Free-Piston Diesel Engine Dynamics and Control

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Abstract

Free-piston diesel engines are characterized by freely moving pistons without any rigid crankshaft or camshaft connected to the pistons. This allows a highly compact and efficient engine design, but requires automatic control of the piston motion. This paper presents a dynamic mathematical model of a free-piston diesel engine, a control oriented dynamic analysis, and a novel computer-based piston motion control system. Experimental evaluation of the results on a full scale diesel test cylinder show feasibility of the suggested control approach.

1 Introduction

The free-piston diesel engine concept was developed by Pescara, and engines of various size were manufactured between 1930-1960 by GM, Ford, Renault, Junker, SIGMA and others [1]. Despite the potential advantages of the Pescara process, 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 thermal efficiency. Today, new materials, modern computer control technology, and high-precision common-rail diesel injection systems allow the implementation of an accurate and reliable electronic control system. This was the motivation for Kværner ASA, who is studying a modern high-speed free-piston diesel engine concept 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 %. In contrast to the original free piston diesel engine with an opposed piston arrangement in the same cylinder, the Kværner design is a more conventional two-stroke arrangement with a single piston per cylinder and a standard diesel cylinder top with a centered injector and four poppet exhaust valves per cylinder.

With the exception of the recent work [2], which is based on a different type of free-piston engine concept, there are no other results on electronic control of free-piston diesel engines reported in the literature. The contribution of the present paper is that it presents dynamic models, energy-based control design, and experimental results on a full scale test cylinder. In [3] we give details of the engine control system, emphasizing valve

and injector timing, piston motion parameter estimation, signal processing, control and its implementation. The main ideas are described in the patent [4], also for an opposed piston arrangement.

2 Free-piston diesel engine operating principles

Figure 1: Sketch of multi-cylinder free-piston engine, indicating mass and energy flows.

Figure 1 provides a principal sketch of a multi-cylinder free-piston diesel engine. For cylinder/piston number n , the air cushion chamber is indicated by subscripts a, n , the combustion chamber by b, n , and the compression chamber by c, n . There is an interstage receiver chamber, denoted r, n , and common intake and exhaust manifolds denoted i and e , respectively. The

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index p refers to a utility air reservoir, and the ambient stagnation state is denoted 0.

The two-stroke diesel cycle, seen from the intake manifold to the exhaust manifold, can be described as follows (see also [5]). *Compression:* The air in the combustion chamber is compressed when the piston moves from its bottom dead center towards its top dead center. *Combustion:* Near the top dead center, the temperature of the compressed air is high enough for auto-ignition, and diesel is injected at high pressure into the combustion chamber through a nozzle, and combustion of the diesel fuel results. *Expansion:* The high pressure in the combustion chamber makes the piston move downwards and the exhaust gas expands. *Scavenging:* During this downwards motion, the combustion finishes and the actively controlled exhaust valves opens at high pressure to deliver high-energy exhaust gas to the exhaust manifold and 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 actively controlled valves connecting the piston compressor chamber and interstage receiver. The high pressure in the compressor chamber and air cushion makes the piston move upwards from its bottom dead center. *Air intake:* During the upwards motion of the piston, fresh air is sucked into the compressor chamber from the intake manifold through passive suction valves.

The free-piston diesel engine differs from a traditional diesel engine in several ways: i) The pistons move freely, only influenced by pressure and friction forces since there is no crankshaft connected to the piston. ii) The exhaust and compressor delivery valves are controlled electronically, since there is no camshaft connecting them to the pistons. iii) The exhaust gas has much higher temperature and pressure than in a traditional engine, as the mechanical power is produced directly from the exhaust gas in the power turbine. The free-piston concept used in the present study is feasible only if the piston motion can be controlled. The requirements in terms on control accuracy and robustness are high as cycle-to-cycle variations in the motion of the piston at stationary running will introduce undesired vibrations and disturbances in the intake and exhaust manifold that will influence the operation, cost and life-cycle of the turbomachinery. The cycle-to-cycle variability in stroke length should be less than 2 mm out of a stroke of about 200 mm. Furthermore, 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.

