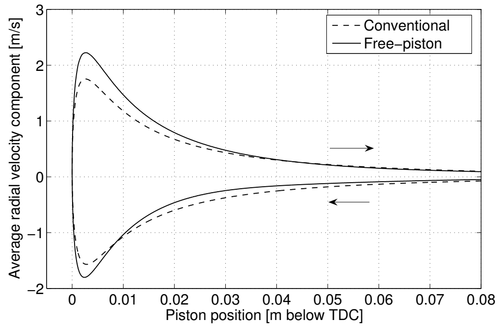
3.1.2. *In-cylinder flow simulations*

A series of simulations were run over a range of engine speeds and for varying initial swirl levels. In addition to the engine design described above, in-cylinder gas flow for a low squish piston with the same basic design as that shown in Figure 4 but with a bowl diameter to bore ratio of 0.75 was investigated. It was found that the differences between the two engine designs were present for all investigated cases, and that the relative differences in gas motion were not largely influenced by these variables.

Figure 5 shows the predicted in-cylinder gas motion in the upper half of the cycle (i.e. around TDC) for the engine described above with a piston motion profile of a free-piston engine and that of a conventional engine. Figure 5a shows the average in-cylinder gas velocity in the two engines. The two main contributors to the gas motion are the swirl and the displacement of gas as the piston moves. At the start of compression the average gas velocities in the two engines are equal, however since the piston velocity in the compression stroke is higher in the conventional engine the average gas velocities are higher. It can further be seen how the swirl decays rapidly around TDC. After TDC the higher piston velocity in the free-piston engine leads to slightly higher average gas velocities.



(a) Average in-cylinder gas velocity.



(b) Average radial velocity of the in-cylinder gases (squish).

Fig. 5. Predicted in-cylinder gas motion.

Figure 5b shows the average radial (inwards) velocity of the in-cylinder gases, or squish. At the start of compression this value is equal to zero but as the cylinder volume deforms, gas flows in the radial direction, into or out of the piston bowl. Slightly higher squish effects are seen in the free-piston engine during the late stages of compression, with the average radial velocity being around 15 % higher than that in the conventional engine. The reverse squish, occuring early in the expansion stroke, is seen to be markedly higher in the free-piston engine, with the peak average radial velocity being around 30 % higher than that in the conventional engine. It was found that in order to achieve a level of squish and reverse squish equal to that of the free-piston engine operating at 1500 rpm (25 Hz), the speed of the conventional engine had to be increased to around 1900 rpm.

3.2. *The combustion process*

Having investigated in-cylinder gas motion, fuel injection was introduced in order to identify effects of the piston motion profile on the combustion process in the free-piston engine. The combustion process depends on a number of factors, one of the most important being the fuel injector properties. A standard common rail injection system was used and appropriate specifications such as injection pressure, nozzle diameter and fuel mass flow profile were chosen. Conventional diesel engine operation with one single injection was assumed and an injection duration of 25 crank angle degrees was used. The fuel used was n-heptane, C_7H_{16}, and the chemistry model consisted of 15 species and 39 reactions. The in-cylinder charge at the start of compression was assumed to be pure air, i.e. no residual products from the previous cycle were retained in the cylinder.

3.2.1. *Fuel injection timing*

In order to evaluate the performance of the engines under comparable operating conditions, the fuel injection timing must be set individually for the conventional and free-piston engines. Earlier work by the authors has indicated that the spark timing in spark ignition free-piston engines has to be significantly advanced compared to the conventional engine due to the faster power stroke expansion. Zero-dimensional modelling of the current engine indicated that the fuel injection timing in the free-piston diesel engine should be slightly advanced when compared to that of a similar conventional engine [24].

Simulations were run to investigate the influence of fuel injection timing and identify the optimum fuel injection timing for both engines, where optimum timing refers to the start-of-injection timing giving the highest indicated efficiency, i.e. maximum net work produced by the cycle. (This is also known as maximum brake torque, MBT, timing.) It was found that the optimum timing differs only very little between the free-piston and the conventional engines, with the free-piston engine requiring a slightly more advanced injection timing. The injection timing found for optimum efficiency were -21 and -19 crank angle degrees ATDC for the free-piston and conventional engines respectively.

In a real engine, the injection timing is likely to be retarded slightly compared to this value in order to reduce exhaust gas emissions formation and avoid excessive in-cylinder gas pressures. This is acceptable because retarding the injection timing can reduce the peak in-cylinder gas temperatures and nitrogen oxides formation significantly, with only minor penalties in fuel economy. In the free-piston engine the effects of retarding fuel injection may, however, be different than in the conventional engine due to the faster power stroke expansion.

