Free-Piston Diesel Engine Timing and Control – Towards Electronic Cam- and Crankshaft

Tor A. Johansen, Olav Egeland, Erling Aa. Johannessen and Rolf Kvamsdal

Abstract— **The free-piston diesel engine replaces the crankshaft of the traditional diesel engine with a power turbine to convert energy from the exhaust gas. Hence, the pistons move freely in the cylinder, mainly influenced by pressure and friction forces. Instead, the motion of the pistons are controlled by an electronic control system –** ***an electronic crankshaft*****. In addition, the camshaft is replaced by an electronic timing system that triggers and actuates the motion of the valves –** ***an electronic camshaft*****. In the present paper we describe a hierarchical multi-rate electronic control system developed for an experimental engine, focusing on piston motion parameter estimation, valve and injector timing, and a piston motion control system. Experimental results with a full scale experimental test engine are included. The results show the effectiveness of the developed control strategy.**

I. Introduction

The free-piston diesel engine concept was developed by Pescara, combining a diesel process with a freely moving piston in the cylinder and a power turbine. Engines of various size were manifactured between 1930-1960 by GM, Ford, Renault, Junker, Sigma and others [1], [2]. Despite the potential advantages of the Pescara process (which includes low weight, simple mechanical design and high thermal efficiency), the mechanical construction and control mechanisms had weaknesses leading to low reliability. Partload operation of the engine was a known difficulty and the partload thermal efficiency was generally poor. Furthermore, lack of suitable materials imposed limitations on the maximum temperatures and thus the thermal efficiency, which was up to about 40 % in these older engines. A good review of the free-piston diesel engine concept and description of some of the older engines can be found in [3], [4].

Today, the material constraints are signficantly less, and the modern computer control technology and high-precision common-rail diesel injection systems allows the implementation of an accurate and reliable electronic control system. This was the motivation for Kværner ASA, who is developing a modern high-speed free-piston diesel engine (KLC) aimed at marine applications as an alternative to both gas turbines and traditional diesel engines. The net output of an 8 cylinder engine is about 8 MW with a thermal efficiency of about 50 %.

The contribution of the present paper is a presentation of the novel KLC engine control system, which is fundamentally different from a traditional diesel engine control system because a traditional diesel engine has mechanical cam- and crankshafts to facilitate timing and constrain the motion of the pistons. Compared to older free-piston diesel engines, computer control offers additional flexibility and degrees of freedom that provides an opportunity to optimize the operation of the engine. The KLC engine control system is experimentally tested on a 1 MW (net power) test cylinder built by Kværner ASA. In contrast to the original free-piston diesel engine with an opposed piston arrangement where two opposed pistons are moving synchronously in the same cylinder, the Kværner design is a more conventional two-stroke arrangement with with a single piston per cylinder and a standard diesel cylinder top with a centered injector and four poppet exhaust valves per cylinder. This leads to further requirements for control than the traditional opposed piston design, since the pistons must be synchronized by automatic control. The companion paper [5] describes a mathematical model of the engine dynamics, and contains a dynamic analysis leading to a piston motion control system structure. The present paper focuses on the details of the engine control system, emphasizing timing, estimation, signal processing and control. Moreover, further details of the implementation and a wider set of experimental results are included here. The main ideas are described in the patent [6], also for an opposed piston arrangement. A description of some similar ideas for a different type of free-piston diesel engine were recently presented in [7], but without any details or experimental results. The advantages of a camless engine in terms of fuel economy and emissions are well known from conventional engines with a crankshaft, e.g. [8], [9], [10], [11].

The outline of the paper is as follows: The operating principles and design of the KLC engine is described in section II. The control objectives and the control structure are presented in section III. The engine control system is then described, focusing on estimation (section IV), timing (section V), piston motion control (section VI), and supervisory control and optimization (section VII). The hardware and software used for the experimental evaluation on the KLC test cylinder is briefly described in section VIII, while section IX contain some experimental results. Conclusions are provided in section X.

II. Description and operating principles of the engine

Figure 1 provides a sketch of a multi-cylinder free-piston diesel engine. The machine consists of a free-piston gas

Tor A. Johansen and Olav Egeland are with the Department of Engineering Cybernetics, Norwegian University of Science and Technology, N-7491 Trondheim, Norway. Erling Johannessen is currently with Rolls-Royce Marine AS, Postboks 924, 5808 Bergen, Norway. Rolf Kvamsdal is with Kværner ASA, Technology Development, Postboks 169, N-1325 Lysaker, Norway.

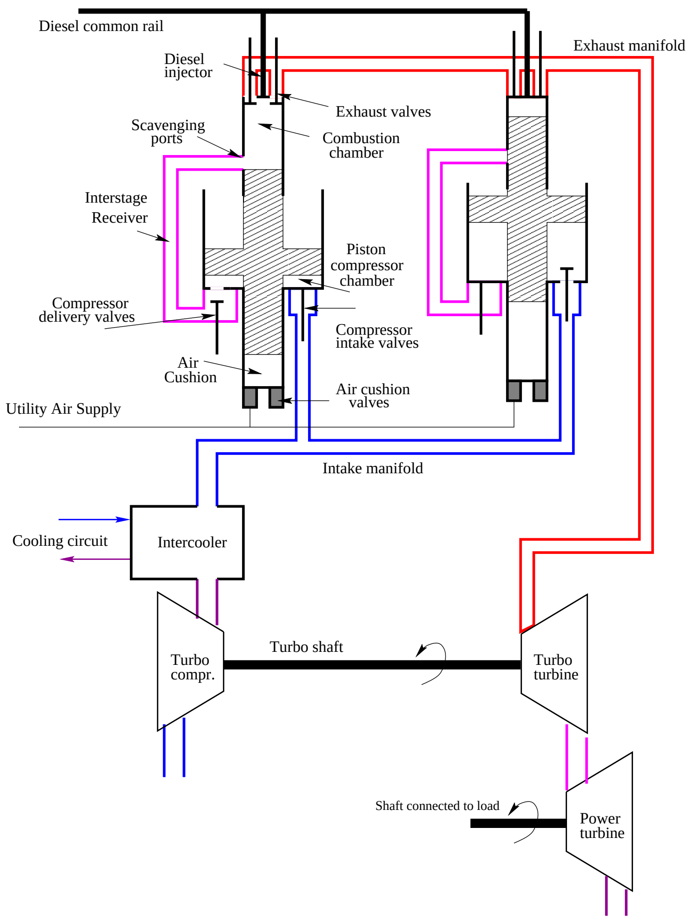

Fig. 1. Sketch of multi-cylinder free-piston engine.

generator, a turbocharger and a power turbine. The air is compressed in 3 stages; in the turbo compressor, the piston compressor chamber (supercharger), and finally in the combustion chamber. After combustion, the exhaust gas expands in 3 stages; first in the combustion chamber just until combustion has completed, and then in the turbo turbine and eventually in the power turbine.

The two-stroke diesel cycle, seen from the intake manifold to the exhaust manifold, can be described as follows (see also [12]). *Compression:* The air in the combustion chamber is being compressed after the scavenging ports are closed above the bottom dead center when the piston moves towards its top dead center. Near the top dead center, the temperature of the compressed air is high enough for auto-ignition and high pressure diesel is injected into the combustion chamber through a nozzle and being combusted. *Air intake:* During the upwards motion of the piston, fresh air is pulled into the compressor chamber from the intake manifold through passive suction valves. *Expansion:* The high pressure in the combustion chamber will now make the piston move downwards and the exhaust gas expands. *Scavenging:* During this downwards motion, the combustion eventually finishes and the hydraulically actuated exhaust valves opens at high pressure to deliver high-energy exhaust gas through the exhaust manifold to the turbines. When the piston is at some distance from its nominal bottom dead center, scavenging ports are uncovered and the fresh air in the interstage receiver flushes the combustion chamber and replaces the exhaust gas. *Compression in piston compressor:* The interstage receiver is at the same time filled by high-pressure air that is compressed in the piston compressor chamber during the downwards motion of the piston and delivered through some hydraulically actuated valves connecting the piston compressor chamber and interstage receiver. The high pressure in the compressor chamber and air cushion will now make the piston move upwards from its bottom dead center and the cycle repeats. The cycle can be illustrated with the experimental pV-diagrams in Figure 2 corresponding to about 20 % load, for the combustion chamber, piston compressor and air cushion. Note that the air is in principle being compressed in three stages: First in the turbo compressor (in the illustrated cycle the turbo compressor was not effective), then in the piston compressor from about 1 bar to about 6 bar, and finally in the combustion chamber from about 6 bar to about 45 bar. The diesel combustion brings the peak pressure to about 125 bar. It is seen that very little net work is consumed by the air cushion. It essentially works as a bounce chamber for the purpose of control. The air cushion is equipped with an active inlet valve connected to a utility air reservior, and an active outlet valve.

The main events during the two-stroke free-piston diesel cycle are illustrated in Figure 3:

- The piston top dead center (TDC) position is the extreme upper postion for the given cycle.
- The piston bottom dead center (BDC) is the extreme lower position for the given cycle.
- The start of combustion (CS) and combustion completed (CC).
- The compressor develivery valves opening position (CVO) and closing position (CVC).
- The exhaust valves opening position (EVO) and closing position (EVC).
- The position where the scavenging ports are being uncovered (IVO) and covered (IVC).

