The control of a free-piston engine generator. Part 2: engine dynamics and piston motion control ⋆

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Abstract

Free-piston engines are under investigation by a number of research groups due to potential fuel efficiency and exhaust emissions advantages over conventional technology. The main challenge with such engines is the control of the piston motion, and this has not yet been fully resolved for all types of free-piston engines. This paper builds on the fundamental investigations presented in the accompanying paper and investigates the dynamics of the engine and the feasibility of classical control approaches. The response of the engine to rapid load changes are investigated using decentralised PID, PDF and disturbance feedforward. It is found that the engine is sensitive to rapid load changes but that in constant power applications standard control techniques provide satisfactory performance. The influence of cycle-to-cycle variations in the combustion process are investigated, but not found to be critical for engine operation.

Key words: free-piston, linear engine, dynamics, control

1. Introduction

A single piston free-piston engine generator currently under development at Newcastle University was presented by Mikalsen and Roskilly [1], and the basic controllability of the engine was investigated in the accompanying paper [2]. It was shown that accurate control of piston dead centre position is critical for the operation of the free-piston engine. Hence, the dynamic response of the free-piston engine to changes in the disturbance inputs, predominantly engine load, is of high interest. If a control system able to maintain the dead centre positions within specified limits cannot be realised, the engine concept itself will not be feasible.

This paper builds on the fundamental analyses in the accompanying paper and investigates engine dynamic characteristics and controller performance. The free-piston engine is a multivariable and non-linear plant, however the successful use of decentralised, single-input single-output (SISO) proportional-integral-derivative (PID) controllers in a similar application has been reported by other authors [3]. This study investigates the use of decentralised PID control, pseudo-derivative feedback (PDF) control, and disturbance feedforward.

2. Controller design

A typical TDC clearance value for an engine such as the one investigated here is 1.5–3 mm. A TDC deviation of ±1 mm is equivalent to a compression ratio range of around 13-18, which is probably acceptable in most cases. Maintaining the TDC position within ±1 mm of the TDC setpoint is therefore chosen as an initial design guideline. The require-

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ments for BDC position control is more relaxed, and a target of ± 3 mm was chosen.

When adding a feedback loop with a controller to the free-piston engine plant, one gets a system as that illustrated in Figure 1.

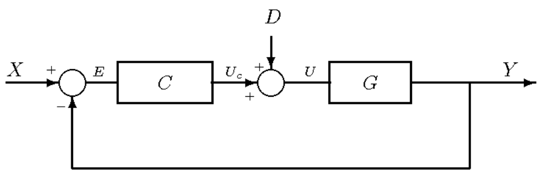


Fig. 1. Feedback control system.

The engine plant, G, has three inputs and three outputs, hence G is a 3×3 matrix. The output vector, Y, contains engine speed, TDC and BDC positions, and can be written as

$$Y = \begin{bmatrix} \text{speed} \\ \text{TDC} \\ \text{BDC} \end{bmatrix}, \tag{1}$$

and the input to the engine, U consists of fuel mass per injection, bounce chamber trapped air mass and electric load force, hence

$$U = \begin{bmatrix} m_{\text{fuel}} \\ m_{\text{BCair}} \\ F_{\text{mag}} \end{bmatrix}. \tag{2}$$

X contains the setpoints for the three output variables, i.e.

$$X = \begin{bmatrix} \text{speed}_{\text{SP}} \\ \text{TDC}_{\text{SP}} \\ \text{BDC}_{\text{SP}} \end{bmatrix}, \tag{3}$$

and the output from the controller, U_c, is the two controllable plant inputs:

$$U_c = \begin{bmatrix} m_{\text{fuel}} \\ m_{\text{BCair}} \\ 0 \end{bmatrix}. \tag{4}$$

Consequently, the disturbance, D, is the electric load force

$$D = \begin{bmatrix} 0 \\ 0 \\ F_{\text{mag}} \end{bmatrix}. \tag{5}$$

The controller matrix will be a 3×3 matrix:

$$C = \begin{bmatrix} C_{11} & C_{12} & C_{13} \\ C_{21} & C_{22} & C_{23} \\ 0 & 0 & 0 \end{bmatrix}.$$

2.1. *Decentralised control*

Depending on the level of interaction in the plant, multiple-input multiple-output (MIMO) systems can in many cases be reduced to a set of single-input single-output (SISO) loops for which standard SISO control systems can be designed. This may simplify the work designing the control system, and the successful application of decentralised control in a free-piston engine was reported by Johansen et al. [3].