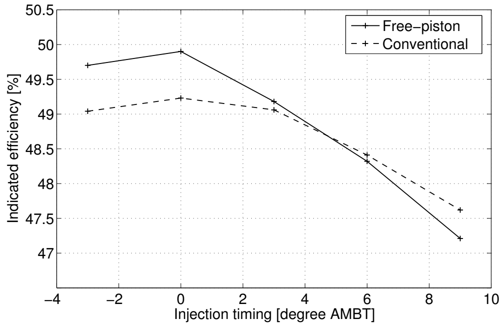


Fig. 6. Effects of variations in fuel injection timing (relative to MBT timing) on engine performance.

Figure 6 shows the predicted effects of variations in the fuel injection timing on engine performance. It can be seen that the free-piston engine is slightly more sensitive to changes in injection timing.

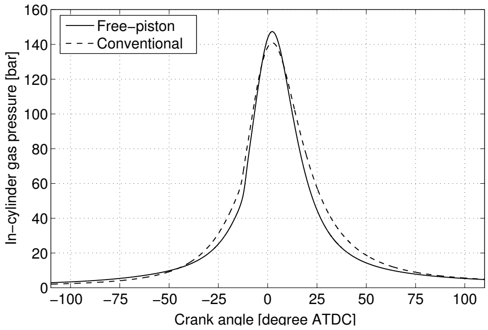
3.2.2. *Engine efficiency*

Figure 6 also shows that a slight indicated efficiency advantage is predicted for the free-piston engine compared to the conventional engine. Differences in efficiency between the two engines are due to the higher piston acceleration around TDC in the free-piston engine, leading to (a) a higher volume change during combustion, which reduces the cycle efficiency, and (b) less time spent in the high-temperature part of the cycle, which reduces the in-cylinder heat transfer losses. Previous work has indicated that the heat lost to in-cylinder heat transfer is approximately 7 % lower in the free-piston engine compared to the conventional engine, giving an indicated fuel efficiency advantage for the free-piston engine of approximately one percentage point [24]. Similar heat transfer reductions were found in the current work, but with a slightly lower predicted fuel efficiency advantage due to the CFD simulation model better accounting for the negative effects of volume change during combustion compared to the less advanced model used previously.

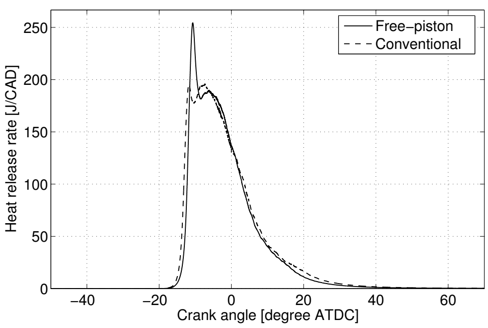
The results confirm that the reduced heat transfer losses outweigh the disadvantages associated with increased volume change during combustion in the free-piston engine.

3.2.3. *Combustion progress*

Figure 7 shows the predicted in-cylinder gas pressure and the heat release rate for the free-piston

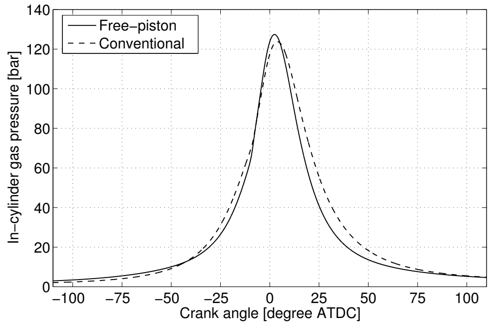


(a) In-cylinder gas pressure.

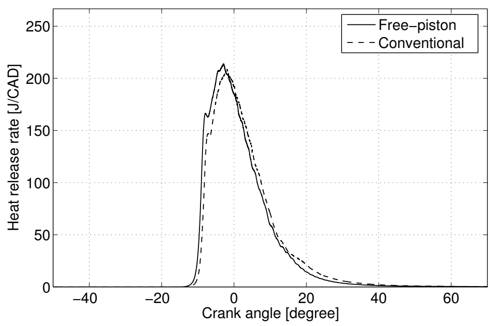


(b) Heat release rate.

Fig. 7. Predicted in-cylinder gas pressure and heat release rate at MBT injection timing.



(a) In-cylinder gas pressure.



(b) Heat release rate.

