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Predictive piston motion control in a free-piston internal combustion engine[☆]

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Abstract

A piston motion controller for a free-piston internal combustion engine is presented. To improve dynamic performance in the control of the piston motion and engine compression ratio, the controller response is determined from a prediction of engine top dead centre error rather than the measured value from the previous cycle. The proposed control approach showed superior performance compared with that of standard PI feedback control known from the literature due to a reduced control action time delay. The manipulation of fuel injection timing to reduce in-cylinder pressure peaks and cycle-to-cycle variations was also studied, indicating that with the the piston motion estimation, the injection timing is a powerful control variable for this purpose.

Key words: free-piston engine, piston motion, compression ratio, control

1. Introduction

Since the invention and commercialisation of the internal combustion engine more than a century ago, numerous alternative configurations have been proposed with the aim of improving engine performance parameters such as fuel efficiency, power density, and exhaust gas emissions formation. Only very few have made it past the prototype stage, and none have become a serious competitor to the conventional engine in the wide range of applications in which it is employed.

In recent years, tightening governmental environmental legislation and consumer demands for high fuel efficiency have led to an increasing interest in unconventional engine configurations both within academia and industry. One factor driving this research is the development of microprocessor-based control systems and modern engine technology such as electronically controlled fuel injection and valve actuation systems. This allows improved operational control of the engine, along with enhanced optimisation possibilities for varying operating conditions.

1.1. Free-piston engines

One unconventional engine configuration under investigation by a number of research groups worldwide is the free-piston engine concept. The free-piston engine is a linear engine, in which the piston is coupled directly to a linear load device and the need for a crank system is eliminated. Free-piston engines were in use in the mid-20th century as air compressors and gas generators and have in recent years attracted attention as an alternative to conventional engines in hydraulic and electric power generation applications [1]. Notable recent developments include the hydraulic free-piston engines described by Achten et al. [2], Brunner et al. [3], and Tikkanen et al. [4], as well as the free-piston engine generators described by Clark et al. [5], Van Blarigan et al. [6], Fredriksson and Denbratt [7], Xiao et al. [8], and Xu and Chang [9]. A comprehensive overview of free-piston engine particular features, history, and reported applications was presented by Mikalsen and Roskilly [1].

The main advantages of such engines are mechanical simplicity, low frictional losses, and high operational flexibility. The absence of the crank mechanism allows the compression ratio to be varied, making the free-piston engine highly suitable for multi-fuel or homogeneous charge compression ignition (HCCI) operation. The performance of such engines compared with conventional engines has previously been studied extensively by the authors [10, 11, 12, 13], and potential ad-

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vantages of improved fuel efficiency and reductions in nitrogen oxides emissions formation have been demonstrated. Challenges associated with the free-piston engine includes starting, valve actuation and timing, and abnormal combustion, however these issues have been successfully resolved in previously reported systems [1]. The main—and currently unresolved—challenge associated with the free-piston engine concept is the control of the piston motion in conjunction with system load handling, due to the endpoints of the motion (the dead centres) having to be controlled within tight limits.

1.2. Engine configuration

A single piston, two-stroke free-piston diesel engine with a variable-pressure bounce chamber was described by Mikalsen and Roskilly [10], and this engine is illustrated in Figure 1. The main components of the engine illustrated in Figure 1 are:

- 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.

Compared to the more common dual piston design, in which the bounce chamber is replaced with a second combustion cylinder, the bounce chamber adds an additional control variable to the system and thereby simplifies the control problem.

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.

The engine design variables are shown in Table 1. Simulations have shown that a power output of 45 kW is achievable at an operating speed of 30 Hz and with a fuel efficiency of approximately 0.42 [10]. The fuel efficiency achievable is 3→5 percentage points higher than that expected from a conventional engine of similar size, mainly due to (a) lower frictional losses; (b) lower in-cylinder heat transfer losses due to a faster power stroke