3 Mathematical Model of N -Cylinder Free-Piston Engine

In this section we derive a mathematical model based on ordinary differential equations describing mass- and energy-balances of the main gas volumes in addition to a force balance for the pistons. The basic equations are derived under the assumption of the gas being ideal and that heat loss and friction are negligible. A somewhat more detailed model with similar structure but based on somewhat more realistic assumptions, including a simple model of the combustion chemistry, has been developed and used for simulation.

Dot notation is used to denote time differentiation so that $\dot{m}$ is the time derivative of m , that is, $\dot{m} = \frac{d}{dt}m$, and not the mass flow as in common in the combustion engine literature. Mass flow from i to j is denoted by M_{ji} . The variable x is the piston position, T denotes absolute temperature, p denotes pressure, V is volume, ρ is density and m is mass. The indices follows the notation introduced in Figure 1.

3.1 Geometric parameters

The inner piston diameter corresponding to the air cushion and combustion chamber is d_{pi} . The outer part of the piston corresponding to the compressor chamber has diameter d_{po} . The mass of each of the pistons is m . The cross section areas are for the air cushion, combustion chamber and compression chambers:

$$A_a = \frac{\pi}{4}d_{pi}^2, \quad A_b = \frac{\pi}{4}d_{pi}^2, \quad A_c = \frac{\pi}{4}(d_{po}^2 - d_{pi}^2) \quad (1)$$

The volumes of the air cushion chambers are $V_{a,n} = A_a(x_{0a} - x_n)$, where x_{0a} is the effective length of each of the air cushions. The volumes of the combustion chambers are $V_{b,n} = A_b(x_{0b} + x_n)$ where x_{0b} accounts for dead volume at the piston upper stop position, corresponding to mechanical contact with the cylinder top. The volumes of the compression chambers are $V_{c,n} = A_c(x_{0c} - x_n)$ where x_{0c} is the effective length of the compression chamber, including dead volume at the piston lower stop position. The volume of the interstage receiver chamber and exhaust manifolds are V_r and V_e respectively.

The position x_n increases when the piston moves downwards, cf. Figure 2. The piston is said to be at the top dead center (TDC) when x_n is at its minimum for the actual cycle. The velocity at this point satisfies $\dot{x}_n = 0$. The position at TDC for the actual cycle is defined to be x_n^{TDC} . Likewise, the bottom dead center (BDC) is denoted x_n^{BDC} . This position is reached when x_n is at its maximum for the actual cycle. Some further geometric parameters are defined in Figure 2.

3.2 Force balances

Equations of motion for pistons:

$$m\ddot{x}_n = -p_{a,n}A_a - p_{c,n}A_c + p_0A_c + p_{b,n}A_b \quad (2)$$

Figure 2: Geometric cycle parameters.

3.3 Mass balances

Conservation of mass gives

$$\dot{m}_{a,n} = M_{ap,n} \quad (3)$$

$$\dot{m}_{b,n} = M_{br,n} - M_{eb,n} \quad (4)$$

$$\dot{m}_{c,n} = M_{ci,n} - M_{rc,n} \quad (5)$$

$$\dot{m}_{r,n} = M_{rc,n} - M_{br,n} \quad (6)$$

$$\dot{m}_e = \sum_{n=1}^N M_{eb,n} - M_{te} \quad (7)$$

The mass flows are expressed as $M_{ap,n} = M_{ap,n}^{\text{in}} - M_{a0,n}^{\text{out}}$ where superscript “in” refers to flow in to the air cushion, while superscript “out” refers to flow out. The variables $M_{ap,n}^{\text{in}}$ and $M_{a0,n}^{\text{out}}$ are both non-negative. Likewise, $M_{br,n} = M_{br,n}^{\text{in}} - M_{br,n}^{\text{out}}$ and $M_{eb,n} = M_{eb,n}^{\text{out}} - M_{eb,n}^{\text{in}}$ where $M_{br,n}^{\text{in}}$, $M_{br,n}^{\text{out}}$, $M_{eb,n}^{\text{out}}$ and $M_{eb,n}^{\text{in}}$ are nonnegative, and the superscript “in” refers to mass flow into the combustion chamber, while the superscript “out” refers to mass flow out of the combustion chamber. Finally, $M_{rc,n} = M_{rc,n}^{\text{out}} - M_{rc,n}^{\text{in}}$ where $M_{rc,n}^{\text{out}}$ and $M_{rc,n}^{\text{in}}$ are non-negative. The superscript “in” refers to mass flow into the compressor chamber, while the superscript “out” refers to mass flow out of the compressor chamber.