Notice that a unique feature of the free-piston engine is that the piston positions when these events occur will be different from one cycle to the next (except IVO and IVC that are fixed by the mechanical design), and that these events are triggered by the engine control system at each cycle, and thus provides additional degrees of freedom that are useful to optimize and adapt the engine operation though parameters of the control system rather than parameters of the mechanical design. The free-piston diesel engine differs from a traditional diesel engine in several other important ways as well:

- The pistons move freely, mainly influenced by pressure and friction forces since there is no crankshaft connected to the piston.
- The valves and injectors are controlled electronically, since there is no camshaft connecting them to the pistons.
- At the combustion chamber exit, the exhaust gas has much higher temperature and pressure than in a traditional engine, simply because the mechanical power is produced

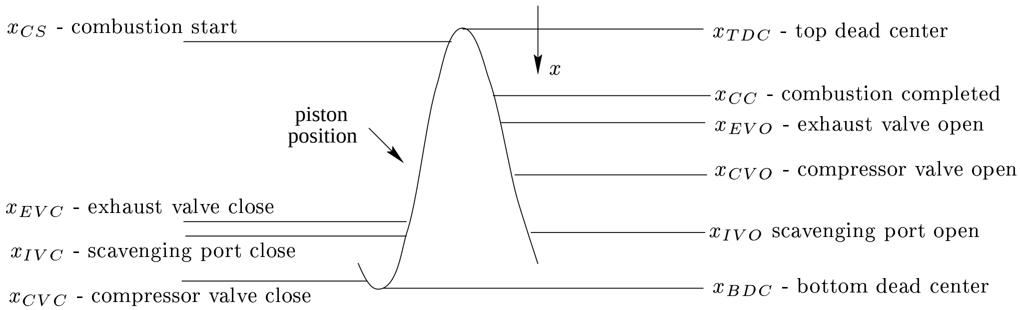

Fig. 3. Main events during the two-stroke free-piston diesel cycle.

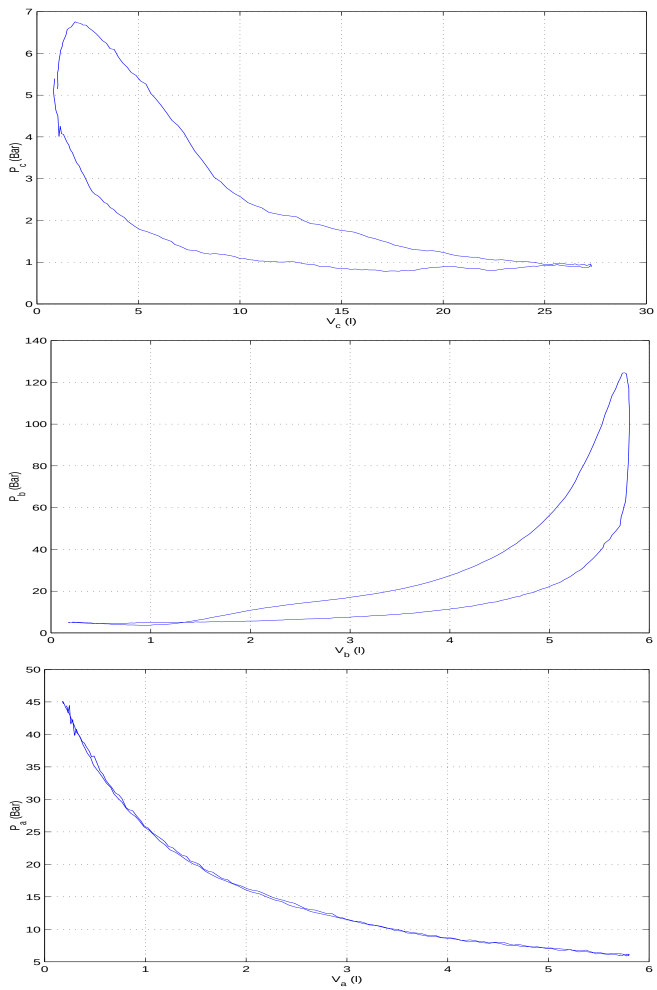

Fig. 2. Experimental pV-diagrams corresponding to about 20 % load: Piston compressor, combustion chamber and air cushion.

directly from the exhaust gas in the power turbine.

Since there is no (mechanical) crankshaft that guides the piston motion, the free-piston diesel engine requires a piston motion control system (electronic crankshaft) in order to control the piston motion. Furthermore, since there is no (mechanical) camshaft that moves the valves, the free-piston diesel engine requires a timing control system (electronic camshaft) to time and control the valves and injectors.

A single cylinder of the diesel combustion unit has been tested in full size. The test unit, which has a mass of about 800 kg (the free piston is about 100 kg), is placed on a foundation. A picture of the test cylinder is shown in Figure 4.

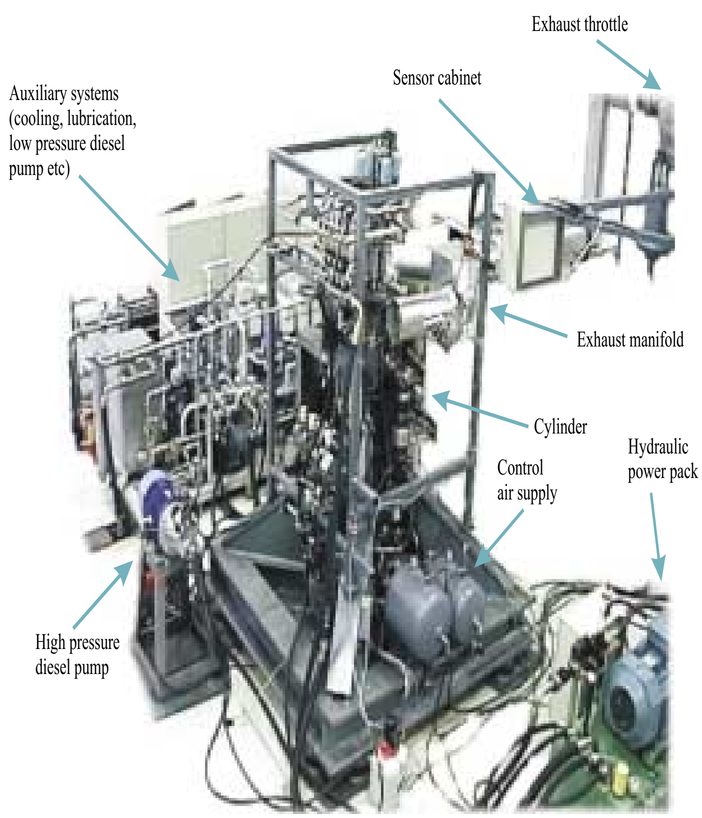

Fig. 4. Picture of test cylinder test unit.

The single cylinder test unit working point is defined by the charge air pressure before the compressor chamber and the exhaust pressure, cf. Figure 5. The charge air pressure is controlled by a valve on the inlet pipe, after the intercooler. The exhaust pressure is controlled by a valve on the exhaust outlet pipe. By adjusting these pressures different compressor and turbine maps may be simulated. In order to remove heat from the hot exhaust gas (up to 900 deg C at full load) in the test rig, water is injected into the exhaust in order to control the exhaust temperature in the receiver.

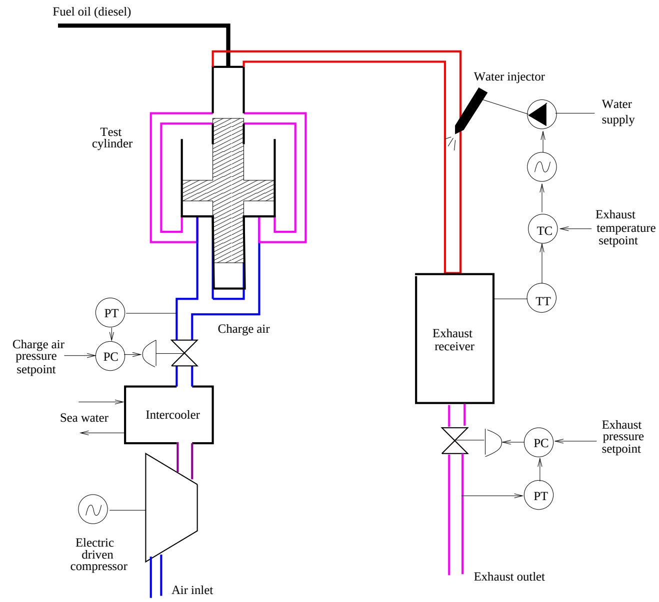

Fig. 5. Overall setup for single cylinder test unit.

The necessary control action for in-cycle control is provided by a set of actuators:

- 4 exhaust poppet valves in the cylinder head are actuated hydraulically. These valves control the exhaust discharge.The exhaust valves open early in the expansion stroke, with a considerable differential pressure for the valve and actuator to overcome. To provide the required force, a combination of hydraulic action and release of energy stored in a spring is used.
- 2 compressor chamber outlet poppet valves are also controlled hydraulically. The hydraulic control emulates the action of a check valve, i.e. air flows when a positive pressure difference exists over the valve.
- 4 cartridge air valves with hydraulic pilot stage control air flow into and out of the air cushion chamber. These valves are used to control the mass of air in the air cushion both during normal operation and in the starting sequence.
- A fuel injector centrally located in the cylinder head is controlled by a solenoid. The fuel injection pressure is adjustable between about 800 and 1600 bar.

A set of auxiliary equipment provides the working condition of the single cylinder test unit. The equipment is placed on a common skid:

- Fresh water is circulated to cool the cylinder unit, i.e. liners and the cylinder head.
- Oil is circulated to cool the piston crown from the inside of the piston.
- Fuel oil is circulated to cool the fuel injector nozzle.