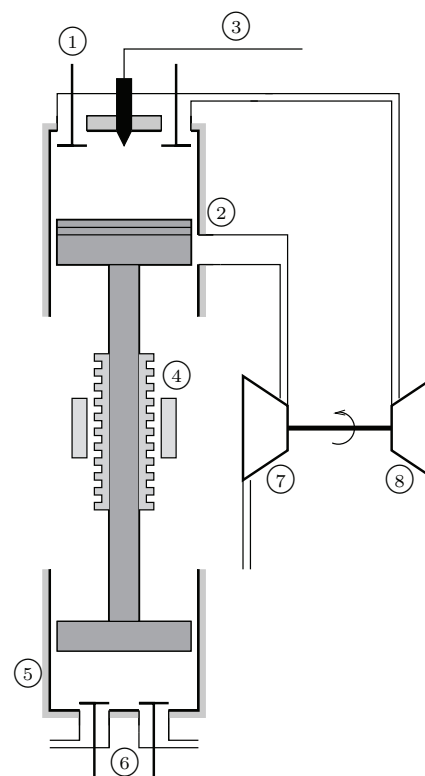


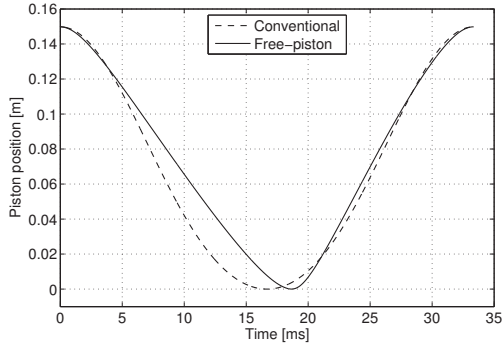
Figure 1: Free-piston engine [10].

expansion; and (c) higher degree of operational optimisation possible. A full-cycle simulation model of the engine was described in [10], and this simulation model has been used in the analysis described in this paper.

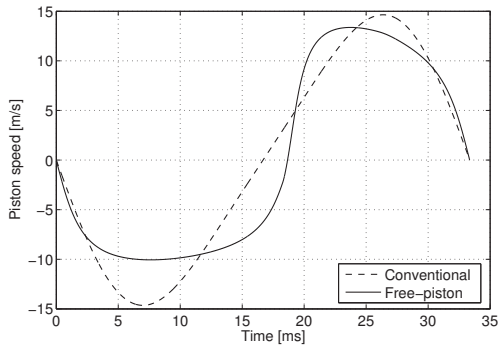
1.3. The piston motion control problem

In conventional engines, the crank system controls the piston motion and the piston motion profile can not be influenced during engine operation. The free-piston engine does not have this system, and the piston dynamics is therefore different to those of conventional engines. Figure 2 shows the piston position and velocity over one engine cycle for the free-piston engine compared with a conventional engine. In particular, the asymmetric motion profile should be noticed, as well as the faster power stroke expansion but lower peak piston velocity. This is due to significantly higher piston acceleration around the endpoints. These particular dynamics of the free-piston engine have been documented by a number of authors [4, 5, 7].

Due to these particular operational characteristics, alternative arrangements must be realised in order to control the piston top dead centre (TDC) and bottom dead



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

Figure 2: Predicted piston dynamics of the free-piston engine [10].

centre (BDC) positions. TDC position must be controlled within a tight range to ensure a compression ratio sufficient for fuel autoignition whilst at the same time avoiding excessive in-cylinder gas pressures. As the engine operates on a two-stroke cycle, accurate control of the bottom dead centre is also required to ensure efficient scavenging of the engine cylinder.

The main design criterion for the control system is the effective rejection of load changes in the electric machine driven by the engine. The available control variables are the fuel mass flow rate and the bounce chamber control valves timing, which allows variation of the air mass trapped in the bounce chamber. The fuel injection and exhaust valve timings can also be controlled, however these variables have little influence on engine dynamics and are therefore reserved for operational optimisation purposes [14, 15]. The optimisation of these, along with the compression ratio setpoint, make up the

secondary control objectives for the engine system.

1.4. State of the art

The volume of published research on free-piston engine control issues is very limited. In the case of single piston, hydraulic free-piston engines, the control is significantly less challenging compared to an engine such as the one presented here, and high operational flexibility and excellent performance has been demonstrated by among others Achten et al. [2].

