

Diesel Cylinder Charge Properties: Feed-Forward Control and Cycle-by-Cycle Analysis Using an In-Cylinder Gas Sampling System

2011-01-0709

Published
04/12/2011Alexander Darlington, Nick Collings and Keith Glover
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doi:10.4271/2011-01-0709

ABSTRACT

Common-rail fuel injection systems on modern light duty diesel engines are effectively able to respond instantaneously to changes in the demanded injection quantity. In contrast, the air-system is subject to significantly slower dynamics, primarily due to filling/emptying effects in the manifolds and turbocharger inertia. The behaviour of the air-path in a diesel engine is therefore the main limiting factor in terms of engine-out emissions during transient operation. This paper presents a simple mean-value model for the air-path during throttled operation, which is used to design a feed-forward controller that delivers very rapid changes in the in-cylinder charge properties. The feed-forward control action is validated using a state-of-the-art sampling system that allows true cycle-by-cycle measurement of the in-cylinder CO_2 concentration.

INTRODUCTION

Calibration of modern light-duty diesel engines to meet the legislated emissions standards is still predominantly carried out under steady-state conditions [1]. The results of this calibration process are typically a set of maps that give the desired boost pressure, mass air flow (MAF) and fuelling parameters for any particular engine speed/load operating point. As long as the engine control system can deliver these demanded conditions, the engine-out emissions should closely match those measured during the calibration process. In taking this approach, it is implicitly assumed that the emissions measured under the steady-state conditions used for calibration will be representative of those measured during the transient testing procedures. The transients found during the New European Drive Cycle (NEDC) are

sufficiently slow for this to currently be a reasonable assumption.

Nevertheless, consideration of the engine/air-path behaviour during transient operation is of significant interest for several reasons. Firstly, the closer the air-path control system can come to delivering the correct air-charge to the cylinders during transient operation, the better the drive-cycle emissions should agree with predictions based on the calibrated steady-state emissions [2, 3]. Also, future legislation may require emissions to be tested under faster, more realistic, transient conditions in Europe (the procedure used for emissions testing in the United States already includes significantly faster transients) [4]. Finally, the demands of future exhaust after-treatment systems may also place an emphasis on improving transient response. In particular, the successful use of Lean NO_x Traps (LNTs) may depend upon being able to rapidly transition the engine from lean to rich operation [5]. It is therefore clear that ideally we would wish to control the actual pre-combustion gas charge in terms of the quantity of air and the quantity of inert burned gas on a cycle-by-cycle basis, rather than the MAF and manifold pressure that act as proxies for the charge properties.

The particular scenario examined in this paper is achieving the fastest possible transition in the cylinder charge properties when the air-path is throttled. In this case, the Exhaust Gas Recirculation (EGR) valve and intake throttle actuators are used to control the composition of the pre-combustion cylinder charge. Throttled operation is commonly used in modern light-duty diesel engines at low speed/load operating points to increase the quantity of EGR, as well as being essential for Diesel Particulate Filter (DPF) regeneration and

LNT regeneration, both of which require significant reductions in air-flow to achieve richer running conditions.

Transient control-oriented modelling of the cylinder charge properties has been discussed extensively in the literature [6, 7, 8, 9, 10], however in this previous work, results have not been validated using actual in-cylinder measurements. Similarly, literature discussing more sophisticated 1-D engine modelling routinely predicts the cylinder charge composition in diesel engines [3, 11] but does not validate them experimentally. The importance of understanding the transient behaviour and its effects on emissions are also investigated in many previous publications [12, 13, 14, 15, 16]. Novelty in this paper comes from the focus of the modelling on the actual charge properties during transient throttled operation and the use of in-cylinder sampling to provide direct measurement of these charge properties.