2.1.1. *Pairing of inputs and outputs*

Analytical methods to find the best pairing of intputs and outputs exist, however in the current system physical reasoning can be used to find the best coupling. Of the two control variables, the fuel mass is the more powerful since it has a higher influence on all the operational variables along with a fast response. Of the control objectives, the TDC position is that with the highest priority, since this is most critical to engine operation and must be controlled within tight limits. The fuel mass should therefore be coupled to the TDC error signal.

This leaves the bounce chamber trapped air to control BDC position. The engine speed is, as discussed in the accompanying paper, not critical for engine operation and having only two control variables available, the engine speed cannot be controlled with conventional methods.

The controller matrix becomes:

$$C = \begin{bmatrix} 0 & C_{12} & 0 \\ 0 & 0 & C_{23} \\ 0 & 0 & 0 \end{bmatrix}$$

where C_{12} is the coupling between the TDC error signal and the fuel mass command signal, and C_{23}

is the coupling between BDC signal and the bounce chamber trapped air demand.

2.2. *Engine dynamics*

The engine dynamic response to a load change depends heavily on the dynamics of the load force, which for the free-piston engine generator are determined by the electric system. Details of this are not known at an early design stage, but it is clear that electric systems have very low time delays and may therefore impose rapid changes in engine load. It is, however, likely that the output power has to be conditioned, which may allow the implementation of energy storage devices to dampen disturbances on the engine if necessary.

A general investigation of the controlled engine's ability to reject disturbances was undertaken, in order to gain insight into the necessity of additional measures in the electric system to aid engine control.

Characteristics of the engine inputs

The characteristics of the inputs to the engine plant and their influence on its response differ in some respects to textbook examples in control systems design. Firstly, for any load change there will be a change of setpoint in TDC position. This setpoint change must take place immediately to avoid excessive in-cylinder pressures. This was implemented in the simulation model such that a load change triggers an instant change of TDC setpoint. Such simultaneous changes of load and system setpoint add to the requirements set on the controller.

Secondly, the characteristics of the free-piston engine generator are such that the timing of the load change will have an influence on the engine response. Since the control variables can be modified only once per engine cycle, if a load change occurs shortly after a control variable is set there may be a significant delay before controller action will take effect. If one considers the TDC control variable, the fuel mass flow demand, which is set at BDC, a series of two engine cycles starting at BDC will include the following events:

(1) BDC.
(2) Compression stroke.
(3) TDC, where TDC position is read.
(4) Power stroke expansion.
(5) BDC, where the controller action for the TDC controller is set.
(6) Compression stroke.
(7) TDC.
(8) Power stroke expansion.
(9) BDC.

If a load change occurs during (1) or (2), the TDC position at (3) will be influenced and trigger a control response at (5) to correct for the error at the next fuel injection (at (7)). While this may be a significant delay, if the load change occurs just after (3), the controller action will not begin until (9) and the correction will not take place until the following TDC, two full cycles after the disturbance occurred.

All the simulations below are done with the load change occurring at TDC, and therefore represent 'worst case' situations.

2.3. *Proportional, integral and derivative feedback control*

Proportional, integral and derivative (PID) control is widely used in industry and, although having some limitations, has proved excellent performance for a wide range of applications. The implementation of PID control is uncomplicated, and initial tuning can be performed using well-known empirical rules. Johansen et al. [3] demonstrated the successful use of PI and PID control for a free-piston application similar to that investigated here.

To investigate the feasibility of PID control in the free-piston engine, such controllers were implemented in both control loops in the simulation model. PID control is implemented by setting the relevant elements of the controller matrix C in the standard feedback control system to a sum of a proportional, an integral and a derivative gain term. I.e. for the controller matrix element mn:

$$C_{mn} = k_p \cdot e(t) + k_i \int_0^t e(t)\,\mathrm{d}t + k_d \frac{\mathrm{d}e(t)}{\mathrm{d}t},$$

where $e(t)$ is the error signal component of the vector E.