3.4 Energy balances

The energy balances for the air cushion, compressor, combustion, interstage receiver and exhaust manifold chambers are for $n = 1, 2, \dots, N$

$$\begin{aligned} \dot{T}_{b,n} = & \frac{M_{br,n}^{\text{in}}}{m_{b,n}} \kappa T_{r,n} + \frac{M_{eb,n}^{\text{in}}}{m_{b,n}} \kappa T_e \\ & - \frac{M_{br,n}^{\text{out}} + M_{eb,n}^{\text{out}} + (M_{br,n}^{\text{out}} + M_{eb,n}^{\text{out}})(\kappa - 1)}{m_{b,n}} T_{b,n} \\ & - \frac{1}{m_{b,n} c_v} A_b p_{b,n} \dot{x}_n + \frac{h_{di} M_{di,n}}{m_{b,n} c_v} \end{aligned} \quad (8)$$

$$\begin{aligned} \dot{T}_{a,n} = & \frac{M_{ap,n}^{\text{in}}}{m_{a,n}} (\kappa T_p - T_{a,n}) - \frac{M_{ap,n}^{\text{out}} (\kappa - 1)}{m_{a,n}} T_{a,n} \\ & + \frac{1}{m_{a,n} c_v} A_a p_{a,n} \dot{x}_n \end{aligned} \quad (9)$$

$$\begin{aligned} \dot{T}_{c,n} = & \frac{M_{ci,n}}{m_{c,n}} (\kappa T_i - T_{c,n}) - \frac{M_{rc,n}^{\text{in}} + M_{rc,n}^{\text{out}} (\kappa - 1)}{m_{c,n}} T_{c,n} \\ & + \frac{M_{rc,n}^{\text{in}}}{m_{c,n}} \kappa T_{r,n} + \frac{1}{m_{c,n} c_v} A_c (p_{c,n} - p_0) \dot{x}_n \end{aligned} \quad (10)$$

$$\begin{aligned} \dot{T}_{r,n} = & \frac{M_{rc,n}^{\text{out}}}{m_r} \kappa T_{c,n} + \frac{M_{br,n}^{\text{out}}}{m_r} \kappa T_{b,n} \\ & - \frac{M_{rc,n}^{\text{out}} + M_{br,n}^{\text{out}} + (M_{rc,n}^{\text{in}} + M_{br,n}^{\text{in}})(\kappa - 1)}{m_r} T_{r,n} \end{aligned} \quad (11)$$

$$\begin{aligned} \dot{T}_e = & \sum_{n=1}^N \left(\frac{M_{eb,n}^{\text{out}}}{m_e} (\kappa T_{b,n} - T_e) - \frac{M_{eb,n}^{\text{in}} (\kappa - 1)}{m_e} T_e \right) \\ & - \frac{M_{te}}{m_e} (\kappa - 1) T_e \end{aligned} \quad (12)$$

where $\kappa = c_p/c_v$. The ambient density, density of the utility pressurized air, and density of the air in the intake manifold are

$$\rho_0 = \frac{p_0}{RT_0}, \quad \rho_p = \frac{p_p}{RT_p}, \quad \rho_i = \frac{p_i}{RT_i} \quad (13)$$

where R is the universal gas constant. The diesel heat release $M_{di} h_{di}$ is initiated at some piston position x_{CS} , and assumed to evolve according to some time-varying function that depends on the diesel pressure and injected amount, see e.g. [5, 6].

3.5 Mass flows

Mass flows through valves and ports are modeled as isentropic flow through nozzles. Let the inlet stagnation pressure be denoted p_1 and the nozzle exit pressure, which is assumed to be equal to the throat pressure, be denoted by p_2 . Then the critical exit pressure corresponding to choked flow is

$$p_C = \left(\frac{2}{\kappa + 1} \right)^{\frac{\kappa}{\kappa - 1}} p_1 \quad (14)$$