EXPERIMENTAL SETUP

The engine air-path configuration used for the experimental work presented in this paper is shown in Fig. 1 below:

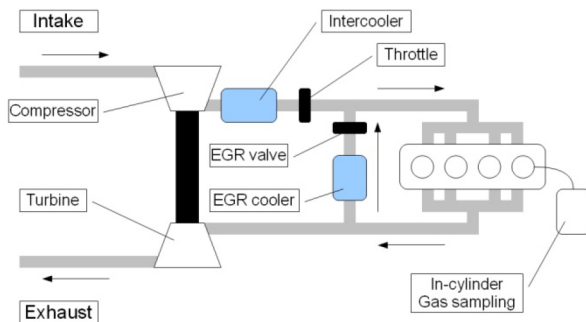


Figure 1. The configuration of the engine used in the modelling and experimental work described in this paper.

The engine is a production 2.0-litre 4-cylinder 16 valve diesel engine (100 kW @ 3800 rpm, 325 N.m @ 1800 rpm) and is equipped with a Variable Geometry Turbocharger (VGT), a cooled high-pressure EGR circuit and a common rail injection system. Three of the cylinders are fitted with pressure transducers and in the fourth, the pressure transducer has been removed to fit the in-cylinder sampling system. Load for the engine is provided by a low inertia (0.46 kg.m²) 125 kW AC electric motor (400 Nm, 6500 rpm).

Controllers for the engine are implemented using Matlab/Simulink® and dSpace® (see table 1), with ECU variable bypassing over the CAN interface. The ECU runs its standard calibration but with the demanded EGR/Throttle actuator positions over-ridden by the rapid prototyping system and communicated at 8ms intervals over the CAN network. Data on CAN latency were not recorded as it was felt unnecessary due to the feed-forward nature of the control implementation.

The rapid prototyping system used event driven tasks (triggered by pulses from the crank/cam encoders) to synchronise the opening and closing of the sampling valve with the engine cycle.

Emissions data were recorded using a Cambustion fast response CO₂ analyser (NDIR 500).

Table 1. Specification of the dSpace hardware setup.

Board	Function	Specification
DS1005	Processor	PowerPC 750, 480MHz
DS2003	Analogue input	32 x 16 bit channels
DS2103	Analogue output	32 x 14 bit channels
DS2201	Multi I/O	20 x 12 bit ADC, 8 x 12 bit DAC, 16 DIO channels
DS4002	Digital I/O	32 DIO channels
DS4302	CAN interface	4 channels

EXPERIMENTAL SETUP

In order to better understand the response of the cylinder charge composition to changes in the actuator positions, a model of the air-path is required. This model will then provide the basis for developing the feed-forward control action used to speed-up the response to changes in the demanded charge properties.

The basic approach adopted here is to use a mean-value, filling and emptying type model for the intake manifold. The intake is assumed to be a fixed volume of an ideal gas whose state is determined by the pressure and mass in the volume. The flows into the intake manifold are given by the effective areas and pressure differences across the EGR valve and throttle. This type of modelling approach has been discussed extensively in the literature [6, 8, 9, 10, 17].

In describing the elements of the model and feed-forward control implementation it will be important to distinguish between measured quantities, demanded quantities and modelled quantities. The notation used is to give modelled quantities no embellishment, demanded quantities a bar and measured quantities a hat.

INTAKE MANIFOLD

The gas in the intake manifold is modelled as ideal gas with constant specific heats, taken to be those of atmospheric air. Hence the state in the manifold is given by:

$$p_i V_i = m_i R T_i \quad (1)$$

Where p_i is the pressure in the intake manifold, V_i is the volume of the intake manifold, m_i is the mass of gas in the intake manifold, R is the gas constant and T_i is the temperature of the gas in the intake manifold. Differentiating [equation 1](#) gives:

$$\frac{V_i}{R} \dot{p}_i = \dot{m}_i T_i + m_i \dot{T}_i \quad (2)$$

Now, assuming the gas in the intake manifold to be ideal, its internal energy is given by:

$$U_i = c_v m_i T_i \quad (3)$$

Where U_i is the internal energy and c_v is the specific heat at constant volume. Differentiating [equation 3](#) gives:

$$\frac{\dot{U}_i}{c_v} = \dot{m}_i T_i + m_i \dot{T}_i \quad (4)$$

Equating [equations 2](#) and [4](#) therefore gives:

$$\dot{U}_i = \frac{c_v V_i}{R} \dot{p}_i \quad (5)$$

Conservation of energy for the intake volume and the ideal gas assumption lead to an alternative expression for the rate of change of internal energy in the intake volume:

$$\dot{U}_i = c_p (w_{th} \hat{T}_{th} + w_{egr} \hat{T}_{egr} - w_{ie} T_i) - \dot{Q}_i \quad (6)$$