Johansen et al. [16, 17] studied the feasibility of decentralised PID control of TDC and BDC position in a single piston free-piston gas generator, manipulating the fuel injection rate and mass of air trapped in the bounce chamber. The engine had similarities to the one investigated in this paper, however the load was extracted entirely from an exhaust gas turbine and therefore load changes had a much less direct impact on engine operation. Experimental results were presented, showing that satisfactory performance could be achieved using this control strategy.

Using a simplified engine model, Tikkanen and Vilenius [18] investigated the control of a dual piston hydraulic free-piston engine in which feedforward propagation of the disturbance was used to improve engine dynamic response. The only available control variable in dual piston engines is the fuel mass flow, however it was demonstrated how satisfactory compression ratio control could be achieved for moderate load changes.

Cycle-to-cycle variations have been reported as a potential challenge in free-piston engines due to the direct influence of one cycle on the next. In experimental studies of dual piston free-piston engines, Tikkanen et al. [4] and Clark et al. [19] found large cycle-to-cycle variations in engine performance parameters, particularly at low loads. However, none of the reports claim that these were critical for engine operation.

Previous work by the authors has investigated the use of decentralised, 2-channel single-input single-output feedback control techniques in the current free-piston engine, linking fuel flow rate to TDC error and bounce chamber trapped air mass to BDC error [14, 15]. The fuel flow rate is a significantly more powerful control variable than the bounce chamber trapped air, since it directly influences the energy level in the system. It is therefore necessary to couple this to the most important operational parameter, which is TDC position. Standard PID and pseudo-derivative feedback (PDF) control approaches were tested, along with disturbance feedforward strategies. Acceptable performance was found for moderate load changes, however if the engine is subjected to highly varying loads, feedforward action was

found to be necessary to reduce the delay in the controller response.

2. Predictive control system design

To improve the dynamic response of the controller, the implementation of a predictive control system was proposed. Consider an upwards stroke of the engine as illustrated in Figure 1, i.e. from BDC to TDC. The energy stored in the bounce chamber will influence the kinetic energy of the piston assembly during the combustion chamber compression stroke, and the piston speed during the compression can therefore be related to the final compression ratio and thereby the TDC position. This can be utilised to predict the TDC position during the compression stroke and provide an estimate of the forthcoming TDC error. If an accurate TDC estimator can be realised, the use of this as opposed to the measured TDC value from the previous cycle will reduce the time delay in the control loop by one full cycle.

2.1. TDC estimation

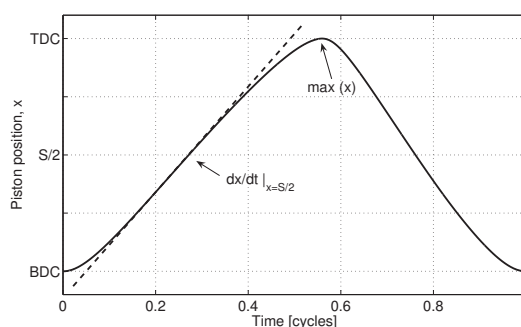
The piston assembly position is measured throughout the cycle, and the piston speed can therefore be calculated at any point in the compression stroke. In this work it was decided to use the speed at the nominal half-stroke point, i.e. when the piston is half way through the compression stroke. The relation between the speed of the piston at this point and the resulting TDC position could then be studied using the simulation model described previously.

Figure 3a shows the piston motion profile of the engine and illustrates the information obtained and calculated from the piston position measurements. For the TDC control circuit, this includes the piston velocity at the half-stroke point in the compression stroke ($dx/dt|_{x=S/2}$) and the outer position of the piston motion, or TDC ($\max(x)$).

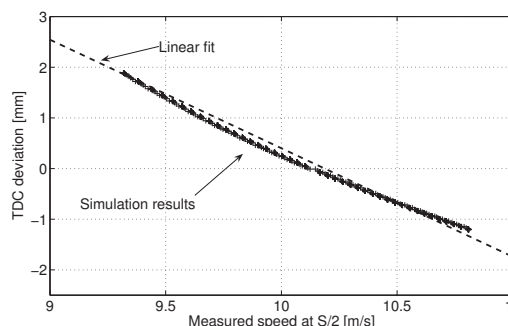
Figure 3b shows the simulated relationship between the piston speed at the half-stroke point and the following TDC position when varying the electric load. A close to linear correlation can be seen, indicating that the TDC can be effectively predicted from the piston velocity in the compression stroke. A prediction routine using a simple linear relation which was fitted to the simulation data was used, to give the TDC prediction as

$$\hat{x}_{\text{TDC}} = a \cdot \dot{x}_{S/2} + b. \quad (1)$$

Coefficients a and b were adjusted to give the fit shown in Figure 3b. In each cycle, this supplies a prediction of the TDC position to the controller during the compression stroke.