Fig. 8. Predicted in-cylinder gas pressure and heat release rate for retarded injection timing.

and conventional engines with the optimum injection timing. It can be seen that the ignition takes place later in the free-piston engine despite the earlier injection, which is due to a longer ignition delay. This results in a higher initial burn rate, since the amount of fuel that burns in the pre-mixed phase depends on the ignition delay.

The reason for the longer ignition delay is the piston motion profile of the free-piston engine, with a higher piston speed and a more rapid compression close to TDC, which lead to a lower compression ratio at the start of injection. It was found that the state of the air within the cylinder at the start of fuel injection was around 35 bar and 820 K for the free-piston engine and 42 bar and 855 K for the conventional one.

The trend of an increased heat release rate in the pre-mixed phase is consistent with the experimental findings of Achten et al. [7], Tikkanen et al. [8] and Fleming and Bayer [14], discussed above. It should be noted that, as the current engine is turbocharged, the ignition delay and the fraction of fuel burnt in the pre-mixed phase is lower than what would be expected in a naturally aspirated engine. Hence, for a non-turbocharged engine one would expect these effects to be more dominant.

With the exception of the longer ignition delay, no significant differences in the combustion process were found. The enhanced squish and reverse squish were not found to have any noticeable influence on the progress of the combustion process, and the fuel burn rate in the late stages of combustion is seen to be similar for the two engines.

Figure 8 shows the pressure plot and heat release rate for the engines with a fuel injection timing retarded by 6 crank angle degrees compared to the optimum one. In this case the compression ratio at the start of injection is similar for the two engines, leading to comparable ignition delays, and Figure 8b shows that no noticeable differences can be found in the combustion progress.

3.2.4. *Engine emissions*

The main exhaust gas emissions from diesel engines are nitrogen oxides and particulates, of which nitrogen oxides, NO_x, are usually regarded the most critical in large scale engines. Both particulates and NO_x are formed mainly during the diffusion combustion phase, and their formation depends heavily on the local temperature and concentration of fuel and oxygen in the reaction zone.

The formation of emissions may be influenced by the operating characteristics of the free-piston engine in two ways. Firstly, the differences in in-cylinder gas motion may influence the fuel-air mixing and temperature distribution within the cylinder. Secondly, the faster power stroke expansion reduces the time available for emissions formation.

A more comprehensive study with a more detailed chemistry model is required to fully investigate the details of emissions formation in the free-piston engine, such as the trade-off between particulates, NO_x emissions and engine efficiency. However, the current chemical mechanism includes the main formation equations for NO_x (the extended Zeldovich mechanism), and some indications of potential differences in the formation of nitrogen oxides between free-piston engines and conventional engines can therefore be obtained with the current simulation setup.

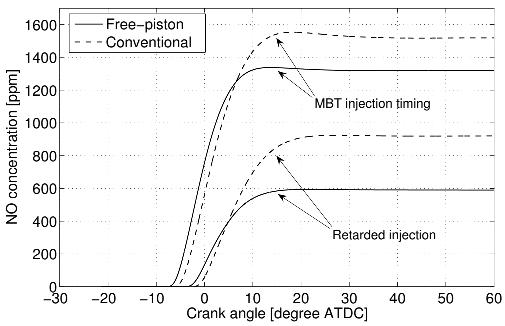


Fig. 9. Predicted average in-cylinder NO concentration.

Figure 9 shows the predicted concentration of nitrogen oxide, NO, within the cylinder for the free-piston and conventional engines. A significant emissions reduction potential can be seen for the free-piston engine. The free-piston engine also appears to benefit more from the retarded injection timing, however it should be noted that the efficiency penalty is higher for this engine, as shown above.

A number of factors influence the NO formation, including the fuel spray timing, the in-cylinder gas motion, and the temperature and pressure reduction from the expansion. A more detailed investigation is therefore required to identify details of the emissions formation in free-piston engines and potential advantages over conventional engines. The formation of other exhaust gas emissions, most importantly particulates, should also be investigated. The results presented here, notably the enhanced in-cylinder gas motion, which is generally known to reduce the level of particulates, suggest that a more detailed study will be worthwhile.

4. Conclusions

A discussion on the basic features and operating characteristics of a turbocharged free-piston diesel engine was presented. A multidimensional simulation model was used to investigate in-cylinder gas motion, the combustion process and nitrogen oxides formation and their effects on engine performance. The results were compared to those predicted for a similar conventional engine in order to identify potential differences in engine performance.

The piston motion profile in the free-piston engine was found to influence in-cylinder gas motion to some degree, with potential advantages of enhanced squish and reverse squish effects. The combustion process was not found to be largely influenced by this, however increased ignition delays were found in the free-piston engine due to a lower compression ratio at the start of fuel injection. A slight fuel efficiency advantage was found for the free-piston engine, which is consistent with earlier findings.