Heat is removed from these circuits through exchange with sea water over plate heat exchangers. The charge air intercooler is realized using an ordinary plate heat exchanger placed on the skid. A tank provides the reservoir for cylinder lubrication and a pump injects lubrication oil into four drillings on the circumference of the cylinder. The combustion fuel is drawn from a tank and supplied to the high pressure pump at around 5 bar. Control of the auxiliaries' skid is handled by commercial PLC with a PC based operator interface.

A considerable number of sensors are installed to provide feedback for closed loop control, for process calculations and for safety monitoring.

Pressure is measured both by traditional transmitters and piezo-electric transducers. The latter provides the bandwidth necessary to record pressure variations over the cycle. However, these transducers have a considerable temperature drift that must be compensated. That is implemented by installing an ordinary transmitter in the same location. Pressure sensors with different frequency characteristics is thus used to give combined measurement. Pressure is measured after electrically driven air compressors, before air inlet to the test cylinder, in the compressor chamber, in the scavenging duct, in the combustion chamber, in the exhaust pipes and the receiver, and in the air cushion.

Temperature is measured by a combination of Pt100 elements and thermocouples. The following main points are instrumented: combustion air (after intercooler, before inlet to test cylinder), scavenging duct, exhaust (in manifold, before and after water injection, in receiver and after silencers), and air cushion. In addition, a number of temperatures are measured on the auxiliary circuits.

Linear displacement transducers (LVDT, linear variable differential transformer) are used for measurement of piston position (2 off, range 250 mm), poppet valve position (6 off, range 15 mm), and fuel injector needle lift (1 off, range 0.45 mm).

III. Control Objective and Structure

The engine control system can be organized in a control hierarchy, see Figure 6. At the most basic level there is *timing*, i.e. injection of diesel and opening and closing of the valves at commanded points during each cycle. This is in itself a nontrivial problem since the actuators and injector have significant time delays corresponding to a large fraction of the cycle period of this high-speed engine. At the next level there is *piston motion control* where commands are given to the timing subsystems in order to control the piston motion. Furthermore, at the upper level there is *supervisory control and optimization* where the objective is to perform logic control, adapt the operating characteristic of the engine to the load, and dynamically optimize the operation of the engine according to some criteria including thermal efficiency, within constraints on emissions and thermal load on the engine. In addition, synchronization of the motion of multiple pistons is required to keep vibrations at a satisfactory level. This will be achieved by commanding setpoints to the piston motion control subsystems, and by actuating on the turbomachinery. Only the most fundamental aspects and opportunities of the supervisory control and optimization system will be discussed in detail in this paper.

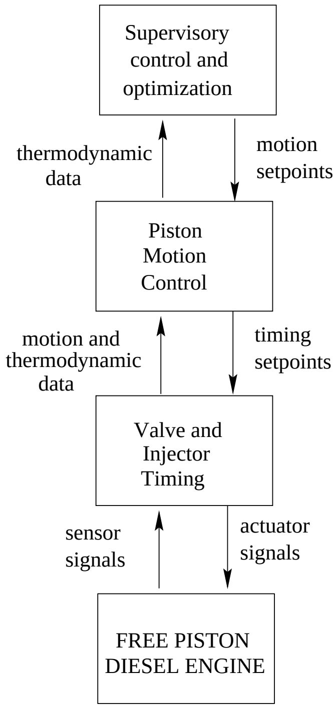

Fig. 6. Control system hierarchy.

A. Timing objectives

The objectives of the different timing subsystems are as follows:
• *Diesel injection timing subsystem:* Transmit signals to the diesel injection electronics such that for each individual cycle the specified amount of diesel is being injected at a specified point in time prior to TDC.
• *Exhaust valve timing subsystem:* Transmit signals to the hydraulic actuators such that the exhaust valves open when the piston is at a specified position in the expansion stroke, and such that they close when the piston is at a specified position in the compression stroke.
• *Compressor delivery valve timing subsystem:* Transmit signals to the hydraulic actuators such that the compressor delivery valves open during downwards piston motion at the point in time when the piston compressor pressure becomes higher than the interstage receiver pressure, and such that the compressor delivery valves close when the compressor chamber pressure eventually drops below the interstage receiver pressure again.
• *Air cushion valve timing subsystem:* Modulate the signals to the air cushion valves in order to avoid having the air cushion outlet valve open when the air cushion pressure is less than atmpospheric, and avoid having the air cushion inlet valve open when the air cushion pressure is higher than the utility air reservior pressure.

Any inaccuracies in the timing will lead to suboptimal operation of the engine. Furthermore, timing inaccuracies will lead to a disturbance on the piston force balance such that the piston motion will vary from cycle to cycle. For example, opening the exhaust valves slightly too late will lead to a higher pressure in the combustion chamber over a part of the specific cycle, which again leads to additional work being made on the piston during the expansion. The consequences of this is an undesired increase in the length of the stroke. Such cycle-to-cycle variations in the motion of the piston at stationary running will introduce unneccesary mechanical vibrations in addition to pressure disturbances in the intake and exhaust manifold that will influence the operation, cost and life-cycle of the turbomachinery. It will also lead to loss of efficiency and, if sufficiently large, malfunction of the engine. The cycle-to-cycle variability in stroke length should be less than 2 mm out of a stroke of about 200 mm.

B. Piston motion control objectives

The piston motion control system may be referred to as an *electronic crankshaft*. The objectives of the piston motion control subsystems are as follows:
• *TDC position control:* Control the TDC position to a specific value (which may depend on the operating point of the engine).
• *BDC position control:* Control the BDC position to a specific value (which may depend on the operating point of the engine).
• *Synchronization:* Control the phase shift between multiple pistons to specified phase angles, in order to minimize mechanical vibrations and pressure variations on the turbine blades.

There are hard constraints on the motion of the piston due to mechanical stop if the pistons hits the top or bottom of the cylinder. The absolute tolerance here is typically less than 8 mm, depending on the operating point of the engine.

The motion of the piston can only be influenced through the mechanical work made by the gas pressures on the piston. These pressures can be indirectly influenced by controlling the mass flow through the engine by actively controlling the point when the hydraulically actuated exhaust and compressor delivery valves open and close. Furthermore, the pressures can be influenced by the amount of diesel being injected into the combustion chamber and by taking air in or out of the air cushion. In fact, the main purpose of the air cushion is to provide a mechanism for control, i.e. balancing the pressure forces on the piston.

The consequences of a malfunction of this motion control system are the following:
• If the TDC position is too low, there will be insufficient compression, and the ignition delay will incerase until the point when temperature is too low for ignition and the piston will stop moving after a few cycles.
• If the TDC position is too high, the compression will be too high, and the combustion chamber peak temperature and pressure will exceed the tolerances of the mechanical design and possible damange the engine. Moreover, emissions may be too high due to the high peak temperature during combustion.
• In addition, and in particular during idle running, the distance from the nominal TDC to the cylinder top is fairly small (about 6 mm). Thus, if the actual TDC exceeds the

nominal TDC by this amount, there will be mechanical contact between the piston and the cylinder top. The consequences of this will depend on the kinetic energy of the piston, the mechanical design and stop springs, but any such crash is clearly undesirable.

- The BDC position is normally required to be as small as possible, in order to make the piston compressor deliver as much air as possible. In order to get sufficient efficiency out of the piston compressor, the nominal BDC is less than 10 mm from the mechanical contact point.
- If the BDC position is too high, too little air will be delivered to the interstage duct and combustion chamber. This is both due to the operation of the piston compressor and because the scavenging ports will be uncovered less and over a shorter time interval. Both these phenomena contribute to reducing the mass flow through the engine. The consequence of reduced mass flow is a lower air-to-fuel ratio which leads to too high exhaust temperature and poor combustion, both being unacceptable from the point of views of emissions and thermal load.

C. Supervisory control and optimization objectives

Some of the key objectives are the following:

- Keep the engine power output at the commanded value given by the operator (for example when the power turbine shaft is connected to propellers), or the power being demanded by the load (for example when the power turbine shaft is being connected to an electric generator on a grid).
- Provide mechanisms for engine starting and stopping, fault detection and monitoring.
- Ensure good conditions for combustions (high enough temperature when combustion starts, sufficient air-to-fuel ratio etc.) and keep the emissions within specified constraints.
- Keep the thermal load on the cylinder top, exhaust manifold and turbines within specified constraints.
- Maximize the thermal efficiency, both at full load and part load operation.

D. Control structure

The control structure that has been implemented to achieve the above mentioned objectives is shown in Figure 7. Further motivation for this structure can be found in [5], which contains a dynamic model and analysis of the engine dynamics to support the design of the piston motion control structure.

A fundamental principle underlying the design of the engine control system is that the timing subsystems are concerned only with the execution of the current cycle, while the piston motion control subsystems are concerned only with controlling the behaviour from one cycle to the next, and not internally within the cycle. The supervisory control and optimization subsystems operates even on a slower time scale corresponding to operating point changes. This corresponds to a separation of time-scales between higher and lower level control task, which is a well known control principle. In this case, notice that the piston motion control subsystem must necessarily operate on variables that are invariant over the cycle (or changes slowly from one cycle to the next). The reason for this is that due to natural constraints on the timing of events through the cycle together with the large dead-time of the actuators, deviations that are detected within a single cycle are difficult to compensate for until the next cycle.

IV. Estimators

A. Piston motion parameters estimator

The purpose of this estimator is to provide parameters that characterizes the piston motion over a single full cycle. The basic parameters are the time instants when the TDC and BDC are reached and the BDC and TDC positions. These parameters are being used in the piston motion control system as feedback variables, being monitored in the supervisory control and optimization system, as well as being used in the timing control to predict the piston motion within a cycle in order to compensate for actuator dead-times using a model-based approach.