The feasibility of PID control in the free-piston engine

PID controllers are generally robust, but the derivative controller term can be sensitive to measurement noise and will produce very large controller responses to step changes in setpoint, since the theoretical derivative response to a step change is infinite. Methods do, however, exist to correct for this and avoid the actuators saturating.

After testing the controller performance, it was found that the particular feature of simultaneous changes in load disturbance and TDC setpoint do represent a challenge for the PID controller. The characteristics of the load and setpoint changes are such that the instant TDC setpoint change will produce an initial error which is opposite to that produced by the load, and therefore trigger an initial control response opposite to the desired one. This limits the gain of the proportional controller term and, in particular, the derivative term due to the initial control response enhancing the error created by the load change. (The setpoint change immediately following a load change can be seen in the plots below.)

For an engine with a constant compression ratio setpoint this will not be a problem, however in the current engine, using only minor changes in compression ratio and TDC setpoint, the derivative gain had to be reduced to a level at which the derivative action did not improve controller performance. If a higher degree of compression ratio control is desired, these problems will be even more serious.

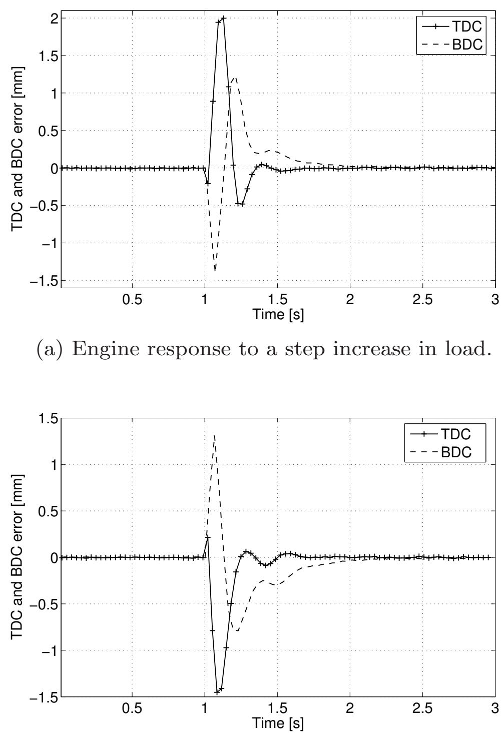
Although methods may exist to improve the performance or correct for these problems, for example delaying the TDC setpoint change (which would increase the risk of excessive in-cylinder pressures during transient operation), the feasibility of a PID controller for the free-piston application is questionable.

Controller performance

Due to the very limited effect of the derivative gain term in the TDC control loop, the performance was investigated using a PI controller only. For the BDC control loop, a PI controller was found to perform satisfactorily. Both controllers were manually tuned to minimise the peak errors in TDC and BDC positions and to provide a fast settling time.

Figure 2 shows the engine dynamic response to a 15 per cent step change in the load at time $t = 1\,\mathrm{s}$, with the engine originally operating at 80 % load. The response of the BDC control loop is seen to be acceptable, and the settings of this controller were therefore adjusted to give a response slightly slower (more overdamped) than that possible, in order to minimise the disturbance on the TDC loop.

For the TDC controller, the setpoint change which creates the initial error, as discussed above, can be seen just after $t = 1\,\mathrm{s}$. A high peak error in the TDC error is further seen, and it was found that this could not be avoided with the current controller due to the delay between the disturbance and the control response. The settling times were found to be acceptable.



(a) Engine response to a step increase in load.

(b) Engine response to a step decrease in load.

Fig. 2. Engine response to a 15 per cent step change in load with PI controller.

2.4. *Pseudo-derivative feedback control*

The concept of pseudo-derivative feedback (PDF) control was proposed by Phelan [4]. Although not in widespread use, PDF control is generally reported to have better load handling capabilities than PI control whereas PI control has better setpoint tracking performance. (See Phelan [4], Setiawan et al. [5], and Ohm [6] for further analyses and examples.) In the current system the load disturbance is the greater challenge due to the high peak error produced by rapid load changes. Rapid setpoint changes will only occur as a response to a load change.