(a) Illustration of data obtained from the piston position measurements.



(b) Relationship between the piston speed at the half-stroke point and the resulting TDC position.

Figure 3: The TDC estimation system.

2.2. Control system structure

A control system as illustrated in Figure 4 is proposed for the free-piston engine, coupling the fuel mass injection rate and fuel injection timing to the TDC error. The mass of air trapped in the bounce chamber can be linked to the BDC error, however only the TDC control loop is considered in this paper as satisfactory BDC control can be achieved using the pressure control valves to modify the bounce chamber air mass [14, 15].

The TDC control system consists of three parts: the measurement system, the predictive system, and the controllers providing the signals to the fuel injection system. Referring to Figure 4, the engine plant has the inputs fuel mass and injection timing, and the output is the continuously measured position of the piston assembly (equivalent to the crank angle in a conventional engine). The actual TDC position is calculated from the extreme position reading and fed back to a standard negative feedback loop with an integral term control element to eliminate any steady state error.

The fuel mass flow control variable has strict limits to its operational range, being limited by the amount of air available in the combustion chamber. It is clear that when operating on high loads, the fuel mass control variable will be close to saturation. This nonlinearity can readily be implemented in a fuzzy controller to avoid a controller demand higher than that deliverable by the actuators. In the current work, a proportional fuzzy controller was used to obtain a linear gain controller response with saturation. Since the influence of changes in the fuel mass and injection timing on engine operation is well understood, control rules can be effectively constructed based on this knowledge. A typical control rulebase for the fuel mass would be:

- **If** TDC error is negative **then** change in fuel mass is to be negative.
- **If** TDC error is zero **then** change fuel mass is to be zero.
- **If** TDC error is positive **then** change in fuel mass is to be positive.

3. Controller performance

Using the simulation model, the feasibility of different control approaches and the controlled engine performance can be tested without the risk of damaging real engine components. The main performance criterion for the controller is the ability to reject load disturbances, since the engine will likely be supplying an electrical grid with varying load demands. A change in the TDC setpoint will occur as a response to a change in engine

load, because the engine compression ratio is varied for different loads. However, these setpoint changes generally have only a small influence compared to that of the load changes [14, 15].

3.1. Fuel mass control

Depending on the characteristics of the electric grid and the units coupled to it, the engine may face rapid changes in load demand. The harshest possible condition faced by the engine will be a step load change; this will be the case if there are no time delays in the load distribution system. However, since there will always be inductive and capacitive components in the electric circuit, a load change, as seen by the engine, will always be smoother than that assumed here. The amount of energy storage in the electric circuit can be modified by design, therefore the worst case (step load changes) has been assumed here in order to obtain more general results.

Due to the particular operating characteristics of the free-piston engine, the timing of a load change will also influence the engine response. The worst case will be a step change in load shortly after TDC, at which point the fuel injection control variable has been set and cannot be changed until the next cycle. All the simulations presented here were run for this situation.

The simulations were run using the predictive control system described above, and repeated with a standard PI controller for comparison. For this first test, the injection timing remained constant and the fuel mass gain coefficients of both controllers were manually tuned to minimise peak error in the TDC position.

Figure 5 shows the dynamic response of the engine for a 15% step change in load at time $t = 1$ s with the engine originally operating at 80% of the nominal full load power. It can be seen that the predictive controller provides a significantly better response than the standard PI controller, due to the shorter time delay in the error response. Both the peak error and the speed of response is enhanced when using the predictive system.