As an approximation, the piston motion over a full cycle is assumed to be piecewise sinusoidal (one sinusoidal piece for downwards motion, and one for upwards motion). The sinusoid describing the downwards part of the cycle has shorter period than the upwards part because diesel is injected near the TDC and the average engine pressure is therefore higher during downwards motion than upwards motion.

A block diagram with the main components of a piston dead center detection algorithm is shown in Figure 8. The piston position is $x(t)$, its sampled signal is $\tilde{x}[k]$, $x[k]$ is the lowpass filtered signal and $v[k] = (x[k] - x[k-1])/\Delta$. The sampling period is $\Delta = 200\mu s$. Since the TDC and BDC positions change from one cycle to the next, the detection algorithm is based on detecting when the piston velocity crosses zero. Due to significant noise in the position sensor, the lowpass filtering is necessary. This filtering, combined with the dynamics of the sensor itself, contributes to a phase shift in the detection signal $d[k]$. This signal is $d[k] = 1$ for the particular sample when the sign of $v[k-1]$ is different from the sign of $v[k]$, and $d[k] = 0$ any other sample. Thus

$$
\begin{aligned}
t_{tdc}[k] &= \begin{cases} t[k] - \tau_f, & \text{if } d[k] = 1 \text{ and } x[k] \leq \frac{x_s}{2} \\ t_{tdc}[k-1], & \text{otherwise} \end{cases} \\
t_{bdc}[k] &= \begin{cases} t[k] - \tau_f, & \text{if } d[k] = 1 \text{ and } x[k] > \frac{x_s}{2} \\ t_{bdc}[k-1], & \text{otherwise} \end{cases}
\end{aligned}
$$

where the time delay due to filtering is

$$
\begin{aligned}
\tau_f &= \Delta + \frac{T_{cycle}}{2\pi} \tan^{-1}\left(\frac{1}{T_{cycle} f_1}\right) \\
&\quad + \frac{T_{cycle}}{2\pi} \tan^{-1}\left(\frac{1}{T_{cycle} f_2}\right) \qquad (1)
\end{aligned}
$$

where $f_1 = 200$ Hz and $f_2 = 500$ Hz. Here, T_{cycle} is the cycle period, x_s is the stroke length, and the dynamics

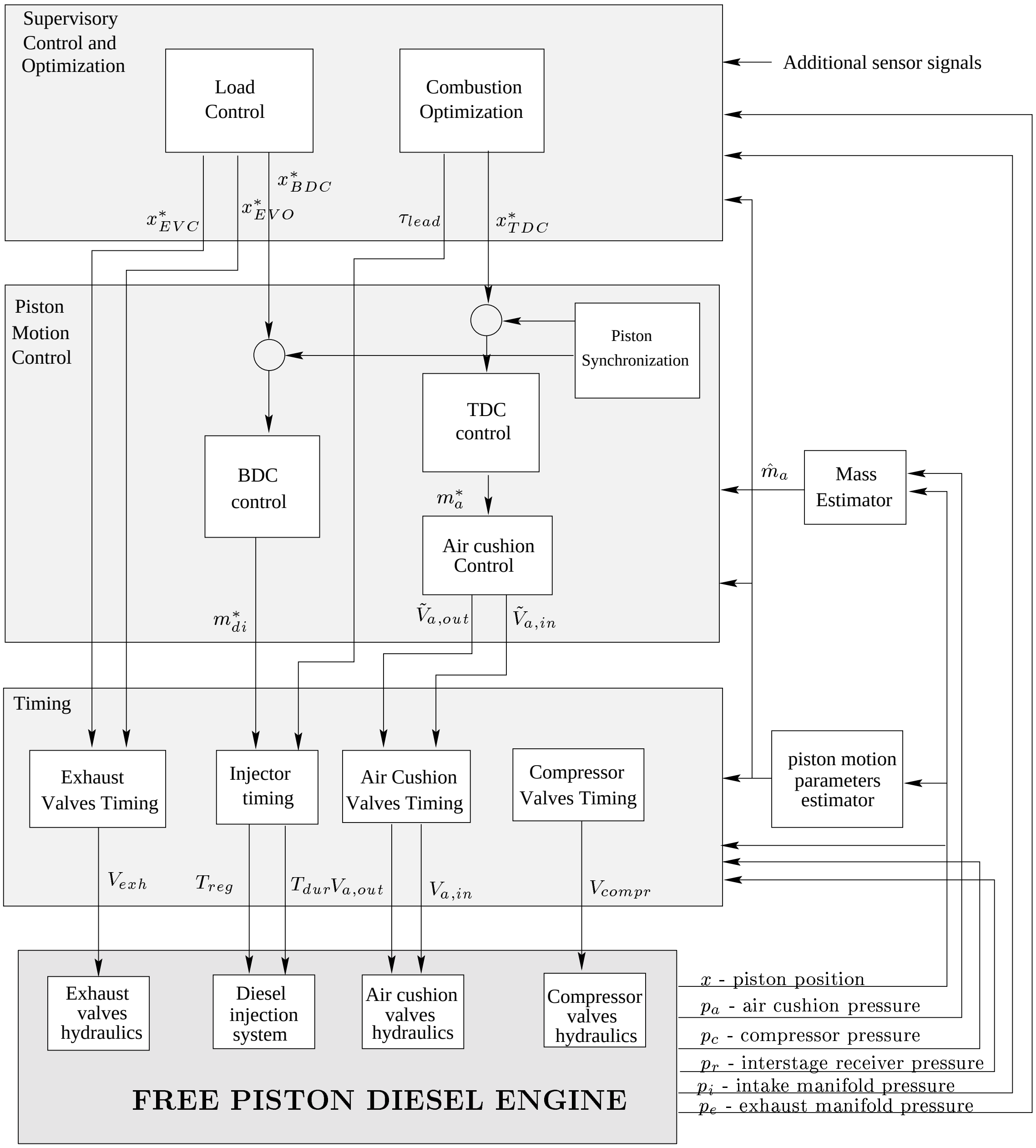

Fig. 7. Control system structure.

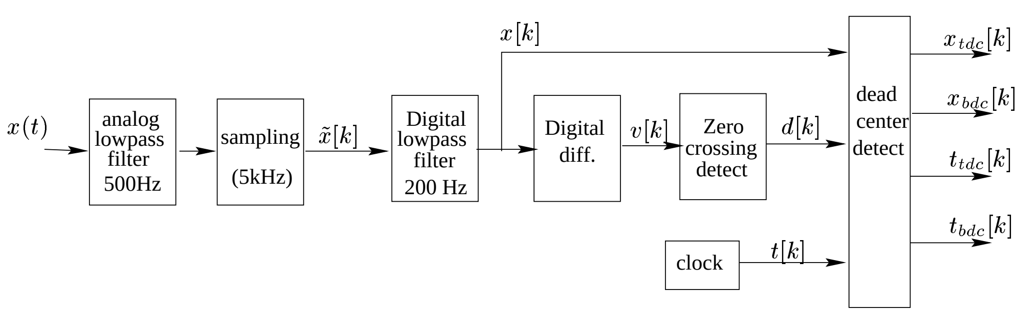

Fig. 8. Algorithm for robust detection of piston dead centers.

in the sensor itself is neglected. The position signal $x[k]$ used to compute the TDC and BDC positions has roughly the same phase shift as $d[k]$, but its amplitude has been damped. Therefore,

$$x_{tdc}[k] = \begin{cases} x[k] - x_{corr}, & \text{if } d[k]=1 \text{ and } x[k] \le \frac{x_s}{2} \\ x_{tdc}[k-1], & \text{otherwise} \end{cases}$$
$$x_{bdc}[k] = \begin{cases} x[k] + x_{corr}, & \text{if } d[k]=1 \text{ and } x[k] > \frac{x_s}{2} \\ x_{bdc}[k-1], & \text{otherwise} \end{cases}$$

Here the term x_{corr} that compensates for the amplitude damping of the filter is given by

$$x_{corr} = \frac{x_s}{2}\left(\sqrt{1+\left(\frac{1}{T_{cycle}f_1}\right)^2}-1\right) \cdot \left(\sqrt{1+\left(\frac{1}{T_{cycle}f_2}\right)^2}-1\right) \quad (2)$$

The following derived parameters are computed

$$T^{up}_{cycle} = t_{tdc} - t_{bdc} \quad (3)$$
$$T^{down}_{cycle} = t_{bdc} - t_{tdc} \quad (4)$$

where it is required that $t_{tdc} > t_{bdc}$ in (3) and $t_{tdc} < t_{bdc}$ in (4).