Figure 3 illustrates the pseudo-derivative feedback control system. The controller in the forward path, C, consists of an integral term only, and a negative feedback of the output state is added after the controller. The feedback gain matrix K is

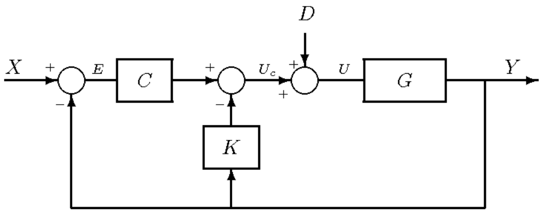


Fig. 3. Pseudo-derivative feedback control system.

$$K = \begin{bmatrix} 0 & K_{12} & 0 \\ 0 & 0 & K_{23} \\ 0 & 0 & 0 \end{bmatrix},$$

where K_{12} and K_{23} contain proportional gain terms only.

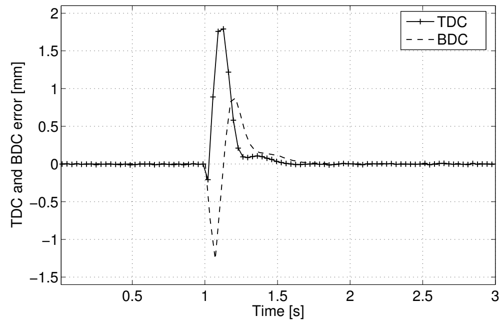
A feature of PDF control that may make it more suitable for the free-piston engine application than PI control is that it is less sensitive to rapid setpoint changes. However, since the controller in the forward path, C, does not contain a proportional or derivative term, the speed of response to setpoint changes may be lower than in the PI controller.

Controller performance

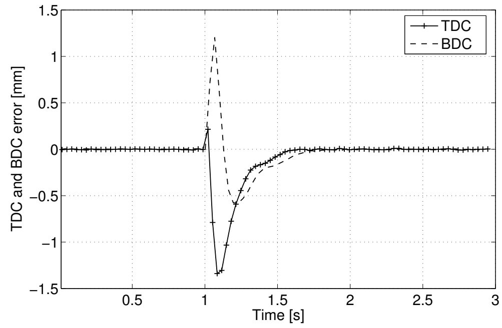
Figure 4 shows the engine dynamic response to a 15 per cent step change in the load with a manually tuned PDF controller, a similar situation to that investigated above. A slightly better response than that obtained with the PI controller can be seen, however the high error peak can not be avoided. This is due to the time delay between the observation of the error and the controller correction, making the feedback control loop too slow to correct for this. In relation to the discussion on controller delay above, it can be seen from the TDC error graph that after two cycles, the error is already nearly 1 mm (the ticks on the graph represent TDC position readings). It is therefore clear that a standard feedback control loop is unable to correct for the initial error peak.

Actuator action

PDF controllers have been reported to give more smooth actuator action than the PI controller. Since the actuator transfers energy to or from the plant, optimising the actuator action to minimise energy consumption is of high importance in many plants. In the free-piston engine, this will mainly be an issue for the fuel mass control variable. In addition to engine fuel consumption, the amount of fuel injected is of high importance for the formation of engine emissions and soot during transient engine operation. Furthermore, the fuel mass control variable relies on there being a sufficient amount of air in the cylinder for the fuel to burn; for high engine loads this control variable will be operating close to saturation.



(a) Engine response to a step increase in load.



(b) Engine response to a step decrease in load.

Fig. 4. Engine response to a 15 per cent step change in load with PDF controller.

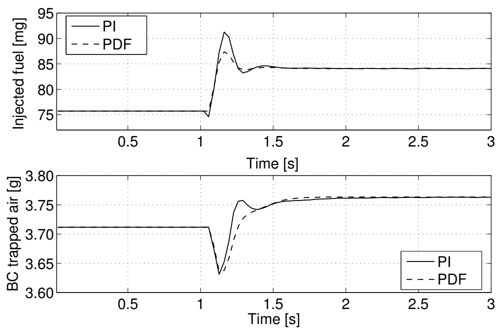


Fig. 5. Actuator action for a 15 per cent step increase in load with PI and PDF controllers.