The nitrogen oxides formation in the free-piston and conventional engines was briefly investigated, and indicated potential advantages for the free-piston engine. It was stated that a more detailed investigation is required to identify details of the emissions formation in such engines. Other free-piston engine topics that should be further investigated include the effects of the variable compression ratio and the potential for operation optimisation, along with the use of alternative– or low-quality fuels.

References

[1] Pescara RP. Motor compressor apparatus, US Patent 1,657,641, 1928.

[2] Toutant WT. The Worthington-Junkers free-piston air compressor, Journal of the American Society of Naval Engineers, 1952:64(3):583–594.

[3] McMullen JJ, Payne WG. Performance of free-piston gas generators, Transactions of the ASME 1954:76:1–13.

[4] Noren OB, Erwin RL. The future of the free-piston engine in commercial vehicles, SAE Transactions 1958:66:305–314.

[5] Underwood AF. The GMR 4-4 "HYPREX" engine – A concept of the free-piston engine for automotive use, SAE Transactions 1957:65:377–391.

[6] Mikalsen R, Roskilly AP. A review of free-piston engine history and applications. Applied Thermal Engineering, 2007:27:2339–2352.

[7] Achten PAJ, van den Oever JPJ, Potma J, Vael GEM. Horsepower with brains: The design of the Chiron free piston engine, SAE Paper 2000-01-2545, 2000.

[8] Tikkanen S, Lammila M, Herranen M, Vilenius, M. First cycles of the dual hydraulic free piston engine, SAE Paper 2000-01-2546, 2000.

[9] Clark N, Nandkumar S, Atkinson C, Atkinson R, McDaniel T, Petreanu S et al. Operation of a small bore two-stroke linear engine Proc. ASME Fall Technical Conference 1998:31:33–42.

[10] Famouri P, Cawthorne WR, Clark NN, Nandkumar S, Atkinson CM, Atkinson RJ et al. Design and testing of a novel linear alternator and engine system for remote electrical power generation, Proc. IEEE Power Engineering Society Winter Meeting, pp. 108–112, 1999.

[11] Hibi A, Ito T. Fundamental test results of a hydraulic free piston internal combustion engine, Proc. Inst. Mech. Eng., 2004:218:1149–1157.

[12] Somhorst JHE, Achten PAJ. The combustion process in a DI Diesel hydraulic free piston engine, SAE Paper 960032, 1996.

[13] Flynn G Jr. Observations on 25,000 hours of free-piston-engine operation, SAE Transactions 1957:65:508–515.

[14] Fleming JD, Bayer RJ. Diesel combustion phenomena as studied in free piston gasifiers, SAE Paper 630449, 1963.

[15] Kleemann AP, Dabadie JC, Henriot S. Computational design studies for a high-efficiency and low-emissions free piston engine prototype, SAE Paper 2004-01-2928, 2004.

[16] Fredriksson J. Modeling of a Free Piston Energy Converter, PhD Thesis, Chalmers University of Technology, 2006.

[17] Golovitchev VI, Bergman M, Montorsi L. CFD modeling of diesel oil and DME performance in a two-stroke free piston engine, Combust. Sci. and Tech. 2007:179:417–436.

[18] OpenFOAM project web site, `http://www.OpenFOAM.org`, OpenCFD Ltd, 2007.

[19] Jasak H, Weller HG, Nordin N. In-cylinder CFD simulation using a C++ object-oriented toolkit SAE Paper 2004-01-0110, 2004.

[20] Nordin N. Complex chemistry modeling of diesel spray combustion. PhD Thesis, Chalmers University of Technology, 2001.

[21] Golovichev V, Nordin N, Jarnicki R, Chomiak J. 3-D diesel spray simulations using a new detailed chemistry turbulent combustion model. SAE Paper 2000-01-1891, 2000.

[22] Reitz RD. Modeling atomization processes in high pressure vaporizing sprays. Atomization and Spray Technology, 1987:3:309–337.

[23] Crowe C, Sommerfield M, Tsuji Y. Multiphase flows with droplets and particles. CRC Press LLC, 1998.

[24] Mikalsen R, Roskilly AP. The design and simulation of a two-stroke free-piston compression ignition engine for electrical power generation. Applied Thermal Engineering, 2008:28:589–600.

[25] Arcoumanis C, Whitelaw JH. Fluid mechanics of internal combustion engines — a review. Proc. Inst. Mech. Eng. 1987:201:57–74.