One of the main purposes of the piston motion parameter estimator is that it provides a model to be used for prediction of the piston motion over periods of time τ corresponding to deadtimes of the actuators. An underlying assumption is that the piston motion parameters do not change much from one cycle to the next. A prediction of the piston position at a time shift τ into the future is given by

$$\hat{x}(t+\tau|t) = \begin{cases} \overline{x} + \underline{x}\sin\left(\omega_{down}(t+\tau-t_{tdc}) - \frac{\pi}{2}\right), \\ \quad \text{if } t \in \left[t_{tdc}, t_{tdc} + \frac{\pi}{\omega_{down}} - \tau\right) \\ \overline{x} + \underline{x}\sin\left(\omega_{up}(t+\tau+\frac{\pi}{\omega_{down}} - t_{tdc}) + \frac{\pi}{2}\right), \\ \quad \text{if } t \in \left[t_{tdc} + \frac{\pi}{\omega_{down}} - \tau, t_{tdc} + \frac{\pi}{\omega_{down}}\right) \\ \overline{x} + \underline{x}\sin\left(\omega_{up}(t+\tau-t_{bdc}) + \frac{\pi}{2}\right), \\ \quad \text{if } t \in \left[t_{bdc}, t_{bdc} + \frac{\pi}{\omega_{up}} - \tau\right) \\ \overline{x} + \underline{x}\sin\left(\omega_{down}(t+\tau+\frac{\pi}{\omega_{up}} - t_{bdc}) - \frac{\pi}{2}\right), \\ \quad \text{if } t \in \left[t_{bdc} + \frac{\pi}{\omega_{up}} - \tau, t_{bdc} + \frac{\pi}{\omega_{up}}\right) \end{cases} \quad (5)$$

where

$$\underline{x} = (x_{bdc} - x_{tdc})/2$$
$$\overline{x} = (x_{bdc} + x_{tdc})/2$$

and $\omega_{up} = 1/T^{up}_{cycle}$, $\omega_{down} = 1/T^{down}_{cycle}$.

In addition, the estimates of T^{up}_{cycle} and T^{down}_{cycle} are enhanced by a feedback from the residual of the predicted piston position (5) with $\tau = 9\ ms$. Since the residual is significantly influenced by the 2nd harmonic of the piston motion, this component is subtracted from the residuals before it is fed back, cf. Figure 9. The parameter vector θ contains the basic parameters t_{tdc}, t_{bdc}, x_{bdc}, x_{tdc} and the derived parameters T^{up}_{cycle} and T^{down}_{cycle}.

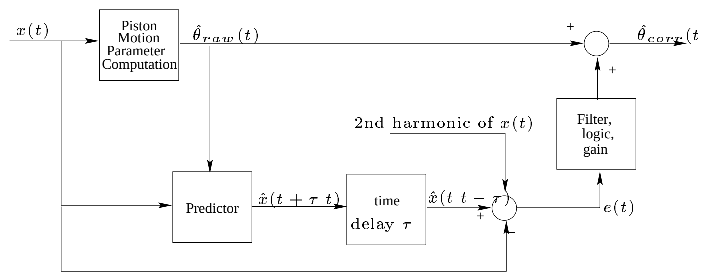

Fig. 9. Structure of piston motion parameter estimator.

B. Air cushion mass estimator

The mass in the air cushion is estimated using the isentropic relation

$$p_a V_a^\kappa = p_{a,avg} V^\kappa_{a,avg} \quad (6)$$

and the ideal gas law

$$p_{a,avg} V_{a,avg} = m_a R T_{a,avg} \quad (7)$$

V is volume, p is pressure, T is absolute temperature, m is mass, R is the gas constant and $\kappa = c_v/c_p$. The index a refers to the air cushion chamber, while avg refers to an average state (averaged over a cycle). Using these equations we arrive at

$$\hat{m}_a = \frac{p_a V_a^\kappa}{R T_{a,avg} V^{\kappa-1}_{a,avg}} \quad (8)$$

In this estimate we use measurements of the air cushion pressure p_a, and measurements of the piston position x is used to compute the volume $V_a = A_a(x + x_{0a})$. It is assumed that the average temperature $T_{a,avg}$ and volume $V_{a,avg}$ are known, but this is not a critical assumption since a significant offset on this estimate can be tolerated.

V. Timing

A description of the basic algorithms underlying the timing subsystems is given below.

A. Injector timing

The desired amount of diesel to inject is converted through a lookup table with interpolation into the duration of pulses T_{reg} and T_{dur} to be given to the diesel injection electronics. This relies on a measurement of the common rail diesel pressure. Accurate injection of diesel is particularly important in the free-piston diesel engine because any deviation will influence the length of the stroke since there is no crankshaft the can take up or provide the extra work. Any offset in the injector characteristic is easily compensated for in the control system, but any unpredictable cycle-to-cycle variations should be less than 5 mm^3/stroke at idle (nominal amount of injected diesel at idle is about 200 mm^3/stroke). At higher load, a proportionally larger variations are tolerated.

The higher level control system specifies a certain point in time prior to the TDC when the combustion should start. The injector timing subsystem must therefore be able to predict the time when the TDC is reached. In addition to the combustion lead time τ_{lead}, one must also take into account the injector delay τ_{inj} (time required to build up the solenoid current) and the ignition delay τ_{ign} (which depends on air temperature and other parameters). Thus, the time from the injector control pulse T_{dur} is initiated until the TDC is reached should be $\tau_i = \tau_{inj} + \tau_{ign} + \tau_{lead}$. Typical values are $\tau_{inj} = 1.0\ ms$, $\tau_{ign} = 1.0\ ms$ and $\tau_{lead} = 1.5\ ms$.

The piston position is predicted according to

$$\hat{x}(t+\tau_i|t) \quad = \quad x(t) + \int_t^{t+\tau_i} v(\sigma) d\sigma \qquad (9)$$

and the injector pulse is initiated at the time t when $\hat{x}(t+\tau_i|t) \approx x^*_{TDC}$. Using the piston motion parameter estimator, we get the piston motion model

$$v(t) \quad = \quad \frac{x_s \pi}{T^{up}_{cycle}} \sin(2\pi t / T^{up}_{cycle} + \phi) \qquad (10)$$

which is valid for the upwards part of the piston motion. Here, x_s and T^{up}_{cycle} (the cycle time corresponding to the upward segment of the piston motion) are estimated online by the piston motion parameter estimaor, and ϕ is a phase angle. Thus, the injector pulse is initiated at the time t when

$$x^*_{TDC} \quad = \quad x(t) - v_{max} \int_0^{\frac{2\pi\tau_i}{T^{up}_{cycle}}} \sin(\theta) d\theta \qquad (11)$$

$$\quad = \quad x(t) - v_{max} \left(1 - \cos\left(\frac{2\pi\tau_i}{T^{up}_{cycle}} \right) \right) \qquad (12)$$

The maximal piston velocity $v_{max} > 0$ of the cycle is computed by combing the piston motion model (10), and the velocity estimated online from the position sensor signal.

Some additional logic and correction factors are used to tune the algorithm and make it reliable with respect to sensor noise and model errors. This is in particular necessary because the next x_{TDC} is unknow (only its nominal value x^*_{TDC} is known a priori).

B. Exhaust valves timing

The hydraulic exhaust valves actuators have a time delay of about 7 ms, and the time to open/close is about 2.5 ms. This is a significant fraction of the total engine cycle, which is about 33 ms at full load and about 80 ms at idle, and the main task of the timing subsystem is to compensate for this delay using a predictive model based on the piston motion parameter estimator. The higher level control system commands a position x_{EVO} where the exhaust valves should start to open during the downwards piston motion (expansion stroke), and a position x_{EVC} where the exhaust valves should be closed during upwards piston motion (compression stroke).

Using the predictor (5), the exhaust valves are commanded open at the first sample where $\hat{x}(t+\tau_e|t) \geq x^{EVO}$ during downwards piston motion, and commanded closed at the first sample where $\hat{x}(t+\tau_e|t) \leq x^{EVC}$ during upwards piston motion, where τ_e corresponds to the dead time plus half the rise time of the actuator. The actuator delay is 7 ms and its rise-time is 2.5 ms, leading to $\tau_e = 8.25$ ms.

In order to improve the reliability of the algorithm, and there is some safety logic that will prevent the exhaust valves from opening at incorrect positions due to sensor noise etc.

C. Compressor delivery valves timing

The compressor delivery valve should normally be open during the part of the cycle when there is a positive pressure difference between the piston compressor chamber and the interstage receiver such that high pressure air is allowed to flow from the compressor through the interstage receiver duct into the combustion chamber. Pressure sensors in both the compressor and interstage receiver makes this possible.

However, due to the large dead time (about 7 ms) and rise-time (about 2.5 ms) of the hydraulic actuators it is necessary to predict the pressure in the compressor $\tau_c = 8.25$ ms time units into the furture, where τ_c corresponds to the dead time plus half the rise time of the actuator. The compressor delivery valve is commanded to be open when it is predicted that the compressor pressure will be larger than the interstage receiver duct pressure τ_c time units into the future. The pressure prediction is based on a prediction of the future piston postion $\hat{x}(t+\tau_c|t)$ using the parameters given by the piston motion parameter estimator and measured current piston position $x(t)$. Assuming isentropic conditions (which is reasonable only prior to opening the compressor delivery valve), the future compressor pressure $\hat{p}_c(t+\tau_c|t)$ is predicted according to

$$\hat{p}_c(t+\tau_c|t)(x_{0c} - \hat{x}(t+\tau_c|t))^\kappa \quad = \quad p_c(t)(x_{0c} - x(t))^\kappa \qquad (13)$$

or

$$\hat{p}_c(t+\tau_c|t) \quad = \quad p_c(t) \left(\frac{x_{0c} - x(t)}{x_{0c} - \hat{x}(t+\tau_c|t)} \right)^\kappa \qquad (14)$$

The piston motion $\hat{x}(t+\tau_c|t)$ is predicted according to equation (5). No prediction of the receiver pressure is required since the volume is so large that it can be considered to be fairly constant over the cycle.

The above pressure predictor assumes isentropic compression, which is not a valid assumption after the compressor delivery valves has opened and there is air flow. The closing of the of the compressor delivery valves is therefore based on predicted piston velocity $\hat{v}(t+\tau_c|t)$, assuming that it is optimal to close the valves when the piston reaches a specified small upwards speed after the BDC. The predicted piston velocity is based on a differentiation of the equation (5).