It should be noticed that, although providing better performance than the PI controller, TDC error values in the order of 1 mm (equivalent to a compression ratio range of approximately 13-18) are probably on the limit of what could be accepted. Clearly, a real system will experience larger load variations, however there will be time delays in the electrical system and the load seen by the engine will therefore depend on the detailed design of the electric machine and electric circuit. For the purpose of this study, i.e. to compare the performance of the predictive control scheme to that of standard meth-

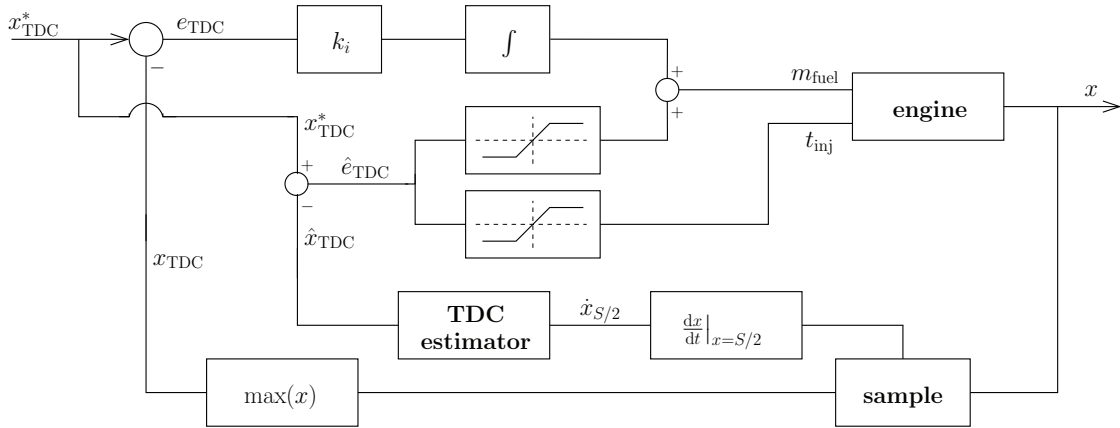
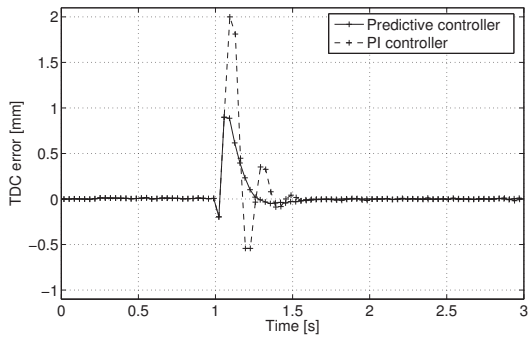
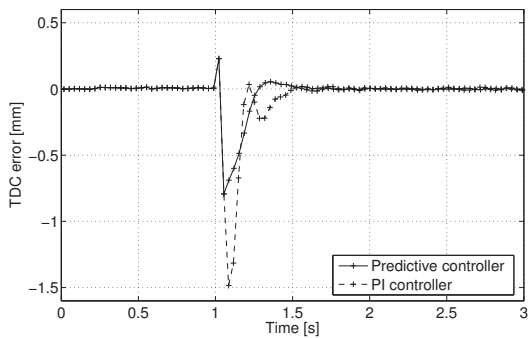


Figure 4: Free-piston engine estimation and control system.



(a) Response to a 15% step increase in load with PI and predictive controllers.



(b) Response to a 15% step decrease in load with PI and predictive controllers.

Figure 5: Engine response to changes in electric load with PI and predictive controllers. (Ticks denote TDC position readings.)

ods, the assumption of a step load change is therefore sufficient.

The initial, minor TDC error visible in the graphs, having an opposite sign compared with that induced by the load, was due to the setpoint changing with load. For the PI controller, this actually generates an initial controller response opposite to that desirable (this can be seen in the figure below), however the effect of this on the controller performance is minor. The predictive controller is insensitive to this setpoint change.

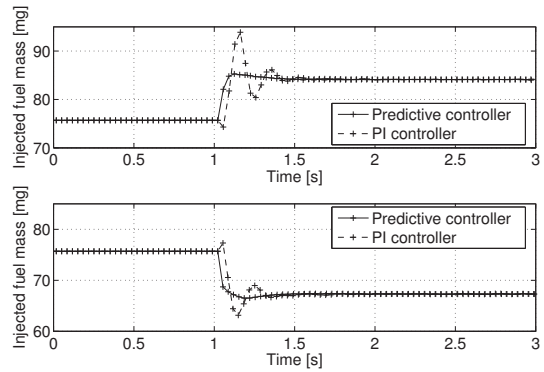


Figure 6: Actuator (fuel injection) action for a 15% step increase (top) and decrease (bottom) in load with PI and predictive controllers.