The mass flow through the valve is [5]

$$M = \begin{cases} A \sqrt{\frac{2\kappa}{\kappa - 1} p_1 \rho_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{2}{\kappa}} - \left(\frac{p_2}{p_1} \right)^{\frac{\kappa + 1}{\kappa}} \right]} & \text{if } p_2 > p_C \\ A \sqrt{\kappa p_1 \rho_1 \left(\frac{2}{\kappa + 1} \right)^{\frac{\kappa + 1}{\kappa - 1}}} & \text{if } p_2 \leq p_C \end{cases} \quad (15)$$

3.6 Numerical values

The parameters in Table 1 corresponds to the Kværner KLC test cylinder with about 1 MW net output. At full load the thermal efficiency is about 50 %, mass flow through the engine is about 1.75 kg/s per cylinder, stroke length about 189 mm, intake pressure (from turbo compressor) about 3.25 bar, exhaust pressure about 20 bar, exhaust temperature about 1050 K, and piston frequency about 1800 rpm.

d_{pi}	0.180 m	d_{po}	0.430 m
x_{0a}	0.2945 m	x_{0c}	0.2546 m
x_{0b}	0.008 m	V_r	$29 \cdot 10^{-3} \text{ m}^3$
$A_{ci,n}$	$24.4 \cdot 10^{-3} \text{ m}^2$	$A_{eb,n}$	$6.4 \cdot 10^{-3} \text{ m}^2$
$A_{rc,n}$	$7.7 \cdot 10^{-3} \text{ m}^2$	$A_{br,n}$	$6.8 \cdot 10^{-3} \text{ m}^2$
m	100 kg		

Table 1: Numerical values of parameters.

The mathematical model was tuned and verified against experimental data to show excellent compliance

with the test cylinder. An example of simulated and experimental results for similar operating conditions are shown in Figure 3.

Figure 3: Model validation: Pressures p_b, p_r, p_a and p_c vs. time (s). Experimental results are left and simulations are right.

4 Model Analysis

The piston motion control subsystem is essentially required to control the TDC and BDC positions of the pistons, with small cycle-to-cycle variations during steady-state operation and with hard constraints during load transients. The purpose of this section is to analyse how the variables x_{TDC} and x_{BDC} , that we want to control, can be influenced by manipulating the free variables we have at hand.

4.1 Control variables

The basic control variables that are available are the signals to the injector (common rail), exhaust valves and piston compressor delivery valves (both hydraulically actuated poppet valves), and in/out valves at the air cushion (electro-hydraulic cartridge valves). These are all on/off type of valves that can be commanded open or closed. These control variables offer considerable degrees of freedom, but there are also certain constraints on their use. The most useful control variables are the air cushion mass m_a and the amount of injected diesel per cycle m_{di} . Notice that the reason why m_{di} is a free variable is that the load of the engine is controlled by a supervisory controller actuating on the turbomachinery or commanding setpoints to the motion control system to achieve the desired exhaust manifold pressure when accelerating or retarding the engine. The variables m_{di} and m_a are convenient to work with since they are more or less invariant over the cycle, in contrast to most other variables in the engine that are periodic. Notice that m_{di} can be specified directly to the diesel injection system through a mapping that maps diesel pressure and desired volume of injected diesel to electronic pulses. The air cushion mass m_a is controllable through valves, and is the setpoint to a low level control system which includes an estimator for the air cushion mass, see Figure 4.

4.2 Energy considerations

Here we establish how the steady-state values of x_{TDC} and x_{BDC} can be influenced by m_{di} and m_a . In order to relate m_{di} and m_a to x_{BDC} we consider the expansion stroke, i.e. piston motion from TDC to BDC. The additional work on the piston needed to change the stroke

length by Δx_{BDC} near the BDC is approximately

$$\Delta W = (p_a^{BDC} A_a + (p_c^{BDC} - p_0) A_c - p_b^{BDC} A_b) \Delta x_{BDC}$$

when assuming Δx_{BDC} is small. Furthermore, one may approximate $p_c^{BDC} \approx p_b^{BDC} \approx p_r$ if the pressure loss over the valves are negligible. This additional work may be produced by additional fuel input Δm_{di} , so

$$\Delta m_{di} h_{di} = \Delta x_{BDC} \cdot \left(m_a A_a R \bar{T}_a \frac{\bar{V}_a^{\kappa-1}}{(\bar{V}_a^{BDC})^\kappa} + (p_r - p_0) A_c - p_r A_b \right) \quad (16)$$

where we have used the ideal gas law and isentropic compression relations

$$p_a^{BDC} = m_a R \bar{T}_a \frac{\bar{V}_a^{\kappa-1}}{(\bar{V}_a^{BDC})^\kappa} \quad (17)$$

where $\bar{T}_a$ and $\bar{V}_a$ are the cycle-averaged volume and temperature of the air cushion. Hence, in order to change the position of the BDC by Δx_{BDC} from one cycle to the next while keeping m_a constant, one must add Δm_{di} defined by (16) to the mass of injected fuel.