D. Air cushion valves timing

The air cushion valves are hydraulic valves which can be commanded to be either fully open or fully closed. The timing subsystem will only modulate the signals from the higher level control system in case the air cushion pressure is being higher than the utility air pressure and the inlet valve is commanded open (then there will be an outflow rater than inflow as required), and in case the air cushion pressure is lower than athmposheric and the outlet valve is commanded open (there will be an inflow rather than outflow). In either of these cases, the signal is blocked such that this control input is simply made unavailable during a certain part of the cycle. Since these valves have fairly short response times (less than 2 ms), just a small margin on the pressure level is used to implement this blocking.

VI. Piston Motion Control

The basic idea of piston motion control is to control x_{TDC} by adjusting the amount of air in the air cushion m_a, and control x_{BDC} by adjusting the amount of diesel m_{di} being injected at each cycle. Based on energy calculations and simulations, the following observations were made [5]:

- x_{TDC} is more strongly influenced by m_a than x_{BDC}, and x_{BDC} is more strongly influenced by m_{di} than x_{TDC}. This motivates a two channel SISO control design.
- There may be a significant offset in the commanded and achieved values of m_{di} and m_a, leading to a need for integral action in both channels.
- The response from m_{di} from x_{BDC} is unstable, requiring derivative action in this channel to get high bandwidth and a suitable phase margin. Furthermore, the gain of this channel is roughly inverse proportional to the engine load. Although a controller with a gain that is proportional to the load is superior to a linear one, it is found that a linear controller gave satisfactory performance over the full operating range.
- The response from m_a to x_{TDC} is stable and fairly linear over the full operating range.

A. TDC control

A setpoint x^*_{TDC} is supplied by the supervisory control and optimization subsystem and a measured parameter x_{TDC} is given by the piston motion parameter estimator. A block diagram with the control structure is given in Figure 10. A PI controller computes the desired m^*_a. In cascade with the TDC controller there is a controller for the air cushion mass. If $\Delta m_a = m^*_a - \hat{m}_a$ is larger than a certain threshold, the air cushion inlet valve is commanded fully open until the air cushion mass reaches the threshold. Likewise, if Δm_a is smaller than a certain threshold, the outlet valve is opened until the air cushion mass reaches the threshold. Notice that the transfer function from the mass flow to the mass is an integrator, so this high-gain strategy does not lead to oscillations.

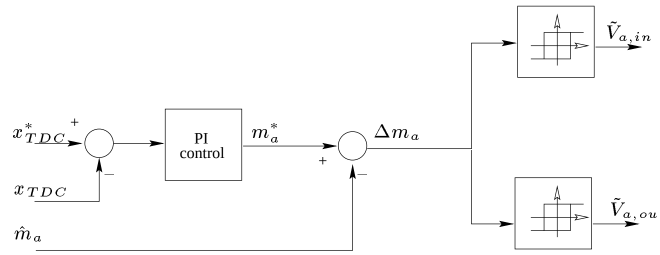

Fig. 10. Block diagram with TDC controller.

B. BDC control

A setpoint x^*_{BDC} is supplied by the supervisory control and optimization subsystem and a measured parameter x_{BDC} is given by the piston motion parameter estimator. As discussed above, a PID controller is applied in this case. Notice that in this case the "derivative" of the $x_{BDC}[k]$ must reflect the change in $x_{BDC}[k]$ from one cycle to the next, rather than the instantaneous derivative of the discrete-time signal $x_{BDC}[k]$.

C. Synchronization

In a multi-cylinder engine it is essential that the motions of multiple pistons are synchronized such that both mechanical vibrations and pressure oscillations in the exhaust manifold and turbines are kept to a minimum. In other words, it is desired that the individual pistons operate on the same frequency with specified phase shifts.

Assume there are two pistons with individual cycle periods $T_1[k]$ and $T_2[k]$ at cycle k. Their cycle times are then $t_1[k+1] = t_1[k] + T_1[k]$ and $t_2[k+1] = t_2[k] + T_2[k]$, and the phase shift is

$$\begin{aligned} \phi[k] &= 2\pi(t_1[k] - t_2[k])/T_1[k] \\ &= \frac{2\pi}{T_1[k]}\left(\frac{q^{-1}}{1-q^{-1}}\right)(T_1[k] - T_2[k]) \end{aligned} \quad (15)$$

where q^{-1} is the unit delay operator. The individual cycle periods $T_1[k]$ and $T_2[k]$ can be influenced by the TDC and BDC positions. In general, the piston speed can be increased (and the cycle time reduced) by increasing the length of the stroke. The reason for this is that the average pressures in the combustion chamber, air cushion and compressor chamber will increase. From a simple harmonic analysis, it is clear that the piston frequency is approximately proportational to the square root of the ratio between the average pressure and the piston mass. The synchronization controller is therefore a feedback from $\phi[k]$ that modifies the TDC and BDC setpoints x^*_{TDC} and x^*_{BDC}.

VII. Supervisory Control and Optimization

A. Logic control, Starting and stopping

Before the engine is started, the piston must be at rest at its lower position and the engine pressure must be athmospheric. The engine is started by simply opening the air cushion inlet valve for a period of 47 ms. The pressure in the air cushion then rises and provides just enough work on the piston to move it to the nominal TDC, where just the right amount of diesel is injected to move the piston down to its nominal BDC. During the first full cycle the timing subsystems are activated, but the piston motion control subsystems are not activated until after the first cycle. From then on, the air cushion mass and amount of injected diesel are adjusted automatically by the control system.

Stopping the engine is done by simply stopping injecting diesel, and keeping all the valves closed. The pressure in the various chambers will then be balancing the forces on the piston, and the piston will stop after a few cycles due to friction and pressure reduction because of air leakage, without being in mechanical contact with the cyclinder top or bottom.

B. Combustion optimization

The air in the combustion chamber just prior to injection of diesel into the combustion chamber has been compressed in three stages: Turbocompressor, piston compressor and diesel cylinder compressor. Prior to the final stage, compression in the diesel cylinder, the temperature of the air in the interstage receiver duct is given by isentropic relations:

$$T_r = T_i \left(\frac{p_r}{p_i} \right)^{\frac{\kappa-1}{\kappa}} \tag{16}$$

where T_i and p_i are the air temperature and pressure in the intake manifold (i.e. after turbo compressor and intercooler), and p_r is the pressure in the interstage receiver. Assume the desired temperature after the final compression stage is $T^\star$. This temperature is typically selected to give optimal conditions for combustion in the sense that it ensures auto-ignition and keeps emissions low. Again, using isentropic relations we have

$$T^\star = T_r \left(\frac{x_{EVC} + x_{0b}}{x^\star_{TDC} + x_{0b}} \right)^{\kappa-1} \tag{17}$$

where x_{0b} is the equivalent cylinder height of the dead volume of the combustion chamber, and $x^\star_{TDC}$ is the required upper dead point in order to achieve the temperature $T^\star$ at TDC. Manipulating the above equation, we get

$$x^\star_{TDC} = -x_{0b} + (x_{0b} + x_{EVC}) \left(\frac{T^\star}{T_r} \right)^{\frac{1}{1-\kappa}} \tag{18}$$

The interstage receiver temperature is not measured, but is estimated from the interstage receiver pressure p_r, and the intake manifold temperature T_i and pressure p_i by substituting (16) into (18).

Depending on the operating environment of the engine, there may be significant variations in the intake air temperature T_i. Using a measurement of T_i, these variations can be completely compensated for, and the total compression ratio will be increased when T_i is low (leading to better combustion), and it will be decreased when T_i is high (leading to reduced emissions and reduced thermal load on the cylinder top, exhaust manifold and turbines). A more sophisticated approach would be to also include the exhaust temperature T_e in some optimization where the x^*_{TDC} is selected such that the thermal efficiency is optimized subject to constraints on the exhaust gas temperature and emissions. The exhaust gas temperature is best estimated from temperature measurements downstream the turbines.

C. Load control

The operating point of a free-piston diesel gas generator is essentially determined by the intake and exhaust manifold pressures, in addition to certain parameters of the control system. Thus, for a certain intake and exhaust pressure, the engine will be stabilized at some steady state of operation with a certain stationary amount of diesel supplied per cycle, and with a certain stationary frequency. The engine's operating point is determined by the intersection between the operating characteristic of the engine (including turbocharger and power turbine) and the operating characteristic of the load.

In the case the load is an electric generator connected to a grid, there is no additional need for a load control system since the amount of injected diesel will be automatically adjusted by the BDC controller to provide the energy needed by the power turbine. Except at very low loads, the power turbine will operate in the choked flow region and thus only appear as a nozzle with fixed area as seen from the gas generator. An increase in the power demanded by the electric generator must be balanced by an increase in the power delivered by the gas generator in terms of increase mass flow and/or pressure and temperature. First, the increased power demand of the power turbine shifts its operating point to a higher exhaust manifold pressure. This leads to an increased minimum pressure in the combustion chamber, and increased pressure in the interstage receiver chamber, which again demands a higher peak pressure of the piston compressor. Consequently, the work on the piston increases and the BDC controller must increase the amount of injected diesel. Due to the general pressure increase, the turbo compressor delivery pressure increases and the mass flow through the engine increases.