Figure 6 shows the mass of fuel injected (actuator action) for the step changes in engine load. The significantly smoother response of the predictive controller is clearly evident. This is of particular importance in the case of an increase in load as the higher fuel injection

rate of the standard PI controller will lead to increased emissions formation and additional thermal load on the engine.

3.2. Injection timing control

The main objective of the TDC control scheme is to avoid excessively low or high in-cylinder gas pressures. Most importantly, for a rapid load reduction there will be an increase in compression ratio and it will be important to minimise the risk of excessive pressure peaks in order to avoid mechanical damage to the engine. Similarly, very high in-cylinder gas temperatures in such a case will lead to high heat loads on the engine and high emissions formation.

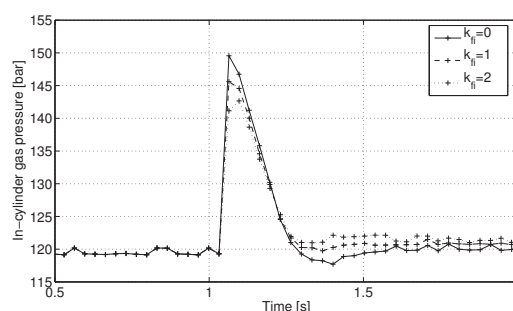
The fuel injection timing has a significant influence on the in-cylinder pressure as it determines the timing for the release of the combustion energy. If the fuel injection timing can be retarded when a high compression ratio is predicted, this may reduce the peak gas pressure. Similarly, for a predicted low compression ratio the ignition delay is likely to increase and the combustion performance may be poorer. An advanced injection timing in such a case could then improve performance. Hence, there are potential significant advantages of having a TDC prediction in terms of engine performance and operational smoothness.

A simple means of implementing injection timing control is therefore to adjust it proportionally to the TDC error. This was implemented in the simulation model and the effect of such injection timing control was analysed by using various values of gain in a proportional controller with saturation. The fuel injection timing was given by

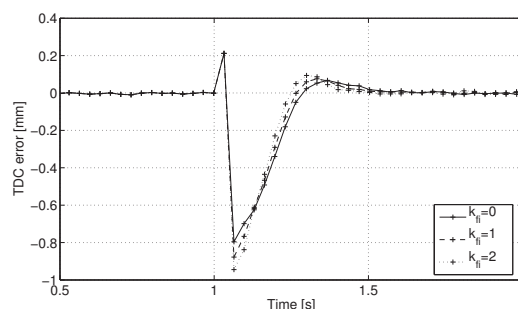
$$\tau_{fi} = k_{fi} \cdot \hat{e}_{TDC} + \tau_{fi,0} \quad (2)$$

where τ_{fi} is the fuel injection timing for the current cycle, k_{fi} is the gain factor, \hat{e}_{TDC} is the predicted TDC error, and $\tau_{fi,0}$ is the fuel injection timing setpoint provided by the supervisory engine control system. The modification of injection timing was limited so as not to allow a start of injection after TDC, as a heavily retarded injection would have negative effects on the combustion process and emissions formation.

Figure 7a shows the in-cylinder pressure for a 15% step reduction in load, using the predictive control scheme with three values of injection timing control gain. The rapid load reduction increases the energy available for compression and leads to an increase in engine compression ratio and high peak in-cylinder pressures. It can be seen that delaying the fuel timing has a advantageous effect on the peak pressure, however a



(a) Peak in-cylinder gas pressure for varying K.



(b) TDC error for varying K.

Figure 7: Effects of fuel injection timing control on engine performance for a 15% step decrease in load.

significant increase in the cylinder pressure following a load reduction is still present.

Figure 7b shows the corresponding TDC error for the same situation. It can be seen that retarding the fuel injection increases the peak error in the TDC position, and thereby also the compression ratio. This is because the combustion process generates adverse work which contributes to slowing down the piston around TDC accelerating it downwards, and hence by delaying the injection this work is reduced. Although the error increases, the reduction in the peak pressure values will be of much higher importance in this case, and this should be taken into account in the design of the control system.