Figure 5 shows the actuator action for the PI and PDF controllers for a 15 per cent step increase in load with the engine originally operating at 80 % load, similar to the situations above. The undesireable initial control response of the PI controller shortly after the load change can be seen, although the magnitude is low in this case since there is only a limited change in TDC setpoint.

The PDF controller is seen to have a significantly smoother controller response, with a peak overshoot in the fuel mass demand, when compared to the final value, of approximately half the value of the PI controller. For this step increase in load from 80 % to 92 %, the PI controller response does in fact surpass the nominal fuel mass per cycle at full load, which is around 89 mg. This further indicates that the PDF controller is a better option for the free-piston engine plant than the PI controller.

2.5. *Disturbance feedforward*

The above investigation indicated that the time delay between the load disturbance and the corrective action from the controller makes the initial error peak difficult to avoid with feedback control only. This suggests the use of disturbance feedforward, which, provided that the disturbance can be accurately measured, allows corrective action to begin before the error is seen by the controller.

Goodwin et al. [7] stated that "[f]eedforward control is generally agreed to be the single most useful concept in practical control-system design, beyond the use of elementary feedback ideas", but warned that it can be sensitive to modelling errors. The very similar influence of the fuel mass control variable and load force disturbance on the free-piston engine plant does, however, suggest that disturbance feedforward is well suited for the free-piston engine controller.

The measurement of the electric load force in the linear electric generator is trivial, and can be obtained by measuring the current and voltage at the generator output. Information on generator load is likely to be needed anyway for engine optimisation purposes by the supervisory control system, and also to apply adaptive control such as gain scheduling which may be necessary.

It should be noted that although the fuel demand signal can be manipulated immediately following a load change by the feedforward loop, there may still be a delay of up to one full cycle between this change and the corrective action due to the fuel injection occurring only once per engine cycle.

Implementation

Disturbance feedforward in the free-piston engine plant is realised by letting the disturbance D influence the fuel mass and bounce chamber air mass demand signals. Figure 6 illustrates the pseudo-derivative feedback control system with an added feedforward gain term.

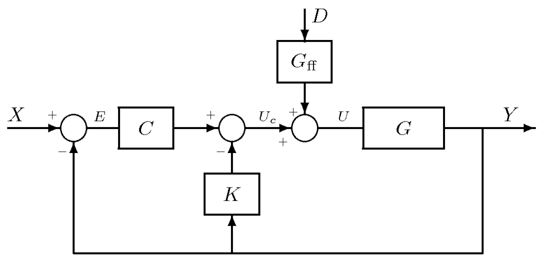


Fig. 6. Pseudo-derivative feedback control system with disturbance feedforward.

The feedforward gain matrix G_{ff} is

$$G_{\text{ff}} = \begin{bmatrix} 0 & 0 & k_1 \\ 0 & 0 & k_2 \\ 0 & 0 & 1 \end{bmatrix},$$

where k_1 and k_2 are constants which regulate the influence of load changes on the fuel mass control variable and the bounce chamber trapped air respectively.

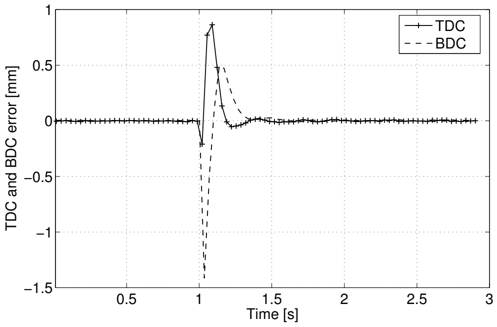
Controller performance

Figure 7 shows the engine response to a 15 % step change in load, similar to that investigated above, with PDF control and disturbance feedforward, both with manually tuned coefficients. A significant improvement when compared to feedback control only can be seen, with a large reduction in the peak error and a reduced settling time.