In the case the operator commands a certain power output of the engine, e.g. when the power turbine shaft is connected to a propeller, a load controller must adapt the operating characteristics to match the desired operating point. The operating characteristic of the engine can be strongly influenced in at least two ways:

- The BDC position has significant effect on the mass flow rate through the engine, since it determines the amount of air being delivered by the piston compressor. For example, increasing the mass flow by increasing x_{BDC} leads to an increase in the exhaust manifold pressure since a fixed geometry turbine is equivalent to a fixed area nozzle (recall that $\dot{m} = \rho v A$ where $\rho = p/RT$, v is the speed of sound in the choked flow region and A is the flow area). As discussed above, such a pressure increase will require additional work to be made on the piston and the BDC controller will increase the supply of diesel to compensate for this, and the power output of the engine has been effectively changed. Similar ideas were also used for load control of the older free-piston diesel engines as discussed in [3].

- A variable geometry turbo or power turbine provides an effective opportunity to adapt the turbine characteristics, since the turbine will now be equivalent to a nozzle with variable area in the choked flow operating region. In combination with adapting the BDC setpoint, variable geometry control provides the opportunity of further improving the partload thermal efficiency.

An effective control strategy is to reduce the BDC position such that the stroke is shorter leading to reduced mass flow and/or to throttle the turbines at part load and idle.

VIII. Prototype Control System Implementation

The implementation of the experimental KLC engine control system is based upon the MATLAB/SIMULINK Real-Time Workshop using dSPACE hardware and real-time interface.

The hardware configuration is illustrated in Figure 11. At the core of the dSPACE real-time system there is an Alpha CPU where the timing and control algorithms are executed. The TMS 320C40 DSP handles all the I/O to the dSPACE system, a total of about 45 analog and digital signals. The system contains some timing circuits than generates high-resolution control pulses to the diesel injector amplifier. The dSPACE system is connected to a host PC from which the operator can start, stop and monitor the engine. The hydraulic valve actuators are operated through an industrial PC with timing circuits that commands pulses to the electromagnetic first stage of the hydraulic valves. There is a separate auxiliary control system that control and monitors the fuel pumps, lube pumps, cooling water and cooling oil circuits, turbo compressor and exhaust handling system etc. In addition, there is a separate data logging system.

The Alpha/TMS application utilizes multi-rate sampling. The core signal processing, timing and estimation algorithms are executed at 5 kHz, fault detection, logic control and monitoring are executed at 500 Hz, while the piston motion control and supervisory control subsystems are executed at 100 Hz.

IX. Experimental Results

A. Startup

Figure 12 illustrates engine startup. Until about $t = 1.280$ s the piston is at rest at its lower position and there is athmospheric pressure throughout the engine. A pulse of pressurized air at 70 bara is injected into the air cushion between $t = 1.280$ s and $t = 1.327$ s. The piston starts moving upwards (notice that x increases when the piston moves downwards) due to the peak pressure of about 8 bara in the air cushion. Diesel is injected near the TDC and the engine is now running. We observe that fairly startionary conditions are achieved already after 5 strokes.

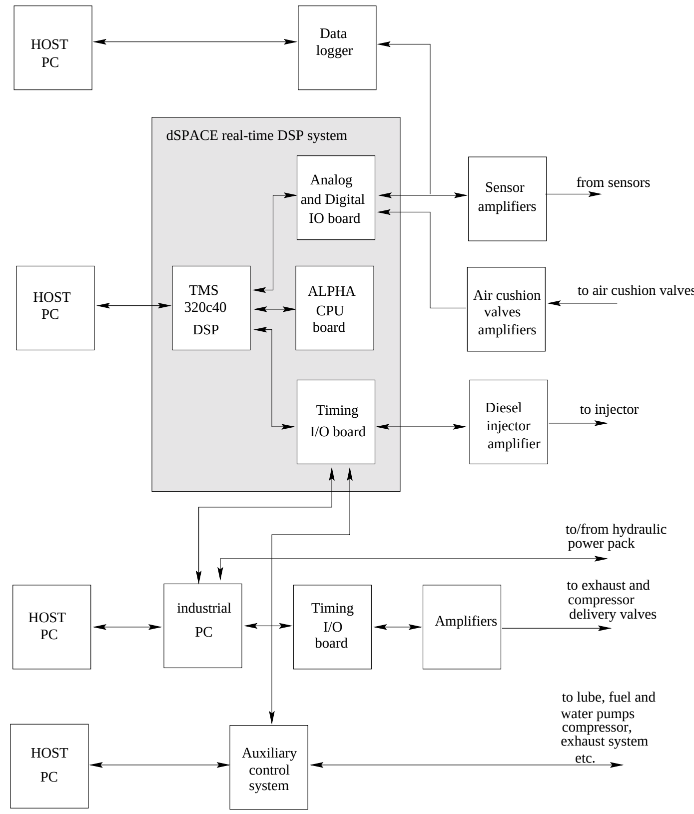

Fig. 11. Hardware configuration.

B. Timing

Figure 13 shows two consequtive cycles corresponding to about 33 % load (i.e. the diesel supply rate is 33 % of full load).

The compressor delivery valves are commanded to open at about $t = 51.529$ s, but due to the time delay they start to open at about $t = 51.536$ s and are not fully open until $t = 51.539$ s, when the compressor pressure crosses the interstage receiver pressure at about 6 bara. The compressor delivery valves are commanded to close at about $t = 51.537$ s, but are fully closed after $t = 51.546$ s, when the compressor pressure falls below the interstage receiver pressure.

A pulse to the injector is initiated at $t = 51.522$ s and one can observe from the slight non-smoothness of the combustion chamber pressure curve that combustion starts at about $t = 51.525$ s, while the TDC is reached at about $t = 51.527$ s.

The exhaust valves are commanded to open at about $t = 51.528$ s, but start to open only at $t = 51.535$ s due to the actuators' time delay. At this point the combustion chamber pressure is about 25 bar and combustion has clearly finished. It can be seen that the combustion chamber pressure curve has a sudden drop at this point due to the exhaust blowdown. At $t = 51.538$ s the combustion chamber pressure is down to the exhaust manifold pressure and the piston is at $x = 190$ mm, while the position where the scavenging ports start to be uncovered is $x = 200$ mm.

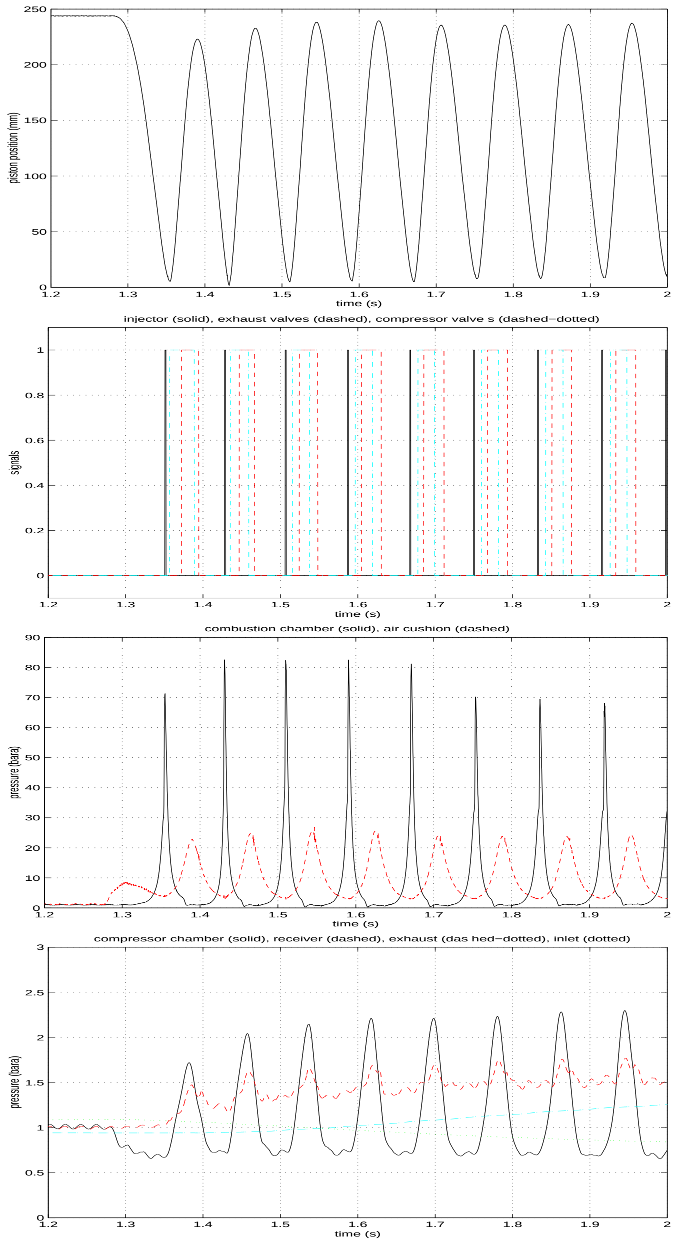

Fig. 12. Experimental results: Engine startup.