The differential influence of air cushion mass m_a and amount of injected fuel m_{di} on x_{TDC} can be found by considering the compression stroke, i.e. the piston motion from BDC to TDC. The additional work on the piston needed to change the stroke length by a small amount Δx_{TDC} near the TDC is approximately

$$\Delta W = (p_b^{TDC} A_b - (p_c^{TDC} - p_0) A_c - p_a^{TDC} A_a) \Delta x_{TDC}$$

Again, one may approximate $p_c^{TDC} \approx p_i$ and $p_b^{TDC} \approx p_r r_{di} \alpha_{di}$, where r_{di} is the combustion chamber compression ratio, and $\alpha_{di} > 1$ is a factor that accounts for the peak pressure increase due to combustion. At idle $\alpha_{di} \approx 2$ while at full load α_{di} is only slightly larger than 1 (because of the long injection period at full load only a small fraction of the diesel will be combusted when the peak pressure is reached). The combustion chamber compression ratio depends on the compression ratio achieved in the piston compressor such that the total compression ratio (piston compressor plus combustion chamber) is constant r_t :

$$r_{di} = r_t \frac{p_i}{p_r} \quad (18)$$

where r_t is a constant. Furthermore, using the ideal gas law and isentropic compression relations we get

$$p_a^{TDC} = m_a R \bar{T}_a \frac{\bar{V}_a^{\kappa-1}}{(\bar{V}_a^{TDC})^\kappa} \quad (19)$$

The change in work ΔW may be caused by a change in the internal energy of the air cushion $\Delta U_a = c_v T' \Delta m_a$ by changing the amount of air by Δm_a

$$c_v T' \Delta m_a = \Delta x_{TDC} \cdot \left(p_i r_t \alpha_{di} A_b - (p_i - p_0) A_c - m_a A_a R \bar{T}_a \frac{\bar{V}_a^{\kappa-1}}{(\bar{V}_a^{TDC})^\kappa} \right) \quad (20)$$

where $T' = T_a$ when $\Delta m_a < 0$ and $T' = T_p$ when $\Delta m_a > 0$.

4.3 Control Structure

Simulations show that the couplings are not too strong since m_{di} influences mainly x_{BDC} and m_a influences mainly x_{TDC} . This can also be seen from the energy considerations in Section 4.2, cf. (16) and (20). One may therefore expect that two SISO controllers may lead to adequate performance, one controller that commands m_{di} on the basis of deviations in x_{BDC} and one that commands m_a on the basis of deviations in x_{TDC} . This leads to the motion control structure suggested in Figure 4.

Figure 4: Motion control system structure.

Integral action is required in both control loops because it is essential that the steady-state error is small, and there may be a significant offset in the diesel injection system such that the actual m_{di} differs from the commanded m_{di}^* . Furthermore, there may be a large offset also in the air cushion mass estimator due to a poor guess of the mean temperature such that the actual m_a differs from its estimate $\hat{m}_a$. Because the response from m_{di} to x_{BDC} is open loop unstable, derivative action is useful to get high performance and robustness.

The results from model analysis suggest the relationships (16) and (20) that can be rewritten in the form

$$\Delta m_{di} = (k_0 + k_1 m_a + k_2 p_r) \Delta x_{BDC} \quad (21)$$

$$\Delta m_a = (g_0 + g_1 m_a + g_2 p_r + g_3 p_i) \Delta x_{TDC} \quad (22)$$

where $k_0, k_1, k_2, g_0, g_1, g_2$ and g_3 are constants. Obviously, this suggests some controller nonlinearities. The variables m_a, p_i and p_r are roughly invariant over each cycle and depends mainly on the load of the engine. At