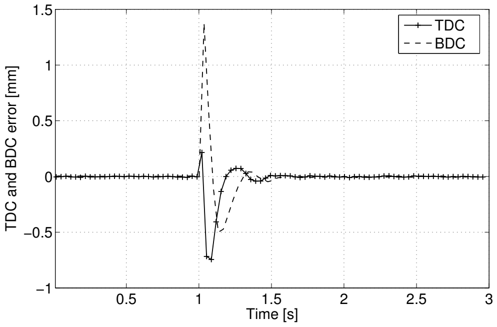
2.6. *Influence of controller choice on engine operational variables*

It has been demonstrated how the different controllers perform in maintaining TDC and BDC positions within certain limits. For the TDC position, the objective of the controller is to maintain the engine compression ratio within a given range in order to (a) ensure fuel autoignition, and (b) avoid excessive in-cylinder gas pressures.

Figure 8 shows the engine compression ratio and peak in-cylinder gas pressure during engine



(a) Engine response to a step increase in load.



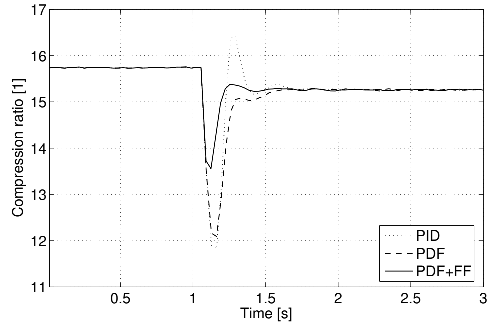
(b) Engine response to a step decrease in load.

Fig. 7. Engine response to a 15 per cent step change in load with PDF control and disturbance feedforward.

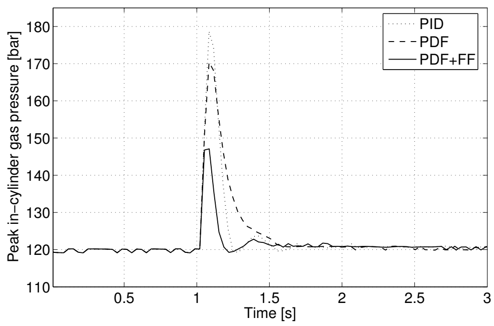
transient operation for a similar load change as that investigated above. The same trends as above can be seen for the different controllers, with the pseudo-derivative feedback controller with disturbance feedforward providing a significantly better response than feedback only control.

Figure 8a shows that the compression ratio drops rapidly following a load increase due to the change in TDC position. This reduction in compression ratio may influence the combustion progress or even lead to failure of the fuel to ignite, which may cause the engine to stop. However, for a turbocharged engine like the one investigated here, even a significant reduction in compression ratio (even down to 10:1) may not represent a problem for engine operation other than leading to a reduced efficiency.

Figure 8b shows the in-cylinder gas pressure following a load decrease. Very high pressure peaks can be seen due to the rapid increase in compression ratio. An increase in compression ratio and peak in-cylinder pressure may, unlike a compression ratio reduction, lead to mechanical damage to the engine and rapid load decreases may therefore be a critical situation. The controller using disturbance feed-



(a) Effect of a 15 % step increase in load on engine compression ratio.



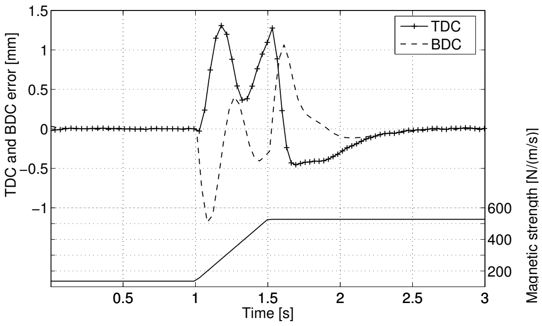
(b) Effect of a 15 % step decrease in load on in-cylinder gas pressure.

Fig. 8. Effects of engine load changes on compression ratio and in-cylinder gas pressure.