Where c_p is the specific heat at constant pressure, w_{th} is the mass flow through the throttle, $\hat{T}_{th}$ is the measured temperature of the gas flowing through the throttle, w_{egr} is the mass flow through the EGR valve, $\hat{T}_{egr}$ is the measured temperature of the gas flowing through the EGR valve, w_{ie} is the mass flow from the intake manifold into the engine and $\dot{Q}_i$ is the heat transfer from the manifold to the environment. If heat loss from the manifold is neglected, [equations 5](#) and [6](#) can be combined to give:

$$\frac{V_i}{\gamma R} \dot{p}_i = w_{th} \hat{T}_{th} + w_{egr} \hat{T}_{egr} - w_{ie} T_i \quad (7)$$

Where γ is the ratio of specific heats. Conservation of mass is used to give:

$$\dot{m}_i = w_{th} + w_{egr} - w_{ie} \quad (8)$$

And [equations 7](#) and [8](#) together form the description of the intake manifold.

ORIFICE FLOW

The mass flow through the intake throttle is described using the orifice flow equation [18]:

$$w_{th} = \begin{cases} \frac{A_{th} \hat{p}_{th}}{\sqrt{R \hat{T}_{th}}} \left(\frac{p_i}{\hat{p}_{th}} \right)^{\frac{1}{\gamma}} \left\{ \frac{2\gamma}{\gamma-1} \left[1 - \left(\frac{p_i}{\hat{p}_{th}} \right)^{\frac{\gamma-1}{\gamma}} \right] \right\}^{\frac{1}{2}} & \text{for } \left(\frac{p_i}{\hat{p}_{th}} \right) > p_{cr} \\ \frac{A_{th} \hat{p}_{th}}{\sqrt{R \hat{T}_{th}}} \gamma^{\frac{1}{2}} \left(\frac{2\gamma}{\gamma+1} \right)^{\frac{\gamma+1}{2\gamma-2}} & \text{for } \left(\frac{p_i}{\hat{p}_{th}} \right) \leq p_{cr} \end{cases}$$

$$p_{cr} = \left(\frac{2}{\gamma+1} \right)^{\gamma/(\gamma-1)} \quad (9)$$

Where A_{th} is the effective area of the throttle and $\hat{p}_{th}$ is the measured pressure before the throttle. The same description is used for the EGR valve but with the upstream conditions obviously changed to be those of the EGR valve.

ENGINE BLOCK

The mass flow from the intake manifold to the engine is given by a constant volumetric efficiency:

$$w_{ie} = \eta_v \frac{\hat{N} V_d p_i}{120 R T_i} \quad (10)$$

Where η_v is the volumetric efficiency, $\hat{N}$ is the measured engine speed in RPM and V_d is the displacement volume of the engine. The mass flow out of the engine is given by:

$$w_{ex} = w_{ie} + \hat{w}_f \quad (11)$$

Where w_{ex} is the mass flow out of the engine block and $\hat{w}_f$ is the measured mass flow of fuel.

CYLINDER CHARGE COMPOSITION

The gas at any point in the air system is considered to be a mixture of fresh air and combustion products (burned gas). It is natural therefore to define the gas composition in terms of the Burned Gas Fraction (BGF). Under normal lean operation the fuel is assumed to burn completely and the BGF will therefore be proportional to the CO_2 concentration. The

burned gas fraction of the gas leaving the engine will be given by:

$$F_{ex} = \frac{F_{ie} w_{ie} + \bar{w}_f (1 + \chi_s)}{w_{ex}} \quad (12)$$

Where F_{ex} is the burned gas fraction of the gas leaving the cylinders, F_{ie} is the burned gas fraction of the gas entering the cylinders and χ_s is the stoichiometric air/fuel ratio. The flow of burned gas from the exhaust back into the cylinders is modelled by assuming plug-flow through the air-system as this gives a better fit with measurements than assuming a well-stirred reactor type description of the gas mixing [2]. The transport delay between the block and the EGR valve is approximated by considering the volumetric flow rate through the exhaust manifold and the volume of the manifold:

$$\Delta t_{ex} = \frac{p_{egr} V_x}{w_{egr} R \hat{T}_{egr}} \quad (13)$$

Where Δt_{ex} is the time delay between gas leaving the engine and arriving at the EGR valve. Therefore the burned gas fraction at the EGR valve is:

$$F_{egr}(t) = F_{ex}(t - \Delta t_{ex}) \quad (14)$$

Where F_{egr} is the burned gas fraction at the EGR valve. The gas flowing through the EGR valve mixes with fresh air and the burned gas fraction of the mixed gas is given by:

$$F_{mix} = F_{egr} \frac{w_{egr}}{w_{ie}} \quad (15)$$

The transport delay between the EGR valve and the cylinders is approximated by considering the volumetric flow rate into the engine and the volume of pipework between the EGR mixing point and the engine:

$$\Delta t_{ie} = \frac{p_i V_{mix}}{w_{ie} R T_i} \quad (16)$$

Where Δt_{ie} is the time delay between the EGR valve and the engine and V_{mix} is the volume of the intake manifold that lies between the EGR valve and the engine (and is substantially smaller than the entire volume between the throttle and the engine). The composition of the gas entering the cylinders can then be given by:

$$F_{ie}(t) = F_{mix}(t - \Delta t_{ie}) \quad (17)$$

Equations 12, 14 and 17 taken together form the model of the composition of the gas entering the cylinders.

FEED-FORWARD CONTROL ACTION

The feed-forward control action is given by an ad-hoc approach based on the physical model of the intake manifold described above. A pre-filter converts the demanded charge composition in terms of burned gas fraction and total mass-flow into a demanded mass-flow through the EGR valve and a demanded pressure in the intake manifold. Two single-input-single-output channels are then implemented: the EGR valve controls the mass-flow through the EGR valve and the intake throttle controls the manifold pressure. The demanded mass-flow of EGR is converted directly to a specific EGR valve area using the orifice flow description in equation 9 and this is converted to an EGR valve position by an experimentally determined look-up table. Following the demanded pressure is achieved by using a high proportional gain on the error between the demanded manifold pressure and the modelled intake manifold pressure. This set-up is illustrated in figure 2 below:

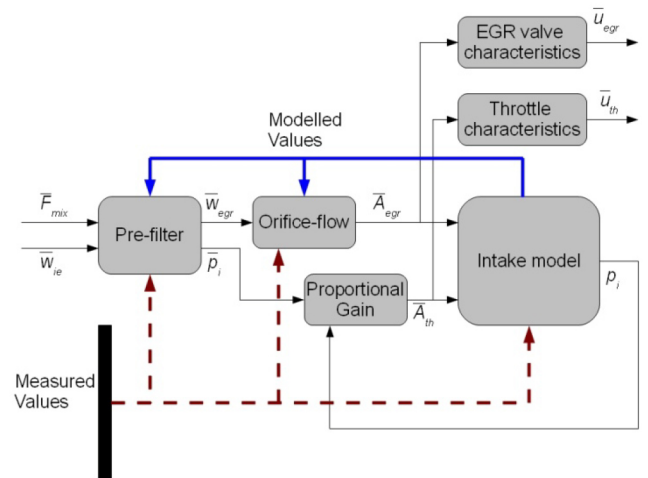


Figure 2. Feed-forward control implementation used in the experimental tests

The ideal throttle control is obvious, the area should be that required to follow the demanded manifold pressure if this is possible and saturate open or closed if this is not possible. In the scenario examined in this paper (a step change in the charge properties from a leaner to a richer condition) the manifold pressure decreases so the expected throttle behaviour is to close fully from its original position while the manifold pressure reduces and then open to the position necessary to deliver the correct cylinder charge properties.