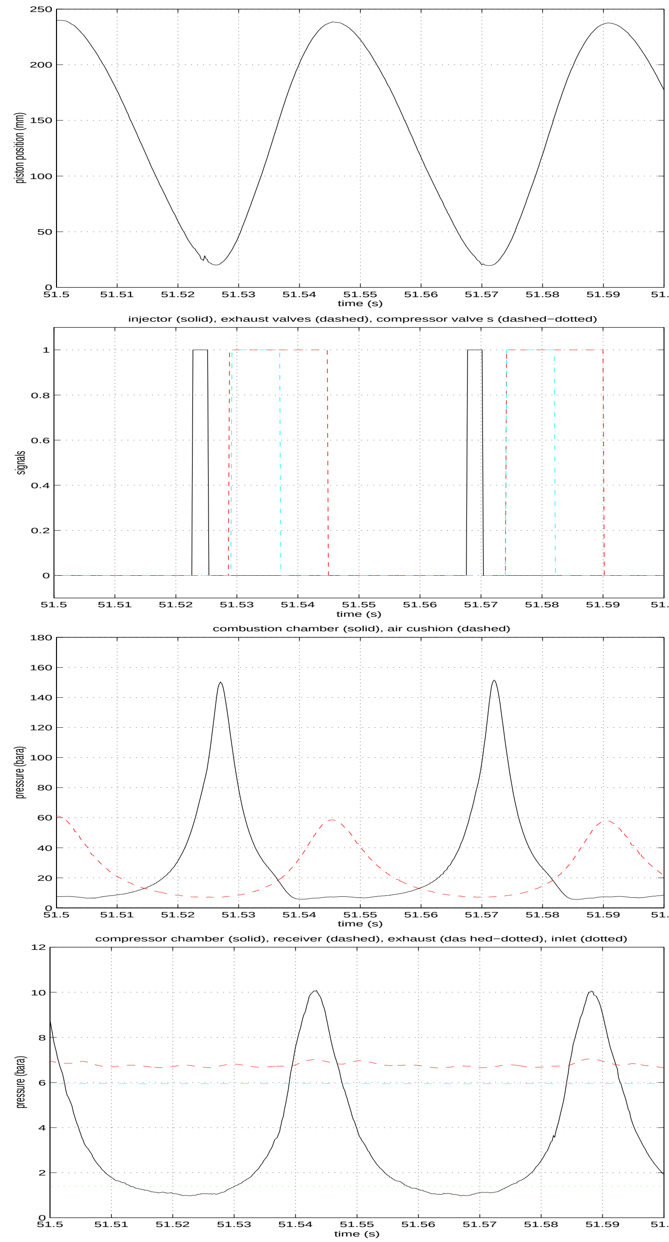

Fig. 13. Experimental results: 33 % load.

C. Piston Motion Control

Figures 14 illustrates the operation of the piston motion control system during fairly startionary conditions corrsponding to about 33 % load on the engine. It can be seen that the TDC and BDC are stabilized at their setpoints, and that the receiver and manifold pressures are stationary. Furthermore, the control inputs m_{di} and m_a shows no significant variations. Figure 15 shows a histogram with statistics of some key variables over 100 cycles. It can be seen that the standard deviation of TDC and BDC are about 1 mm.

X. Conclusions

The poor reliability and flexibility as well as difficulty in implementing purely mechanical control mechanisms was one of the main reason why the first generation free-piston engines was abandonen around 1960. The present results shows that todays electronic control technology (sensors, actuators and computer technology) is not a limiting factor anymore and provides the required processing capacity and resolution to implement the required control system functionality of modern high-speed free-piston diesel engines. A major challenge is still to design the engine and control system to get sufficiently high realiability, fault tolerance and robustness.

References

[1] P. Klotch, "Ford free-piston engine development", in *SAE Paper 590045*, 1959.

[2] A. F. Underwood, "The GMR 4-4 Hyprex engine - a concept of the free-piston engine for automotive use", in *SAE Paper*

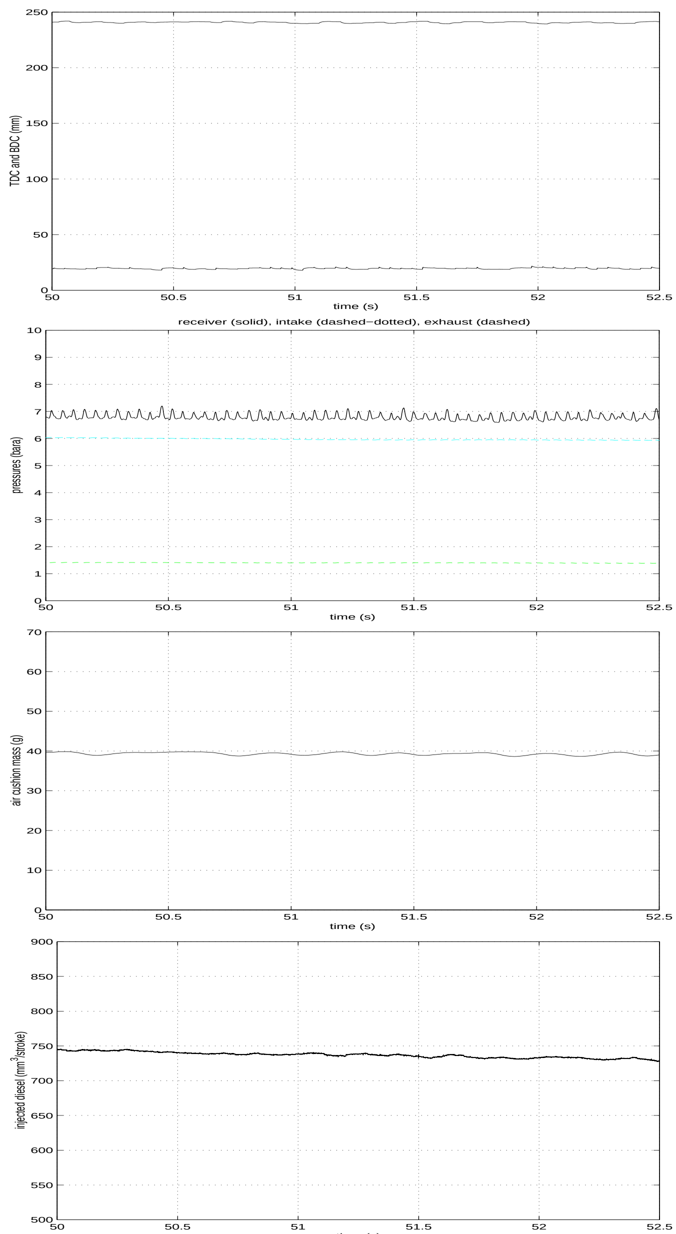

Fig. 14. Experimental results: Startionary running at 33 % load.

570032, 1957.

[3] A. F. Moiroux, “Free-piston engine possibilities”, in *ASME Oil and Gas Power Conference*, 1958, pp. 58–OGP–7.

[4] P. A. J. Achten, “A review of free piston engine concepts”, in *SAE Paper 941776*, 1994.

[5] T. A. Johansen, O. Egeland, E. A. Johannessen, and R. Kvamsdal, “Dynamics and control of a free-piston diesel engine”, Submitted for publication, 2000.

[6] M. J. Førde, T. A. Johansen, R. Kvamsdal, and O. Egeland, “A method for controlling the stroke of a diesel free-piston gas generator”, Patent WO9728362, 1997.

[7] S. Tikkanen and M. Vilenius, “Hydraulic free piston engine - challenge for control”, in *Proc. European Control Conference, Karlsruhe*, 1999.

[8] G. B. Meacham, “Variable cam timing as an emission control tool”, in *SAE Paper 700645*, 1970.

[9] T. H. Ma, “Effects of variable engine valve timing on fuel economy”, in *SAE Paper 880390*, 1988.

[10] C. Gray, “A review of variable engine valve timing”, in *SAE Paper 880386*, 1988.

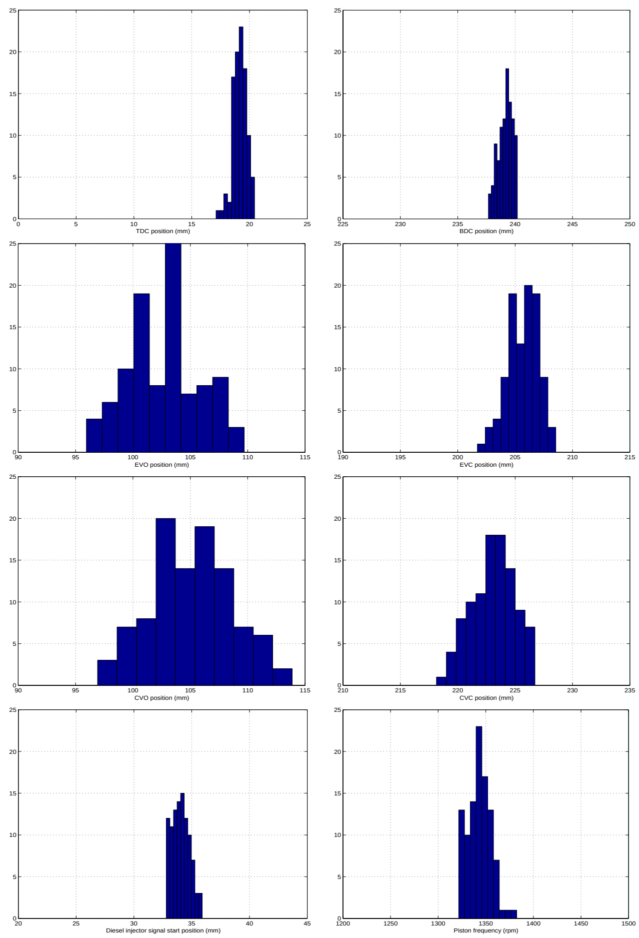

Fig. 15. Experimental results: Histograms with some key variables over 100 cycles, at about 33 % load.

[11] A. G. Stefanopoulou, J. S. Freudenberg, and J. W. Grizzle, “Variable camshaft timing engine control”, *IEEE Trans. Control Systems Technology*, vol. 8, pp. 23–34, 2000.

[12] J. B. Heywood, *Internal Combustion Engine Fundamentals*, McGraw-Hill, 1988.