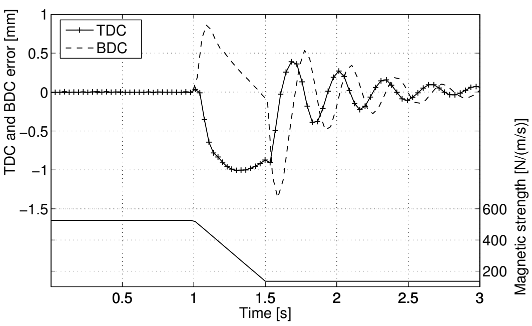
forward is seen to perform significantly better than the feedback-only controllers in this respect, however there is clearly a potential for further improvement. Figure 8b also shows that despite the engine load decrease, there is actually a minor increase in the peak cycle pressure. This is due to the change in compression ratio setpoint and underlines the operational flexibility of the free-piston engine.

2.7. *Ramp load changes*

In the investigations above it was found that even a relatively small load change of 15 % produced significant TDC and BDC position errors due to the harsh characteristics of a step change. In a real system, the load changes may be larger, but will occur over a finite time, which may be possible to influence in the operation of the engine or in the design of the electric system. A better representation of a change in the load demand may be a ramp change between two load levels, where the slope of the ramp determines how rapid the load change is.



(a) Engine response for a linearly increasing load.



(b) Engine response for a linearly decreasing load.

Fig. 9. Engine response to a ramp change in load between 20 % and 100 % over 0.5 s.

Figure 9 shows the engine response to a linear change in engine load between 20 % and 100 % (the load is varied by changing the electric load force) over 0.5 s using the PDF controller with disturbance feedforward. The effects of the high nonlinearities over the load range can be seen, with the engine exhibiting a significantly more oscillatory behaviour at low loads. This indicates that a nonlinear controller should have better performance. The peak error values are seen to be lower for the ramp load changes, as would be expected. With the implementation of load storage devices in the electric circuit, the slope of the ramp can be influenced and the control challenge can thereby be eased.

2.8. *Further improving engine controlled performance*

Clearly, there are other approaches to the control problem that can be investigated, both multivariable control and decentralised, SISO control beyond standard techniques. Here, a basic investigation was presented based on well-known methods, which showed acceptable performance for moderate disturbances. The most obvious method of ensuring stable engine operation for dynamically varying loads is to implement load storage devices (batteries and/or capacitors) in the electric circuit, and thereby smooth the load demands on the engine.

Improvements in the proposed controller are possible. In the above investigations, the controller was manually tuned to achieve acceptable performance. With controller coefficients in both the feedback and disturbance feedforward loops, and two controller loops in the system, the tuning is a multivariable optimisation problem. There is clearly a potential for improvement in the tuning of the controllers. Furthermore, the implementation of gain scheduling is a powerful and uncomplicated method to improve controller performance over the full load range.

3. Cycle-to-cycle variations

Another control-related issue that needs addressing is the smoothness in the operation of the free-piston engine. Cycle-to-cycle variations can occur in for example the amount of fuel injected and the progress of the combustion process, and such variations will have effects on engine performance. Unlike in conventional engines, any such variations will have a direct influence on the following cycle in the free-piston engine, and this type of engine may be more prone to cycle-to-cycle variations and potentially instability if such errors accumulate.

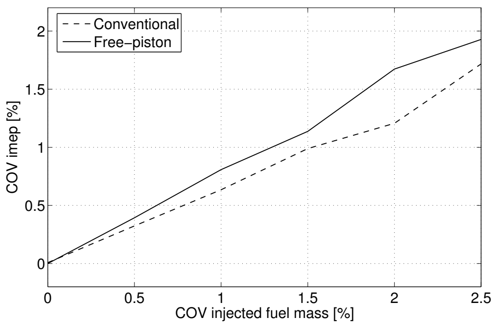
Some experimental reports have described high cycle-to-cycle variations in the operation of bouncing-type free-piston engines. However, it is not clear whether this is due to the free-piston engine characteristics or to inaccuracies in the experimental apparatus. Using a detailed simulation model such as the one developed in this work, the influence of single variables can be investigated without the disturbance of variations in other operational variables.