idle, $p_i = 1$ bar and it increases gradually to $p_i = 3.25$ bar at full load. Moreover, $p_r = 2$ bar at idle and it increases to $p_r = 22$ bar at full load. Moreover, $m_a = 16$ g at idle and it increases to about $m_a = 80$ g at full load. Figure 5 illustrates theoretical dependencies of $x_{TDC}^*, x_{BDC}^*, m_a^*, m_{di}^*, p_r$ and p_i as a function of load. These are computed in order to achieve an exhaust gas temperature that is independent of the load. Thus, one may rewrite (21) and (22) as

$$\Delta m_{di} = k(L) \Delta x_{BDC} \quad (23)$$

$$\Delta m_a = g(L) \Delta x_{TDC} \quad (24)$$

where L is load varying between approximately 5% at idle to 100%. The gains are illustrated in Figure 6. It should be noted that even though the individual terms of (22) are highly nonlinear (for example $g_2 p_r$ will change with a factor of more than 10 from idle to full load) it is clear that (24) are much less nonlinear as a function of L because the individual terms will dominate at different loads.

Figure 5: Main engine parameters as a function of load (theoretical calculations).

Figure 6: Nonlinear gain as a function of load.

The main contributions to the dynamics is the volume of the interstage receiver chamber. A rough estimate of the dominant time-constant might be found by computing the ratio of the volume V_r to the volumetric flow rate $q_r = \overline{M}_r / \rho_r$, where $\overline{M}_r$ is the mass flow through the interstage receiver averaged over a cycle. Notice that V_r is constant, while q_r may show some variation

between idle and full load. However, this variation is neglectable since the air-to-fuel ratio does not vary too much (it is somewhat higher at idle than at full load). Hence, $\tau = V_r/q_r$ can be considered to be operating point independent. In fact, experimental results shows that adequate performance can be achieved with linear controllers (PI control of the air cushion and PID control of the diesel injection) over the full operating range, even though a nonlinear diesel controller where the gain is scheduled on the load of the engine consistent with (23) might give consistently better performance over the full operating range.

5 Experimental Results

Results of transient running of the 1 MW test cylinder are shown in Figure 7, see [3] for details on the experimental setup. Initially, the piston is at rest at the PBS position and the pressure in the engine is atmospheric. The engine starts at $t \approx 2.0$ s by injecting the exact amount of air m_a into the air cushion to generate the pressure required to make the piston move to the desired TDC position. At the TDC exactly the required amount of diesel m_{di} is injected to take the piston to the desired BDC position. At this point, the piston motion controller takes over, and command m_a and m_{di} for each cycle. The engine load is increased from idle to about 40 % during this time interval, by manually choking the exhaust gas flow. We observe from the curves that the piston motion control system operates satisfactory with variability of the dead center position of about 1 mm after the initial transient. Somewhat larger variability must be accepted during startup.

6 Conclusion

A detailed dynamic model of a free-piston diesel engine is derived based on thermodynamics relations and the equations of motion, leading to a set of ordinary differential equations. The dynamic analysis reveal that the piston motion can be stabilized and the top and bottom dead center positions can be controlled by manipulating the air cushion mass and the amount of injected diesel per cycle. A simple control structure for this purpose is suggested and verified experimentally on a full scale test cylinder.

References

- [1] A. F. Moiroux, "Free-piston engine possibilities", in *ASME Oil and Gas Power Conference*, 1958, pp. 58–OGP–7.
- [2] S. Tikkanen and M. Vilenius, "Hydraulic free piston engine - challenge for control", in *Proc. European Control Conference, Karlsruhe*, 1999.
- [3] T. A. Johansen, O. Egeland, E. A. Johannessen, and R. Kvamsdal, "Free-piston diesel engine timing and control – towards electronic cam- and crankshaft", *IEEE Trans. Control Systems Technology*, vol. 9, 2001.
- [4] 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.
- [5] J. B. Heywood, *Internal Combustion Engine Fundamentals*, McGraw-Hill, 1988.
- [6] G. Borman and K. Ragland, *Combustion Engineering*, McGraw-Hill, 1998.

Figure 7: Piston motion (x_{BDC} and x_{TDC} positions) and control input m_{di} and m_a when the load of the engine is increased from idle to about 40 % using a throttle in the exhaust pipe. Curves for the exhaust manifold pressure p_e , scavenging pressure p_r and intake manifold pressure p_i are also shown.