3.3. Cycle-to-cycle variations

Combustion engines exhibit fluctuations in the work output and in-cylinder gas pressure levels between cycles due to variations in the injected fuel mass, the scavenging process performance, and other operational variables. If such variations are too high, the engine will not run smoothly and operational (or ‘driveability’) problems may occur. The free-piston engine will be more prone to these variations, which may have a serious impact due to the variable compression ratio and the direct influence one cycle has on the next.

A highly desirable feature of the predictive control scheme is that it may be able to dampen cycle-to-cycle variations by adjusting the fuel mass and timing on a cycle-to-cycle basis prior to TDC, based on the predicted TDC position. The standard PI controller will be unable to correct for cycle-to-cycle variations and will instead add to the problem by generating a control signal in response to variations from previous cycles. In an extreme case, with very high cycle-to-cycle variations, a standard PI controller may therefore even compromise system stability.

Figure 8 shows the peak in-cylinder pressures during an operational sequence for the free-piston engine with the three tested control schemes: standard PI, predictive control, and predictive and injection timing control. For the operation shown, a random variation in the fuel mass was introduced by varying this uniformly by $\pm 2\%$ of the nominal value. This is slightly higher variation than what one would expect from a modern injection system, however in a real engine other variables could also contribute to cycle-to-cycle variations.

The predictive controller with injection timing control was found to reduce significantly the variation in peak in-cylinder pressure, p_{\max} , compared with the conventional PI controller. The variations in peak pressure

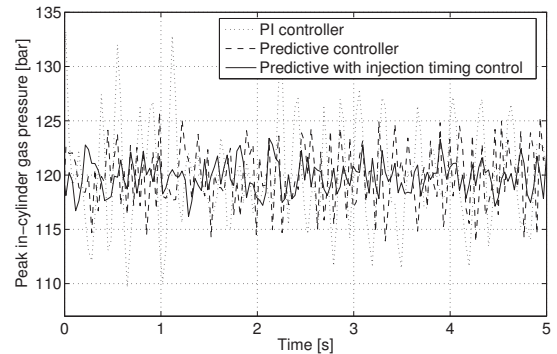


Figure 8: Peak in-cylinder gas pressure for the engine with different controllers.

were measured using the coefficient of variation, defined as

$$\text{COV}_{p_{\max}} = \frac{\sigma_{p_{\max}}}{\bar{p}_{\max}} \quad (3)$$

where $\sigma_{p_{\max}}$ is the standard deviation of p_{\max} and \bar{p}_{\max} is the mean value. It is generally accepted that COV values of more than approximately 5% are undesirable, as this prevents the engine from running smoothly and stable.

Coefficient of variation values in the peak pressure under these conditions were found to be 3.8 per cent with the PI controller, 2.5 per cent with the predictive fuel mass controller, and 1.2 per cent with the predictive fuel mass and injection timing controller. This indicates that the modification of injection timing based on the piston assembly kinetic energy in the compression stroke is a very powerful method to reduce in-cylinder pressure variations in the free-piston engine.

4. Conclusions

The development of a predictive piston motion controller for a free-piston internal combustion engine was presented. Engine top dead centre position was predicted during the compression stroke and the control action, fuel mass flow, was based on this information rather than the measured value in order to reduce the controller time delay. Significantly improved load disturbance rejection was observed with the predictive control scheme, including a lower peak error and faster response. The modification of ignition timing in combination with the fuel mass flow was found to reduce peak in-cylinder pressures during rapid load decreases

to some degree and significantly reduce engine cycle-to-cycle variations. The proposed controller showed superior performance to that of a standard PI controller, which has been studied previously by these and other authors.

Clearly, there is significant potential to improve the predictive controller performance compared to the simplified approach adopted in this study. The control scheme could make use of a more advanced fuzzy control system to account for the non-linear and multi-variable nature of the control problem. The use of adaptive prediction and control elements can improve the TDC predictions and account for changes in engine operational parameters. The implementation of an adaptive prediction system should be straight-forward, since the measured TDC value will be available for comparison shortly after the prediction.

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