A common measure of engine operational smoothness is the variation in indicated mean effective pressure and peak in-cylinder pressure between cycles. This is commonly measured in terms of a coefficient of variation, COV, defined as [8]:

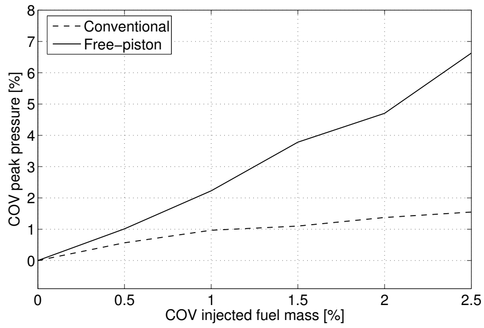
$$\mathrm{COV}_{\mathrm{imep}} = \frac{\sigma_{\mathrm{imep}}}{\overline{\mathrm{imep}}}$$

where imep denotes the engine indicated mean effective pressure, σ_{imep} is the standard deviation of imep and $\overline{\mathrm{imep}}$ is the average value, or mean. It can be defined similarly for other operational variables.

Heywood [8] stated that vehicle driveability problems usually occur for COV_{imep} higher than around 10 per cent.



(a) Effect of variations in fuel mass on indicated mean effective pressure.



(b) Effect of variations in fuel mass on peak in-cylinder gas pressure.

Fig. 10. Effects of variations in injected fuel mass.

Simulations were run for the free-piston engine and for an equivalent conventional engine, using a similar strategy as that described in [1]. The engines were run in steady state operation with no controllers active and random variations in the injected fuel mass was imposed in order to investigate the relation between variations in fuel heat input and engine peak and mean effective pressures.

Figure 10 shows the effects of variations in the injected fuel mass on the indicated mean effective pressure and the peak in-cylinder gas pressure. Figure 10a shows that the indicated mean effective pressure varies with the amount of fuel injected, as one would expect, but that the variations in the free-piston engine are only slightly higher than those of the conventional engine. With modern fuel injection systems, one would expect a variability in the fuel mass in the lower half of the investigated range and operational problems due to such variations are therefore not expected.

Figure 10b shows that the variations in peak in-cylinder gas pressure are significantly higher for the free-piston engine than for the conventional one. This is due to the variations in combustion energy from one cycle influencing the compression ratio for the next. The combination of variations in both compression ratio and injected fuel mass gives significantly higher peak pressure variations in the free-piston engine.

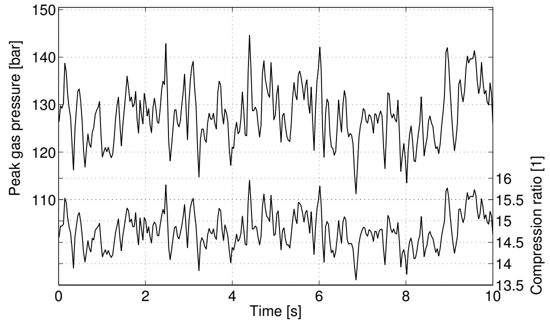


Fig. 11. Example of operational sequence for the free-piston engine.

Figure 11 shows an example of a operational sequence for the free-piston engine with a COV in injected fuel mass of 2 %. It is seen that even with the relative high COV used here, the peak pressures obtained are not critical for the engine. The figure also shows the variations in engine compression ratio, which means that there is some variation in engine TDC position. The variation is low but may influence the engine controller and this should therefore be taken into account in the design of the control system.

4. Conclusions

The control of piston dynamics in a free-piston engine generator was investigated using a full-cycle simulation model. A proposed control strategy, based on standard feedback ideas, was found to give adequate performance for moderate load changes. Pseudo-derivative feedback control was found to be more suitable for the free-piston engine than conventional PID control, and it was shown how the use of disturbance feedforward is crucial to achieve a satisfactory response for rapidly changing loads. The use of energy storage devices in the electric circuit was discussed, and the significantly better controller performance for less harsh load changes was shown.

Finally, the effects of cycle-to-cycle variations on engine operation was studied. It was found that the variations in peak in-cylinder gas pressure is significantly higher than that in conventional engines. Although the magnitude of the variations in the current engine was not at a level which would be critical for engine operation, the issue of such variations should be taken into account in the design of free-piston engines.

It is clear that much work remains before the free-piston engine can provide a realistic alternative to conventional technology. This paper has, together with the accompanying paper, discussed the basic features of the free-piston engine plant and its controllability. Further research into free-piston engine control is required to solve the significant control challenges associated with such engines.

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